Bundesministerium
für Verkehr,
Innovation und Technologie

NACHHALTIGwirtschaften
konkret

IEA Solar Heating and Cooling Programm

Task 38: Solare Kühlung und Klimatisierung

Subtask A: Kleine und mittlere Kompaktsysteme

D. Jähnig

Berichte aus Energie- und Umweltforschung

40/2012

Dynamik mit Verantwortung

Impressum:

Eigentümer, Herausgeber und Medieninhaber:
Bundesministerium für Verkehr, Innovation und Technologie
Radetzkystraße 2, 1030 Wien

Verantwortung und Koordination:
Abteilung für Energie- und Umwelttechnologien
Leiter: DI Michael Paula

Liste sowie Downloadmöglichkeit aller Berichte dieser Reihe unter
http://www.nachhaltigwirtschaften.at

IEA Solar Heating and Cooling Programm

Task 38: Solare Kühlung und Klimatisierung

Subtask A: Kleine und mittlere Kompaktsysteme

DI Dagmar Jähnig, DI Dr. Alexander Thür
Charlotta Winkler, Martin Vukits, Dietmar Amtmann
AEE - Institut für Nachhaltige Technologien

Erich Podesser
Podesser Consulting

Philip Ohnewein, Nicole Olsacher, Sabine Putz,
Moritz Schubert, Christian Holter
S.O.L.I.D.

Gleisdorf, Juni 2011

Ein Projektbericht im Rahmen der Programmlinie

IEA FORSCHUNGS KOOPERATION

Impulsprogramm Nachhaltig Wirtschaften

Im Auftrag des Bundesministeriums für Verkehr, Innovation und Technologie

Vorbemerkung

Der vorliegende Bericht dokumentiert die Ergebnisse eines Projekts aus dem Programm FORSCHUNGSKOOPERATION INTERNATIONALE ENERGIEAGENTUR. Es wurde vom Bundesministerium für Verkehr, Innovation und Technologie initiiert, um Österreichische Forschungsbeiträge zu den Projekten der Internationalen Energieagentur (IEA) zu finanzieren.

Seit dem Beitritt Österreichs zur IEA im Jahre 1975 beteiligt sich Österreich aktiv mit Forschungsbeiträgen zu verschiedenen Themen in den Bereichen erneuerbare Energieträger, Endverbrauchstechnologien und fossile Energieträger. Für die Österreichische Energieforschung ergeben sich durch die Beteiligung an den Forschungsaktivitäten der IEA viele Vorteile: Viele Entwicklungen können durch internationale Kooperationen effizienter bearbeitet werden, neue Arbeitsbereiche können mit internationaler Unterstützung aufgebaut sowie internationale Entwicklungen rascher und besser wahrgenommen werden.

Dank des überdurchschnittlichen Engagements der beteiligten Forschungseinrichtungen ist Österreich erfolgreich in der IEA verankert. Durch viele IEA Projekte entstanden bereits wertvolle Inputs für europäische und nationale Energieinnovationen und auch in der Marktumsetzung konnten bereits richtungsweisende Ergebnisse erzielt werden.

Ein wichtiges Anliegen des Programms ist es, die Projektergebnisse einer interessierten Fachöffentlichkeit zugänglich zu machen, was durch die Publikationsreiche und die entsprechende Homepage **www.nachhaltigwirtschaften.at** gewährleistet wird.

Dipl. Ing. Michael Paula
Leiter der Abt. Energie- und Umwelttechnologien
Bundesministerium für Verkehr, Innovation und Technologie

Inhalt

1 Kurzfassung

Übergeordnete Ziele der IEA SHC Task 38 "Solare Kühlung und Klimatisierung" waren die Entwicklung und Erprobung von Kompaktsystemen zur solaren Heizung und Kühlung im kleinen und mittleren Leistungsbereich; die Entwicklung und Erprobung von kundenspezifischen großen solaren Klimatisierungs- und Kühlanlagen für den Nicht-Wohnbereich sowie die Entwicklung von neuen Konzepten für solare Kühlmaschinen.

Über den technologischen Schwerpunkt hinaus war es das Ziel des Vorhabens, die Ergebnisse der Forschungs- und Entwicklungsarbeiten möglichst rasch an die möglichen Nutzer und Anwender sowie an die Industrie weiterzugeben, um die Implementierung der Ergebnisse am Markt voranzutreiben.

Die Arbeiten in der IEA SHC Task 38 basierten auf den Ergebnissen von zwei bereits abgeschlossenen IEA SHC Tasks, Task 25 (Solare Klimatisierung von Gebäuden) und Task 26 (Solare Kombianlagen). Darauf aufbauend wurden Anlagen zur solaren Heizung und Kühlung entwickelt, getestet und erprobt:

- Kompaktsysteme für solares Heizen und Kühlen für den kleinen und mittleren Leistungsbereich (2-20 kW)
- Kundenspezifische große solare Klimatisierungs- und Kühlanlagen für den Nicht-Wohnbereich
- Neue Konzepte für solare Kühlmaschinen und andere Komponenten

Die Arbeit in der Task 38 wurde in vier sogenannten Subtasks durchgeführt:

Subtask A: Kleine und mittlere Kompaktsysteme
Subtask B: Große kundenspezifische Anlagen
Subtask C: Simulation und theoretische Grundlagen
Subtask D: Know-how Transfer

Der Schwerpunkt der Arbeiten der an diesem Projekt teilnehmenden Partner lag auf Subtask A, der von der AEE INTEC geleitet wurde.

Hier wurden Systemkonzepte für kleine und mittlere Kompaktsysteme entwickelt, miteinander verglichen und in Pilotanlagen erprobt. Ziel war es, Systempakete anbieten zu können, die eine solare Kombianlage zur Warmwasserbereitung und Heizung umfasst und die überschüssige Solarenergie im Sommer zum Kühlen verwendet. Solche Systeme sollten als Komplettpaket angeboten werden, so dass sie von Installateuren ohne aufwändige Einzelstudien geplant und gebaut werden können (sog. pre-engineered systems).

Ein wesentlicher Input für die Entwicklung von vorkonfigurierten solaren Kühlsystemen sind die praktischen Erfahrungen von ersten Demonstrationsanlagen. Neben dem Austausch betriebstechnischer Erfahrungen auf persönlicher Basis in Form von Vorträgen bei den Expertentreffen wurde eine einheitliche Auswertung der Messdaten koordiniert. Dazu wurde ein einheitliches Prozedere zur Messdatenerfassung bzw. Messdatenauswertung definiert, um letztendlich vergleichbare Kenndaten zu bekommen.

Dazu wurde eine Excel-Tabelle entwickelt, in die die Messdaten von einem ganzen Jahr eingetragen werden können und in der dann automatisch eine ganze Reihe von Kennzahlen berechnet wird.

Diese Excel-Tabelle wurde in weiterer Folge verwendet, um Messdaten von 11 kleinen Anlagen zum solaren Heizen und Kühlen (Subtask A) und 11 großen Anlagen (Subtask B) zu analysieren und miteinander zu vergleichen. Davon waren drei kleine und drei große Anlagen in Österreich.

Desweiteren waren die Projektpartner dieses Projektes auch in den anderen Subtasks aktiv. Dies betrifft insbesonders

Subtask B: Analyse der Anlage im Rathaus/Service Center der Stadt Gleisdorf
Subtask C: Neuentwicklungen (Parabolrinne und Dampfstrahlkältemaschine), hygienische Rückkühleinrichtungen, Analyse bestehender Simulationstools
Subtask D: Handbuch für Planer, Mitarbeit bei der kompletten Neuauflage des Handbuchs aus dem vorangegangenen Task 25.

Die IEA SHC Task 38 wurde mit dem Kick-Off Meeting im Oktober 2006 gestartet, wurde einmal verlängert und im Dezember 2010 abgeschlossen.

Die Ergebnisse des Task 38 sind in einer Reihe von „Technical Reports" nachzulesen. Ein Teil dieser Berichte ist bereits jetzt von der IEA-SHC Homepage www.iea-shc.org herunterzuladen, einige Berichte wurden bzw. werden erst im Laufe des Jahres 2011 fertiggestellt und werden dann ebenfalls von der genannten Homepage heruntergeladen werden können.

Ein Großteil der Taskergebnisse ist außerdem in das Handbuch für Planer eingeflossen, das im Laufe des Jahres 2011 im Buchhandel erhältlich sein wird.

2 Summary

The overall goal of IEA SHC Task 38 „Solar Air Conditioning and Refrigeration" is the development and testing of compact solar heating and cooling systems with low nominal capacity, custom-built high capacity systems for non-residential applications and the development of new thermally-driven cooling machine concepts and components for solar applications.

Beyond the technological focal point, the goal of the collaborative research project is an efficient know-how transfer to the users of this technology and to the relevant industry in order to promote market implementation.

IEA SHC Task 38 is based on the results of two previous IEA-SHC tasks, Task 26 Solar Combisystems and Task 25 Solar assisted air conditioning of buildings. Based on the results of these two tasks, concepts and systems for solar heating and cooling will be developed and tested:

- Compact systems for solar heating and cooling in a capacity range of 2 – 20 kW.
- Custom-built high capacity systems for non-residential applications
- New concepts of thermally-driven cooling machines and components

IEA SHC Task 38 is divided into 4 subtasks:

Subtask A: Small and medium pre-engineered systems
Subtask B: Large sized custom-made systems
Subtask C: Modeling and fundamental analysis
Subtask D: Market transfer activities

The main activities within Task 38 of the partners participating in this project are in Subtask A, which is led by AEE INTEC.

In this subtask, system concepts for small and medium compact systems have been developed, compared and tested in pilot applications. The goal was to come to system packages that contain a solar combisystem for domestic hot water and space heating and use the surplus solar energy in summer for cooling applications. Such systems should be offered as pre-engineered packages so that they can be designed and built by installers without the need of customized design studies.

Practical operation experiences from demonstration systems are an important input for the development of pre-engineered solar heating and cooling systems. Exchange of experiences was achieved on a personal level by means of presentations and discussion at the expert meetings. In addition a common procedure for monitoring of demonstration plants as well as for the analysis of these data was defined. This makes monitoring data of different plants comparable.

An Excel-Tool was developed where data of a whole operating year can be entered. Automatically the tool will then calculate a large number of key figures to compare.

This Excel-Tool was used to analyze monitoring data of 11 small and medium scale plants (Subtask A) and 11 large custom-made plants (Subtask B) and to compare them with eachother. Three of the small and medium scale plants and three of the large plants were in Austria.

The partners of this project were also involved in other activities of Task 38. This concerns mainly the following work packages

Subtask B: Analysis of the solar heating and cooling system of the town hall / service center of the city of Gleisdorf
Subtask C: New developments (parabolic trough and steam jet ejector chiller), hygienic heat rejection units, analysis of simulation tools
Subtask D: Handbook for planners, co-authors for the completely new edition of the handbook of Task 25.

IEA SHC Task 38 started with the kick-off meeting in October 2006, was prolonged and ended in December 2010.

The results of the task are published in a series of technical reports. A part of these reports can already be downloaded from the IEA-SHC hompage www.iea-shc.org, some reports are only being completed during 2011 and can then also be downloaded from this homepage.

The main part of the task results will be included in the new edition of the "Handbook for Planners" that will be available in bookshops by the end of 2011.

3 Einleitung

3.1 Die Rahmenbedingungen

Nach dem Jahrhundertsommer 2003 stellte sich zunächst die Frage, ob Wohn- und Verwaltungsgebäude auch in Mitteleuropa zukünftig aktiv gekühlt werden müssen. Anhand von Messungen, die von der AEE INTEC an einem optimal gedämmten Passiv-Verwaltungsbau (Christophorus Haus) durchgeführt wurden, konnte zunächst gezeigt werden, dass bei gut gedämmten Gebäuden und geringen internen Lasten bei passiver Nachtlüftung und „Direct Cooling" von wasserdurchströmten Deckenpaneelen und Fußbodenelementen über Tiefensonden extreme Spitzen über 27°C nicht auftreten.

Bei weniger optimierten Gebäuden, hohen internen Lasten oder hohen Komfortansprüchen werden weltweit jedoch zunehmend mehr Klimaanlagen eingebaut – und das nicht nur im Verwaltungsbau. Jährlich werden alleine etwa 43 Millionen Raumklimageräte verkauft mit den Hauptmärkten in China (12 Millionen) und den USA (11,8 Millionen). Die Marktdurchdringung in Europa ist geringer (2,8 Millionen Einheiten 2002), weist jedoch eine hohe Wachstumsrate auf. Gleichzeitig werden in China jährlich mehrere Millionen Quadratmeter Solarkollektorfläche installiert und Österreich ist in Europa führend auf dem Solarthermiemarkt.

Wenn es gelingt, die elektrischen Kompressionskältemaschinen mit ihrer sommerlichen Stromnetzbelastung durch thermische Kälte zu ersetzen, dann eröffnet sich ein großes, weltweites Wachstumsfeld für die österreichische Solarindustrie.

Sommerliche Gebäudeklimatisierung stellt sowohl in Wohngebäuden als auch in kommerziell genutzten Gebäuden einen weltweit wachsenden Markt dar. Hauptgründe hierfür sind steigende interne Kühllasten und wachsende Komfortansprüche sowie architektonische Trends, beispielsweise hin zu Gebäuden mit hohem Glasanteil in der Fassade.

Für große, klimatechnische Anlagen mit Kälteleistungen oberhalb von rund 50 kW stehen verschiedene thermisch angetriebene Techniken zur Verfügung, die in Verbindung mit thermischen Solarkollektoren für die sommerliche Gebäudeklimatisierung verwendet werden können. Neben den erhöhten Investitionskosten ist das Haupthindernis für eine breite Nutzung dieser Technik der Mangel an praktischer Erfahrung in der Auslegung, der Regelung und dem Betrieb dieser Systeme. Für kleine Anlagen mit Leistungen unter 30 kW gibt es kaum marktverfügbare Komponenten. Deshalb ist die Entwicklung von thermisch angetriebenen Klimatisierungsverfahren im kleinen Leistungsbereich für die solar unterstützte Klimatisierung von besonderer Bedeutung.

In den USA und Japan gab es in den 80er Jahren des vergangenen Jahrhunderts große F&E-Aktivitäten zur Entwicklung von Verfahren der solar unterstützten Klimatisierung, die allerdings wegen der ungünstigen Wirtschaftlichkeit eingestellt wurden. Mittlerweile hat sich jedoch der Markt für thermische Kollektoren erheblich entwickelt und es sind viele Systeme verschiedener Anbieter aus zunehmend industrieller Fertigung im Markt verfügbar. Vor diesem Hintergrund wurden in den vergangenen Jahren verschiedene Aktivitäten gestartet, um das Thema der solar unterstützten Klimatisierung erneut aufzugreifen.

3.2 Verfügbare Technologien

Aus thermodynamischer Sicht gibt es etliche Verfahren zur Umwandlung von Solarstrahlung in Kälte bzw. gekühlte und entfeuchtete Luft. Eine Übersicht zeigt Abb. 1. Dabei wird hier die Umwandlung von Solarstrahlung in Elektrizität mit Solarzellen und die

anschließende Kältebereitstellung auf der Basis von Kompressionskälteanlagen nicht weiter betrachtet. In aller Regel ergibt sich der größte Nutzen von Photovoltaik-Anlagen bei Einspeisung der gewonnenen elektrischen Energie in das Stromnetz. Dies gilt insbesondere, wenn eine Einspeisevergütung wie beispielsweise in Österreich gewährt wird. Insofern spielt die systemtechnische Betrachtung solar-elektrischer Verfahren für die Komfort-Klimatisierung – außer in Nischenanwendungen – in der Praxis nur eine untergeordnete Rolle.
Von den Verfahren, die Solarkollektoren für den thermischen Antrieb verwenden können, sind alle sorptionstechnischen Verfahren am weitesten entwickelt.

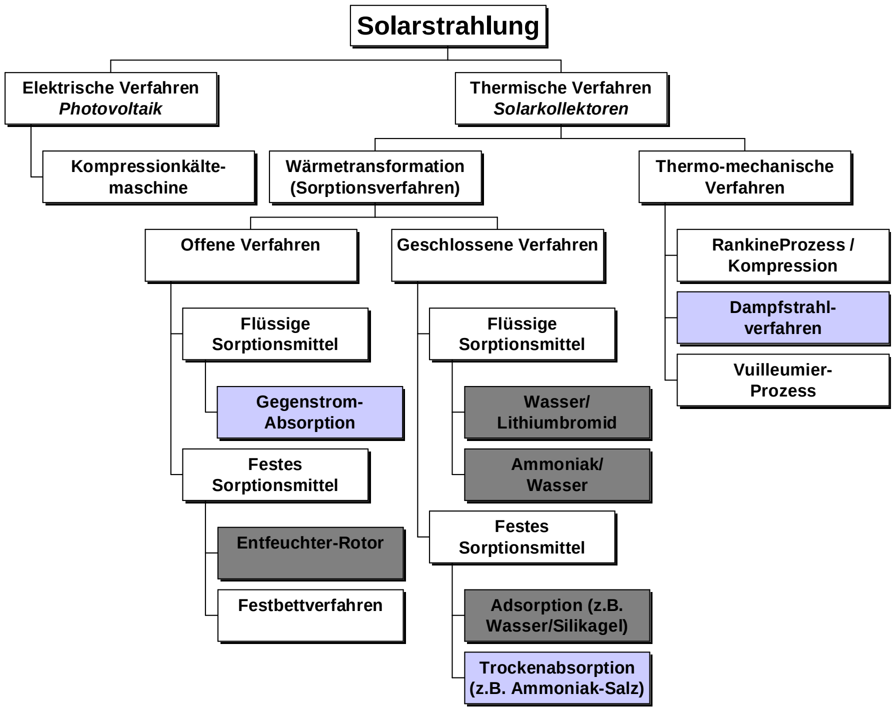


Abb. 1: Übersicht über physikalische Wege der Umwandlung von Solarstrahlung in Kälte bzw. konditionierte Luft.

Dunkelgrau markierte Prozesse stellen marktverfügbare Technologien dar, die derzeit in der Praxis am häufigsten für solar unterstützte Klimatisierung Verwendung finden. Schwach grau markierte Verfahren befinden sich in einem Status von Pilotanlagen oder Systemtests. (Quelle: Henning, Hans-Martin (Hrsg.): Solar-Assisted Air-Conditioning in Buildings – A Handbook for Planners, Springer Verlag, 2004)

Im Folgenden werden die heute am häufigsten eingesetzten Techniken sowie die Solar-kollektoren, die für die Wärmebereitstellung eingesetzt werden können, kurz beschrieben.

3.2.1 Thermisch angetriebene Kaltwassererzeugung

Die heute dominante Technik thermisch angetriebener Kaltwassererzeugung basiert auf Absorptionsverfahren. Absorptionskälteanlagen sind am Markt in einem großen Leistungsbereich verfügbar, wobei allerdings nur wenige Anlagen im Leistungsbereich unter 100 kW als marktreife Serienprodukte angeboten werden, die mit Heißwasser – beispielsweise von Solarkollektoren – betrieben werden können. Für Anwendungen der Komfortklimatisierung in Gebäuden werden hauptsächlich Absorptionskälteanlagen, die mit dem Stoffpaar Wasser-Lithiumbromid arbeiten, genutzt. Der COP (Coefficient of Performance), definiert als die nutzbare Kälte pro eingesetzter Wärmemenge, liegt bei einstufigen Anlagen – je nach aktuellen Betriebsbedingungen - im Bereich 0,65-0,8 und die Antriebstemperaturen liegen typischerweise bei rund 75-100°C (Vorlauftemperatur des Heizmediums).

Neben den einstufigen Anlagen sind auch zweistufige Maschinen erhältlich, bei denen zwei Austreiber, die bei unterschiedlichen Temperaturen arbeiten, genutzt werden. Dabei wird die Kondensationswärme des bei hoher Temperatur kondensierenden Kältemittels zum thermischen Antrieb des zweiten Austreibers genutzt. Somit kann ein höherer COP im Bereich von 1,0-1,4 – je nach Betriebsbedingungen – erzielt werden, wofür typischerweise eine Antriebstemperatur im Bereich 140-160°C erforderlich ist. Hier sind allerdings nur Anlagen im größeren Leistungsbereich deutlich oberhalb 100 kW erhältlich.

An Stelle des flüssigen Sorptionsmittels können auch feste Sorbentien zur Aufnahme des verdampften Kältemittels eingesetzt werden. Seit Anfang 2007 wird von einem deutschen Hersteller die erste Kältemaschine mit dieser Technologie im Leistungsbereich unter 20 kW in Feldtestanlagen auf dem Markt eingeführt.

3.2.2 Offene Verfahren – sorptionsgestützte Klimatisierung

Während thermisch angetriebene Kältemaschinen Kaltwasser bereitstellen, das zur Versorgung aller Arten klimatechnischer Geräte verwendet werden kann, wird in offenen Verfahren direkt Luft konditioniert. Letztlich basieren alle offenen Klimatisierungsverfahren auf einer Kombination von Verdunstungskühlung und thermisch angetriebener Luftentfeuchtung mittels eines Trocknungsmittels, d.h. einem hygroskopischen Material. Wiederum können entweder flüssige oder feste Sorptionsmittel zum Einsatz gelangen. Der heute überwiegend angewandte Prozess nutzt hierfür Sorptionsrotoren, die entweder mit Silikagel oder Lithiumchlorid als Sorptionsmittel ausgestattet sind. Marktübliche Anlagen mit Sorptionsrotoren können Antriebstemperaturen im Bereich 50-85°C zur Regenerierung des Sorptionsrotors nutzen.

Das heute am weitesten verbreitete offene Verfahren verwendet Sorptionsrotoren.

3.2.3 Solarkollektoren

Die wichtigsten Bauformen von Solarkollektoren, die für die solar unterstützte Klimatisierung genutzt werden können, sind die folgenden:

- Solarluftkollektoren dienen der direkten Lufterwärmung
- Flachkollektoren sind in Europa die üblichste Bauform und erwärmen einen flüssigen Wärmeträger, in der Regel Wasser-Glykol oder Wasser; die Hauptanwendung liegt in der solaren Brauchwassererwärmung
- Vakuum-Röhren-Kollektoren gibt es in vielen unterschiedlichen Bauformen; wegen der geringeren Wärmeverluste sind sie insbesondere für die Bereitstellung höherer Temperaturen interessant
- Parabolrinnenkollektoren und Fresnelkollektoren werden derzeit im Rahmen der IEA SHC und Solar Paces Task 33/IV in mehreren europäischen Ländern entwickelt. Sie sind durch ihre höheren Wirkungsgrade bei Temperaturen deutlich über 100°C für Kälteverfahren mit hohen Antriebstemperaturen interessant.

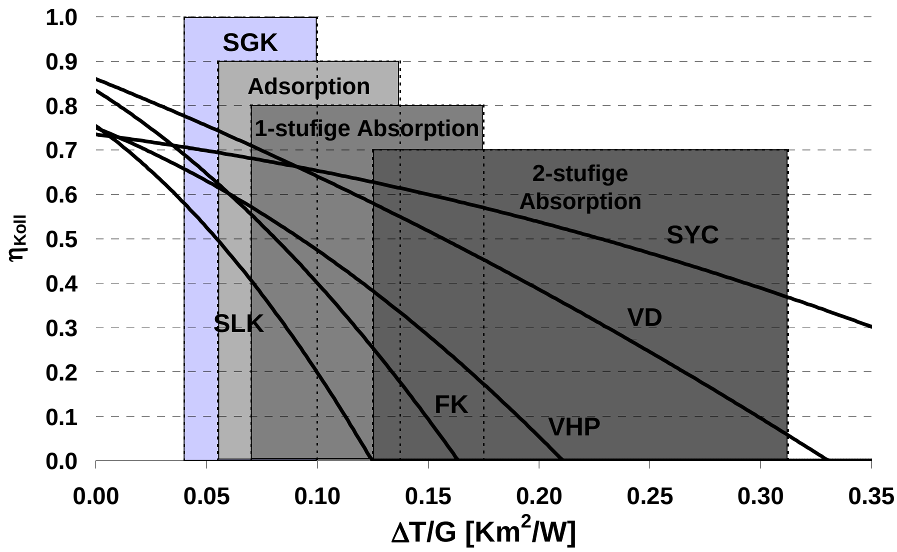


Abb. 2: Typische Wirkungsgrad-Kennlinien unterschiedlicher Solarkollektoren als Funktion der Temperaturdifferenz zwischen Kollektorbetriebstemperatur und Umgebungstemperatur, ΔT, dividiert durch die Einstrahlung auf den Kollektor, G.

Die Abkürzungen bedeuten: SLK = Solar-Luft-Kollektor, FK = Flachkollektor, VHP = Vakuum-Röhren-Kollektor mit Heat-Pipe, VD = direkt durchflossener Vakuum-Röhren-Kollektor, SYC = Ganzglas-Vakuum-Röhren-Kollektor mit Konzentrator, SGK = sorptionsgestützte Klimatisierung. Die markierten Flächen charakterisieren den typischen Betriebsbereich unterschiedlicher thermisch angetriebener Verfahren zur Kühlung/Klimatisierung.

(Quelle: Henning, Hans-Martin (Hrsg.): Solar-Assisted Air-Conditioning in Buildings – A Handbook for Planners, Springer Verlag, 2004)

In Abb. 2 sind die heute wichtigsten kälte- und klimatechnischen Verfahren für die solar unterstützte Klimatisierung in einem Diagramm mit typischen Kennlinien unterschiedlicher Solarkollektoren dargestellt. Dabei stellt die Ordinate die Temperaturdifferenz aus der Betriebstemperatur T_{op} des Solarkollektors und der Außentemperatur, T_{umg}, dividiert durch die Einstrahlung G auf den Kollektor dar.

Der jeweils markierte Bereich für eines der Verfahren ist durch die jeweilige Betriebstemperatur sowie durch die Einstrahlung auf den Kollektor charakterisiert. Der linke Rand gilt dabei für eine hohe Einstrahlungsleistung von 1000 W/m^2 und der rechte Rand für eine Einstrahlungsleistung von 400 W/m^2. Die Abbildung gibt einen Hinweis darauf, welche Kollektoren für welche Verfahren geeignet sind.

3.3 Rahmenbedingungen in Österreich

Bis Ende 2006 waren in Österreich insgesamt 3,3 Mio. m² Kollektorfläche mit einer installierten Leistung von 2,3 GW_{th} in Betrieb. Die breite Anwendung von solarthermischen Anlagen konzentrierte sich bisher nahezu ausschließlich auf die Bereiche Schwimmbäder sowie Warmwasserbereitung und Raumheizungsunterstützung für Einfamilien- und Mehrfamilienhäuser sowie im Bereich der Nahwärmeversorgung.

In diesen Anwendungsbereichen wurden seit Mitte der 80er Jahre erhebliche Wachstumsraten bei der installierten Kollektorfläche erzielt. Damit ist Österreich in Europa bei der installierten Kollektorfläche pro Einwohner im Spitzenfeld.

Der Bereich der solaren Raumheizungsanlagen verzeichnete seit 1998 hohe Zuwachsraten. Rund 50% der jährlich installierten Kollektorfläche werden für solare

Raumheizungsanlagen eingesetzt. Einer der Gründe dafür liegt auch in der ständigen Weiterentwicklung der Komponenten und Systeme für dieses Anwendungssegment.
Ein wesentlicher Impuls war dafür auch die österreichische Beteiligung an der Task 26 „Solar Combisystems" im Rahmen des Solar Heating and Cooling Programme der IEA.
Unter Leitung der AEE INTEC wurden in Task 26 von 1998 bis 2002 solare Raumheizungsanlagen weiterentwickelt und optimiert.

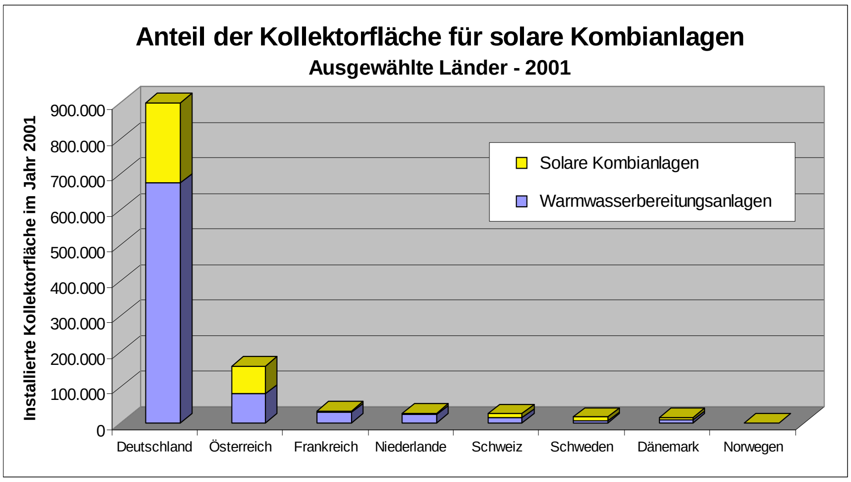


Abb. 3: Anteil der solaren Kombianlagen für Warmwasser und Raumheizung an der installierten Gesamtkollektorfläche im Jahr 2001

Österreichische Unternehmen sind seit einigen Jahren auch im Export von Anlagen sehr erfolgreich. Im Jahr 1998 betrug der Exportanteil, der in Österreich gefertigten Kollektoren lediglich 24 % (58.100 m²), bis zum Jahr 2000 verdoppelte sich die exportierte Kollektorfläche auf 112.000 m², bis 2004 konnte eine Steigerung des Exportvolumens auf 320.000 m² und bis 2006 auf 844.000 m^2 oder 76 % der Produktion erzielt werden (Faninger, 2004 bis 2006). Österreich ist damit der weltgrößte Exporteur von Sonnenkollektoren.

Nicht nur durch Standardprodukte, wie Flachkollektoren, sondern auch durch die Spezialisierung auf neue Nischenmärkte wie CPC-Kollektoren oder Fassadenkollektoren konnten österreichische Unternehmen beachtliche Verkaufszahlen in Österreich und im Export erzielen.

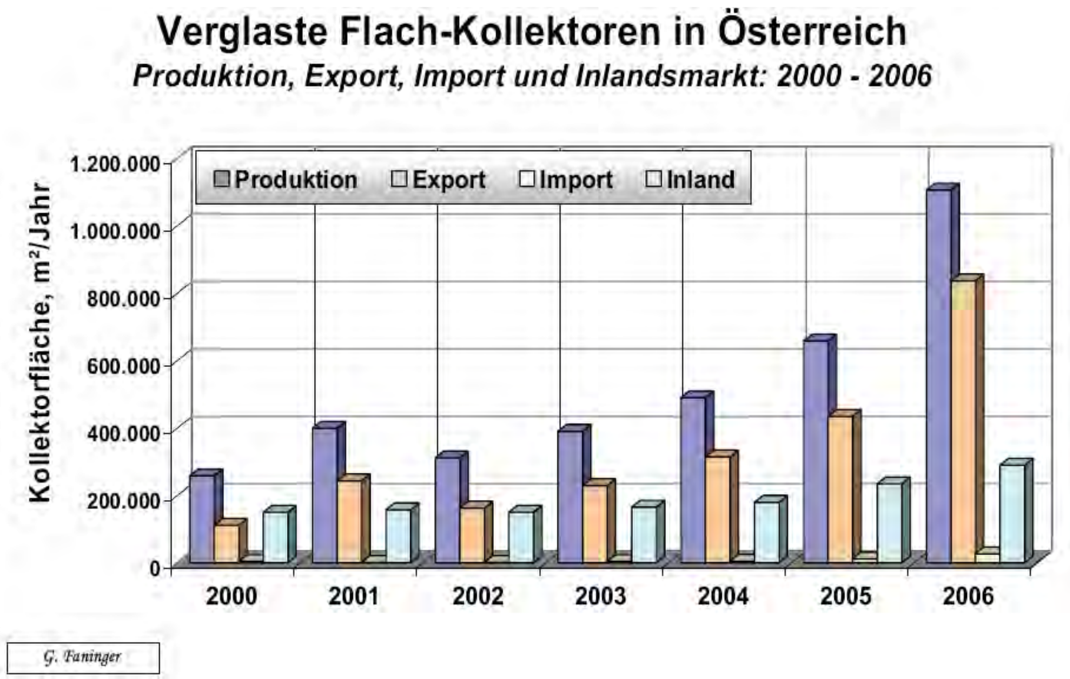


Abb. 4: Der Flachkollektormarkt von 2000 bis 2006 in Österreich (Faninger, 2006)

Der Vorteil österreichischer Unternehmen war bisher vor allem durch ein gutes Preis-Leistungsverhältnis geprägt. Zur Absicherung der hervorragenden Marktposition österreichischer Unternehmen bedarf es allerdings permanenter technischer Innovationen und der Erschließung neuer Anwendungsbereiche. Solare Kühlung und Klimatisierung stellt unzweifelhaft einen solchen neuen Anwendungsbereich dar.

3.4 Stand der solaren Kühlung und Klimatisierung in Österreich

Der Stand der Entwicklung von Komponenten und Systemen sowie die Realisierung von Pilotanlagen ist in Österreich in einem Anfangsstadium im Vergleich zu Deutschland, wo eine größere Zahl sowohl von F&E Projekten, als auch von Pilot- und Demonstrationsprojekten durchgeführt wurden.

Innerhalb der Tasklaufzeit ist die Anzahl der installierten Anlagen zum solaren Heizen und Kühlen in Österreich stark gestiegen. Im Subtask B wurden die bekannten Anlagen zusammengetragen. Für Österreich finden sich hier 20 Einträge, inzwischen mögen auch noch einige hinzugekommen sein.

Tabelle 1: Anlagen zum solaren Heizen und Kühlen in Österreich

Name der Anlage, Baujahr, Nom. Kälteleistung, Hersteller und Typ der Kältemaschine

5 Austria Weinbaubetrieb Peitler Leutschach wine cellar 2003 10 Pink (Ammonia/Water)

6 Austria Fa. SOLID Graz office building 2003 2 Kunze (Ammonia/Water)

7 Austria Fa. SOLution Sattledt office building 2005 15 EAW (Water/LiBr)

8 Austria Privathaus Jungreithmayr Sattledt residential building 2007 5.5 Sortech (Water/Silicagel)

9 Austria Fa. Bachler Gröbming training center 2007 9 Pink (Ammonia/Water)

10 Austria Fa. Manschein Gaweinstal office and industrial building 2008 7.5 Sortech (Water/Silicagel)

11 Austria Fa. Kreuzroither Schörfling office and industrial building 2008 15 Sortech (Water/Silicagel)

12 Austria Fa. SOLID Graz office building 2008 17 Yasaki (Water/LiBr)

13 Austria Municipality of Vienna Section 34 Vienna office building 2009 7 Sortech (Water/Silicagel)

14 Austria Oköpark Hartberg Office 30.4

15 Austria BH Rohrbach Rohrbach Office 30

16 Austria General Solar St. Veit a.d. Glan Office 35

17 Austria Ferngas OÖ Haid Office 70

18 Austria Gasokol Saxen Office 30

19 Austria Rathaus Gleisdorf Gleisdorf Office 2008 35 6000 m³/h

20 Austria SOLution Solartechnik, new site Sattledt Office 60

21 Austria Fa. Paar Graz Office 105

22 Austria Raiffeisenbank Trofaiach Trofaiach Office (bank) 70

23 Austria Feistritzwerke Gleisdorf Gleisdorf Office 24

24 Austria Sunmaster Eberstalzell Office 80

25 Austria ENERGYbase Wien Office

Von der Grazer Solartechnikfirma S.O.L.I.D GmbH wurden im Ausland bereits einige größere Anlagen installiert:

- Eine Absorptionskälteanlage mit einer Kälteleistung von 90 kW wurde in Pristina, Kosovo für das Gebäude der EAR (European Agency for Reconstruction) errichtet.
- Eine Absorptionskälteanlage mit einer Kälteleistung von 35 kW wurde in Brüssel, für das REH – Renewable Energy House errichtet.
- Eine Absorptionskälteanlage mit einer Kälteleistung von 600 kW wurde in China, im „Olympic Sailing Village" errichtet.
- Eine Absorptionskälteanlage mit einer Kälteleistung von 70 kW wurde in Arizona, USA im „Desert Outdoor Center" errichtet.
- Weitere Anlagen sind in Planung

Kältemaschinenentwicklungen in Österreich:
Entwicklung einer 10 kW Ammoniak/Wasser Absorptionskältemaschine der Fa. Pink, Langenwang, es existiert eine ganze Reihe von installierten Anlagen z.T. mit Solarenergie betrieben, teilweise mit Fern- oder Abwärme.

Die Entwicklung einer Absorptionskältemaschine mit dem Kältemittel Ammoniak von SolarFrost/Dr. Kunze ist im Laborstadium.

3.5 Aufbau des Berichts

Kapitel 4 dieses Berichts gibt eine kurze Übersicht über das Implementing Agreement (Solar Heating and Cooling) sowie den Task 38, bevor dann in Kapitel 5 näher auf die Ziele des Task 38 eingegangen wird.

In Kapitel 6 werden dann die Arbeitspakete, an denen Partner aus diesem Projekt beteiligt waren, und deren Ergebnisse kurz beschrieben. Eine ausführliche Form der Ergebnisse ist in den jeweiligen „Technical Reports" zu finden, die unter jedem Abschnitt angegeben sind. Soweit diese bereits verfügbar sind, sind sie außerdem in Anhang diesem Bericht beigefügt.

4 Übersicht über das Implementing Agreement / Task

Eines der ersten Implementing Agreements im Rahmen der IEA war das 1977 ins Leben gerufene "Solar Heating and Cooling Programme" (SHC). Seitdem wurde eine Vielzahl von Forschungsprojekten zur aktiven und passiven Solarenergienutzung, Tageslichtnutzung und die Anwendung dieser Technologien in Gebäuden und anderswo (z.B. Landwirtschaft und Industrie) durchgeführt.

Das SHC Exekutivkommittee hat sich folgende Ziele gesetzt:

* die Leistungsfähigkeit von Technologien und Konzepten zum solaren Heizen und Kühlen zu verbessern.

* Industrie und Regierungen dabei zu unterstützen, den Marktanteil von Technologien und Konzepten zum solaren Heizen und Kühlen zu erhöhen.

* Informationen zu Technologien, Konzepten und Anwendungen zum solaren Heizen und Kühlen zur Verfügung zu stellen.

* Entscheidungsträger und die Öffentlichkeit über den Stand der Technik und Nutzen von solarem Heizen und Kühlen zu informieren.

Übergeordnete Ziele der IEA SHC Task 38 "Solare Kühlung und Klimatisierung‟ sind die Entwicklung und Erprobung von Kompaktsystemen zur solaren Heizung und Kühlung im kleinen und mittleren Leistungsbereich; die Entwicklung und Erprobung von kundenspezifischen großen solaren Klimatisierungs- und Kühlanlagen für den Nicht-Wohnbereich sowie die Entwicklung von neuen Konzepten für solare Kühlmaschinen.

5 Ziele des Projektes

Übergeordnete Ziele der IEA SHC Task 38 "Solare Kühlung und Klimatisierung“ sind die Entwicklung und Erprobung von Kompaktsystemen zur solaren Heizung und Kühlung im kleinen und mittleren Leistungsbereich; die Entwicklung und Erprobung von kundenspezifischen großen solaren Klimatisierungs- und Kühlanlagen für den Nicht-Wohnbereich sowie die Entwicklung von neuen Konzepten für solare Kühlmaschinen.

Über den technologischen Schwerpunkt hinaus ist es das Ziel des Vorhabens, die Ergebnisse der Forschungs- und Entwicklungsarbeiten möglichst rasch an die möglichen Nutzer und Anwender sowie an die Industrie weiterzugeben, um die Implementierung der Ergebnisse am Markt voranzutreiben.

Ziele, die im Rahmen der Beteiligung an der Task 38 von der den Antragstellern vorrangig verfolgt werden, sind die gemeinsame Weiterentwicklung von Anlagen zur solaren Kühlung und Klimatisierung sowie von Anlagen zur solaren Raumheizung. Basierend auf den Ergebnissen der IEA SHC Task 25 (Solare Klimatisierung von Gebäuden) und Task 26 (Solare Kombianlagen) sollen Kompaktsysteme (Pre-engineered Systems) für solares Heizen und Kühlen entwickelt, getestet und in Pilotanlagen erprobt werden.

Eines der wichtigsten Ergebnisse des Tasks 38 ist die Entwicklung eines einheitlichen Monitoringkonzeptes, mit dessen Hilfe installiert Anlagen detailliert vermessen werden und die Ergebnisse miteinander verglichen werden können. In der Vergangenheit wurde bei Anlagen zum solaren Kühlen oft nur das Funktionieren der Kältemaschine als solche messtechnisch erfasst. Der Task aber hat allen Teilnehmern deutlich gemacht, wie wichtig es ist, das Gesamtsystem und alle Energie- auch Hilfsenergieverbräuche zu messen, um eine Aussage über den energetischen Nutzen des Systems treffen zu können.

Weitere Ziele und Ergebnisse des Tasks werden im folgenden Kapitel dargestellt.

6 Inhalte und Ergebnisse

6.1 Subtask A

6.1.1 Koordination der Subtask A

Die AEE INTEC hatte die Aufgabe, die Subtask A als Subtask-Leader zu leiten und zu koordinieren. In der gesamten Task 38 waren rund 70 Personen aktiv beteiligt, davon waren ca. 20 bis 25 Personen schwerpunktmäßig in der Subtask A aktiv, wobei auch die Industrie an dieser Subtask besonders interessiert war.

Inhaltlich wurden von den Projektpartnern in der Subtask A bei folgenden Teilbereichen mitgearbeitet:

6.1.2 A1: Market Overview

Basierend auf den Erkenntnissen aus der IEA-SHC Task 26 bzw. dem ALTENER Projekt „Solar Combisystems“ wurde eine Zusammenfassung von bestehenden Systemkonzepten für Solar-Kombianlagen erstellt. Darauf aufbauend wurde eine Arbeitsgruppe initiiert und geleitet, mit dem Ziel die Eignung existierender Konzepte von Solar-Kombianlagen als Basis für „Pre-engineered Solar Cooling Systems“ zu prüfen. Das Ziel ist es letztendlich jene Basiskonzepte zu finden und die Randbedingungen zu spezifizieren, um sie um die Funktion Solares Kühlen erweitern zu können. Das Ergebnis dieser Arbeit hat dann auch als Input für das Arbeitspaket A2, der Definition von „Generic Systems“ Verwendung gefunden.

Auch die Erstellung der Marktübersichten der anderen Teilbereiche „Kältemaschinen“, Rückkühlsysteme“ und „Kältespeicher“ durch Fraunhofer ISE, Freiburg bzw. ITW Uni Stuttgart wurde von der AEE INTEC koordiniert. Der Gesamtbericht, der alle vier Teilbereiche enthält, wurde von der AEE INTEC erstellt. Für den Bereich Rückkühlsysteme wurde von der AEE INTEC eine Übersicht der in den im Rahmen der Task 38 vermessenen Systemen verwendeten Rückkühleinrichtungen zusammengestellt. Diese Übersicht, noch ergänzt durch weitere Marktdaten anderer Taskteilnehmer, wurde dann auch für das Task 38 Handbuch für Planer verwendet.

Der Endbericht für dieses Arbeitspaket (D-A1) ist auf der Task 38 Homepage verfügbar und im Annex zu diesem Bericht zu finden.

6.1.3 A2: Generic Systems

Als Grundlage für vorgefertigte Systemkonzepte wurden Hydraulikschemata und Regelungskonzepte als „Generic Systems“ entwickelt, d.h. Systemkonzepte die firmen- bzw. produktunabhängig beschrieben werden. Basierend auf der langjährigen Erfahrung mit solaren Kombisystemen für Warmwasserbereitung und Heizungsunterstützung (unter anderem auch aus der Task 26) war die AEE INTEC maßgeblich an der Aufbereitung und Anpassung der „heißen“ Seite dieser Systeme beteiligt. Im weiteren Verlauf der Task 38 wurden diese Aktivitäten in starker Kooperation mit dem IEE Projekt „Solar Combi^{+}“ durchgeführt.

Bei der Erstellung des Endberichts für dieses Arbeitspaket wurde der Arbeitsgruppenleiter (ZAE Bayern, Deutschland) von der AEE INTEC durch Korrekturlesen des Berichtsentwurfs und zahlreiche Verbesserungsvorschläge unterstützt.

Der Endbericht für dieses Arbeitspaket (D-A2) ist auf der Task 38 Homepage verfügbar und im Annex zu diesem Bericht zu finden.

6.1.4 A3: Monitoring of small systems

Die IEA-SHC Task38 (Solar Air Conditioning and Cooling) des Programms „Solares Heizen und Kühlen" der Internationalen Energieagentur hatte ein Monitoringprogramm für installierte solare Heiz- und Kühlsysteme als ein zentrales Ziel [1]. Es sollte die Funktionsweise von so vielen Systemen wie möglich dokumentiert und ein Vergleich der Ergebnisse durchgeführt werden, um optimale Systemkonfigurationen und Regelstrategien der verschiedenen Anwendungen zu identifizieren. Der „Subtask A" der Task38 behandelt vorgefertigte solare Heiz- und Kühlsysteme mit dem Ziel, die Entwicklung vorgefertigter Systeme zu unterstützen, die serienmäßig produzierbar sind und nur mehr einen geringen Planungsaufwand aufweisen. Damit können solare Heiz- und Kühlsysteme von Installateuren installiert werden, ohne dass diese die einzelnen Komponenten extra dimensionieren müssen, was zu einer Reduktion von Kosten und Installationsfehler führt. Diese Art der Vorfertigung eignet sich speziell für kleinere Systeme mit einer Kühlleistung von bis zu ca. 20 kW. Diese Pakete sollen die Kollektoren, Kältemaschine, Rückkühleinheit, Pumpengruppen mit Verrohrungen sowie den Regler beinhalten. Auf Grund dessen, dass es nur einige wenige komplett vorgefertigte Paketlösungen auf dem Markt gibt, wurden im Rahmen der Subtask A generell Systeme im kleinen Leistungsbereich unter 20 kW Kälteleistung einem Monitoring unterzogen. Das Ziel ist es, optimale Systemkonfigurationen zu identifizieren, welche schließlich durch die Hersteller zu vorgefertigten Paketlösungen weiter entwickelt werden können.

Monitoringkonzept

Die Methodik basiert auf der Messung sämtlicher relevanter Energieströme (Wärme, Kälte, elektrischer Strom), um damit Kennzahlen wie Primärenergieverhältnis, elektrische Arbeitszahl, Effizienz der solarthermischen Anlage, Primärenergieeinsparung für Heizen und Kühlen im Vergleich zu einem konventionellen (nicht solarem) Heiz- und Kühlsystem errechnen zu können.
Abb. 5 zeigt ein allgemeines Schema mit Energieflüssen für verschiedene Konfigurationen von solaren Heiz- und Kühlsystemen. In konkreten Anlagen ist dann typischerweise nur mehr eine deutlich reduzierte Anzahl von Energieflüssen vorhanden.

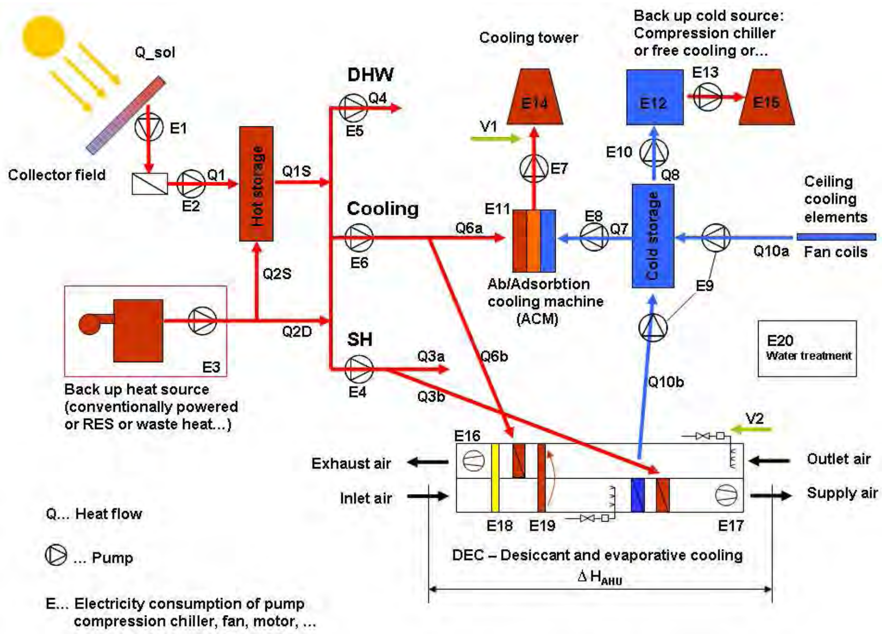


Abb. 5: Allgemeines Monitoring Schema für solare Heiz- und Kühlsysteme

Als „Monitoring Tool" wurde eine Berechnungsvorlage als Excel Datei ausgearbeitet, worin sämtliche verfügbare Messdaten auf monatlicher Basis nach der Nomenklatur des allgemeinen Monitoring Schemas (Abb. 5) eingesetzt werden können.

Für Lüftungsanlagen mit Sorptionsgestützter Klimatisierung (SGK), (Desiccant Evaporative Cooling - DEC), wurde eine zusätzliche Berechnungsvorlage in einer eigenen Excel Datei entwickelt. Darin ist es möglich mit zeitlich hoch aufgelösten Messdaten der SGK monatliche Enthalpiedifferenzen sowohl für den Heiz- als auch den Kühlbetrieb zu berechnen. Zusätzlich dazu werden Enthalpiedifferenzen basierend auf verschiedenen Referenzszenarien berechnet. In all den Szenarien wird eine konventionelle Kompressionskältemaschine dazu verwendet, die Zuluft so zu konditionieren (abkühlen und entfeuchten), dass die (rechnerische) absolute Luftfeuchtigkeit (g pro kg trockene Luft) der tatsächlich gemessenen absoluten Luftfeuchtigkeit entspricht.

Die Zulufttemperatur der Referenzsysteme kann wie folgt gewählt werden:

a) Identisch mit der gemessenen Zulufttemperatur der SGK, basierend auf Nachheizung durch einen mit Ergas befeuerten Kessel.
b) Ein Fixwert kann manuell für jedes Monat separat gewählt werden und wird erreicht mittels Nachheizung durch einen Gaskessel.
c) Keine Temperaturvorgabe, da keine Nachheizung durch einen Gaskessel angenommen wird, bzw. die Nachheizung durch eine Maßnahme erreicht wird (z.B. durch Wärmerückgewinnung innerhalb der Lüftungsanlage oder der Nutzung von Abwärme einer Kältemaschine), welche keinen zusätzlichen Energieaufwand bedeutet.

In diesem „Monitoring Tool" sind Kennzahlen in 3 verschiedenen Ebenen definiert, welche aus den in das Eingabearbeitsblatt eingegebenen Daten errechnet werden.
In der ersten Ebene wird eine Bewertung des gesamten Systems mittels Berechnung von Arbeitszahlen (Coefficient of Performance – COP) und Primärenergieverhältniszahlen (Primary Energy Ratio - PER) durchgeführt. Die elektrische Arbeitszahl (COPel,tot) ist das Verhältnis der gesamten Nutzenergie (Wärme und/oder Kälte) zum elektrischen Stromaufwand, jedoch ohne den Stromverbrauch von Pumpen und Ventilatoren der

Energieverteilung im Gebäude (dieser wird im Referenzsystem als gleich angenommen). Der COPel,tot wird also für die produzierte Nutzenergie berechnet. Die elektrische Arbeitszahl (COPel,overall) inkludiert zusätzlich den Stromverbrauch der Energieverteilung im Gebäude.
Die Primärenergieverhältnisse (PER) werden für Systeme mit fossilen und/oder erneuerbaren (Renewable Energy Source - RES) Energieträgern berechnet. Es ist dies jeweils das Verhältnis von erzeugter Nutzenergie im Verhältnis zu der dafür aufgewendeten Primärenergie, also wieder exklusive der Verteilung im Gebäude.
Als Primärenergiefaktoren [kWh Endenergie / kWh fossiler Primärenergie] werden als Standard verwendet: 0,9 für fossile Brennstoffe (Heizöl, Erdgas), 10 für Biomasse (Pellets, Hackschnitzel) und 0,4 für elektrischen Strom für den Vergleich auf internationaler Ebene (auch eine Berechnung des Primärenergiefaktors für Blockheizkraftwerke ist integriert). Diese Werte können im „Monitoring Tool" aber auch entsprechend den nationalen oder lokalen Rahmenbedingungen durch den Nutzer angepasst werden. Der fossile, CO2-relevante Primärenergiebedarf eines Systems mit erneuerbaren Energieträgern kann im Vergleich zu einem System mit fossilen Energieträgern trotz schlechter Systemeffizienz sehr niedrig sein. Grund dafür ist der günstige Primärenergiefaktor von 10 für erneuerbare Energieträger im Gegensatz zu 0,9 bei fossilen Brennstoffen.
Um auch erneuerbare und fossil betriebene Systeme hinsichtlich Systemeffizienz miteinander vergleichen zu können, wird für Systeme mit erneuerbaren Energieträgern zusätzlich zu dessen erneuerbarem Primärenergiebedarf (PERres) ein fossiler Primärenergiebedarf (PERfossil) berechnet. Damit kann die Qualität der verschiedenen Systeme unter gleichen Rahmenbedingungen verglichen werden.

In Ebene zwei des „Monitoring Tools" wird hauptsächlich die Qualität der „heißen" Teilsysteme Solarkreis und Energiespeichermanagement innerhalb des Systems evaluiert. Diese werden durch Kennzahlen wie „Kollektorkreiseffizienz", „Kollektorertrag pro m²", „Solarenergienutzung" (solar energy utilization), „Speichereffizienz" und „Systemeffizienz" bewertet.

In der dritten Ebene wird eine tiefer gehende Analyse durchgeführt. Einige spezifisch definierte Arbeitszahlen bewerten die Leistungsfähigkeit der Teilsysteme Sorptionskältemaschine und SGK. Neben der thermischen Arbeitszahl (COPth) der Sorptionskältemaschine [kWh erzeugte Kälte/ aufgewendete thermische Antriebsenergie] werden einige elektrische Arbeitszahlen (COPel - kWh erzeugte Kälte/ aufgewendete elektrische Energie) berechnet. Dies sind der COPel der Kältemaschine selbst, COPel der Kältemaschine plus Austreiber-, Kaltwasser- und Rückkühlpumpe inklusive Kühlturmventilator sowie der COPel noch einmal erweitert um die Kollektorkreispumpen. Analog dazu werden dieselben Kennwerte bei SGK Einheiten berechnet.
Auch der Wasserverbrauch des Nasskühlturms und des SGK Systems wird in verschiedenen Kennzahlen evaluiert: Wasserverbrauch im Verhältnis zur erzeugten Kälte, Strombedarf der Wasseraufbereitung im Verhältnis zum Wasserverbrauch oder im Verhältnis zur erzeugten Kälte.
Schließlich wurde für die Berechnung ein Referenzsystem definiert, um bewerten zu können ob das solare Heiz- und Kühlsystem bessere Ergebnisse liefert als ein konventionell betriebenes System, welches dieselben Heiz- und Kühllasten abdeckt.
Dieses Referenzsystem besteht aus einem Gasbrennwertkessel (95% Kesselnutzungsgrad), welcher den Heizenergiebedarf abdeckt und die Warmwasserbereitung bewerkstelligt. Es wurde ein kleiner Warmwasserspeicher mit typischen Wärmeverlusten angenommen, welche auf den gemessenen Warmwasserverbrauch basieren.

Der Kühlenergiebedarf wird im Referenzsystem von einer Kompressionskälte-maschine mit einer Jahresarbeitszahl von 2,8 abgedeckt. Die benötigte bzw. „verbrauchte" Primärenergie wird sowohl für das solare Heiz- und Kühlsystem als auch für das Referenzsystem berechnet. Der Quotient dieser beiden Werte bildet die Primärenergieeinsparung fsav,shc des solaren Heiz- und Kühlsystems und wird schließlich gemäß Gleichung 1 berechnet.

$$f_{sav,shc} = 1 - \frac{\dfrac{Q_{boiler}}{\varepsilon_{fossil} \cdot \eta_{boiler}} + \dfrac{Q_{RES}}{\varepsilon_{RES} \cdot \eta_{RES}} + \dfrac{E_{el}}{\varepsilon_{elec}} + \dfrac{Q_{cooling,missed}}{SPF \cdot \varepsilon_{elec}}}{\dfrac{Q_{boiler,ref}}{\varepsilon_{fossil} \cdot \eta_{boiler,ref}} + \dfrac{E_{el,ref}}{\varepsilon_{elec}} + \dfrac{Q_{cooling,ref}}{SPF_{ref} \cdot \varepsilon_{elec}}} \qquad \text{Gl. 1}$$

Legende: ε…Primärenergiefaktor, Q…Wärme, E…elektrische Energie, η… Nutzungsgrade, SPF…seasonal performance factor – Arbeitszahl

Ergebnisse

Es wurden im Rahmen der IEA-SHC Task38 11 solare Heiz- und Kühlanlagen im kleinen Leistungsbereich (unter 20 kW Nennkälteleistung) einem Monitoring unterzogen und mit dem Monitoring Tool analysiert [5]. Es sind Messdaten für mindestens ein vollständiges Jahr pro Anlage verfügbar und anonymisiert in den folgenden Diagrammen für die Monate Juni bis September der Jahre 2008, 2009 oder 2010 dargestellt.

In Abb. 6 sind gemessene COPth dargestellt. Die Werte liegen meist zwischen 0,5 und 0,7 nahe den Herstellerangaben und hängen vom System und der Betriebsweise der Maschine ab. Hohe Austreiber- und Kaltwassertemperaturen, niedrige Rückkühltemperaturen sowie ein geringer Taktbetrieb resultiert in günstige COPth.

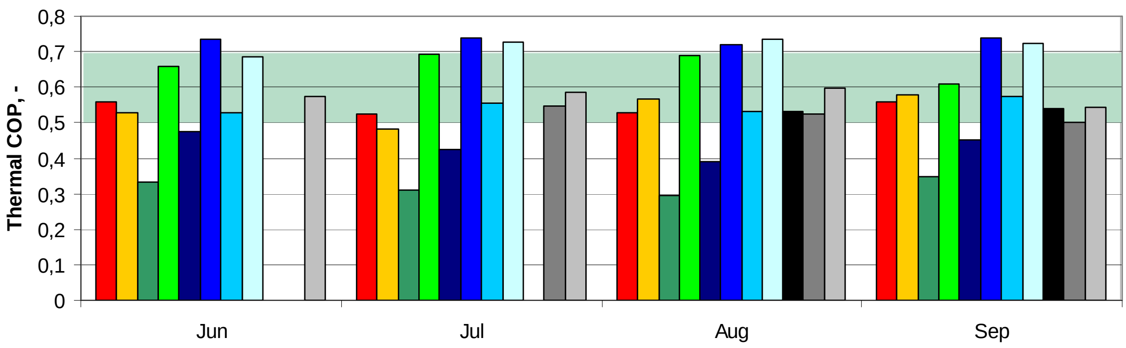


Abb. 6: COPth diverser Anlagen

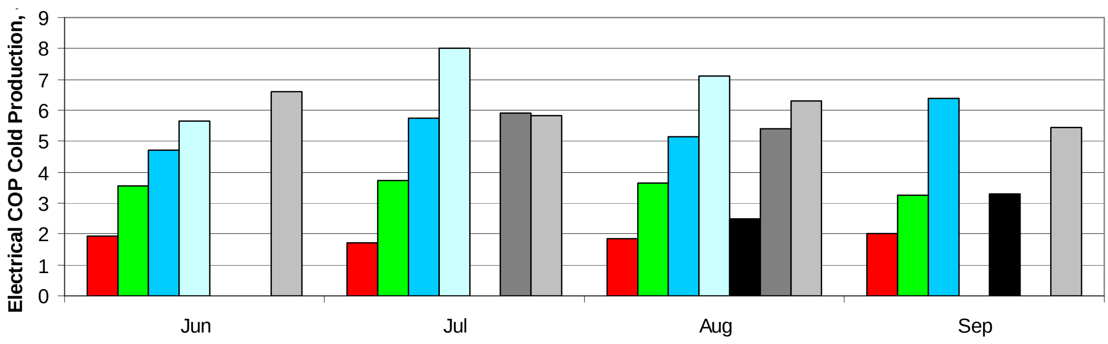


Abb. 7: COPel der Kälteproduktion diverser Anlagen

In Abb. 7 sind gemessene COPel der Kälteerzeugung dargestellt. Dieser Wert inkludiert den Stromverbrauch der Kältemaschine selbst, der Rückkühl-, Austreiber- und Kaltwasserpumpe (wenn ein Kaltwasserspeicher vorhanden ist) sowie des Kühlturmventilators. Das beste System dieses Monitorings erreicht den Wert 8, der Großteil der Systeme liegt im akzeptablen Bereich zwischen 5 und 6, wo jedoch noch Potential für Verbesserungen vorhanden ist.

Der Wasserverbrauch pro kWh Rückkühlenergie der verwendeten Hybrid- bzw. Nasskühltürme ist in Abb. 8 dargestellt. Es ist zu erkennen, dass die Werte zwischen den Systemen und Monaten sehr unterschiedlich sind. Gründe für diese Streuung sind zum einen die spezifischen Klimabedingungen der Standorte bzw. der unterschiedlichen Monate und zum anderen die verschiedenen Kühlturm Technologien. Nichts desto trotz zeigt diese große Streuung, dass hinsichtlich des Wasserverbrauchs noch Optimierungspotential bestehen dürfte.

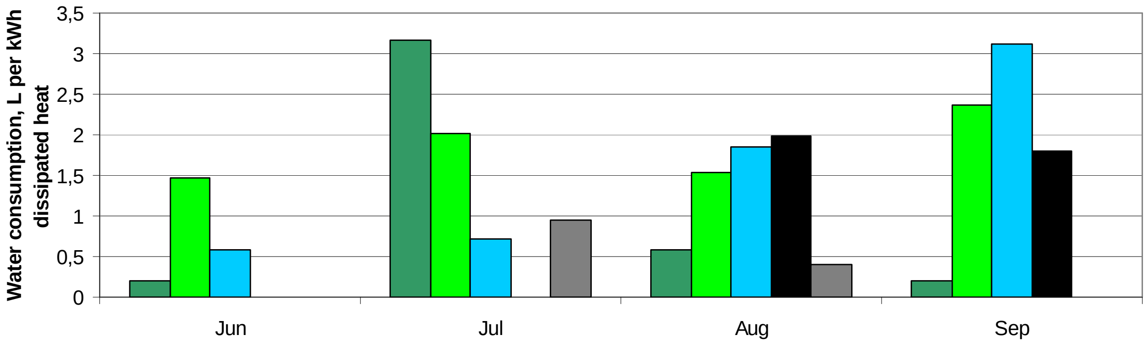


Abb. 8: Kühlturm - Wasserverbrauch pro kWh Rückkühlenergie (erste 2 Säulen Hybrid- und 3x Nasskühlturm)

Schlussfolgerung

Die analysierten Systeme zeigen, dass gute bzw. sehr gute Kennzahlen erreicht werden können, auch wenn in einzelnen Fällen die thermische Arbeitszahl (COPth) unter der nominellen liegt. Die Gesamteffizienz hängt stark vom Systemkonzept und der Regelungsstrategie des gesamten Systems sowie dem Hilfsstromverbrauch der einzelnen Komponenten wie Pumpen und Ventilatoren ab. Die thermische Arbeitszahl der Sorptionskältemaschine ist wichtig, hat jedoch eher einen geringen Einfluss auf die Gesamteffizienz des Systems. In den meisten Fällen besteht aber noch deutliches Optimierungspotential: sowohl das Systemkonzept, effizientere Komponenten (Pumpen, Ventilatoren) als auch die Regelungsstrategien können verbessert werden, um noch bessere Werte zu erreichen.

Endberichte:

Die Beschreibung der Monitoring Prozedur als gemeinsames Ergebnis von Subtask A und Subtask B wird als D-A3a bzw. D-B3b im Laufe des Jahres 2011 auf der IEA-SHC Homepage verfügbar sein.

Der Bericht zu den Ergebnissen des Monitorings der Subtask A Anlagen (D_A3b) ist im Anhang dieses Berichts zu finden und wird im Laufe des Jahres 2011 auch auf der IEA-SHC Homepage verfügbar sein.

6.1.5 A4: Evaluation procedures

Dieses Arbeitspaket wurde in Subtask D verschoben. Das im Rahmen des oben schon erwähnten Monitoringkonzeptes entwickelte Kennzahlensystem bildet hierbei die Grundlage für die Evaluierung von Anlagen zum solaren Heizen und Kühlen.

6.1.6 A5: Installation and Maintenance Guidelines

In der zweiten Hälfte der Task 38 wurden die Erfahrungen aus Labor- und Demonstrationsprojekten in enger Zusammenarbeit mit der Industrie in den „Installation and Maintenance Guidelines" zusammengefasst. Dazu wurde eine Arbeitsgruppe initiiert, die dann vom AIT geleitet wurde. In dieser Arbeitsgruppe wurden Befragungen von Nutzern von Anlagen zum solaren Heizen und Kühlen einerseits und von Anbietern von Paketlösungen andererseits durchgeführt. Die Ergebnisse sind im Bericht D-A5 zusammengefasst worden. Außerdem sind hier die Erkenntnisse aus den EU Projekten „SOLAIR" und „SolarCombi+" eingeflossen.

6.2 Subtask B

6.2.1 B1: Market Overview

Im Arbeitspaket B1 wurden bestehende Anlagen in aller Welt zusammengetragen. Ziel dieses Arbeitspaket war es, einen Überblick über die derzeit verwendeten Technologien und Systemkonzepte zu bekommen. Dies ist als Hilfestellung für Planer gedacht, die Unterstützung bei der Auslegung von Anlagen für verschiedene Anwendungsbereiche und Klimate benötigen.

Der Endbericht für dieses Arbeitspaket geht insbesondere auf die großen kundenspezifischen Anlagen ein, innerhalb von Arbeitspaket A5 wurden aber auch die bestehenden Anlagen im kleinen Leistungsbereich zusammengetragen und in diese Studie integriert.

Der Endbericht für dieses Arbeitspaket (D-B1) ist auf der Task 38 Homepage verfügbar und im Annex zu diesem Bericht zu finden.

6.2.2 B2/B3: Selection of System Designs and System Control

In diesem Arbeitspaket wurden für ausgewählte Anlagen die System- und Regelungskonzepte zusammengetragen und miteinander verglichen. Wie schon oben erwähnt konnte AEE INTEC in Kooperation mit der Firma S.O.L.I.D. die Anlage beim Rathaus und Service Center der Stadt Gleisdorf einbringen.

Der Endbericht zu diesen Arbeitspaketen enthält ausführliche Beschreibungen von 14 Anlagen sowie deren System- und Regelungskonzepte. Da dieser Bericht viele Details der einzelnen Anlagen enthält, ist er nicht veröffentlicht worden sondern nur den Teilnehmern der Task 38 zugänglich (interner Bericht D-B2).

6.2.3 B4: Monitoring of Custom-Made Systems

Das unter Subtask A3 erwähnte Monitoringkonzept wurde in Kooperation mit Subtask B erstellt und wurde auch für die Analyse der Messdaten von großen kundenspezifischen Anlagen verwendet.

Die AEE INTEC hat in Kooperation mit der Firma S.O.L.I.D. die Anlage am Rathaus und Service Center Gleisdorf über 2 Jahre messtechnisch begleitet. Mithilfe des im Task entwickelten Monitoring Excel-Tools wurden die Daten ausgewertet und ein ausführlicher Bericht als Beitrag für den internen Bericht D-B3 erstellt. Im Rahmen des nationalen HdZ+ Projektes SolarCoolingMonitor werden die Ergebnisse im Endbericht Ende 2011 präsentiert werden.

6.2.4 B6: Soft tool package for successful projects

Drei Planungshilfen wurden innerhalb der Task 38 entwickelt:

- Die **Checkliste** ist für die ersten Schritte innerhalb eines neuen Projektes gedacht. Hier werden mit einfachen Mittel die wichtigsten Randbedingungen abgeklärt, ob ein neues Projekt für solares Kühlen geeignet erscheint.
- Als nächster Schritt kann ein **Entscheidungsbaum** genutzt werden, um zu entscheiden, welche Technologie für das Projekt besonders geeignet ist.
- Anschließend kann man mit dem ***TDC tool*** (Thermally Driven Cooling tool) schon eine erste Abschätzung über Erträge und Leistungszahlen des gewählten Systems machen.

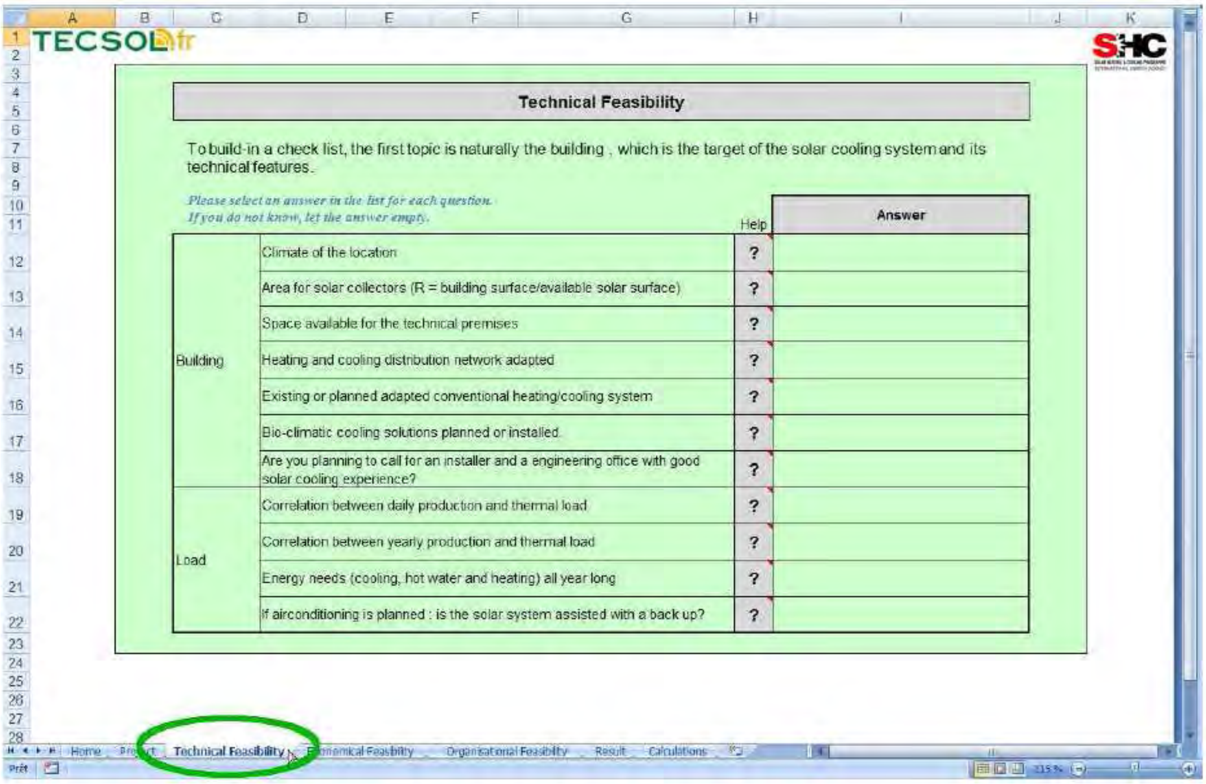


Abb. 9: Beispielseite der Checkliste

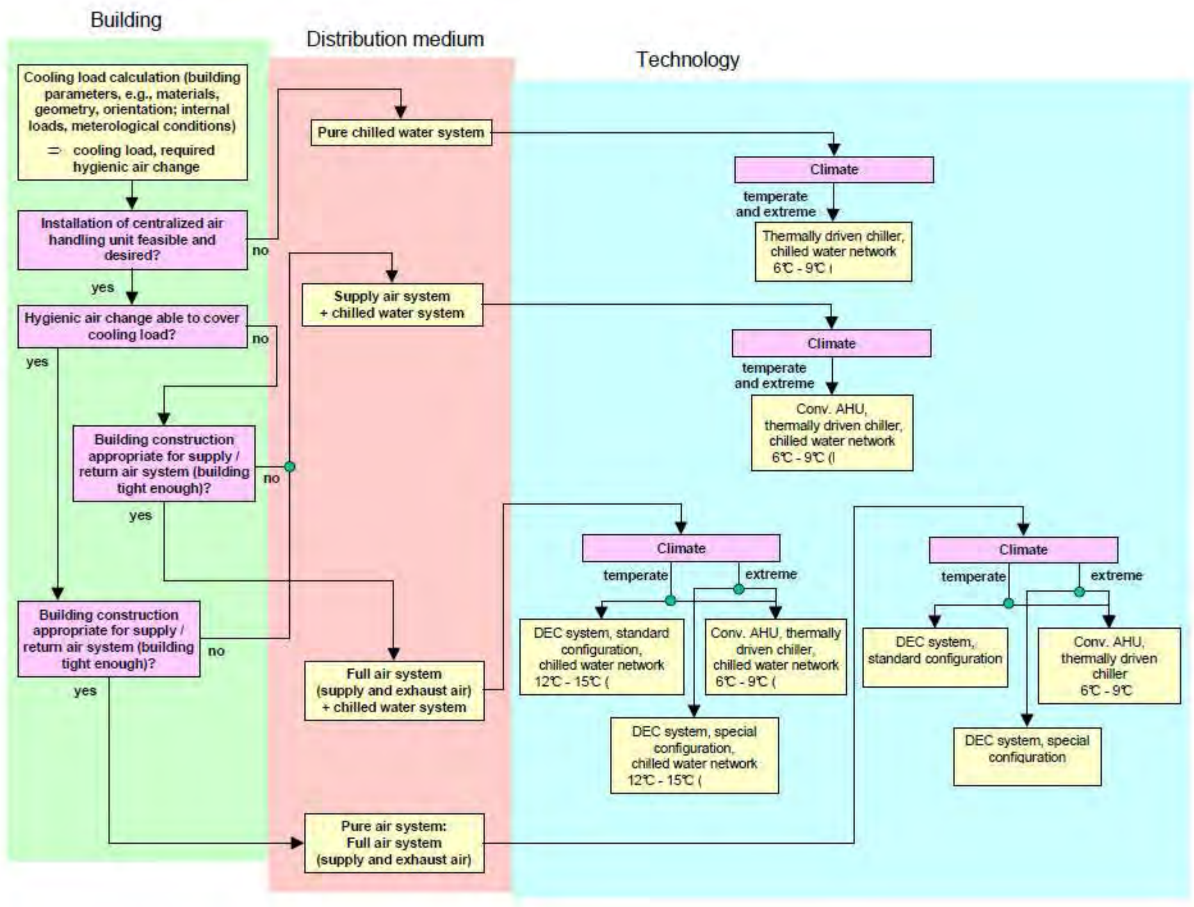


Abb. 10: Entscheidungsbaum

Endergebnis ist ein Bericht über alle drei Planungshilfen (D-B4b), der detaillierte Anleitungen der jeweiligen Tools enthält. Dieser Bericht ist allerdings nur den Taskteilnehmern zugänglich und nicht frei verfügbar.

Zur Checklistenmethode gibt es einen frei verfügbaren Bericht (D-B4a) sowie ein Online-Tool. Beide sind von der Internetseite der Firma TECSOL herunterzuladen (http://www.tecsol.fr/checklist/).

6.3 Subtask C

6.3.1 C2: Technical review and report of new SAC developments

In diesem Arbeitspaket wurden der Stand der Technik sowie aktuelle Entwicklungen auf dem Gebiet des solaren Kühlens zusammengetragen.

Ergebnis ist ein Bericht mit ausführlichen Technologiebeschreibungen der verschiedenen Technologien des solaren Kühlens

- Absorption (Wasser-LiBr und Ammoniak-Wasser)
- Adsorption
- Flüssigsorption
- Trockensorption
- Thermomechanisch
- Dampfstrahlkälte

Außerdem enthält der Bericht aktuelle Entwicklungen sowie Kontaktdaten von Instituten und Firmen, die auf diesem Gebiet forschen.

Diese Beiträge sind Teil des Berichts D-C1, der im Anhang diesem Bericht beigefügt ist und von der IEA-SHC Homepage heruntergeladen werden kann.

6.3.2 C4: Simulation tools – analysis and development

Das TDC-Pre-Feasibility Tool aus dem EU Projekt SACE (EaSYCOOL) wurde von S.O.L.I.D. getestet. Handhabung, Funktionalität und Bedienerfreundlichkeit wurden getestet.
Es ist positiv aufgefallen, dass man sehr viele relevante Klima-, Gebäude-, Anlagendaten etc. eingeben bzw. aus Dateien laden kann. Auch ist das Laden der verschiedenen Einstellungsdateien recht einfach und übersichtlich.
Bei der Evaluierung des Programms gingen wir davon aus, dass es während der Akquisephase für einen potentiellen Kunden einer Anlage zur solaren Kühlung verwendet wird. Dafür wäre es sehr nützlich, wenn das Programm generell eine Druckfunktion hätte und man dann die wichtigsten Parameter einer interessant erscheinenden Lösung ausdrucken könnte. So eine Gesamtübersicht haben wir auch nicht gefunden.
Da das Programm ja im Kundenkontakt verwendet werden soll, ist es sehr wichtig, dass alle Werte, Parameter möglichst exakt erläutert sind. So kommt der Benutzer nicht in Verlegenheit und macht keine widersprüchlichen Aussagen zum Kunden.
Da u.U. mehrere Alternativen nacheinander berechnet werden, wäre es sinnvoll, wenn die wichtigsten Parameter der gerade berechneten Konfiguration in einem separaten Fenster sichtbar sind.

Vorschläge zur Überarbeitung des Tools wurden den verantwortlichen Arbeitsgruppenleiter ISE Fraunhofer weitergegeben. Eine Beschreibung des Tools ist im Bericht D-C2a zu finden, im Anhang dieses Berichtes beigefügt ist..

6.3.3 C5: Heat rejection concepts

Hier wurden wesentliche Beiträge von Erich Podesser zum Thema Hygiene bei kleinen Nasskühltürmen geleistet. Der 41-seitige Bericht wurde gemeinsam mit der Technischen Universität Graz, Harald Moser, erstellt. Die Beiträge von Erich Podesser betrafen vor allem die Themen

a. Wet Cooling Tower – Process Calculation
b. Legionella
c. Avoidance of Legionella and Micro-Organism
d. Water Disinfection by UV-radiation
e. Water Disinfection by Metal Ions

Zu a.: Der Rückkühlprozess durch nasse Kühltürme wurde erklärt und in Anlehnung an die bekannte Literatur mit einer einfach anzuwendenden Vorgangsweise rechnerisch dargestellt. Die präsentierte Methode kann sowohl zur Vorausberechnung neuer Kühlturmformen als auch zur Produktprüfung im Rahmen von Übernahmen angewendet werden.

Zu b. und c.: In diesem Kapitel wurden vor allem Informationen über die Bakterienfamilie der Legionellen gegeben, günstige Bedingungen für deren Vermehrung genannt und die Methoden der Vermeidung beschrieben.

Zu d.: Hier wurde der Zerstörungsmechanismus bei Legionellen durch die UV-Strahlen beschrieben, die praktischen Anwendungen und vorhandenen Produkte genannt.

Zu e.: Die Desinfektion durch Metallionen, insbesondere durch die Silber-Ionen wurde praktisch und theoretisch behandelt. Im Rahmen von Laborversuchen wurde bakterienverseuchtes Wasser mit Silber-Ionen behandelt und es wurde mit Hilfe von Nährböden in Petrischalen die erfolgreiche Desinfektion durch die Silber-Ionen praktisch nachgewiesen. Mit den Erkenntnissen wurde ein entsprechendes elektronisches Gerät gebaut. Die Fa. ECONICsystems hat bereits einige Geräte gefertigt, und verschiedene praktische Tests durchgeführt. Unter anderen wurde das Gerät im Rahmen von Vergleichstests im Forschungsprojekt „InnoCool" bei JOANNEUM RESEARCH eingesetzt. Ergebnisse dazu werden im Projektbericht verfügbar gemacht.

Die Ergebnisse wurden in einem Bericht zusammengefasst, der Teil des Deliverables D-C5 ist. Der Gesamtbericht ist derzeit noch nicht fertiggestellt, wird aber im Laufe des Jahres auf der IEA-SHC Hompage verfügbar sein. Im Anhang dieses Berichtes ist der österreichische Beitrag zu diesem Bericht zu finden.

6.4 Subtask D

6.4.1 D1: Proposals for performance assessment methodology

Siehe: A4 „Evaluation procedures"

6.4.2 D4: Handbook 2nd Edition

Im Rahmen des Tasks wurde das ursprünglich im vorangegangenen Task 25 entstandene „Handbook for Planners" komplett neu überarbeitet.

Die AEE INTEC war Hauptautor für das Handbuchkaptitel 10 über „Umgesetzte Beispiele und Erfahrungen mit kleinen vorgefertigten Anlagen zum solaren Heizen und Kühlen". Das gut 6.000 Wörter zählende Kapitel wurde hauptsächlich von der AEE INTEC verfasst, mit Inputs von 5 Koautoren. Die Inputs der Koautoren wurden so überarbeitet, dass insgesamt ein in sich schlüssiges Gesamtkapitel entstanden ist.

Auch zu anderen Kapiteln wurden von der AEE INTEC Beiträge geliefert, insbesonders Kapitel 11, in dem sich eine Beschreibung der Erfahrungen mit der Anlage beim Rathaus und Service Center der Stadt Gleisdorf befindet.

6.4.3 D5: Policy Measures

Die Firma S.O.L.I.D. lieferte Inputs für das Policy Paper (D-D5.1)_1, das derzeit noch überarbeitet wird und erst im Laufe des Jahres 2011 auf der IEA-SHC Homepage verfügbar sein wird.

6.4.4 D6: Training Materials

Um den Wissensstand von Planern, Architekten und Installateuren über die Techniken des solaren Kühlens signifikant zu heben, wurden im Workpackage D5.2 „Training material for installers and planners" Schulungsunterlagen erarbeitet und zusammengestellt, um diese dann gegen Ende des Tasks in workshops vorstellen und einzusetzen zu können. Die AEE INTEC bereitete Daten auf, die österreichischer Produkte (Ammoniak-Wasser Absorptionskältemaschine der Firma Pink) bzw. realisierte Gesamtanlagen betreffen.

Weiters wurden Statistikdaten den österreichischen Markt betreffend zusammengestellt, um die Marksituation zu analysieren bzw. notwendige Maßnahmen zu entwickeln, welche die Marktbedingungen für solares Kühlen verbessern sollen.

6.4.5 D8: Industry Newsletters

Ein Newsletter wurde während der Tasklaufzeit verfasst. Darin wurden die geplanten Aktivitäten beschrieben sowie einige Demonstrationsanlagen vorgestellt.

Der Newsletter wurde in 5 Sprachen veröffentlicht, die deutsche Version ist im Anhang zu diesem Bericht zu finden.

7 Schlussfolgerungen

Der Task 38 hat generell viel zum Know-How Transfer zwischen den teilnehmenden Ländern beigetragen.

Ein besonders wichtiges Ergebnis des Tasks ist das entwickelte Monitoringkonzept, das es ermöglicht, Anlagen miteinander zu vergleichen. Vor Beginn des Tasks wurden Anlagen zum solaren Kühlen häufig nur teilweise vermessen. Das heißt, es wurde nicht eine komplette Energiebilanz einschließlich der Hilfsenergieverbräuche erstellt, sondern zum Beispiel nur die Kältemaschine an sich vermessen. Um eine Aussage, über eine tatsächliche Energieeinsparung gegenüber einem Referenzsystem treffen zu können, ist aber die Gesamtbilanz notwendig.

Durch die Arbeit im Task ist jetzt eine ganze Reihe von Anlagen (sowohl große als auch kleine) detailliert vermessen und analysiert worden. Die Ergebnisse zeigen, dass häufig die Kältemaschine zwar hervorragend arbeitet, die Anlagen aber Mängel im Systemkonzept oder in der Komponentenauswahl aufweisen, und damit die Energieeinsparung stark reduziert ist oder sogar mehr Energie verbraucht wird als bei einem konventionellen Referenzsystem.

Diese Erkenntnisse haben dazu geführt, dass in Österreich ein neues e2020 Projekt entworfen wurde (SolarCoolingOpt), indem seit Oktober 2010 praktisch alle in Österreich in diesem Bereich aktiven Institute und Firmen zusammenarbeiten, um die genannten Probleme zu lösen.

Inzwischen wurde auch auf internationaler Ebene als Nachfolge die IEA SHC Task48 „Quality assurance and support measures for Solar Cooling" mit Start im Oktober 2011 vom Exekutivkomitee bewilligt.

8 Know How Transfer

Wie schon erwähnt, fand das abschließende Taskmeeting in Graz statt. Außerdem fand im Anschluss die internationale Tagung Eurosun 2010 ebenfalls in Graz statt, bei der ein Großteil der Taskergebnisse in zahlreichen Tagungsbeiträgen präsentiert wurde. Da österreichische Experten (z.B. Technische Büros, Installateure, Solarfirmen) dadurch ausreichend Gelegenheit hatten, sich über die Taskergebnisse zu informieren, wurde – anders aus ursprünglich vorgesehen – darauf verzichtet, noch einmal zusätzlich einen Industrieworkshop in Österreich zu veranstalten.

Wie schon erwähnt, wurde stattdessen aufbauend auf den Ergebnissen des Tasks ein neues nationales Forschungsprojekt gestartet: SolarCoolingOpt. Bei diesem Projekt sind praktisch alle österreichischen Forschungsinstitute und Solarfirmen beteiligt, die sich mit dem Thema Solares Kühlen beschäftigen. Für dieses Projekt fand am 3./4.11.2010 ein Auftaktworkshop statt, bei dem die wichtigsten Taskergebnisse von den Teilnehmern präsentiert wurden. Dadurch fand ein Know-how Transfer zu den österreichischen Solartechnikfirmen statt.

Die Ergebnisse des Tasks wurden in einer ganzen Reihe von „Technical Reports" dokumentiert, die (soweit bereits verfügbar) von der IEA-SHC Homepage heruntergeladen werden können. Einige Berichte sind derzeit noch in Arbeit und werden im Laufe des Jahres ebenfalls auf der genannten Homepage verfügbar sein.

Außerdem wird ebenfalls im Laufe des Jahres 2011 eine komplette Neuüberarbeitung des Handbuchs für Planer im Springer Verlag erscheinen. Diese wird dann im Buchhandel erhältlich sein.

9 Abbildungsverzeichnis

10 Tabellenverzeichnis

11 Anhang

Technische Berichte der Task38 mit maßgeblicher Beteiligung des Konsortiums:

Generell sind sämtliche frei zugänglichen Berichte auf der IEA SHC Homepage verfügbar unter:
http://www.iea-shc.org/publications/task.aspx?Task=38

D-A1: „Market Available Components for Systems for Solar Heating and Cooling with a Cooling Capacity <20kW - A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)"

D-A2: „Collection of selected systems schemes Generic Systems - A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)"

D-A3a bzw. B3b: "Monitoring Procedure for Solar Cooling Systems - A joint technical report of subtask A and B" (Achtung: hier im Anhang als 99% draft Version; die offizielle Endfassung wird im Herbst 2011 auf der Task38 Homepage verfügbar sein)

D-A3b:" Monitoring Results - A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)"

D-A5: "Installation, Operation and Maintenance Guidelines for Pre-Engineered Systems - A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

D-B1: "State of the art on existing solar heating and cooling systems
A technical report of subtask B"

D-C1: "State of the art – Survey on new solar cooling developments - A technical report of subtask C"

D-C2a: "Description of simulation tools used in solar cooling - New developments in simulation tools and models and their validation: Solid desiccant cooling + Absorption chiller - A technical report of subtask C

D-C5: "Hygienic Aspect of Small Wet Cooling Towers - A technical report of subtask C

D-D8a: "Industrie Newsletter – Erste Ausgabe: 01 – 2009 IEA – SHC Task 38 Solar Air-Conditioning and Refrigeration"

Task 38
Solar Air-Conditioning
and Refrigeration

D-A1:

Market Available Components for Systems for Solar Heating and Cooling with a Cooling Capacity < 20 kW

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: October 2010

Edited by Dagmar Jaehnig[1]

[1]Institution	**AEE INTEC**
Address	**Feldgasse 19, A-8200 Gleisdorf, Austria**
Phone	**+43-3112-5886-28**
Fax	**+43-3112-5886-18**
e-mail	**d.jaehnig@aee.at**

Contributions from Torsten Koller / Alexander Eichhorn[2], Tomas Núñez[3], Wolfgang Streicher[4], Harald Moser[5], Erich Podesser[6], Daniel Mugnier[7] and Lars Reinholdt[8]

[2]Institution	**Stuttgart University, Institute for Thermodynamics and Thermal Engineering (ITW)**
Address	**Pfaffenwaldring 6, 70550 Stuttgart, Germany**
Phone	**+49 711 685 69446**
Fax	**+49 711 685 63503**
e-mail	**koller@itw.uni-stuttgart.de**

[3]Institution	**Fraunhofer Institute for Solar Energy Systems (FhG-ISE)**
Address	**Heidenhofstraße 2, 79110 Freiburg, Germany**
Phone	**+49 761/4588-5533**
Fax	**+ 49 761/4588-9533**
e-mail	**tomas.nunez@ise.fraunhofer.de**

[4]Institution	**Graz University of Technology, Institute of Thermal Engineering (IWT TU-Graz)**
Address	**Inffeldgasse 25 B, A-8010 Graz**
Phone	**+43 316 873 7306**
Fax	**+43 316 873 7305**
e-mail	**w.streicher@tugraz.at**

[5]Institution	**Graz University of Technology, Institute of Thermal Engineering (IWT TU-Graz)**
Address	**Inffeldgasse 25 B, A-8010 Graz**
Phone	**+43 316 873 7304**
Fax	**+43 316 873 7305**
e-mail	**harald.moser@tugraz.at**

[6]Institution	**Podesser Consulting**
Address	**Graz, Austria**
Phone	**+43-316 672605**
Fax	**+43-0316 672605**
e-mail	**erich.podesser@utanet.at**

[7]Institution	**TECSOL**
Address	**Perpignan, France**
Phone	**+33/6/67/52/41/06**
Fax	**+33/4/68/68/16/41**
e-mail	**daniel.mugnier@tecsol.fr**

[8]Institution	**Danish Technological Institute**
Address	**Denmark**
Phone	**(+45) 7220 1270**
e-mail	**lars.reinholdt@teknologisk.dk**

Contents

1 Introduction

Systems for solar heating and cooling with a cooling capacity below 20 kW should be as much pre-engineered as possible. That means that the entire system layout and the size of all components is pre-defined by the manufacturer or seller of the system. To install such a system there is no detailed planning process necessary. It can be bought “off the shelf” designed for a given heating and cooling load and can be installed by an HVAC installer. Unfortunately, only few companies offer this kind of pre-engineered system up to now.

To help companies in putting together such a system, this report aims at showing components for (small) solar heating and cooling systems that are on the market and can be used to design pre-engineered systems.

The report is divided into 4 subchapters:

1) Solar combisystems

Solar combisystems without cooling function are the basis for each solar heating and cooling system. The report shows the variety of systems on the market and discusses which characteristic makes them more or less suitable for coupling with a thermally driven chiller.

2) Thermally driven chillers

In this chapter, an overview of chillers on the market up to a cooling capacity of 20 kW is given. Only chillers that are commercially available or at least in a pre-commercial stage, are considered. Tables dealing with different chiller technologies show the most important technical data of each chiller.

3) Heat rejection units

As there is a large number of companies offering heat rejection units, there is not a complete list of manufacturers and models included in this report. Instead, there is a description of the most important technologies for heat rejection for small scale systems. In addition, there is a list of heat rejection units that is used in systems included the Task 38 (Subtask A) monitoring program. This list shows some key data of these systems.

4) Cold stores

Currently the technology for cold storage that is used for small systems are simply cold water stores. Because there are a lot of manufacturers on the market that offer this kind of stores, a list of manufacturers and models is not included.

However, ice storage is becoming increasingly interesting also for small-scale systems (for research and demonstration systems). Therefore, a short description of the available technologies is included in this report as well as a list of companies that is engaged in ice storage. Although small units are not available on the market they may be available on request.

2 Solar Combisystems

By Dagmar Jaehnig, AEE INTEC, Austria and Wolfgang Streicher, Graz University of Technology, IWT, Austria

2.1 Scope

This section aims at identifying solar combisystems for domestic hot water preparation and space heating that may be suitable for integration into a solar cooling system. The analysis is mainly based on the generic systems that were elaborated within IEA SHC Task 26 on Solar Combisystems. An overview of these generic systems was published in 2000 (Suter et al., 2000). In total, 21 generic systems were identified within Task 26.

2.2 Generic Solar Combisystems Without Cooling Function from IEA SHC Task 26

The 21 generic solar combisystems identified by IEA SHC Task 26 differ from one another in many ways.

Two systems (#20 and #21) are designed for seasonal heat storage. The excess solar heat in the summer months is stored to be used in the winter months. Therefore, a combination with a solar cooling system does not make much sense and these systems are excluded from this analysis of suitability of solar combisystems for integration of a cooling system.

Fig. 1 shows a classification of the remaining 19 systems according to three categories which will be explained in detail in the following three subchapters. The letters and numbers in the figure refer to the IEA SHC Task 26 lettering scheme of system categories and the IEA SHC Task 26 system numbering respectively.

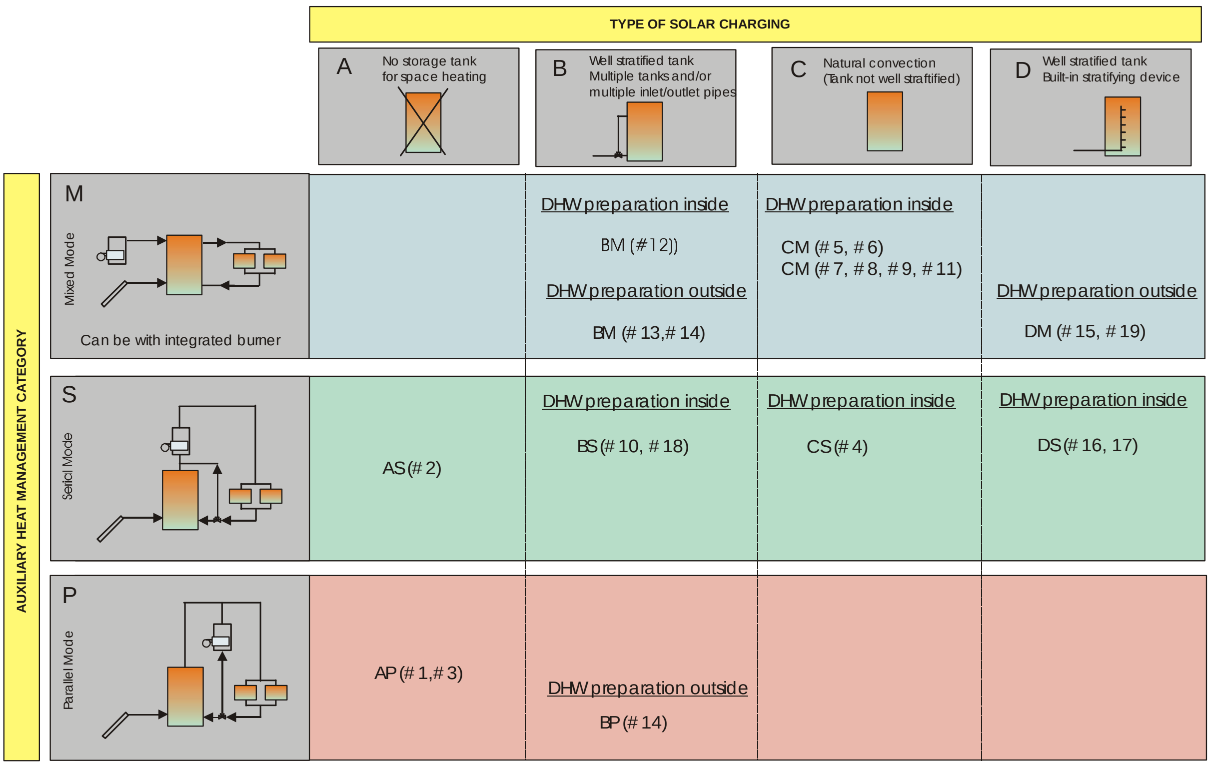


Fig. 1: Generic solar combisystems according to IEA SHC Task 26.

2.2.1 Heat Storage

There are 4 basic categories regarding heat storage.

- Category A: No controlled storage device for space heating. There is no tank for storing energy for space heating. These systems use either the floor slab of the building with an underfloor heating system for storage purposes or there is no possibility at all to store heat for space heating.
- Category B: Heat management and stratification enhancement by means of multiple tanks ("distributed storage") and or multiple inlet/outlet pipes and/or 3- or 4-way valves to control the flows through the inlet/outlet pipes.
- Category C: Tanks using natural convection only to maintain stratification to a certain extent but without built-in stratification devices ("stratifiers") for further stratification enhancement.
- Category D: Heat management using natural convection in storage tanks and built-in stratification devices

2.2.2 Auxiliary Heat Management

- Mixed mode (M): The heat storage tank is charged by both solar collectors and the auxiliary heater. Some of these systems have an auxiliary burner integrated into the storage tank; some use an external auxiliary heater.
- Serial mode (S): The space heating loop may be fed by the auxiliary heater, or by both the solar collectors (or a storage unit for solar heat) and the auxiliary heater connected in series in the return line of the space heating loop.
- Parallel mode (P): The space-heating loop may be fed alternatively by the auxiliary heater and by both the solar collectors (or a storage unit for solar heat); or there is no hydraulic connection between the solar-heat distribution and the auxiliary-heat emission.

The mixed mode is by far the most common mode. However, the serial mode is more common in Spain where the mixed mode is prohibited by law.

2.2.3 Location of DHW Preparation Unit

The domestic hot water preparation unit can either be located inside or outside of the heat storage tank.

- Outside the storage tank: This category includes external flat plate heat exchangers that are heated from the heat storage tank. In many cases, the heat exchangers are protected from high temperatures by a tempering valve. This protects the heat exchanger from scaling. Another option is to use a separate tank to store the domestic water. The temperature of this store can be limited to lower temperatures to prevent scaling.
- Domestic hot water preparation units inside the tanks are often used in solar combisystems. These include: Tank-in-tank systems where a small domestic hot water tank is integrated in the top part of the heat storage tank, different types of immersed heat exchangers such as corrugated-tubes that contain a relatively large amount of potable water that is kept at a temperature high enough to be able to deliver hot water to the tap at all times. For these systems scaling may also be a problem if the tank is used for storing heat at very high temperatures for chiller operation in summer.

2.2.4 Advantages and Disadvantages of DHW Preparation Methods

Separate DHW Tank

In Fig. 2 a picture shows a possible situation of a large space heating tank and a small hot water tank. The hot water tank (200-300 ltr) is typically heated by one or two immersed heat exchangers with heat either from the solar thermal collectors, from the space heating tank or from the boiler. Alternatively, when a lot of heating power is necessary, a solution with a flat plate heat exchanger is possible to heat the hot water tank. In principle many hydraulic concepts are possible. The hot water tank can be heated by the conventional boiler, by the solar collectors or both or only indirectly by transferring heat from the main energy store. Based on this flexibility, a proper design of the specific system and the used components has to be done and strongly influences the comfort, performance and efficiency of the system.

Fig. 2: Separate DHW-tank

Advantages

- With low temperature space heating systems (floor heating, wall heating) best “thermal stratification” can be achieved between the two very different required temperature levels (space heating loop: 25-40°C and hot water demand: 45-60°C with cold water supply from the mains of 10-5°C).
- Low return temperature to the collector when the collector loop is also directly connected to the hot water tank.
- Scaling problems occur mainly if a flat plate heat exchanger between main store and hot water tank is used. This can be avoided by limiting the flow temperature to the heat exchanger at 60°C.
- High hot water peak power on demand side.
- Maintenance and replacing of the hot water tank is facilitated.
- If the boiler directly heats the auxiliary volume of the domestic hot water tank, the set point temperature for the boiler can be low.

Disadvantages

- Two stores require more space and piping.
- Hot water tank must be resistant against corrosion.

- Very high ratio of surface to volume leads to relatively high heat losses.
- A pump is necessary to transfer energy from the one tank to the other.
- If an immersed heat exchanger only in the top of the hot water tank is used, to transfer heat from the main heat store to the domestic hot water store, the main heat store cannot be cooled to very low temperatures (mains temperature) leading to lower collector efficiencies. A concept with a flat plate heat exchanger between space heating tank and hot water tank performs better, but is much more expensive.
- A circulation loop connected to the hot water tank leads to very bad thermal stratification in the space heating tank due to the high circulation return temperature.
- Parasitic energy (electricity for the pump) is required for hot water preparation.
- Relatively large volume on a temperature level that favors legionella growth.
- If the boiler is only heating the auxiliary volume of the space heating tank, the boiler set point temperature must be relatively high.

Tank in Tank

The tank-in-tank concept was developed in order to reduce space requirements. The DHW tank (100-200 ltr) is integrated in the heat store as shown in Fig. 3.

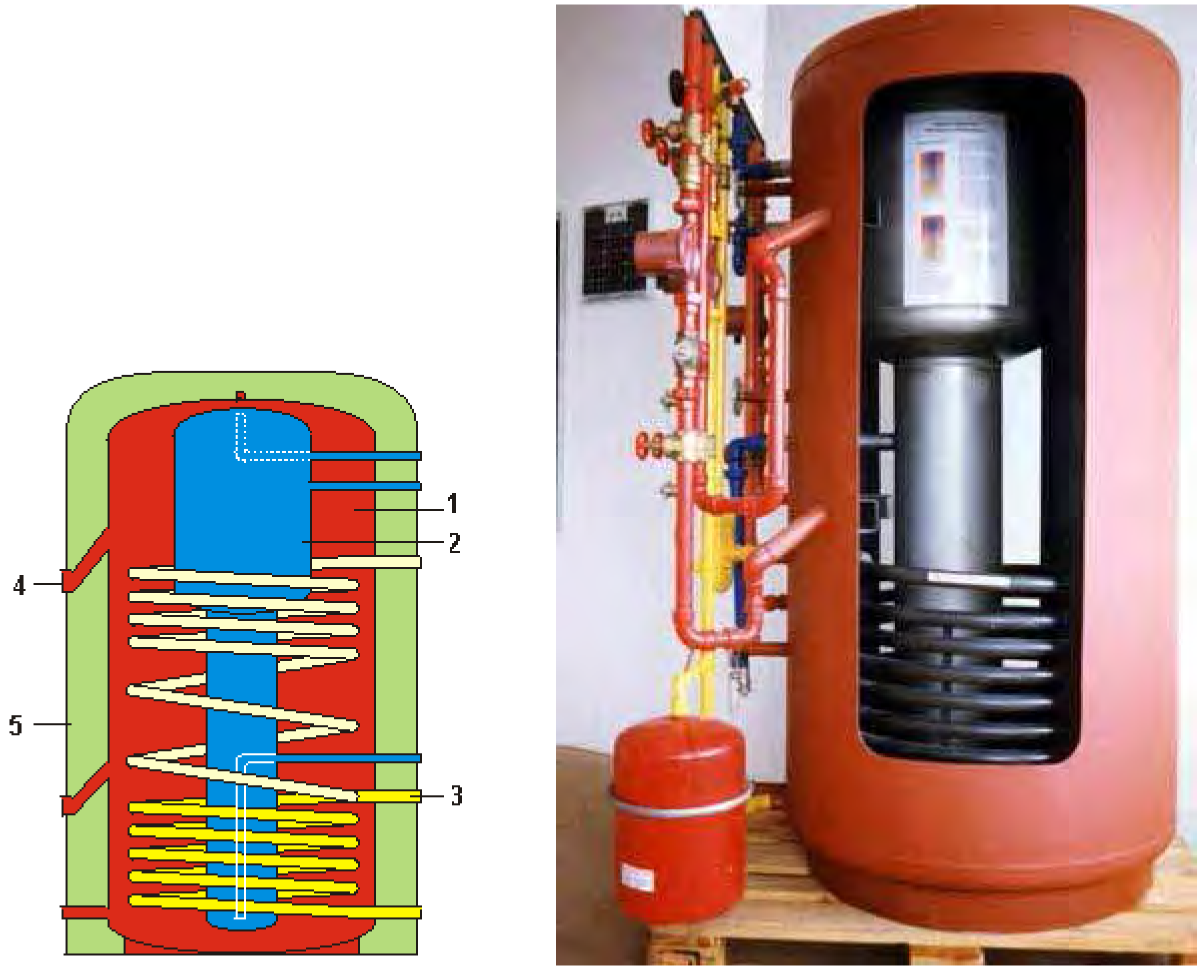


Fig. 3: Tank-in-tank system (Jenni, CH)

Advantages

- High peak power on demand side.
- Little effect of the circulation circuit if the circulation return line is connected at the height that corresponds to the temperature level of the circulation return line
- Scaling problems are rare. Because the diameters of the integrated tank are large compared to a heat exchanger, scaling usually has little effect. Lime can accumulate at the bottom end of the DHW tank, if the temperature of the heat store is above 60℃.
- No parasitic (additional electric) energy for hot water preparation.
- Little space requirement.
- Much smaller hot water volume compared to the hot water tank which reduces the legionella problem in the tank.
- Little heat losses due to the compact design.
- The set point temperature for the boiler can be relatively low.

Disadvantages

- Depending on details of how the hot water tank is designed, the temperatures in the tank cannot be reduced close to the mains temperature. This depends mainly on how far down the integrated tank reaches. If the hot water tank is only integrated in the top part of the store, the bottom of the tank cannot be cooled down properly.
- Some volume in the lower part is on a temperature level that favors legionella growth.
- Maintenance and replacement is almost impossible, but depends on the tank design.
- The top part of the space heating tank must be heated to hot water set temperature all the times.

Internal Heat Exchanger for DHW-preparation

The tank-in-tank concept was further developed to a DHW preparation system with immersed heat exchanger (DHW volume: 30-70 ltr) shown in Fig. 4 to reduce the legionella risk and the cost. Different systems are on the market, mainly differing in how the parts of the heat exchanger are situated at different heights of the heat store.

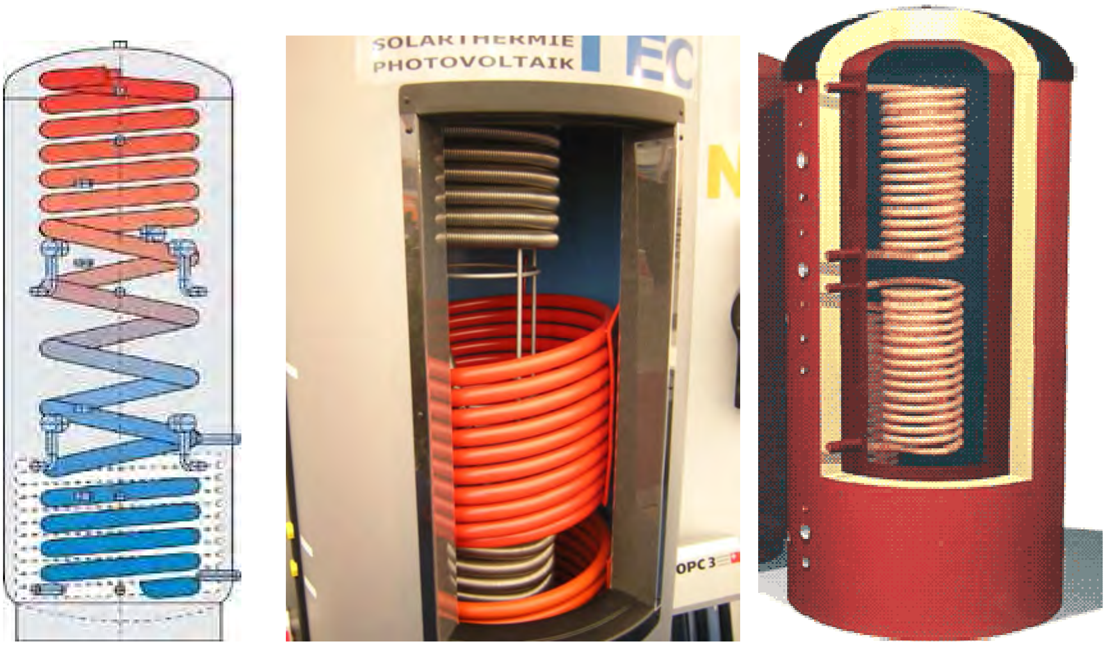


Fig. 4: Internal heat exchanger for DHW-preparation (left and middle: Feuron, CH / right: Solentek, S)

Advantages

- No parasitic (additional electric) energy for hot water preparation.
- Low legionella risk because only a very little volume is kept warm.
- Little space requirement.
- Little effect of the circulation circuit if the circulation return line is connected at the height that corresponds to the temperature level of the circulation return line
- Little heat losses due to the compact design.

Disadvantages

- Low peak power, therefore depending on comfort requirements, the auxiliary set temperature must be significantly higher than the domestic hot water set temperature.
- Depending on details of how the heat exchanger is designed and where in the store it is located, the minimum temperature in the store that can be reached due to the incoming mains temperature can be relatively high.
- Scaling problems possible if temperature in heat store exceeds 60℃.
- Maintenance and replacement is almost impossible, depending on the design.

Flat Plate Heat Exchanger Unit

The last step of the development to reduce the volume of DHW (1-2ltr) which is kept warm is to prepare DHW in the continuous-flow principle by an external flat plate heat exchanger.

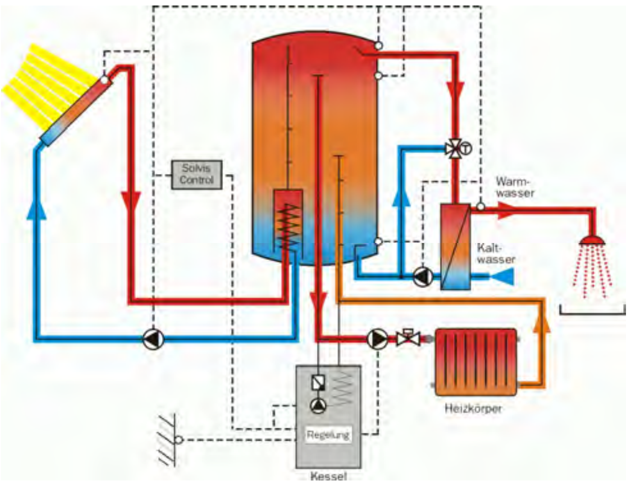


Fig. 5: External flat plate heat exchanger unit for DHW (left: SOLVIS, GER/right: Sonnenkraft, AUT)

Advantages

- Peak power is limited, but the power can easily be adapted using a correctly designed heat exchanger, depending on the requirements.
- Low return temperature to the heat store.
- Best usage of the stored energy in the tank.
- No scaling problems if flow temperature is limited.

- Lowest legionella risk because almost no volume is kept warm.
- Little space requirement.
- Maintenance and replacement is easily possible.

Disadvantages

- Circulation loop has negative influence on the thermal stratification in the heat store due to high return temperature if no stratification device for the return flow is used in the tank.
- Advanced control system is needed for good behavior of this concept.
- A pump is necessary to run the hot water preparation unit.
- Parasitic energy (electricity for the pump) is required.
- Depending on system design, increased heat losses due to the external installation of the hot water heat exchanger.
- The top part of the space heating tank must be heated to a temperature higher than hot water set temperature all the times to enable enough peak power during hot water preparation, if the boiler peak power is not high enough.

2.3 Suitability for Integration of a Solar Cooling System

Connecting a thermally driven cooling machine to such a combisystem means to add a heat sink in summer that needs a low temperature difference (5-10 K) at high absolute temperature level (60-90℃). This temperature level is well above the temperature needed for DHW preparation. Space heating normally does not occur at the same time as cooling is needed.

In a first step, some of the solar combisystems defined in IEA SHC Task 26 were excluded because their system concept is not suitable for integration of a solar cooling system for the following reasons:

- Three systems do not include a storage tank for storing heat for space heating. The heat is stored in the concrete slab of the building itself. Although it may be possible to operate a solar cooling system without a hot storage tank, by delivering the generated heat from the solar thermal collectors directly to the chiller, the heating and cooling systems would in this case not be truly integrated but rather two separate systems that use the same collector field. Therefore these systems are excluded from this report**. (Systems #1, #2, #3)**
- Three systems are very small (small storage tank and small collector area). Such systems are typically used e.g. in the Netherlands and Denmark. These systems are designed for relatively small solar fractions and the small tank sizes are not suitable for use as hot storage tank for a solar cooling system. **(Systems #4, #5, #6)**

After excluding these systems, there are 13 systems remaining: #7, #8, #9, #10, #11, #12, #13, #14, #15, #16, #17, #18, #19 (see Fig. 6).

To analyze which of these systems are more suitable to integrate solar cooling, the main aspect is how the domestic hot water preparation is done. The key problem here is to avoid excessive scaling if the tank is regularly heated above the 80℃ required to power a thermally driven chiller.

Four of these systems use an external domestic hot water preparation. Two use separate tanks and two others use an external flat plate heat exchanger. In both cases, the flow temperature to the store/HX can easily be limited to 60℃ which avoids scaling.

For the systems with integrated domestic hot water preparation the situation is more difficult. In the case of tank-in-tank systems, the diameters inside the integrated tank are much larger than in integrated heat exchangers. Therefore, scaling may not cause a great problem. Integrated heat exchangers on the other hand can be blocked completely by lime formation if the temperatures in the store are high and hard potable water is used. Some manufacturers recommend limiting the temperature in the store to e.g. 60℃ which avoids scaling. In that case, the system cannot be used for solar cooling application anymore because temperatures well above 60℃ are needed for the coo ling machine.

Therefore, a big question mark has to be put behind systems with integrated heat exchangers for domestic hot water preparation and a small question mark for tank-in-tank systems. For both system types experiences with installed systems will show whether they are suitable for integration of solar cooling or not.

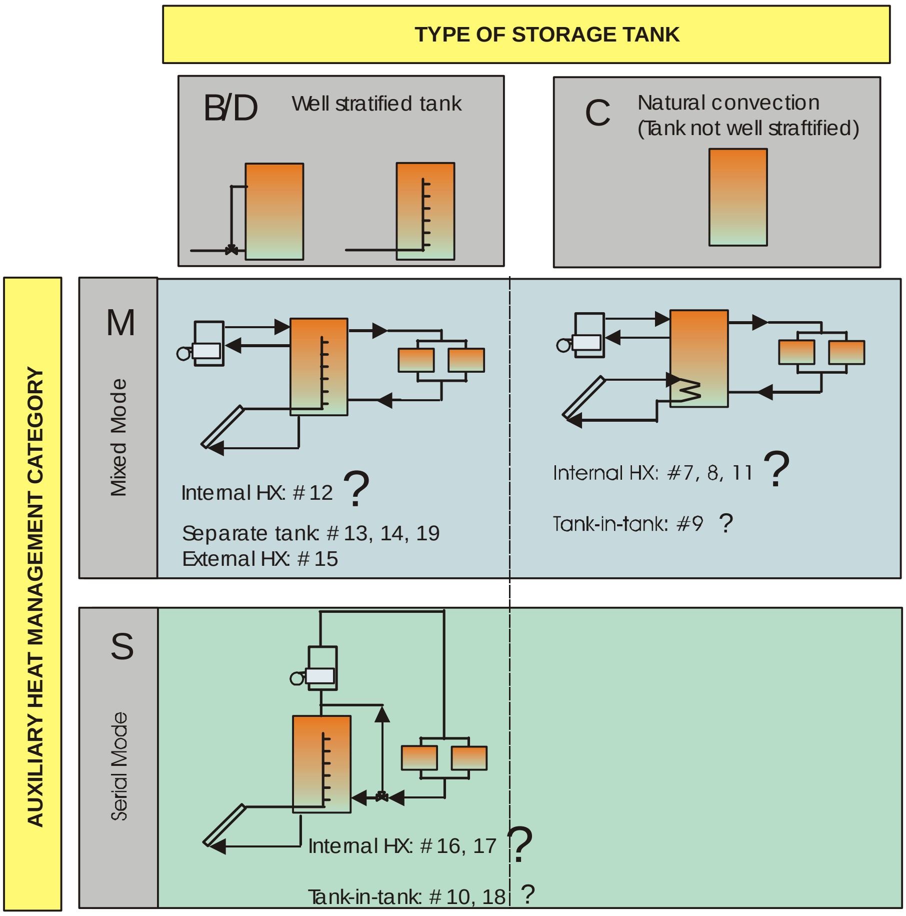


Fig. 6: Remaining 13 solar combisystems (from IEA SHC Task 26) after excluding all those that are not suitable for integration of solar cooling

2.4 Other Features Relevant for Solar Cooling Integration

After having described different solar combisystems without cooling function we now need to look at topics that are important for solar cooling into applications but are not relevant for the solar combisystem for heating only.

2.4.1 Possibility to Reduce Tank Size for Summer Operation

If the store size of the solar combisystem is relatively large, it may be necessary to reduce the tank size in summer to ensure that cooling operation can start relatively early in the day without having to heat the entire storage tank to high temperatures first. This can be done by including an extra outlet at mid height of the storage tank that is used in summer. Of course this outlet can also be used in winter to reach temperatures for space heating operation faster. This second outlet can be an additional integrated heat exchanger or an additional outlet to the external heat exchanger with a mixing valve. This feature is already included in system # 10, 12 and 18 but could also be included in other systems if necessary.

2.4.2 Stratification in the Tank

According to the high temperature level, low temperature lift and high mass flow needed to drive solar cooling machines, principally a new volume above the DHW preparation could be placed in the store. This would lead to a strong increase of the storage volume. Therefore in summer normally the whole storage volume is used for solar cooling and DHW preparation needs to be mixed down to lower temperatures. The auxiliary heat remains on the upper part of the store, as the power of the heater should be sufficient for driving the cooling machine and the DHW production. Additionally DHW production is for e.g. office buildings the smaller part compared to space heating, so the main focus of the hydraulics for summer operation can be laid on the cooling part. This is maybe different in hotels, where the DHW demand can be significantly higher. Here a solar preheating zone for DHW in the bottom of the tank may be used. In this case, stratifiers for the solar inlet may be used to allow preheating of the water at low temperatures and automatically switching back to the high temperature level for solar cooling. In spring and autumn, where DHW production, space heating and cooling may occur in the same time periods, the optimization of the volumes has to be done by simulation.

2.4.3 High-flow / Low-flow

In most cases a collector field connects several single collectors. These collectors can be either connected in series or in parallel; mainly combinations of the two forms of connection are used. To assure turbulent flow and therefore a high heat transfer rate in each collector tube, the volume flow through each collector should be kept above a certain level. On the other hand the volume flow should not be too high in order to avoid unnecessary pressure drop and therefore high electricity demand for the circulation pumps.

By connecting the collectors in series, the total mass-flow through the whole collector field is usually reduced (low-flow) and the temperature rise in the collector is increased. The advantage of this is that hot water can be supplied quickly. The disadvantage is a higher thermal loss of the absorber to the environment, which is due to the larger temperature difference. The pump electricity demand decreases due to the lower level of total mass flow in the collector loop but increases due to the higher pressure drop in the collectors. The higher pressure loss of collectors connected in series can be overcome partly by sizing the collector loop pipes bigger and therefore lower the pressure losses in these tubes. When connecting in series, there is a more regular flow through the collector area due to a higher driving pressure drop over the field. The hydraulic layout has to be adjusted to the total mass flow. Low-Flow systems normally use stratification units in the tank in order to prevent mixing the hot temperature from the collector fluid with colder temperature at the inlet of the store.

High-flow installations with collector areas below 15 m² are most often combined with internal heat exchangers in the storage tank. For bigger fields external heat exchangers with fixed inlets are used. There can be two or more internal heat exchangers or pairs of inlets located at different heights in the storage tank to allow stratification even with high-flow systems.

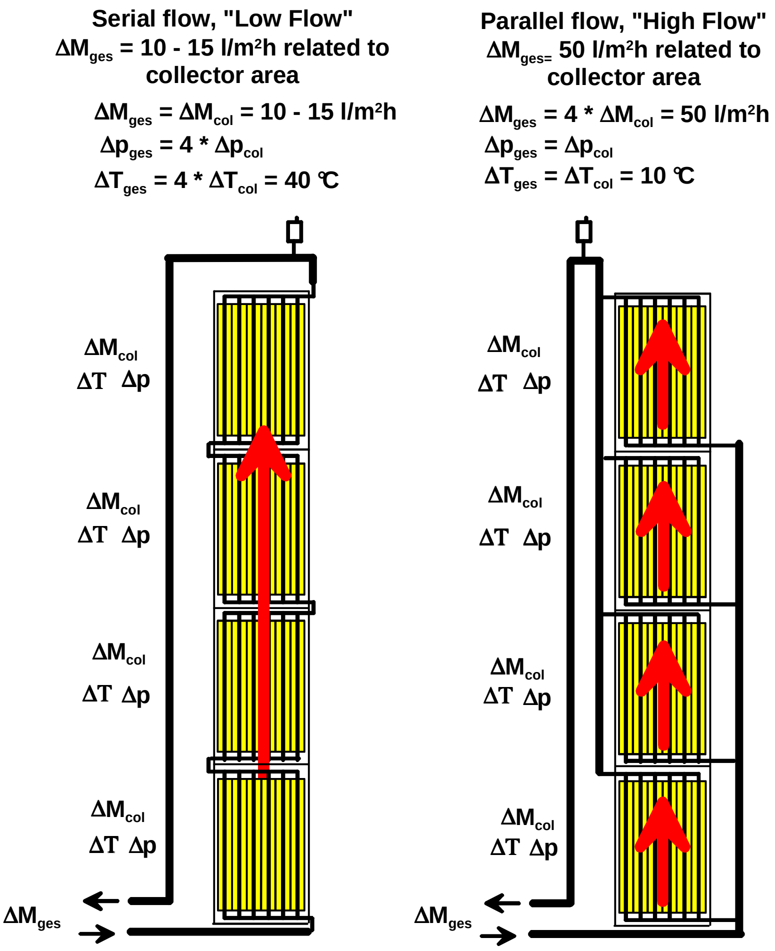


Fig. 7: Hydraulics for the collector field for low flow serial (left) and high flow parallel (right) systems. In all cases care has to be taken to maintain an equal resistance in all parallel lines. This can be partly achieved by a "Tichelmann" arrangement where the added length of the inlet and outlet pipes is the same for all parallel collectors (Streicher 2008).

Many existing solar combisystems (without cooling function) work with the low-flow principle in the collector loop. The advantage is that the necessary temperature lift for domestic hot water preparation and space heating can be reached in one step in the solar collector if the solar radiation is high enough. Therefore, the auxiliary heater has to be operated less frequently.

When connecting such a system to a thermally driven chiller, this can be a problem because the temperature difference between flow and return of the hot side of the chiller is typically small (~ 6 K). Therefore it would make more sense to operate the collector loop (at least for the cooling mode) at this small temperature difference rather than the roughly 30-40 K that are typical for low-flow solar combisystems due to the domestic hot water (DHW) demand. During the summer high flow operation would be favourable for solar cooling but low flow operation would be favourable for DHW production.

One option to overcome this dilemma is to switch the collector loop at least between low-flow operation in winter to high-flow operation in summer. However, care has to be taken to ensure proper turbulent flow and acceptable pressure drops in the collector circuit as well as on the balanced flow through all parallel collector tubes.

There are in principle four options:

- The collector hydraulics stay the same in summer and winter but the mass flow through the collector varies by switching the pump speed or the type of pump. In this case there is either a high pressure drop in the collector loop during high flow operation, that results in a high electricity demand of the collector pump or the collector loop is designed for high flow and during low flow operation the flow regime in the collector is laminar which reduces the collector efficiency by a few percentage points and may additionally result in non-uniform flow in the parallel collector tubes. As high flow has about four times the volume flow of low flow, the pressure drop is about 16 times (4^2) higher in high flow than for low flow (according to Bernoulli's law with dp ~ (ρv^2)/2. The electricity demand is even 64 times higher (P_{el} = (dV * dP)/ η ~ v^3. Here also the question arises, that maybe two different pumps for high flow and low flow have to be chosen (high pump head and high volume flow and low pump head at low volume flow).

- The other possibility is to change the hydraulics of the solar collector. This can be done by installing switching valves that connect more collectors in series for low-flow operation. Again probably two pumps are needed (one for high volume flow and low delta T and one for low volume flow and high pump head).
- A third possibility would be to drive the collector field only in high flow and adjust the hydraulic system for space heating accordingly (e.g. no stratifiers in the tank etc.), which gives less useful solar gain in winter but makes the plant cheaper.
- A fourth possibility would be to drive the collector between high and low flow. This assures equal flow in the collectors but gives slightly less solar yield during the heating and cooling season.

Concluding, it becomes obvious, that there is room for plenty of optimization for solar thermal driven cooling systems coupled to solar combisystems.

3 Thermally Driven Chillers

By Tomas Núňez, Fraunhofer ISE, Germany

A survey of small capacity thermally driven chillers was carried out in the frame of the PolySMART project. The intention of the survey was to assess the present developments in the field taking into account laboratory developments, prototypes, small series products and commercially available machines. The survey was limited to the capacity range of about 70 kW cooling power. A total of 34 developments have been registered. The majority of the surveyed machines were prototypes that have been modified and changed in the last three years. Some developments were discontinued or are no longer available. In the tables below an extraction of the survey is presented. The criteria for this extraction are:

- The machine should be available as commercial product, as a small series product or a small series product is to be expected in short time
- The chilling power is up to 30 kW
- Performance data is available.

The machines are classified according to the technology: adsorption systems, including the pairs water / silica gel and water / zeolite, absorption systems with the working pairs ammonia / water and water / lithium-bromide and other technologies which include jet cycles and a three-phase Water / Lithium-chloride absorption chiller. This last chiller is also a thermo-chemical store for heat and cold and therefore a different technology.

3.1 Adsorption Systems

Table 1: Adsorption chillers below 20 kW chilling capacity

Manufacturer			**SorTech AG**	**SorTech AG**	**ECN**	**InvenSor GmbH**	**InvenSor GmbH**
country			Germany	Germany	Netherlands	Germany	Germany
Model name / number			ACS08	ACS15	SoCOOL	HTC 10	LTC 07
Contact data of supplier			SorTech AG Weinbergweg 23 D - 06120 Halle (Saale) Fon: +49 (0)345 279 809-0		ECN POBOX1 NL-1755 ZG Petten the Netherlands tel +31-224 564871	InvenSor GmbH Gustav-Meyer-Allee 25 D-13355 Berlin Email: info@invensor.de Telefon: +49 (0)30 - 46 307 - 396	
internet			www.sortech.de	www.sortech.de	www.ECN.nl	www.invensor.de	www.invensor.de
other supplier / model name			SolarNext / chillii® STC8	SolarNext / chillii® STC15		SolarNext / chillii® ISC7	SolarNext / chillii® ISC10
Basic information		**unit**					
technology			water-silica gel	water-silica gel	water-silica gel	water zeolite	water zeolite
nominal chilling power		[kW]	8	15	2.5	10	7
nominal COP			0.6	0.6	0.5	0.5	0.54
heat rejection type			external	external	external	external	external
intended area of application			air-conditioning	air-conditioning	residential cooling	air-conditioning	air-conditioning
development stage [4)]			pre-commercial	pre-commercial	ID	pre-commercial	pre-commercial
specification sheets available (yes/no)			yes	yes	no	yes	yes
(expected) investment costs		[€]	n.a.	n.a.	1500-2500 (expected)	n.a.	n.a.
Nominal operation conditions							
Driving circuit	power	[kW]	13.4	25	5.5	20	13
	operating temperature (in/out)	[℃]	72 / 65	72 / 65	85 / 79	85 / 77	65 / 59.5
	temperature range (from-to)	[℃]	55..95	55..95	60-95	65-95	55-85

	heat transfer fluid[5)]		water	water	water	water	water
	flow rate	[l/h]	1600	3200	600	2200	2200
	operating pressure	[bar]	4	4	1--2	4	4
	pressure drop	[mbar]	230	260	500 (indicative)	230	230
Heat rejection circuit	power	[kW]	21.4	40	8	30	20
	Temperature (in/out)	[℃]	27 / 32	27 / 32	30 / 40	27 / 33	27 / 31
	temperature range (from-to)	[℃]	22..37	22..37	15-40	27-41	22-37
	heat transfer fluid[5)]		water	water	water	water	water
	flow rate	[l/h]	3700	7000	600	4500	4500
	operating pressure	[bar]	4	4	1--2	4	4
	pressure drop	[mbar]	350	440	500 (indicative)	500	500
chilling circuit	nominal chilling power	[kW]	8	15	2.5	10	7
	chilling temperature (in/out)	[℃]	18 / 15	18 / 15	16 / 12	18 / 15	18 / 15
	temperature range (from-to)	[℃]	6..20	6..20	5--20	8-18	15-18
	heat transfer fluid[5)]		water	water	water	water	water
	flow rate	[l/h]	2000	4000	500	2900	2000
	operating pressure	[bar]	4	4	1--2	4	4
	pressure drop	[mbar]	300	500	300	240	130
parasitic demand	electrical power	[kW]	0.007	0.014	<0.3	0.02	0.02
	water consumption	[l/s]	none	none	none	none	none
Dimensions & weight	Length /width/height	mm	790 / 1060 / 940	790 / 1340 / 1390	n.a.	1300 /650 / 1650	1300 / 650 / 1650
	Weight (empty / operation)	Kg	265 / 295	530 / 590	n.a.	- / 370	- / 370

3.2 Absorption Systems

3.2.1 Ammonia – Water Systems

Table 2: Ammonia - water absorption systems below 20 kW chilling capacity

Manufacturer		**AOSOL**	**Pink GmbH**	**Robur**
country		Portugal	Austria	Italy
Model name / number			chillii® PSC12	ACF60-00 LB
Contact data of supplier		Ao Sol, Energias Renováveis, SA. Parque Industrial do Porto Alto, Portugal. Email: aosol@aosol.pt Telf: +351 263 651 305	Pink GmbH Bahnhofstraße 22 8665 Langenwang Austria Tel: +43 (0)3854/3666	Robur S.p.A. Via Parigi 4/6 24040 Verdellino/Zingonia (Bg) Italy
internet		www.aosol.pt	www.pink.co.at	www.robur.com
other supplier / model name			SolarNext / chillii® PSC12	
Basic information	**unit**			
technology		ammonia-water	ammonia-water	ammonia-water
nominal chilling power	[kW]	8	12	12
nominal COP		0.6	0.65	
heat rejection type		internal	external	internal
intended area of application		air-conditioning	air-conditioning	process cooling
development stage		pre-commercial	commercial	pre-commercial
specification sheets available (yes/no)		not yet	Yes	restricted
(expected) investment costs	[€]	5000	n.a.	

Nominal Operating conditions					
Driving circuit	driving power	[kW]	13.3	18.5	n.a.
	operating temperature (in / out)	[℃]	96 / 86	75 / 68	240
	temperature range (from-to)	[℃]	80-110	75..85	180. .240
	heat transfer fluid[5)]		water	water	diathermic oil
	flow rate	[l/h]	850	2300	3500
	operating pressure	[bar]	2	n.a.	n.a.
	pressure drop in circuit	[Pa]	0.5	n.a.	n.a.
Heat rejection circuit	re-cooling power	[kW]	21.3	30.5	n.a.
	re-cooling temperature (in / out)	[℃]	35 (air temperature)	24 / 29	35 (air temp erature)
	temperature range (from-to)	[℃]	30-42	24	-12 ... 4 5
	heat transfer fluid[5)]		air	water	air
	flow rate	[l/h]	6300	5200	
	operating pressure	[bar]	-		
	pressure drop in circuit	[Pa]	-		
Chilling circuit	chilling power	[kW]	8	12	12
	chilling temperature (in/out)	[℃]	14 / 9	18 / 15	0 / -5
	temperature range (from-to)	[℃]	7-16	6..15	-10 ... 45
	heat transfer fluid		water	water	brine 40% glycol
	flow rate	[l/h]	1030	3400	2600
	operating pressure	[bar]	2	n.a.	3
	pressure drop in circuit	[Pa]	500	n.a.	400
Parasitic demand	electrical power	[kW]	0.5	0.3	0.84
	water consumption	[/s]	-		none
Dimensions & weight	Length / width / height	mm	n.a.	800 / 600 / 2200	890 / 1230 / 1290
	Weight in operation	kg		350	370

3.2.2 Lithium – Bromide / Water Systems.

Table 3: Lithium-bromide / water absorption systems below 20 kW chilling capacity

Manufacturer		**Sonnenklima GmbH**	**EAW**	**Yazaki**	**Rotartica**	**Rotartica**
country		Germany	Germany	Japan	Spain	Spain
Model name / number		suninverse	Wegracal SE 15	WFC-SC 5	045V	045
Contact data of supplier		SK SonnenKlima GmbH Am Treptower Park 28-30 D 12435 Berlin Tel: +49 30 53 0007 700 Fax: +49 30 53 00 07 17	EAW Energieanlagenbau Westenfeld GmbH Oberes Tor 106 98631 Westenfeld Telefon: 036948 84-132 Telefax: 036948 84-152 info@eaw-energieanlagenbau.de	Yazaki Europe Ltd. Environmental and Energy Equipment Operations Robert-Bosch-Strasse 43, 50769 Köln (Cologne), Germany Phone: (49) 221-59799-0 Fax: (49) 221-59799-197 Email: info@yazaki-airconditioning.com	Avda. Cervantes 45, 48970 Basauri (Bizkaia) Spain Tel: (+34) 94 402 51 20 Fax: (+34) 94 402 51 21 E-mail: rotartica@rotartica.com	
internet		www.sonnenklima.de	www.eaw-energieanlagenbau.de	www.yazaki-airconditioning.com	www.rotartica.com	
other supplier / model name			SolarNext / chillii® ESC15	SolarNext / chillii® WFC18		
Basic information	**unit**					
technology		absorption water-LiBr	absorption water-LiBr	absorption water-LiBr	absorption water-LiBr	absorption water-LiBr
nominal chilling power	[kW]	10	15	17.6	4.5	4.5
nominal COP		0.78	0.71	0.7	0.62	0.62
heat rejection type		external	external	external	integrated	external
intended area of application		domestic, commercial	air-conditioning	air-conditioning	domestic	domestic
development stage		Not available	commercial	commercial	Not available	Not available
specification sheets available (yes/no)		yes	yes	yes	yes	yes
(expected) investment costs	[€]	n.a.	15,000		n.a.	n.a.

Nominal Operating conditions							
Driving circuit	driving power	[kW]	13.6	21	25.1	6.7	6.7
	operating temperature (in / out)	[°C]	75 / 65	90 / 80	88 / 83	90 / 83	90 / 83
	temperature range (from-to)	[°C]	75 - 95		70 - 95	8 0 - 105	80 - 105
	heat transfer fluid		water	water	water	water	water
	flow rate	[l/h]	1200	1800	4320	1200	1200
	operating pressure	[bar]	<= 2,5	< 6	< 5.88	1.5	1.5
	pressure drop in circuit	[mbar]	200	400	770	200	200
Heat rejection circuit	re-cooling power	[kW]	24	35	42.7	11.7	11.7
	re-cooling temperature (in/out)	[°C]	27/35	30 / 36	31 / 35	40	40
	temperature range (from-to)	[°C]	20 - 35 (approx.)			25-45	25-45
	heat transfer fluid		water, open cycle	water	Water	Air	water
	flow rate	[l/h]	2600	5000	9180		1980
	operating pressure	[bar]		6	5.88		1.5
	pressure drop in circuit	[mbar]	320	900	383		1116
Chilling circuit	chilling power	[kW]	10	15	17.6	4.5	4.5
	chilling temperature (in/out)	[°C]	18 / 15	17 / 11	7 / 12.5	16	16
	temperature range (from-to)	[°C]	6 - 15			8-22	8-22
	heat transfer fluid[5)]		water	water	Water	water	water
	flow rate	[l/h]	1300 - 2900	1900	2770	1200	1200
	operating pressure	[bar]	< 2,5	< 6	< 5.88	1.5	1.5
	pressure drop in circuit	[mbar]	350	400	526	300	300
parasitic demand	electrical power	[kW]	0,12	0,3	0.048	1.2 incl. fan	0.4
	water consumption	[/s]	none	none	None	none	none
Dimensions & weight	Length / width / height	mm	795 / 1130 / 1960	1750 / 760 / 1750	594 / 744 / 1736	1092 / 760 / 1150	1092 / 760 / 1150
	Weight in operation	Kg	550	660	420	290	290

3.3 Other Technologies

Table 4: Other technology chillers below 20 kW chilling capacity

Type of technology		**absorption water-LiCl**	
Manufacturer		**Climatewell**	
country		Sweden	
Model name / number		Climatewell 10	Climatewell 20
Contact data of supplier		ClimateWell AB Instrumentvägen 20 126 53 Hägersten Stockholm Sweden info@climatewell.com	
internet		www.climatewell.com	
other supplier / model name			
Basic information	**unit**		
nominal chilling power	[kW]	4	n.a.
nominal COP		0.68	0.68
heat rejection type 3)		external	external
intended area of application		residential	residential
development stage 4)		commercial	commercial
specification sheets available (yes/no)		yes	yes
(expected) investment costs	[€]	7500	n.a.

Nominal Operating conditions				
driving circuit	driving power	[kW]	n.a.	n.a.
	operating temperature (in/out)	[℃]	80 / 70	80 / 70
	temperature range (from-to)	[℃]	60-120	60-120
	heat transfer fluid[5)]		Water	Water
	flow rate	[l/h]	900	1500
	operating pressure	[bar]	10 (max)	10 (max)
	pressure drop in circuit	[Pa]	200	450
Heat rejection circuit	re-cooling power	[kW]	n.a.	n.a.
	re-cooling temperature (in/out)	[℃]	30 / 40	30 / 4 0
	temperature range (from-to)	[℃]	20-40	20-40
	heat transfer fluid[5)]		Water	Water
	flow rate	[l/h]	1800	3000
	operating pressure	[bar]	10 (max)	10 (max)
	pressure drop in circuit	[Pa]	250	580
chilling circuit	chilling power	[kW]	4	n.a.
	chilling temperature (in / out)	[℃]	18 / 13	18 / 1 3
	temperature range (from-to)	[℃]	8-18	8-18
	heat transfer fluid[5)]		Water	Water
	flow rate	[l/h]	900	1500
	operating pressure	[bar]	10 (max)	10 (max)
	pressure drop in circuit	[Pa]	200	450
parasitic demand	electrical power	[kW]	0.03	0.03
	water consumption	[/s]	-	-
Dimensions & weight	Length / width / height	mm	1685 / 1211 / 807	1940 / 1211 / 807
	Weight in operation	Kg	835	1078

4 Heat Rejection

There is a number of different heat rejection units also for small thermal capacities available on the market. Therefore, not all units on the market will be presented in this report. However, this chapter should give an overview of the available technologies.

In the second part of this chapter, some key data of the heat rejection units that are used in systems being monitored within the framework of Task 38. The choice of heat rejection solution is often critical to the electrical power consumption of the thermally driven chiller. As it can be seen in chapter 3, most of the chillers are water cooled which means that the cooling water supplied to chiller has to be connected to some type of equipment rejecting the heat to the environment.

The third part of this chapter gives some general considerations on the electricity consumption of heat rejection units and shows ways how to reduce it.

4.1 Heat Rejection Technologies

By Harald Moser, Graz University of Technolgy, IWT, Erich Podesser, Podesser Consulting, Tomas Núñez, Fraunhofer ISE, Daniel Mugnier, TECSOL and Lars Reinholdt, DTI

Generally speaking, different heat sinks are possible to reject the heat, e.g. air, ground or water. While the use of ground and water depends strongly on the local conditions, air is available for almost all applications.

For rejection of heat to the ambient air, in principle two types of systems are available: open cooling towers (or wet cooling towers) and closed cooling towers or (dry coolers). As a combination of these, adiabatic pre-cooling of the air in the dry cooler and hybrid cooling towers should be mentioned.

The main difference between these technologies is that in the dry cooler the cooling water rejects the heat to the air via a heat exchanger and in wet cooling towers the cooling water is sprayed into the air and direct heat and mass transfer takes place. Thus in dry coolers only sensible heat and in wet cooling towers mainly latent heat is exchanged.

A further option is to use ground coupled systems like vertical boreholes or horizontal ground coupled heat exchangers for heat rejection. These systems are well known as low temperature heat sources for ground coupled heat pumps and as heat sink for non-mechanical cooling systems. The performance strongly depends on the ground characteristics and an accurate dimensioning.

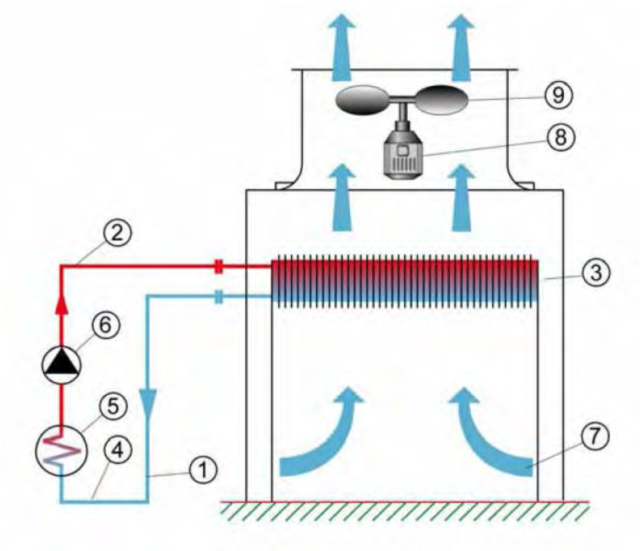
4.1.1 Dry Cooler

Dry coolers consist generally of finned heat exchangers (air to water), fans and a casing. The water circulates in a closed circuit and by passing ambient air over the finned surfaces the heat is rejected to the air (compare Fig. 8).

With air-cooled heat exchangers, it is not possible to cool the medium to below the ambient dry bulb temperature. In this case the approach temperature between the medium outlet temperature and the inlet temperature of the dry air depends mainly on the size and capacity of the dry cooler - typical values of approach temperatures are 5 to 9 K (SWKI, 2003).

Dry coolers are often used for cooling refrigerants, oils or water/glycol mixtures. Compared to wet cooling towers they have lower operational and maintenance cost and because the cooling water does not come in direct contact to the air they have no hygienic problems or legionnella risks. Further advantages are little noise, easy installation and a low profile.

The main disadvantages compared to wet cooling towers are higher heat rejection temperatures, much higher investment costs, parasitic energy consumption for the fan and space requirement.



1. cooling circuit
2. inlet flow
3. cooling element (heat exchanger)
4. return flow
5. heat source
6. circulating pump
7. cooling air
8. fan drive
9. fan

Fig. 8: Sketch of a dry cooler (SWKI, 2005)

4.1.2 Wet Cooling Towers

Cooling towers are characterized by the primary cooling being evaporation of water. The lowest achievable temperature is the wet bulb temperature for the ambient air. The wet bulb temperature is depending on both the dry bulb temperature and the moisture content of the air. At rising dry bulb temperature and constant moisture content the wet bulb temperature will rise. As a consequence the capacity of a cooling tower will go down as the ambient temperature raise doing the day.

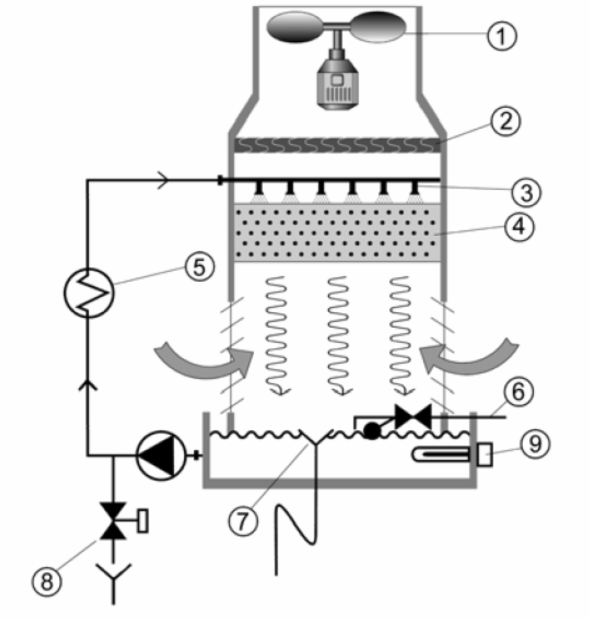
Cooling towers can be

- open type, having direct contact between cooling water and the air stream in the tower
- closed type, not having direct contact between cooling water and the air stream in the tower

The open wet cooling tower (open loop evaporative cooling tower) consists of a shell containing packing/fill material with a large surface area. Nozzles arranged above the packing, spray and distribute the cooling water onto the packing. The water trickles through the packing into a basin from which it is pumped back to the chiller. The water is cooled by air and drawn or blown through the packing by means of a fan. The air flow, which is either in counter or cross flow to the water flow, causes some of the water to evaporate, thus latent heat, is exchanged from the water to the air.

The evaporated water is continuously replenished by make-up water. However, evaporation also increases the concentration of the dissolved solids in the cooling water and blow down of the cooling water is therefore necessary. In wet cooling towers the wet-bulb temperature determines the degree of cooling and thus cooling below the ambient dry bulb temperature is possible. The characteristic approach temperature, which is the difference between the water outlet temperature and the ambient wet-bulb temperature, of open wet cooling towers lies between 4 to 8 K (SWKI 2005).

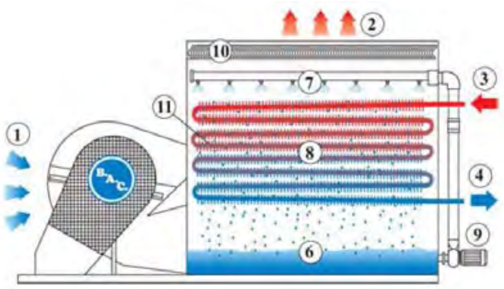
Compared to dry coolers wet cooling towers are able to cool the cooling water to a lower temperature level, require less space and have much lower investment costs. The main disadvantages of wet cooling towers are hygienic problems, water consumption and high maintenance effort.



1. fan with drive
2. drift eliminator
3. spray nozzles
4. trickle packing
5. heat source
6. float valve and fresh water inlet
7. overflow
8. bleed off
9. frost protection heating

Fig. 9: Sketch of an open wet cooling tower (Jaeggi, http://www.guentner.ch/pdfs/Evaluation of Air-cooled Cooling Systems.pdf, 26.03.2009)

The open loop wet cooling tower has the risk of fouling the heat transfer surfaces as dirt and dust from the air incl. biological material will be rinsed out of the cooling air stream. This is eliminated in closed cycle wet cooling towers as the cooling water is cooled in pipes over which water is distributed and will evaporate. Compared to open type wet cooling towers the approach temperature is higher but still lower than the dry coolers. Due to the more complex design the investment cost is higher, but the running cost is lower. An example of a closed cycle cooling tower can be seen in Fig. 10.

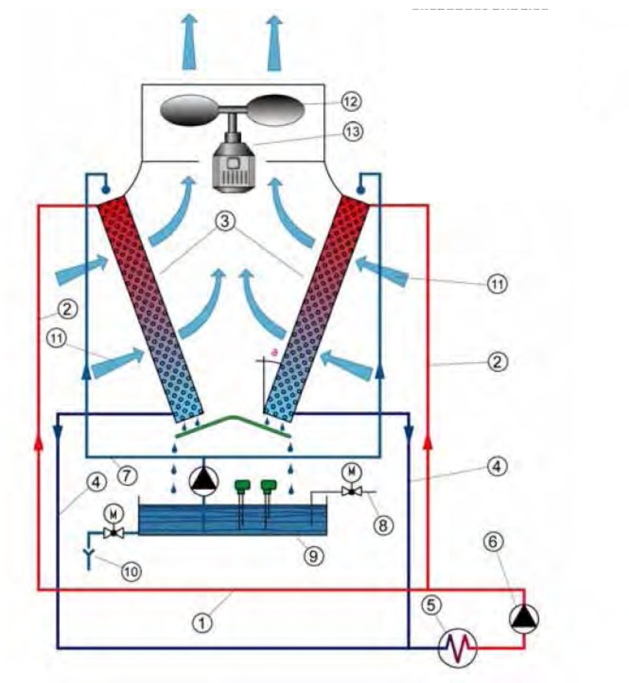


1. Air In;
2. Air Out;
3. Fluid In;
4. Fluid Out;
6. Cold Water Basin;
7. Water Distribution System;
8. Coil;
9. Spray Water Pump;
10. Eliminators;
11. Optional Extended Surface.

Fig. 10: Sketch of a closed cooling tower (BAC type VFL, www.baltimoreaircoil.be)

4.1.3 Hybrid

The hybrid dry cooler combines the two methods of dry cooling and evaporative cooling. The cooling water is circulated by a pump in a closed primary cooling circuit from the heat source to cross current air to water heat exchanger.



1. primary cooling circuit
2. inlet flow
3. finned tube heat exchanger
4. return flow
5. heat source
6. circulating pump
7. deluging water circuit
8. make up water inlet
9. water collector
10. bleed off
11. cooling air
12. fan
13. fan drive

Fig. 11: Sketch of a hybrid dry cooling system (Jaeggi, http://www.guentner.ch/pdfs/Evaluation of Aircooled Cooling Systems.pdf, 26.03.2009)

In cool weather conditions, this process cools down the cooling water sufficiently and the hybrid cooler operates like a dry cooler. At high air temperatures the hybrid cooler uses the principle of evaporative cooling in order to achieve lower cooling temperatures. Therefore, a pump circulates water from a basin to the cooling element where the water flows back via the finned surface of the air to water heat exchanger. The air flowing past the heat exchanger causes the water to evaporate on the fin surface, and takes the heat from the fins.

Comparing a hybrid dry cooler to common dry cooler we see that it has the advantage of using evaporative cooling at hot weather conditions and therefore cools down the cooling water below the dry bulb temperature, it has a higher capacity and lower energy consumption. On the other hand the hybrid dry cooler has higher investment costs, maintenance effort and water consumption. Furthermore, hygienic measures have to be taken as for the wet cooling tower.

4.1.4 Boreholes

Boreholes are vertical ground coupled heat exchangers. They are of special interest in cases of a little available surface area. The vertical boreholes use the relative low and constant temperature in the soil. Temperature fluctuations can be measured only down to a depth of 15 m. Below 15 m depth the temperature is constant with 10 °C over the season. This temperature increases every 30 m depth with 1 °C. For four kinds of soils the guideline VDI 4650 (VDI 2009) provides the following specific heat transfer values: 30 W/m in dry soil, 55 W/m in schist and similar stone, 80 W/m in solid rock and 100 W/m in a soil with significant ground water flow. This classification of soil shows that the heat transfer depends strongly on the soil condition. The distance between two or more boreholes should be a minimum of 5 m.

Boreholes are carried out as single u-type, double u-type or coaxial tube heat exchangers made of a plastic material (high density polyethylene tubes HDPE). They are installed in vertical holes drilled into the ground. In order to improve the thermal contact to the ground and seal the borehole, a special sealing material (cement or bentonite) with high thermal conductivity is used. Boreholes are normally drilled down to a depth of about 100 m, but this depends on the geology of the ground and the intended use, and in most cases they are used as a low temperature heat source for heat pump applications. The following explanations refer to heat pumps (heat source): In this case, the heat extraction capability is the key factor for the dimensioning of the borehole system: if the system is too small for the intended heat extraction power the temperature of the borehole will decrease in time and the ground may even freeze (for chillers utilizing water as refrigerant the risk of freezing limits the lowest soil temperature to 4-5℃). The main par ameter for the evaluation of the borehole performance is the heat conductivity of the ground, which depends on the geology and ground water flows. This parameter can be experimentally determined through a 'Thermal Response Test'. In this experiment a constant power is injected into a finished borehole and the mean fluid temperature is measured. The measurement has to be carried out over a period of more than 50 hours in order to avoid transient and capacitive effects in the measurement.

An important figure is the specific heat capacity of the borehole. However, this is not the only important characteristic: the number of operation hours and thus the total amount of heat extracted annually is a decisive factor for the long term performance of a borehole system. While the specific heat capacity is a factor that is important during the actual operation of the borehole, the total extracted heat is the factor that determines the long term reliability and thus if the borehole can be considered as a renewable resource. Thus, the real specific power has to be calculated for a long period of time (15 to 30 years) taking into account the hours of full load operation per year. As a result the specific power that can be expected without depletion of the source decreases with the number of full load hours per year.

This is also valid for the use of a borehole system as heat sink: a specific power may be defined but the total amount of heat absorbed by the ground has to be considered. A borehole system can only work as a renewable heat sink if the heat absorbed can be dispersed sufficiently within the ground in order to not significantly affect the undisturbed temperature distribution of the ground. This factor determines the long-term suitability of the system as a heat sink. As a conclusion, monovalent systems (either as heat source or heat sink) can only be operated a limited number of hours a year.

Thus, bivalent systems which use the ground as heat source as well as heat sink are convenient. In this case the heat extracted from the ground and the heat rejected into the ground may be counterbalanced. The dimensioning of such a system with a reversible thermally driven chiller which can also be used as a heat pump is not straight forward. Since the ratio of heat to be rejected in the chilling mode to low temperature heat extracted in the heat pumping mode is typically around 2.25 to 2.5, the dimension of the borehole system can only be calculated with suitable simulation tools which take into account the heat extraction and heat rejection powers and temperatures as well as the expected operation hours in each mode over the whole expected lifetime of the system in order to avoid long-term disturbance of the ground.

It is recommended that a qualified advice and expertise should be obtained before the drilling of a borehole. Information about imposed conditions such as the expected kind of soil and the heat transfer should be obtained. The drilling has to be applied by a concessionary company. The specific cost of a borehole down to 150 m in Austria is in the range of 55 to 60 €/m. More information can be obtained, for example, in German in (Ochsner 2009).

4.1.5 Horizontal Ground Heat Exchangers

The horizontal ground heat exchanger is designed to the use of the cooling storage capacity of the ground. Heat exchanger made of polymer tubes (PER for example) is put at 0.5 to 2 m depth into the ground in order to reject heat to the ground. It is connected to the heat rejection circuit of the thermally driven chiller.

Fig. 12: Example of horizontal ground probe heat rejection system in INES office, Chambery (source: INES RDI)

This heat rejection technology is interesting for the following reasons: no need of any wet or dry cooling tower leading to far less electricity consumption, low heat rejection temperature (less than 30°C) if the geothermal heat exchanger is well designed, ease of implementation of the horizontal probe network for new buildings during the civil works phase, and the possibility to use these heat exchangers during winter in heat pump mode.

The heat transfer in horizontal ground heat exchangers for heat rejection systems depends above all on the kind of soil. In all cases of applications the soil should be natural and not a man made earth deposit. Regarding the guideline VDI 4640 Part 2 (VDI 2001) the following specific heat extraction values can be expected: 10 W/m^2 soil surface for dry solid soil, 20 to 30 W/m^2 in moist solid soil, and up to 40 W/m^2 for a water saturated soil. For the realization of horizontal ground heat exchanger it is recommended to use a 0.75“ or 1.0” PE-tube for a maximum pressure of 10 bar. The PE-tubes of the horizontal ground heat exchanger should be piped in a depth of 0.5 to 2 m with a horizontal distance of 50 cm in moist soil and with about 80 cm in dry soil. For having the desired surface it is normally necessary to pipe parallel loops with not more than a length of 100 m.

Thanks to a horizontal ground heat exchanger, it is possible to use very little electricity consumption to run the pump connecting the chiller to the heat rejection loop (only 25 $W/kW_{heat\ rejection}$ for a 4.5 kW chiller). Economically speaking, this solution is more expensive than a traditional wet cooling tower system (more than double cost) due to the significant length of polymer pipes but on a 20 years global cost calculation (avoidance of water treatment and water consumption), this investment is more interesting, especially for new buildings (civil works to burden the pipes more or less free) and for countries where legionella protection legislation consequences make wet cooling tower management expensive.

Table 5 shows example data of an installed horizontal ground heat exchanger

Table 5: Example data of a horizontal ground heat exchanger, source: Bengt Hedestam, ECONICsystems, Gars am Kamp, Austria

heat rejected	5 kW
horizontal ground area	540 m^2
PE-tube length	270 m
tube diameter	32 mm
tube wall thickness	3 mm
piping depth	2 m in a dry rocky ground
heat transfer area	27.14 m^2
specific heat transfer density	185 W/m^2 tube surface
specific heat transfer	18.5 W/m of tube
specific cost	12 to 15 €/ m PE-tube
water temperature of the heat rejection system	22/27 °C in the morning and 28/33 °C in the late afternoon
heat rejection temperature	starts every morning around 22/27°C

4.2 Examples of Heat Rejection Components

The following table shows key data of the wet, dry and hybrid heat rejection systems that are employed in the monitoring installations of subtask A.

Table 6: Key data of heat rejection systems installed in Task 38 monitoring installations

		Number of systems	Nominal thermal capacity	Nominal flowrate	Nominal Flow/Return Temperature	Nominal Outdoor Conditions (Dry bulb / Wet bulb)	Air volume flow	Nominal electricity consumption (max)	Heat Exchanger area	U-Value
			kW	kg/hr	°C/°C	°C/°C	m³/hr	W	m²	W/(m² K)
Wet Cooling Towers	AXIMA EWK 036 / 06	3	35	5000	32 / 26	/ 21		330		
Dry Air Coolers	CIAT AIRIAL 7023 HI 680	1	25	2905	35 / 27	25 /		600	133	
	Güntner GFH 080.2B/1-S(D)-F4/8P	1	28,5	5422	45 / 40	32 /	10500	340	197,6	25,63
	Güntner GFH 067B/2-S(W)-F6/12P	1	24	3351	41 / 34	37,8 / 32	12900	800	270,6	28,6
	Güntner GFH 052A/2-L(D)-F6/12P	1	27	2930	43 / 35	30 /	9620	570	168,7	
Hybrid Coolers	SORTECH RCS 08	3	21	3685	31.8 / 27	24.5 / ??	13000	650	221,4	32,65
	SORTECH RCS 15	0	42	7000				1200		
	Baltimore Air Coil 1 VXI 9-3X	1	67	7360	35 / 29.5	32 / 22	9000	2200	19	

In addition, one system uses a dry air cooler integrated in the chiller. Two installations use ground coupled heat rejection systems – one with a horizontal ground heat exchanger, one using boreholes.

The table shows that the characteristic numbers of the different heat rejection units vary significantly. One of the important figures is the electricity consumption of the unit. It influences significantly the operating costs of a system and also the primary energy efficiency. Fig. 13 shows the relative electricity consumption (i.e. the electricity consumption per kW rejected heat) as a function of the nominal thermal capacity of wet, dry and hybrid heat rejection units used systems in the Task 38 (Subtask A) monitoring program. In addition, a few other wet cooling towers were added where data on the electricity consumption was available.

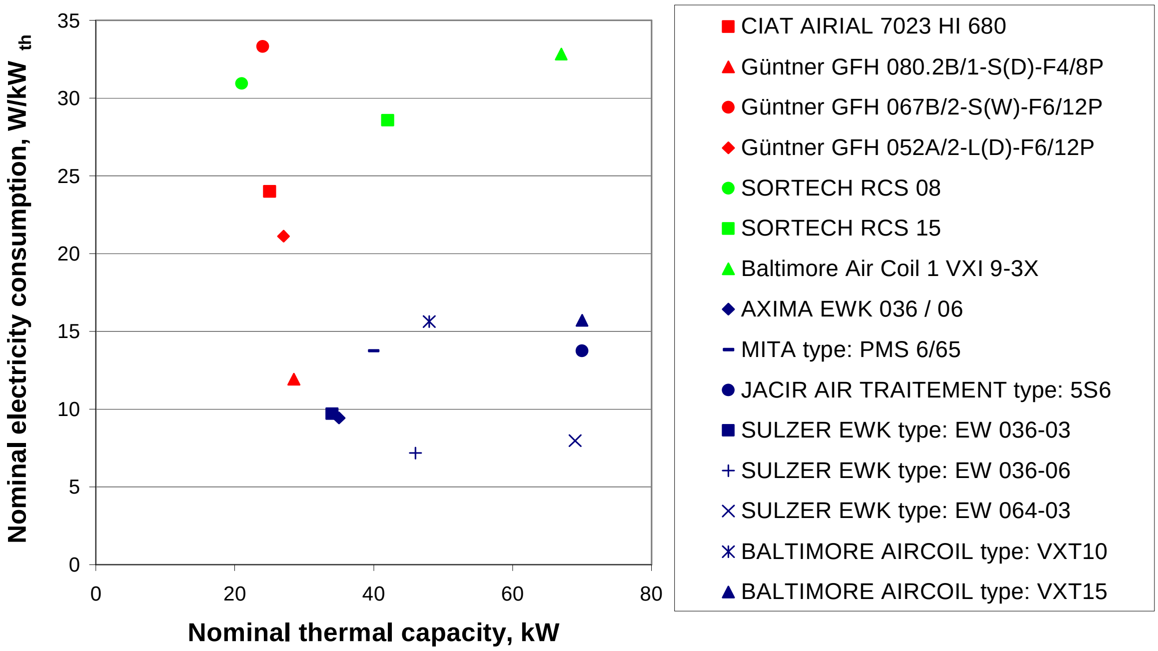


Fig. 13: Nominal electricity consumption of heat rejection systems installed in Task 38 monitoring installations and a few other cooling towers up to a heat rejection capacity of 70 kW_{th} in Watts per kW rejected heat

The figure shows that data points scatter significantly. For approximately the same nominal thermal capacity (25-35 kW) the electricity consumption varies from roughly 9 W/kW_{th} to 34 W/kW_{th}. The wet cooling towers typically have relatively low electricity consumption. However, the figure shown includes only the electricity consumption of the fan(s) and not for water treatment. In addition, water consumption has to be taken into account. The red dots show the dry heat rejection units and the green ones the hybrid systems.

This is not meant to be an exhaustive analysis of the topic. It only shows that it is important to pay attention to the choice of heat rejection unit for a specific system in order to reduce the primary energy consumption of the system.

4.3 Electricity Consumption

By Lars Reinholdt, Danish Technological Institute

If not carefully designed, the electrical consumption can grow to a level making the competition to traditional electrical driven vapor compression systems very hard. Of the total electricity consumption of the system the heat rejection system often can count for more than 50% making that a major focus point in the design phase. Three points have to be addressed

1. Flow rate and pressure drop in the water loop
2. Fans in the heat rejecting device
3. Control strategy

The electrical power for a pump can be estimated by the simple equation

$P = \eta_p * V * \Delta p_p$

η_p being the electrical efficiency of the pump, V being the volume flow of water and Δp_p being the pressure difference across the pump.

Both the volume flow and the pressure drop have a direct impact. As a rule of thumb the power consumption increase in a power of 3 of the flow rate, as the frictional pressure drop increases in a quadratic of the flow velocity.

The control strategy also has an impact on the electrical power consumption: Traditionally thermally driven chillers are controlled by controlling the driving temperature level keeping the cooling water temperature constant: If the cooling capacity is to be reduced the driving temperature is decreased.

At fixed driving and cold temperature the most thermally driven chillers have a strong dependence between the cooling capacity and the heat rejection temperature whereas the thermal COP is only affected a little. This opens for optimization of the electrical power consumption for the heat rejection in part load: By decreasing the fan speed (power savings) the heat rejection temperature will increase. As the temperature difference between air and cooling water goes up and the cooling capacity drops (constant COP) the heat rejection device will be oversized which normally results in higher efficiency and makes further power savings possible. Running in part load will normally also make it possible to decrease the cooling water flow rate resulting in reduction of the power for pumps. Further optimization can be done by letting the driving temperature increase until the limit for the respective chiller: Increasing driving temperature will increase the cooling capacity again making further savings on the pumps (lowering the flow rate) and heat rejection device. Savings up to 50% have been shown. Further this control strategy can make it possible to reach the cooling capacity at lower driving temperature by running the heat rejection fans at full (or over) speed resulting in a lower heat rejection temperature when needed (although resulting in higher power consumption).

5 Cold Stores

Some solar cooling systems using a thermally driven chiller use a cold store to be able to deliver cold at times when there is not enough solar radiation available to power the thermally driven chiller.

5.1 Cold Stores Used in Small-Scale Solar Cooling Systems

All small-scale solar cooling systems currently on the market use a water filled storage tank that stores cold at temperature between roughly 4 and 18℃. Such tanks are available from almost all manufacturers of heat storage tanks. As there are so many products available, it is not necessary to list manufacturers in this document.

5.2 Alternative Cold Storage Technologies

As an alternative to water filled storage tanks, latent heat storage can be used. The most popular option is an ice storage tank. Such tanks are used quite frequently to shift cooling loads for compression chillers to times when electricity prices are low. They are typically used for large capacity systems. However, some manufacturer would also be able to produce small tanks and some small size systems are currently under development.

In the following sections, first an overview of the different ice storage technologies that are available will be given. Then, a database of manufacturers of ice storage systems (although most of them typically manufacture large store sizes) is given. The idea behind this database is to give a starting point for companies or institutes interested in these technologies.

5.2.1 Ice Storage Technologies

By Torsten Koller and Alexander Eichhorn, ITW, University of Stuttgart, Germany

Ice storage systems are categorized according to the method of discharging the stored ice. If the return flow of the cooling medium is directly mixed with the storage fluid (water/ ice) it is a so-called “direct” ice storage system. If cooling fluid and storage mass are separated by a heat exchanger it is called “indirect” system. A “hybrid” system is a combination of the direct and the indirect system.

In the following, the different design options for ice storage systems are summarized [Gasser, Kegel, 2005] [Urbanek et al, 2006]:

- **Direct ice storage systems**
  - Ice-on-coil (external melt)
  - Sheet ice harvester / silo-ice storage systems
  - Ice slurry

- **Indirect ice storage systems**
  - Ice-on-coil (internal melt) also called ice bank systems
  - Ice-on-coil (external melt using a water circuit which is directly coupled with the discharging circuit of the ice store)
  - Encapsulated ice

- **Hybrid ice storage systems**
  - Ice-on-coil (combination of external and internal melt)

5.2.1.1 Ice-on-coil system

The major component of an ice-on-coil system is a heat exchanger (e.g. tube coils or plate heat exchangers made out of copper, stainless steel or polypropylene) which is implemented into a non-pressurized storage tank filled with water. These heat exchangers are either directly connected to the refrigerant circuit of the chiller or have their own secondary circuit which is coupled to the refrigerant circuit. In the first case the heat exchanger in the ice store is working as an evaporator of a chiller. No additional heat exchanger is needed and temperature differences are minimized.

At “indirect cooling” the cooling capacity is transferred by a secondary circuit. For example water/glycol mixtures are used as heat transfer medium. The main advantage is the reduction of the required amount of refrigerant and the reduced impact of leaks.

In ice-on-coil systems the storage water is freezing on the outer surface of the heat exchanger tubes and a constantly growing ice layer is established. The growth rate of this ice layer and the transferred cooling capacity decrease as the layer grows thicker due to the increased thermal resistance of the ice layer.

External melt ice storage systems

For external melt ice storage systems (EMISS) two different, completely separated circuits for charging and discharging are used. The ice store is charged by a heat exchanger which is implemented into the storage vessel. This heat exchanger works either directly as an evaporator or is coupled to the chiller by a secondary circuit.

During discharging mode the ice adhered to the heat exchanger melts from the outside of the ice mass to the inside. The water of the discharging circuit is returned to the top of the storage tank and removed from the bottom, and in-between pumped for example, through chilled ceilings at a lower temperature level. The water of the discharging circuit is the storage mass.

It is possible to sustain high discharging rates and constant water outlet temperatures of 1℃ to 2℃ with this type of system if a steady flow th rough the storage tank is guaranteed.

To ensure even and steady flow through the ice store the formation of ice bridges between the individual heat exchanger coils and consequentially ice block generation (complete freezing of all water in the storage tank) has to be prevented. By the use of agitators and bubblers, the flow within the ice store can be significantly enhanced. The ice melts more evenly because of the enhanced mixing.

Since ice block generation has to be prevented the maximum capacity compared to internal melt systems is significantly reduced. Fig. 13 shows a schematic diagram of the lay-out of an external melt ice storage system setup.

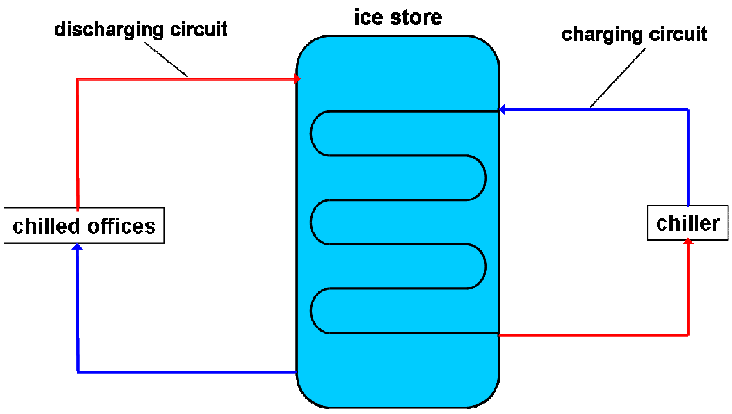


Fig. 14: Schematic diagram of an external melt ice storage system

Internal melt ice storage systems

Internal melt ice storage systems (IMISS) use the same heat exchanger for charging and discharging. During discharging the heat transfer medium inside the coils of the heat exchanger has a higher temperature than the melting temperature of the phase change material (PCM, e.g. water). The ice adhered to the heat exchanger starts to melt from the outside of the heat exchanger surface to the outer boundaries of the ice layer (Fig. 14). A growing layer of liquid water between the heat exchanger surface and the ice is generated. Hence the discharging rate decreases as the driving temperature difference decreases.

The most important advantage of internal melt systems is the possibility to realize very high storage capacities since ice block generation does not have to be prevented.

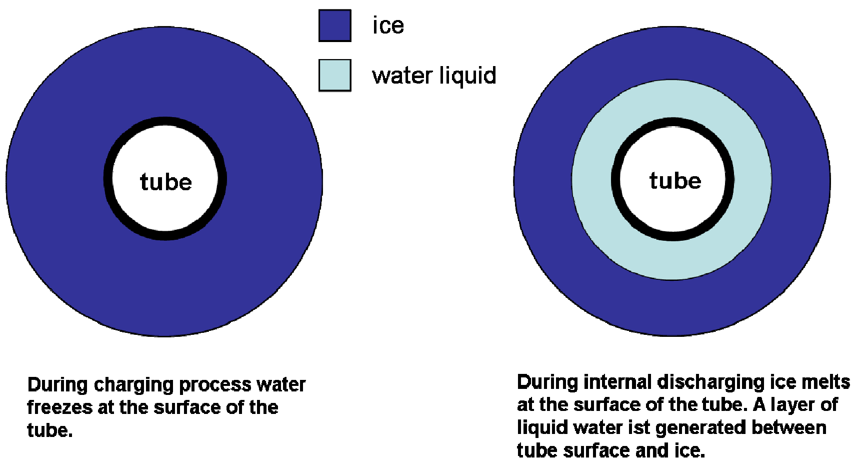


Fig. 14: Charging and discharging process for internal melt system

A disadvantage of internal melting is that simultaneous charging and discharging is not possible. A simultaneous charging and discharging could be useful in cases where the absorption chiller generates less cooling capacity than needed to meet the cooling demand. With external melt systems cooling can be maintained under these conditions. The absorption chiller then continues to charge the ice store while the cooling demand is met by discharging the ice store.

Fig. 15 shows the schematic lay-out of an internal melt ice storage system.

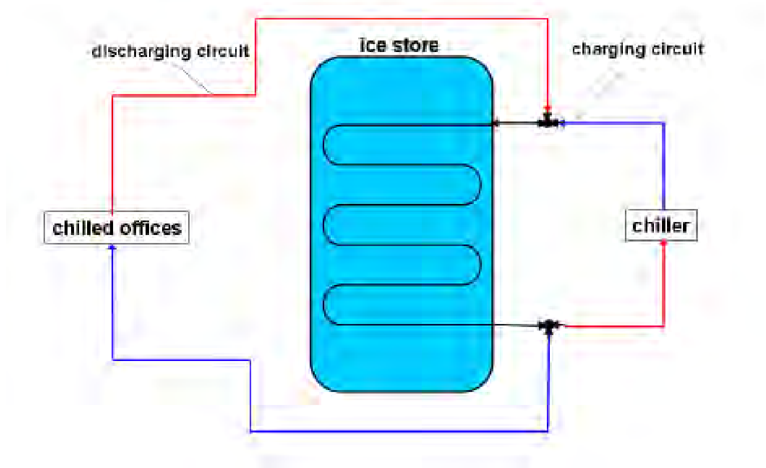


Fig. 15: Schematic lay-out of an internal melt ice storage system

Hybrid systems

If an ice store can be discharged by external and internal melt simultaneously it is called a hybrid system. The advantages of both systems are combined. The internal discharging circuit melts the ice from the outer surface of the heat exchanger. Channels of liquid water are generated between the heat exchanger surface and the ice mass. Simultaneously the external discharging circuit circulates the liquid water.

Therefore hybrid systems allow a high discharging rate at a constant temperature level. Fig. 16 shows a schematic lay-out of a hybrid system.

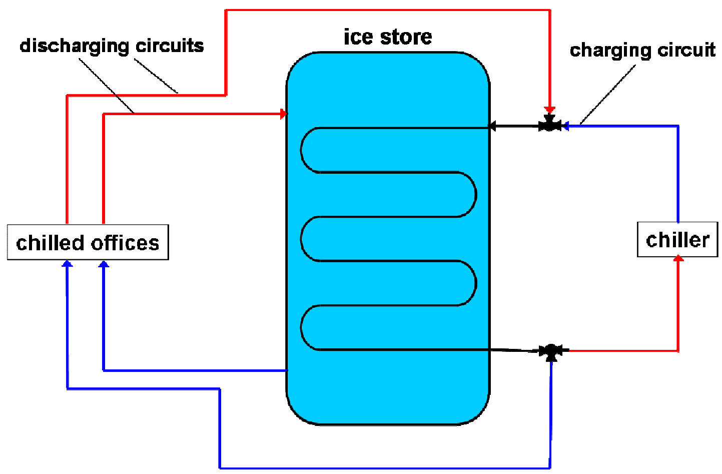


Fig. 15: Schematic lay-out of a hybrid ice store system

5.2.1.2 Encapsulated Ice

An encapsulated Ice Storage System (EISS) consists of a steel containment filled up to 50-70% with special plastic containers [Siemens Schweiz AG]. The containers have various shapes and usually consist of polyethylene. The containers are filled with water or brine. They are not completely filled to account for the volume expansion during the freezing process. The containers are in direct contact with the heat transfer medium flowing through the tank. For non pressurized systems a barrier has to be installed to keep the plastic containers from buoying upwards once they are frozen. Fig. 16 shows a schematic diagram of an encapsulated ice storage system.

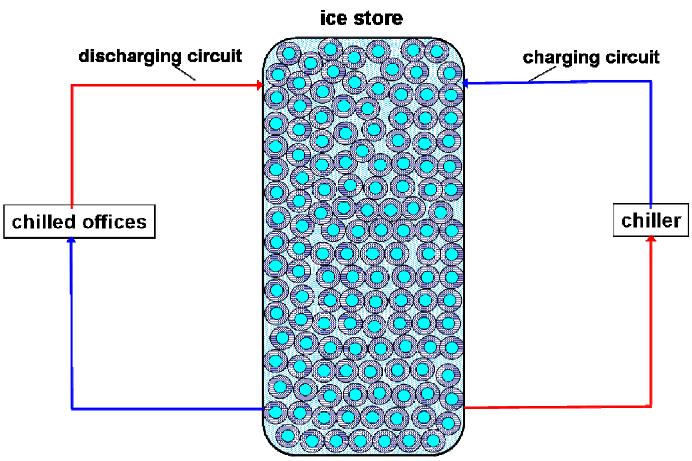


Fig. 16: Schematic diagram of an encapsulated ice storage system

During the charging process the plastic containers are floating in the heat transfer medium (e.g. water/glycol mixture) which has a temperature between −3 °C and −6 °C. The ice grows from the inside of the plastic container walls to their centre.

During the discharging process the ice inside the plastic containers starts to melt, at first at the container walls. Once the ice is completely detached from the container wall the heat transfer rate decreases since there is no direct contact between ice and plastic container wall anymore (Fig. 17). Because of this effect the discharging rate decreases and the discharging temperature increases.

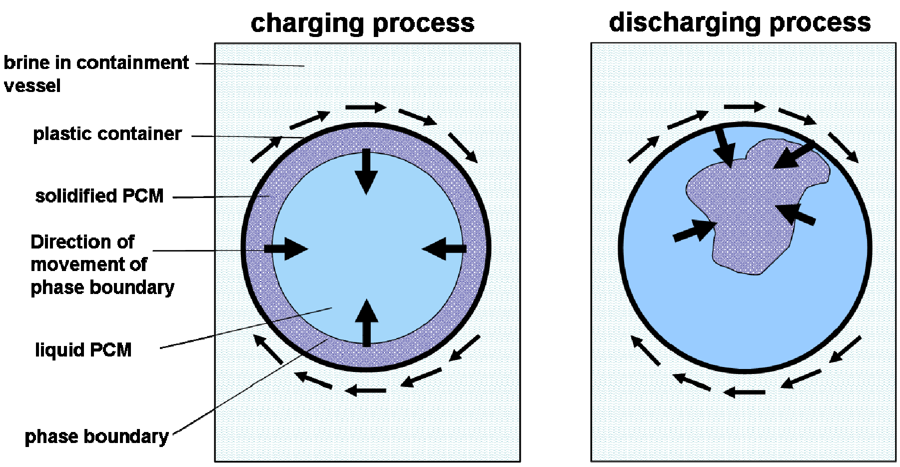


Fig. 17: Solidification and melting processes in a plastic container during charging and discharging of an encapsulated ice system

5.2.1.3 Sheet Ice Harvester System

A sheet ice harvester system (SIHS), shown in Fig. 18, generates the ice outside of the storage tank. Water extracted from the bottom of the storage tank is sprayed over evaporator plates. An additional recirculation circuit is needed. A thin ice layer is formed at the evaporator plates. Once a thickness between 6 mm and 10 mm is reached the ice is removed by a mechanical process if the evaporator consists of tubes or by a periodical injection of hot gas into the evaporator plates. The ice falls into the storage tank below the ice generator. Water flows through this tank and the system is discharged in the same way as an external melt ice-on-coil system. Likewise, low and constant discharging temperatures can be achieved. Water is sprayed over the crushed ice to guarantee a sufficient wetting of the ice. Due to this and the large surface of the small ice particles high discharging rates are achievable.

Compared to ice-on-coil systems the necessary heat transfer area needed is small, since the ice is directly generated at the evaporator of the chiller at a very low temperature level. The disadvantage of this system is the high complexity of controlling the SIHS due to the third recirculation circuit and the complicated process of evaluating the charging condition. Since hot gases are injected into the ice generation process, the overall efficiency is lowered.

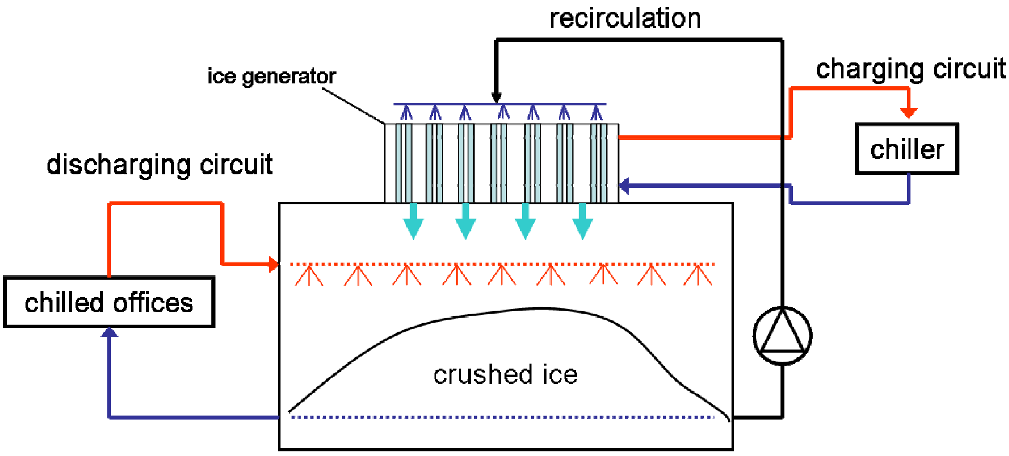


Fig. 18: Schematic diagram of a sheet ice harvester system

5.2.1.4 Ice Slurry Systems

Ice slurry is a non-Newtonian fluid (Bingham fluid) which consists of ice crystals in an aqueous solution. The main advantage of ice slurry systems is the high specific energy content due to the latent stored heat. Ice slurries are pumpable up to an ice content of about 40%. These systems can be used within a wide temperature range. The ice content can be influenced by the use of additives. The heat transfer within the machinery is very good since the heat transfer between the fluid and the ice particles is very high and thereby ensures a fast phase change. The working principle of an ice slurry system is shown in Fig. 19.

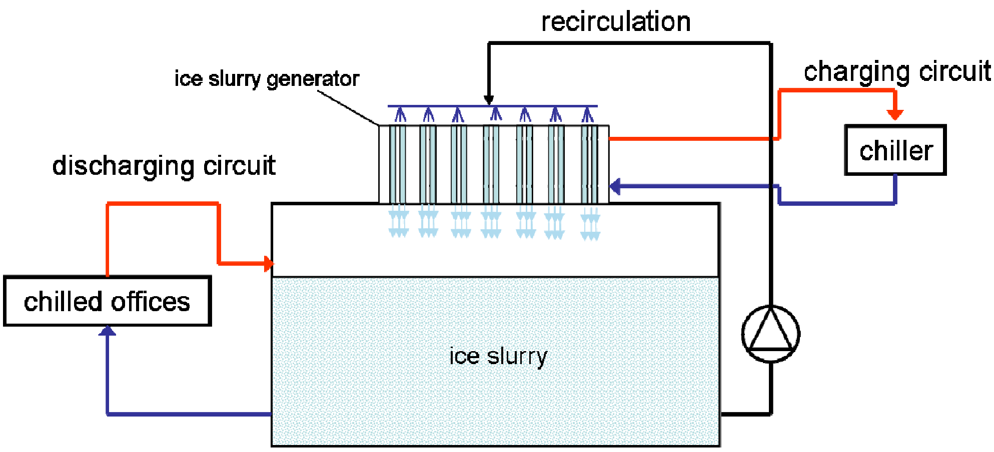


Fig. 19: Schematic diagram of an ice slurry system

5.2.1.5 Summary of all discussed ice storage systems

The following Table 7 summarizes the discussed ice storage systems.

Table 7: Summary of different ice storage systems [Hilliweg, Hofmann, 2003]

physical storage principle	storage system	storage material	material carrying heat during charging	material carrying heat during discharging	specific storage capacity
sensible	chilled water	water	water	water	7 kWh/m^3 at $\Delta T = 6$ °C
latent	ice-on-coil (external melt)	ice	coolant or brine	water or brine	40-44 kWh/m^3
latent	ice-on-coil (internal melt)	ice, paraffin or eutectic mixture	brine	brine	53 kWh/m^3
latent	hybrid system	ice	brine	water or brine	53 kWh/m^3
latent	encapsulated ice	ice or eutectic mixture	brine	brine	53 kWh/m^3
latent	sheet ice harvester	crushed ice	coolant	water	–
latent	ice slurry	ice or water with additives	ice or water with additives	ice or water with additives	–

5.2.2 Addresses of Manufacturers

This is a non exhaustive list of ice storage manufacturers.

Baltimore Aircoil Company, USA P.O. Box 7322, Baltimore, MD 21227 7600 Dorsey Run Road, Jessup, MD 20794 Phone: +1 (410) 799-6200 Fax: +1 (410) 799-6416 http://www.baltimoreaircoil.com
BUCO WÄRMEAUSTAUSCHER INTERNATIONAL GMBH, Germany Sandstrasse 31 D-21502 Geesthacht Phone: +49 (0)41528082-0 +49 (0)4152-8082-43http://www.buco-international.com
CALMAC, USA Manufacturing Corporation Headquarters 3-00 Banta Place Fair Lawn, NJ 07410 Phone: +1 (201) 797-1511 Fax: +1 (201) 797-1522 http://www.calmac.com
CLIMATIC GfKK mbH, Germany Gradestraße 113-119 D-12347 Berlin (Britz) Phone +49 (0) 30 / 60 09 94 - 0 Fax: +49 (0) 30 / 60 09 94 – 99 http://www.gfkk.de
CRISTOPIA Energy Systems, France 78, chemin du Moulin de la Clue Quartier Cayrègues F- 06140 VENCE (France) Phone : +33 (0) 4 93 58 40 00 Fax: +33 (0) 4 93 24 29 38 info@cristopia.com http://www.cristopia.fr/
FAFCO S.A., Switzerland Johann Renfer-Strasse 4-6 CH-2504 Biel/Bienne Phone: +41 (0)32 / 342 33 52 Fax: +41 (0)32 / 342 39 47 Email: info@fafco.ch http://www.fafco.ch
GP Klimatechnik GmbH, Germany Hohlstr. 25 D-76275 Ettlingen Phone: +49-(0)7243-526868 Fax: +49-(0)7243-526869 Email: vertrieb@gpklimatechnik.de http://www.gpklimatechnik.de
Integral Energietechnik GmbH, Germany LISE-MEITNER-STR. 2 D-24941 FLENSBURG Phone: +49 (0)461-9993-33
Paul Mueller Company, USA 1600 W. Phelps

Springfield, Missouri 65801-0828 Phone: +1 (417) 575-9000 · 1-800-MUELLER Fax: +1 (417) 575-9669 http://www.muel.com/
Raffel - Searle GmbH Kältetechnik, Germany Auf dem Kummgraben 17 D-53343 Wachtberg Phone: +49 (0)2 28 95 29 8-0 Fax: +49 (0)2 28 95 29 8-10 Email: info@raffel-kaeltetechnik.de http://www.raffel-kaeltetechnik.de
Rössler Kühltechnik, Germany Uracher Str. 6 D-71229 Leonberg Phone: +49 (0)7152/73305 Fax: +49 (0)7152/73864
SOREMA, France Z.I. Sud Ouest - Bd du Cormier BP 455 F - 49304 Cholet Cedex Phone. +33 (0)2 41 62 30 29 - Fax : +33 (0)2 41 62 81 25 http://www.sorema.com/web/froid-industriel/
Sunwell Technologies Inc., Canada Phone :+1.905.856.0400 Fax :+1.905.856.1935 Email : inquiries@sunwell.com http://www.sunwell.com/
Tankki Oy, Finland Oikotie 2 FIN-63700 ÄHTÄRI Phone: +358-(0)6-510 1111 Fax: +358-(0)6-510 1200 Email: tankki@tankki.fi http://www.tankki.fi/
TH. WITT Kältemaschinenfabrik GmbH, Germany Lukasstr. 32 D-52070 Aachen Phone: +49 (0)2 41 1 82 08-0 Fax: +49 (0)2 41 1 82 08-190 Email: info@th-witt.com http://www.th-witt.com
VRITHERM ®, Germany Finkenweg 1 D-70771 Leinfelden-Unteraichen Phone: +49 (0)7 11 / 754 50 18 19 Fax: +49 (0)7 11 / 754 50 10 Email: info@vritherm-energie.de http://de.vritherm-energie

6 Bibliography

Gasser, B.; Kegel, B. (2005): Gebäudetechnik: Faktor 10; Bau + Architektur 4/05; p. 34-35.

Grandegger, K. (2005): Einsatz von Eisspeichern in der Gebäudetechnik; FAFCO Deutschland; Leinfelden-Echterdingen; 2005

Hilliweg, A; Hofmann, P. (2003): Die Größe von Kälteanlagen minimieren; TGA Fachplaner 6-2003; p.29-35.

Juran, H. (2000).: Eis im Rohr; Chemie Technik Nr. 5; Jahrgang 29, 2000; S.92-93

Ochsner, K. (2009): Wärmepumpen in der Heizungstechnik, Praxishandbuch für Installateure und Planer, 5. überarbeitet Auflage, C.F. Müller.

Siemens Schweiz AG: Kältetechnik; Siemens Building Technologies; p.76-85.

Streicher, W. (2008): Solar Thermal Heating Systems, lecture book, Institute of Thermal Engineering, Graz University of Technology

Suter, J.-M., Letz, T., Weiss, W., Inäbnit, J. (ed.), 2000: IEA SHC – Task 26, Solar Combisystems in Austria, Denmark, Finland, France, Germany, the Netherlands and the USA, Overview 2000, Büro n+1, Bern, Switzerland, www.iea-shc.org

SWKI (2005): Richtlinie 2003-3 „Rückkühlung“, Schweizer Verein für Wärme und Klimaingenieure

Urbaneck, T.; Uhlig, U.; Platzer, B.; Schirmer, U.; Göschel ,T.; Zimmermann, D. (2006): Machbarkeitsuntersuchung zur Stärkung der Kraft-Wärme-Kopplung durch den Einsatz von Kältespeichern in großen Versorgungssystemen; Stadtwerke Chemnitz AG und Technische Universität Chemnitz Technische Thermodynamik; ISBN 3-00-015770-0.

VDI (2001): VDI 4640 – part 2:2001: Thermal use of the underground, Verein Deutscher Ingenieure, 2001.

VDI (2009): VDI 4650 – 1: 2009: Calculation of heat pumps - Simplified method for the calculation of the seasonal performance factor of heat pumps - Electric heat pumps for space heating and domestic hot water

Task 38
Solar Air-Conditioning
and Refrigeration

D-A2:

Collection of selected systems schemes "Generic Systems"

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: November 2009

By Michael Becker[1], Martin Helm[1] and Christian Schweigler[1]

[1]Institution	**ZAE Bayern, Abtl.1: Technik für Energiesysteme und Erneuerbare Energien**
Address	**Walther-Meissner-Straße 6, D-85748 Garching**
Phone	**+49-89/329442-19**
Fax	**+49-89/329442-12**
e-mail	**schweigler@muc.zae-bayern.de**

Contents

1. Scope

For wide-spread application of solar cooling, compact systems shall be installed by professionals from the heating and plumbing sector without requiring a detailed planning procedure on a case-by-case basis. Thus along with the required equipment – i.e. thermally driven chiller, solar thermal system, and heat rejection device - well-proven system configurations have to be available for selecting an appropriate system concept with regard to the given specific requirements.

Work in subtask A, work package A2, focussed on providing “generic system schemes” representing standardized solar cooling systems and facilitating comparison of system concepts suggested by manufacturers and professionals in the field. Major objective is a favourable system operation in terms of optimised performance and availability

For this purpose a screening of the system technology applied in current solar cooling installations has been carried out. Apart from information furnished by component manufacturers and system providers, experience from pilot installations and recent academic work has been incorporated.

The survey is focussed on systems with the following characteristics:

- Pre-engineered systems, i.e. no case-specific planning is required
- Chilled water capacity < 20 kW
- Year-round operation for heating & cooling

This documentation of the state-of-the-art of small scale solar cooling systems is addressed to professionals, i.e. plant manufacturers, planners, and installers. It contributes to the background and common understanding of experts in the field and provides basic information required for implementation of the technology.

The generic solar cooling systems are composed of typical components and represent distinct system topologies applicable for various operating conditions and system boundaries. The given information is not product-specific or referring to a certain brand.

The main aspect is the hydraulic structure of the systems; in addition applicational aspects of the system components and control issues are discussed.

2. Basic system topology

Starting point for the definition of the system topology is the solar thermal system (Figure 1): The solar heat collected by the collector is transferred via a heat exchanger from the primary solar loop to the secondary loop. There, a heat store serves for balancing heat generation and heat consumption by the load. In times of insufficient solar gain a backup boiler is operated in order to provide additional heat input. The solar heating system may serve for both, space heating and tap water heating.

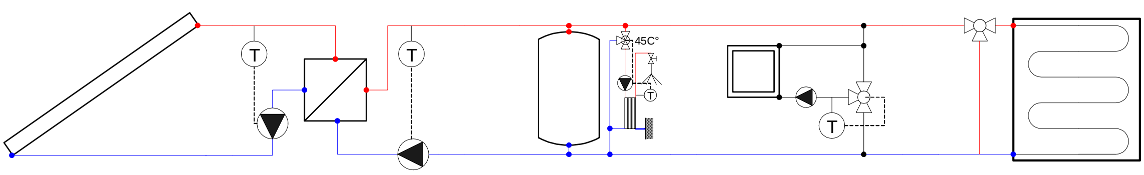


Figure 1: Solar thermal system, starting point for the definition of the solar cooling system.

In solar cooling systems a thermally driven sorption chiller is applied for the provision of useful cooling. In this case the heat generated by the solar thermal system serves as driving force for the thermally driven chiller. Taking into account the supply of useful cooling and the transfer of the chiller's reject heat to the ambient, a system with three sub-systems is formed, as shown in Figure 2:

- The solar thermal system provides heat to the desorber G of the chiller.
- Reject heat of condenser (C) and absorber or adsorber (A) are released via cooling water loop and cooling tower.
- Useful cooling is provided by the evaporator (E) and supplied to the consumer via the chilled water loop.

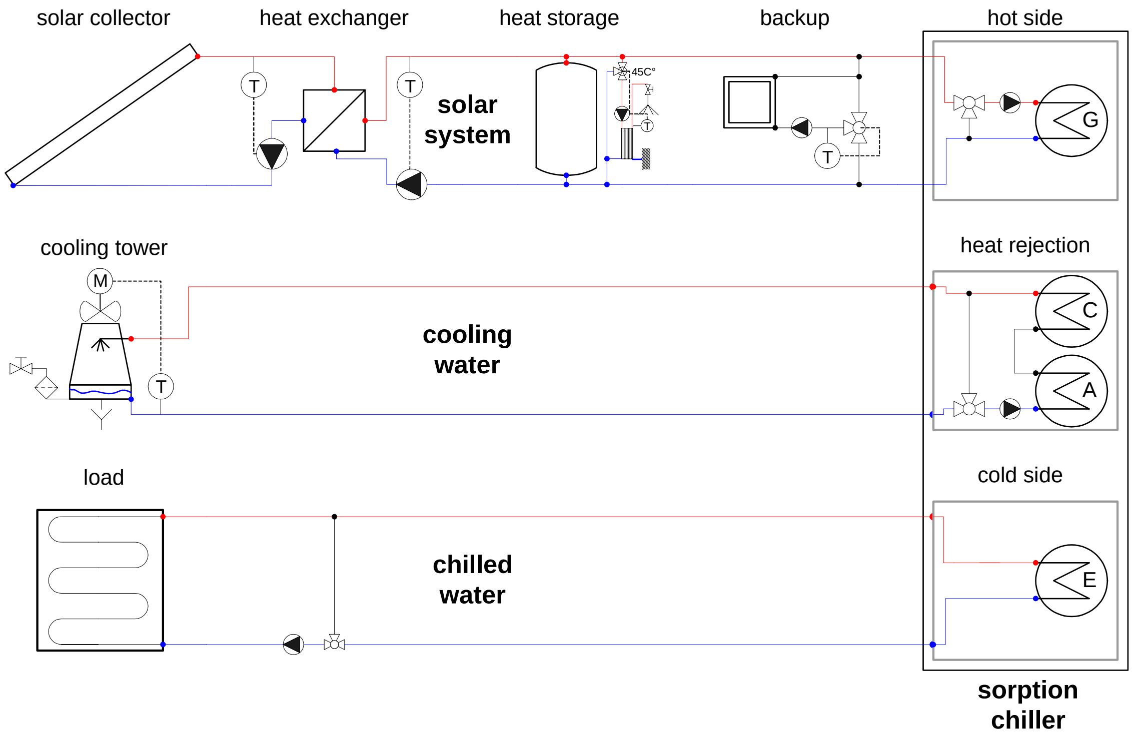


Figure 2: Basic solar cooling system: Thermally driven chiller coupled to solar thermal system, cooling water loop, and chilled water loop.

3. Composition of generic systems

In order to cope with different ambient conditions, specific demand characteristics of heating and cooling, and the room-side appliances for heating and cooling, in real applications a variety of technical options is available for all three sub-systems.

The different options are presented on basis of a standardized system topology according to the basic structure of the "generic system" discussed above as shown in Figure 3. For each sub-system a structural template has been defined containing placeholders for the integration of additional components.

In detail, the following options are to be specified in the three sub-systems.

Solar sub-system:

- Solar collector

- Heat exchanger
- Heat storage
- Backup-heater

Heat rejection sub-system:

- Main cooler
- Auxiliary cooler
- Heat exchangers for separation of primary and secondary cooling loop

Chilled water sub-system:

- Load: Room-side cooling appliance
- Distribution of chilled water to the cooling appliances
- Cold (chilled water) storage
- Backup chiller

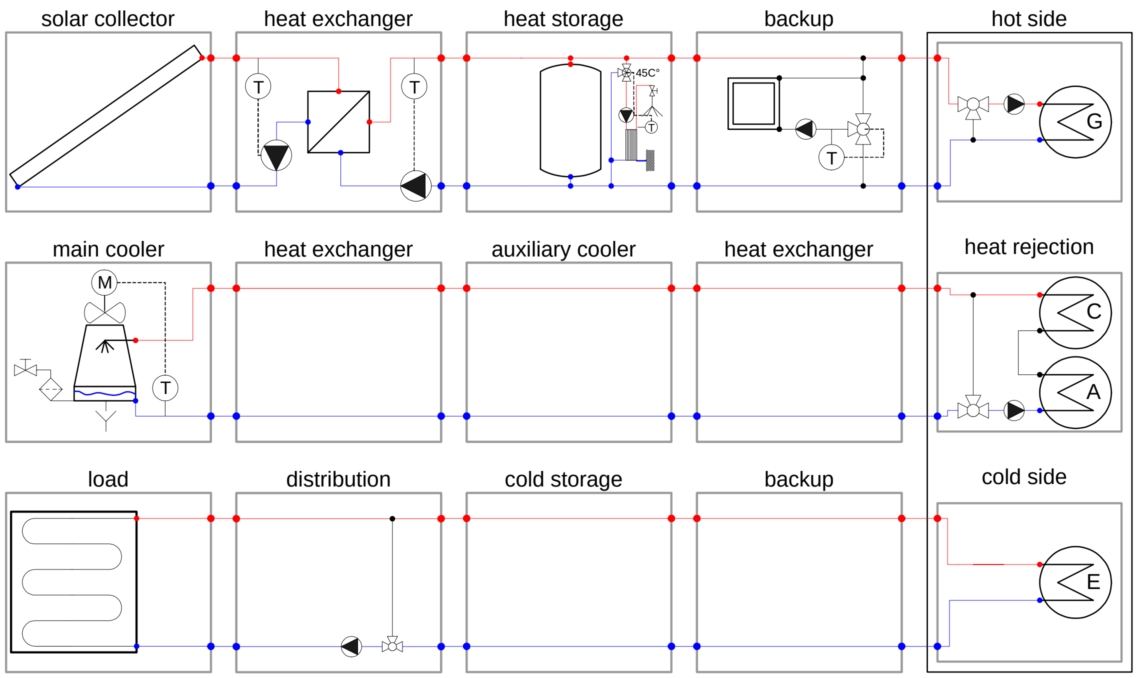


Figure 3: Generic system: standardized system topology with placeholders for the integration of optional system components.

In the following sections, typical constellations and alternative components for the three sub-cycles are presented. For each placeholder all relevant options together with a characterisation of technical and operational aspects are given.

3.1 Solar thermal sub-system

All types of **solar collectors** – i.e. flat plate collectors, vacuum tube collectors and concentrating collectors – are applicable. Vacuum tubes and concentrating collectors often exhibit higher collector efficiencies at elevated temperatures of the solar loop. Thus higher availability of the system is accomplished during times of limited solar irradiation.

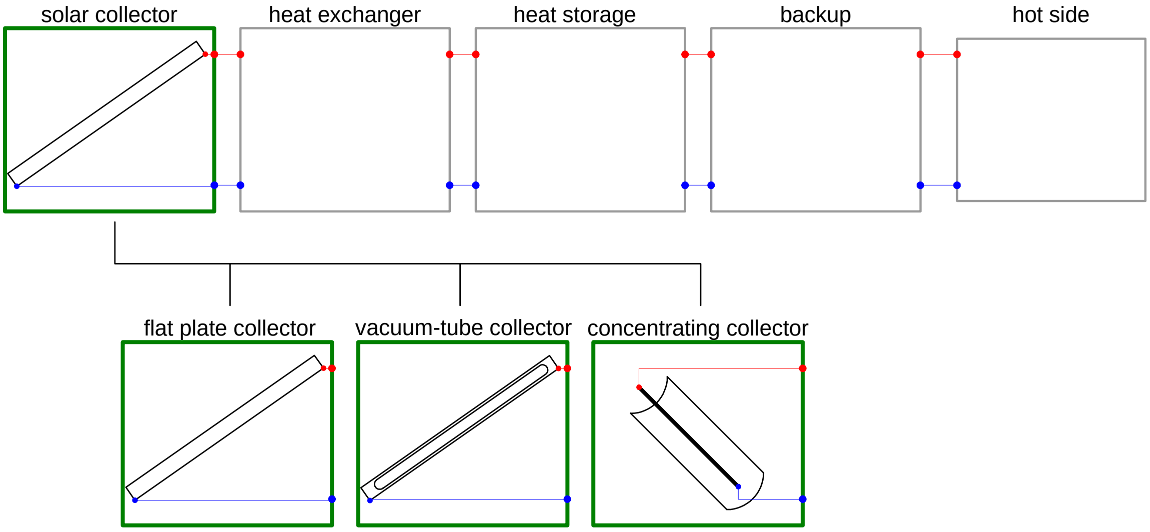


Figure 4: Solar sub-system: options for the solar collector

In the solar thermal system a **heat exchanger** can be integrated for separation of primary and secondary loop. In conventional systems a water/glycol mixture together with a separating heat exchanger is used, assuring trouble-free operation throughout the year at the expense of a reduction of the temperature level in the secondary loop due to the heat transfer in the heat exchanger. In order to avoid the reduction of the temperature level a "direct" utilization of the solar heat may be applied. In this case either freezing of the heat carrier water has to be avoided by means of heat input from the backup-heater or an electrical heater, or a drain-back concept has to be chosen. When the later is applied, the collector loop is filled by the circulating pump only when substantial solar contribution for

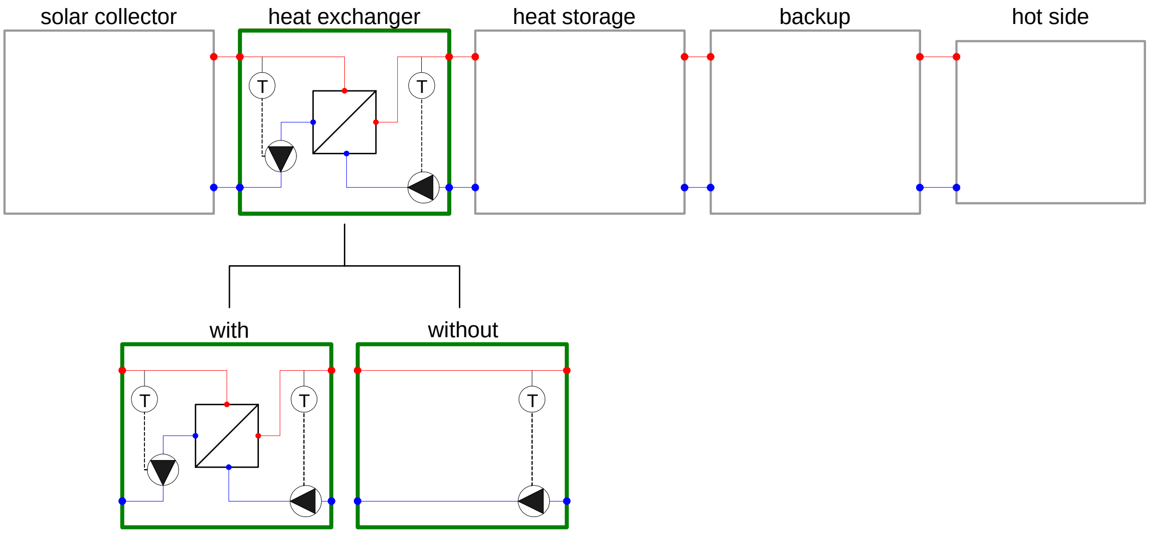


Figure 5: Solar sub-system: heat exchanger for separation of primary and secondary solar loop.

operation of the system in heating or cooling mode is to be expected. A third option would be to drive the chiller directly with the water-glycol mixture, taking into account changing heat transfer rates in the desorber of the chiller.

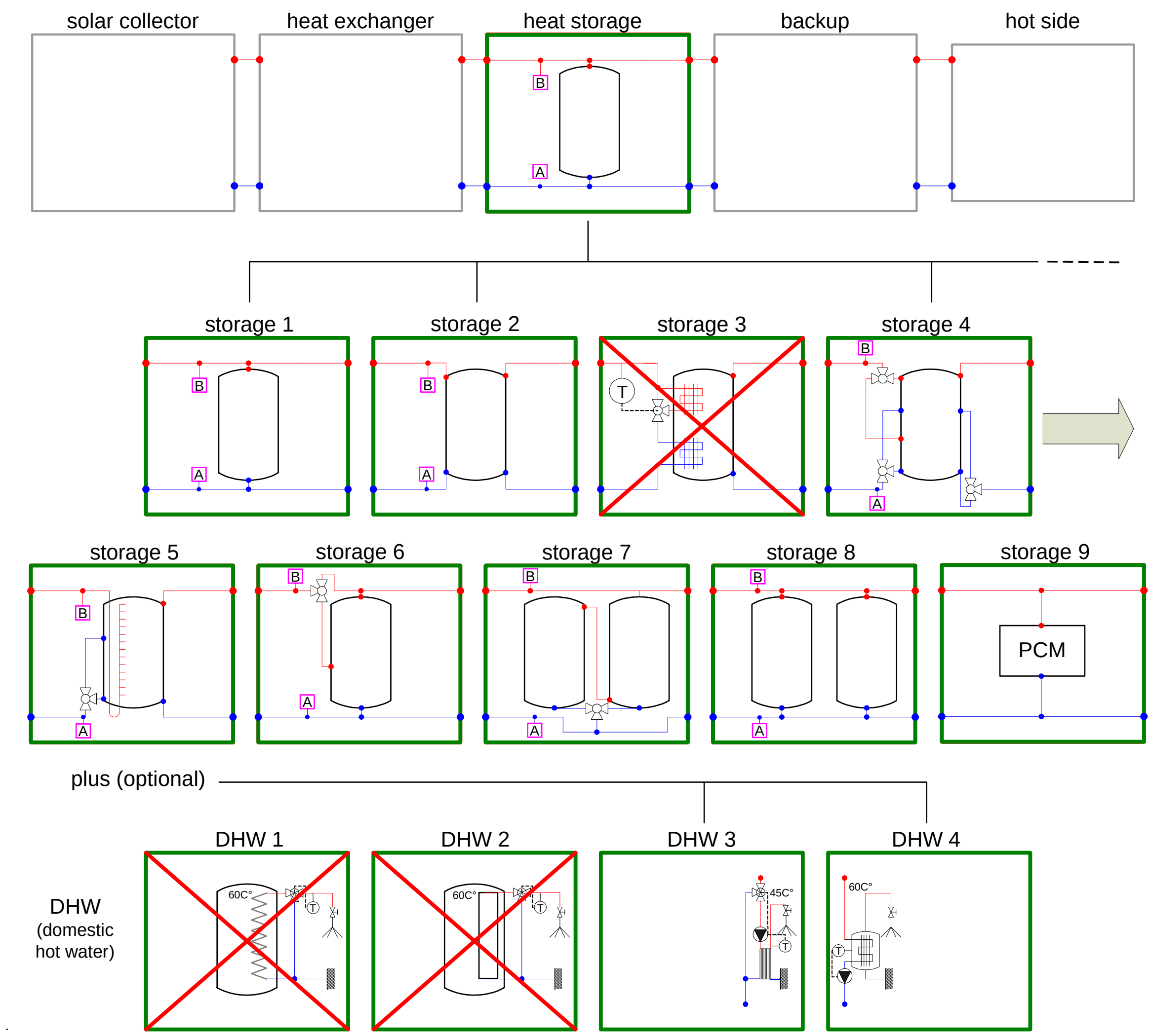


Figure 6: Solar sub-system: options for the heat storage and domestic hot water preparation.

In order to balance solar gain and the profile of the heat consumption, solar thermal systems comprise a **hot water heat storage**. For operation of the thermally driven chiller solar heat at a sufficiently high temperature level is required and any devaluation (cooling by mixing or heat exchange) of the solar heat should be avoided. Thus a direct integration of the storage without heat exchanger is favourable, eliminating a temperature drop during charging and discharging of the store. Therefore, option “storage 3” in Figure 6 is not suitable for solar cooling. In addition the active storage volume should be variable in order to achieve a quick increase of the solar loop temperature for minimum delay between onset of solar irradiation and start of the production of chilled water by the chiller. For this purpose a direct link between solar heat generation and supply of driving heat to the chiller can be chosen (“storage 1, 6, 7, and 8”). For these cases it must be assured that the store still properly serves as a hydraulic switch between heat generation and load. Therefore only minimum pressure drop between supply and return line of the solar thermal system across the store is allowed in order to avoid parasitic flows. Parasitic flows might occur when only the solar collector is charging the store (parasitic flow may occur at the auxiliary heater or the desorber) or when the store is only discharged (parasitic flow may occur on the collector side). In addition, the return line feeding cold water from the storage to the collector inlet can be switched to a lower outlet port of the store when the top layer of the storage has reached the desired driving temperature for the operation of the chiller ("storage 4 and 5"). By that means a reduced thermal inertia of the solar cooling system is achieved with shorter start-up time in the morning. In larger systems two storage tanks connected in series to the solar heat supply can be used ("storage 7"). In this configuration charging of the second store is started only when the first store has been heated completely. "Storage 8" with two parallel storage tanks may serve for reduced flow speed in the storage facilitating a stratified loading with optimum temperature in the top layer of the storage. For the same purpose storage tanks with a distributed feeding system for stratified charging can be applied ("storage 5"). Latent heat stores ("storage 9") offer the advantage of high thermal capacity in a limited temperature interval. Accordingly, a substantial reduction of the thermal inertia of the system could be achieved; yet these systems are still under development.

Apart from space heating and the supply of driving heat for the sorption chiller, solar heat serves for the generation of domestic hot water. In solar combisystems without cooling function, tap water is either heated inside the main heat store by means of an integrated heat exchanger ("DHW 1") or a tank-in-tank system ("DHW 2") or an external flat plate heat exchanger ("DHW 3") or a separate domestic hot water tank ("DHW 4") is used. For the first two options with tap water preparation inside the main heat store, most manufacturers recommend to limit the tank temperature to about 60°C in order to avoid scaling and calcification in the tap water system. This limitation is not compatible with the requirements for the operation of the thermally driven chiller which is operated with about 60 to 90°C hot water supply temperature. Consequently, a solution with external DHW preparation ("DHW 3" or "DHW 4") is more favourable.

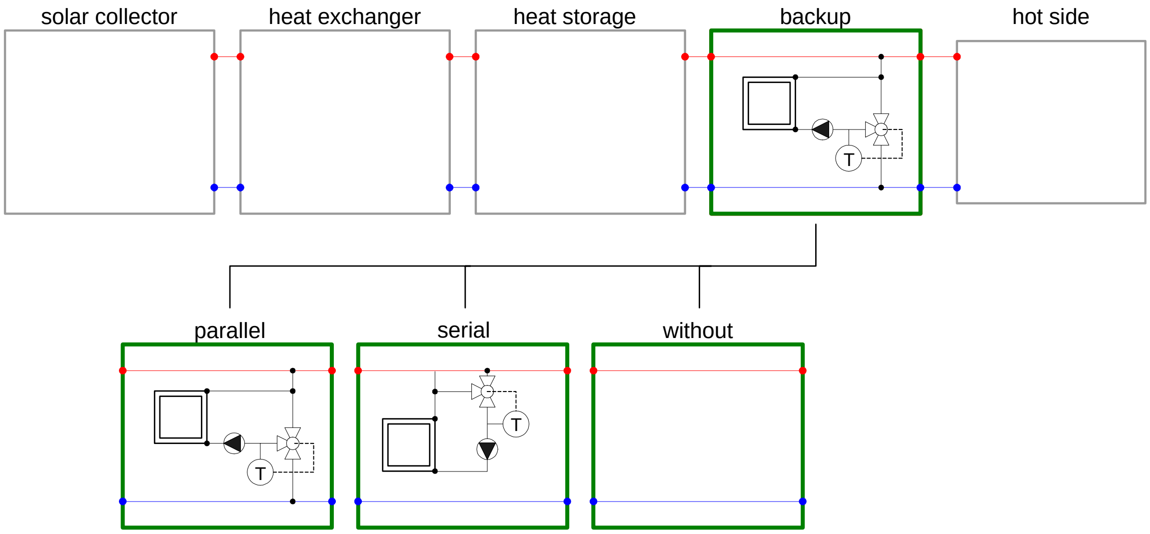


Figure 7: Solar sub-system: options for the integration of the backup-heater.

For supplementing the solar collector system in conventional solar heating systems a **backup-heater** is applied for continuous heat supply during periods of insufficient solar gain. Analogously the backup heater can provide driving heat for the sorption chiller, guaranteeing unrestricted availability of the solar cooling system independently from the output of the solar collector system. Yet, with regard to the required primary energy input only marginal contribution of the backup heater should be allowed. When large amounts of cooling shall be provided without solar heat input, e.g. during the late evening or during night time, a backup chiller integrated into the chilled water sub-system is a promising alternative in terms of primary energy utilization. Then for the solar sub-system no backup heater is required as long as the system serves for solar cooling only. In Figure 7 two options for the integration of the backup-heater are shown. The backup heater is either placed in parallel to the heat storage tank (see “parallel” in Figure 7) or adds heat to the hot water supply line approaching the desorber of the thermally driven chiller (“serial”). Apart from direct integration into the hot water loop driving the chiller, the backup-heater can also provide heat to the heat storage tank. From there the heat can be distributed to all heat consumers. These details are not shown in the graph.

3.2 Heat rejection sub-system

In the heat rejection sub-system different cooling tower types are applicable as **main coolers** (see **Fehler! Verweisquelle konnte nicht gefunden werden.**). For given ambient conditions lowest cooling water temperatures are accomplished by utilizing an open wet cooling tower. A closed cooling water loop allows for less maintenance effort. In this case either a closed wet cooling tower or a dry air cooler may be chosen. The dry air cooler eliminates the formation of fog and legionella bacteria growth at the expense of an increase of the cooling water temperature compared to the wet cooling tower options. Apart from the heat transfer to the ambient air alternative heat sinks may be used in specific situations: A swimming pool offers both a reasonably low temperature level and the option of re-utilizing the reject heat of the sorption cooling process. As an alternative, geothermal systems may be used for dumping the reject heat: Options are ground heat exchangers, boreholes or ground water wells. Heat transfer to these geothermal installations during cooling mode of the system may have a positive impact on the system performance during the inverse operating mode, i.e. during heating operation with the thermally driven chiller operating in heat pump mode. When reject heat is stored underground during the cooling season the average annual ground temperature stabilizes at a higher value facilitating the extraction of ambient heat during the heating season.

The heat rejection via the main cooler may be assisted by an **auxiliary cooler** (see Figure 9), allowing for re-utilization of the reject heat for heating of domestic hot water (“DHW

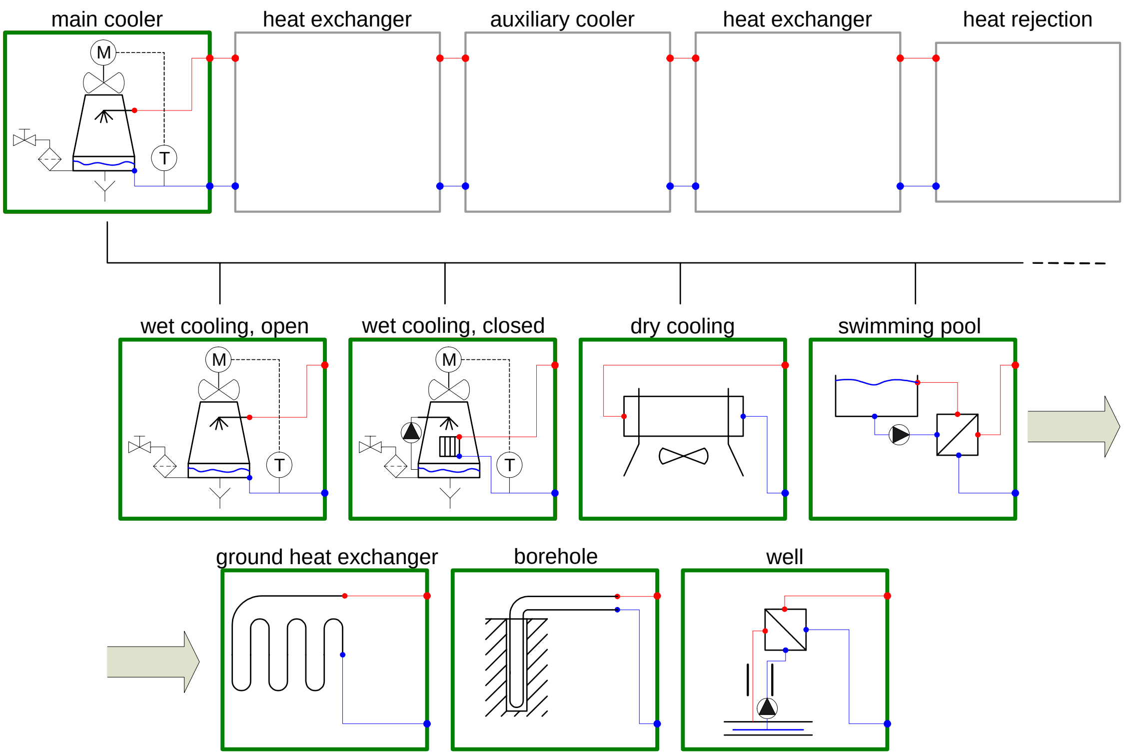


Figure 8: Heat rejection sub-system: open and closed wet and dry cooling towers and alternative heat sinks.

preheating") or a swimming pool. Other alternatives are the heat transfer to the exhaust air of an air-handling unit ("AHU") or a latent heat store ("PCM storage"). The latter two options may offer reduced effort for the transfer of the reject heat to ambient in terms of parasitic energy consumption or operating cost. While the main cooler is continuously available, the capacity and availability of the auxiliary cooler may be limited. Therefore, the auxiliary coolers are installed in the cooling water line leaving the sorption chiller or in parallel to the main cooler. In any case the main cooler assures the desired temperature drop of the cooling water, either by adjusting the flow rates through main cooler and auxiliary cooler or by setting the capacity of the main cooler accordingly.

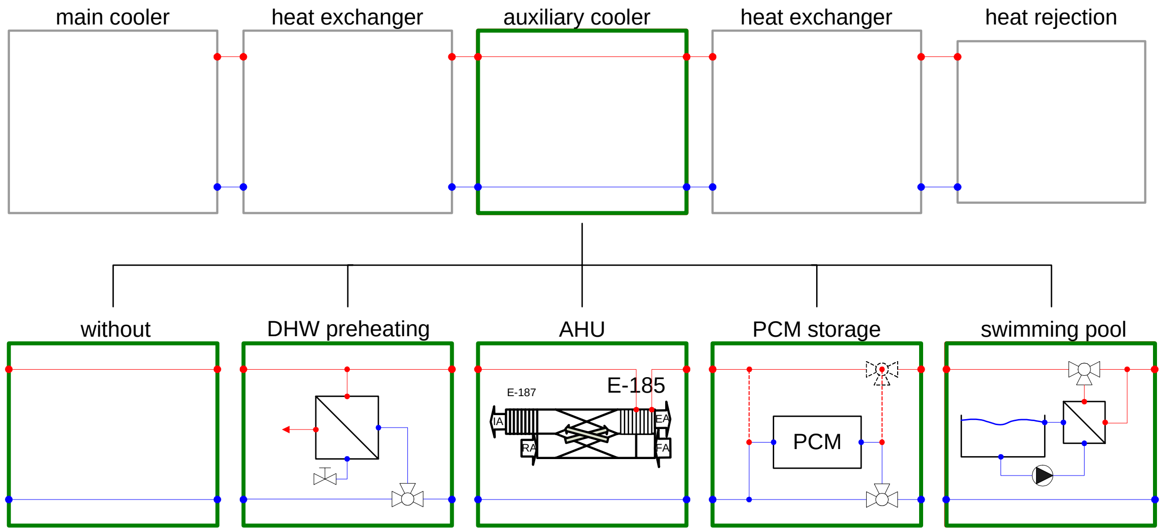


Figure 9: Heat rejection sub-system: auxiliary coolers.

In conventional cooling installations open wet cooling towers are directly coupled to absorber and condenser of the chiller. In order to avoid fouling of the cooling water system due to intake of any pollutants from the open cooling tower, a **heat exchanger** may be installed. Although the thermally driven chiller itself may to some extent tolerate this fouling effect, a heat exchanger is required when additional components like an auxiliary cooler are integrated into the cooling water system. Under certain circumstances a heat

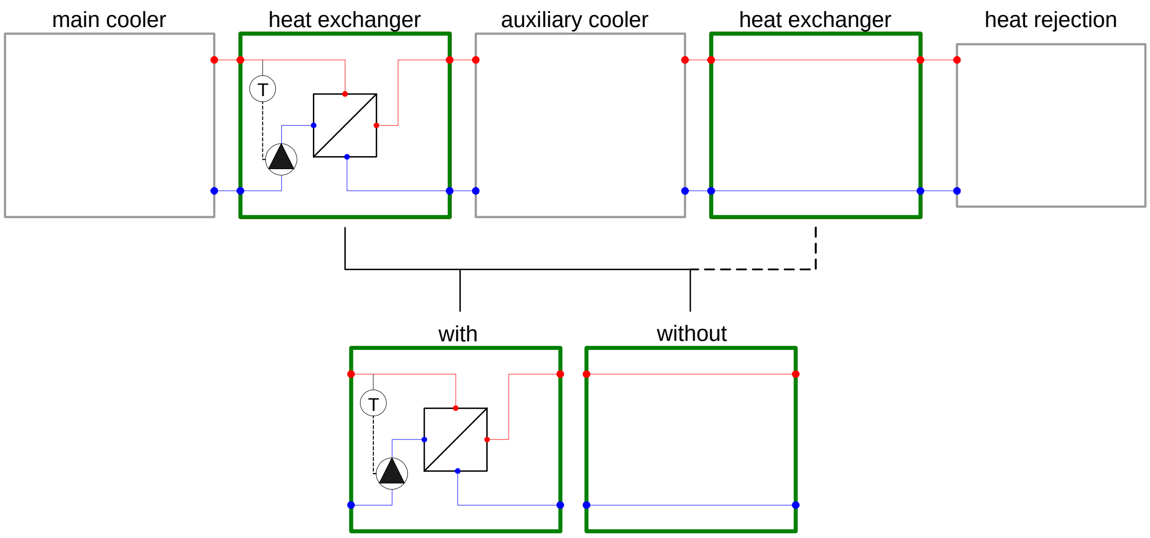


Figure 10: Heat rejection sub-system: optional heat exchangers for separation of thermally driven chiller, auxiliary cooler and main cooler.

exchanger has to be placed between auxiliary heater and chiller.

3.3 Chilled water sub-system

For the transfer of the cooling effect different **room-side installations** are available (see Figure 11). For cooling and dehumidification of the room air either air-handling units ("AHU") or fan coils may be chosen. For dehumidification chilled water temperatures below the dew point of the supply air are required. If only sensible cooling is desired elevated chilled water temperature is sufficient for the operation of radiative cooling surfaces ("radiative heating/cooling"). The same installation can be used for radiative heating during the heating season.

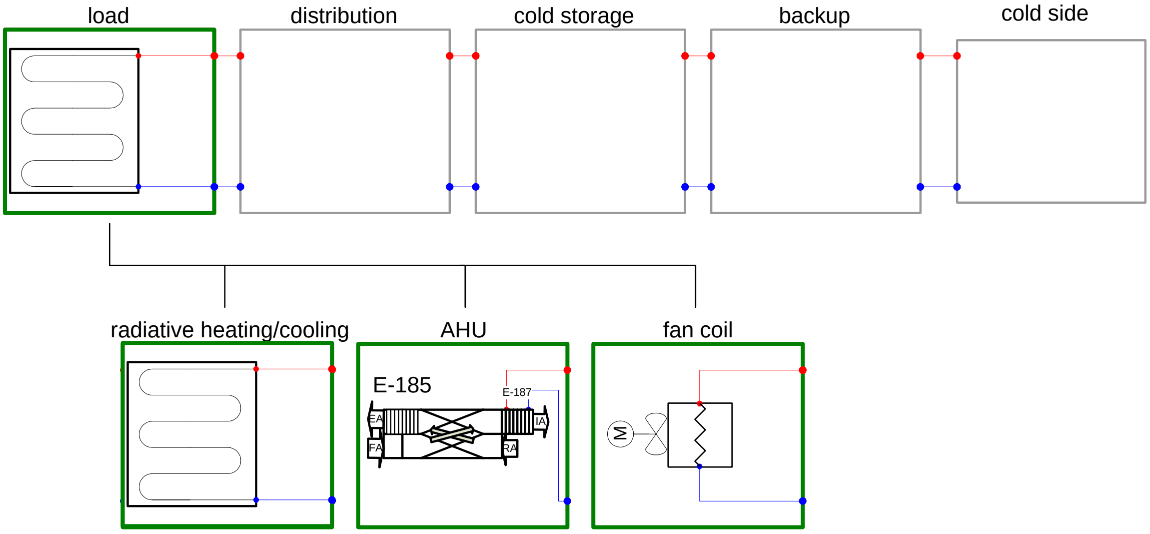


Figure 11: Chilled water sub-system: Room-side installations for the transfer of the cooling effect.

For the **distribution of the chilled water** to the above cooling appliances different hydraulic configurations are feasible (Figure 12): In general a mixing valve between chilled water supply and return line is applied for adjusting the operating temperature of the room-side appliance ("single"). In distributed systems coolers of the same type are installed in parallel in order to assure equal operating conditions. In the graphs a second

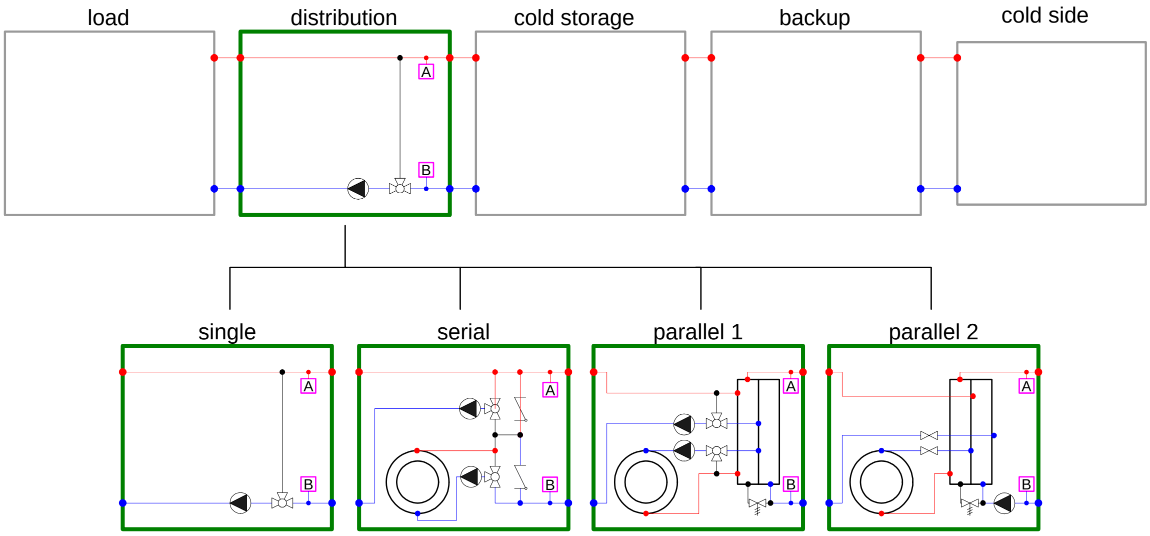


Figure 12: Chilled water sub-system: Options for the distribution of the chilled water.

load is depicted by two concentric circles. In the configuration “parallel 1” for each loop a pump together with a mixing valve is installed. In configuration “parallel 2” the main chilled water pump at the inlet port of the distributor serves for circulation of the chilled water throughout the whole chilled water system. Temperature adjustment is accomplished by means of flow control by two-way valves in each sub-loop. If coolers of different types shall be operated with different chilled water supply temperature a serial configuration (“serial”) allows for stepwise utilization of the cooling effect. As a consequence, the serial concept allows for higher chilled water return temperature and enhanced performance and capacity of the chiller.

In the case of larger fluctuations of the cooling demand (which results in fluctuations of the volume flow between distribution and storage) a **cold storage** serves for levelling the chilled water consumption by decoupling the volume flow of the load and the chiller, and facilitates stable operation of the thermally driven chiller (Figure 13). Predominantly chilled water is used as storage medium. The storage is either directly loaded (“cold storage 1” and “2”) or an internal heat exchanger (“cold storage 3”) serves for the separation of the primary and secondary cooling loop. In analogy to the solar sub-system, the configuration “storage 1” with direct link from chilled water production to the load allows for bypassing the cold storage. Consequently, the chilled water flow transiting the storage vessel is reduced and mixing of the storage volume is avoided. Again hydraulic integration with minimum pressure drop is essential for avoiding parasitic flows in the chilled water system. The configuration “storage 3” induces a temperature loss between primary and secondary chilled water loop and is therefore not recommended. Yet, a separation of the hydraulic loops by a heat exchanger may be required if an ice storage shall be applied. In the future the use of other phase change materials as storage medium or PCM slurries may gain increasing importance. As a result due to the latent heat effect a reduction of the storage volume in comparison to a chilled water tank is accomplished.

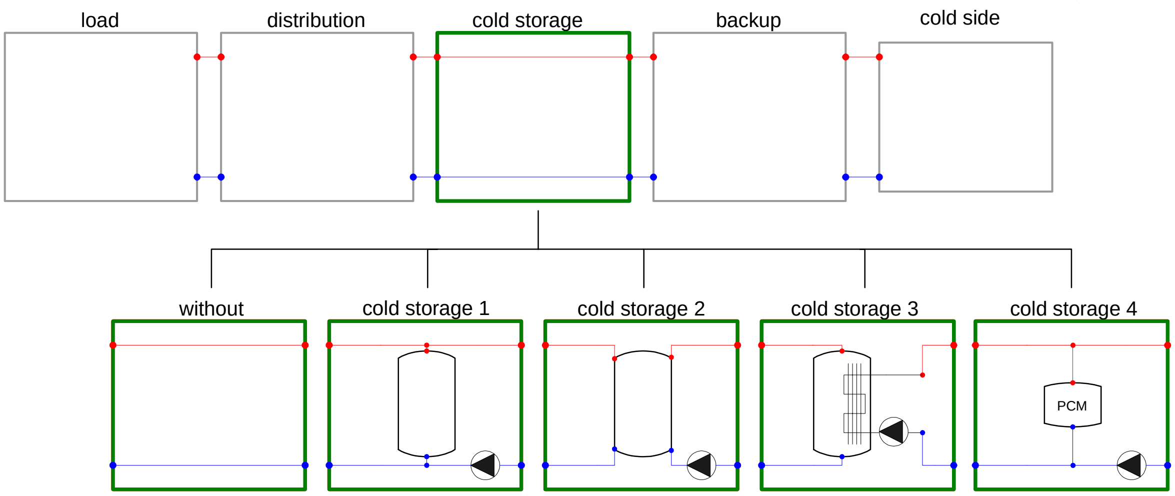


Figure 13: Chilled water sub-system: Chilled water storage and latent cold storage (PCM, phase change material).

If a reliable chilled water supply has to be guaranteed – independently from the availability of the solar driving heat – a **backup cooling source** may be provided. For this purpose a compression chiller may be installed in the chilled water supply line. If the thermally driven chiller does not provide sufficient cooling capacity, the mechanical chiller is controlled in order to reach the desired chilled water supply temperature. As an alternative a ground water well or any other geothermal installation, e.g. a borehole or a ground heat exchanger, can serve as backup cooler. During the winter the ground water well may be used as geothermal heat source for the system operating in heat pump mode.

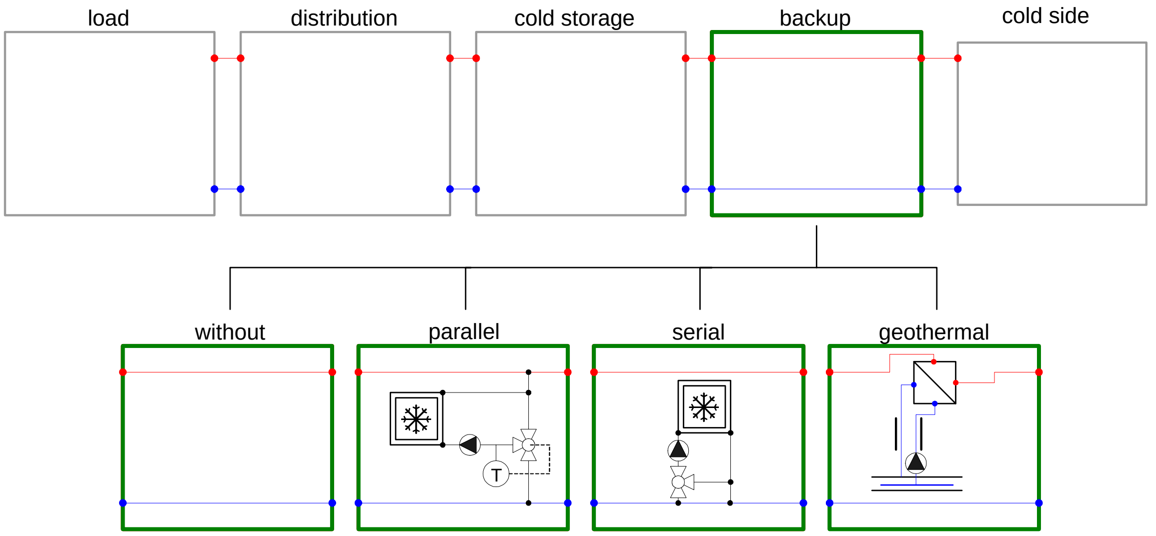


Figure 14: Chilled water sub-system: Integration of a compression chiller or a ground water well as backup cooling source.

4. System control and hydraulics

4.1 Solar thermal system

The thermo-physical properties of the sorbent used in the thermally driven chiller set the required temperature level of the driving heat which is applied to regenerate the sorbent in the desorber of the chiller. Consequently, when the solar heat from the collector system serves as driving heat for the chiller the solar collector loop is to be operated with limited temperature lift and with high volume flow of the heat carrier. Thus, in low-flow collector installations, which are needed for combisystems (combined space heating and domestic hot water production), a change of the pump setting together with a change of the hydraulic configuration may be required for the cooling season to avoid extremely high collector temperatures resulting in inefficient collector performance with the risk of stagnation of parts of the collector circuit due to boiling of the water-glycol mixture.

In solar heating mode, the solar thermal system may operate at moderate temperatures providing heat input to a low temperature heating system or serving for preheating of a tap water flow. Given a favourable hydraulic integration the system reaches an operational state after a short start-up phase. During operation surplus solar heat results in a gradual increase of the system temperature with a heat storage effect within a rather large temperature interval. As stated above in solar cooling mode the solar heat has to exceed a certain minimum temperature level required to drive the sorption chiller. Consequently, after start of the solar thermal system with incipient solar radiation in the morning hours a substantial amount of heat has to be collected in order to reach the required operating temperature. To achieve a quick response of the system with only short delay between the start of the solar thermal system and the operation of the chiller the thermal inertia of the system has to be limited or stratifiers for charging the heat store in combination with a matched flow system have to be applied. For this purpose the volume of the heat storage, its modular design and hydraulic integration, the loading strategy of the heat storage, and the collector hydraulics and control have to be chosen accordingly, as discussed in section 3.1.

4.2 Capacity control

For the purpose of capacity control the chilled water supply temperature serves as the controlled variable. Conventionally, the hot water supply temperature represents the manipulated variable. Actuating device is a three way mixing valve at the inlet port of the desorber of the thermally driven chiller. For reducing the chilled water capacity of the chiller the entering hot water temperature is reduced by recirculating cooled hot water leaving the desorber back to the desorber inlet. As an alternative the chilled water capacity can be controlled by manipulating the hot water flow through the desorber by either a speed control of the hot water pump or by means of a two-way regulating valve.

According to the state-of-the art, cooling systems operate with fixed chilled water supply temperature. For standard comfort air-conditioning the chilled water supply temperature is set to 6°C or 45°F with a supply-return span of 6 K or 10°F. In part load operation with fixed chilled water supply temperature the return temperature drops accordingly. Deviating from this established convention, i.e. operation with fixed chilled water supply temperature, a positive effect on the performance of the solar cooling system would be achieved by controlling the chilled water return temperature to a fixed value. Consequently, the thermally driven chiller would be operated with increasing chilled water supply temperature during part load operation. Thus a lower driving hot water temperature would be sufficient resulting in a higher contribution of the solar collector and reduced need for backup operation of the boiler or the compression chiller, respectively.

4.3 Auxiliary power demand

With regard to the desired primary energy saving effect, the operation of all system components has to be optimized. In order to reduce the pumping energy demand highly efficient pumps should be used together with a reasonable control strategy avoiding circulation of the solar thermal system with only marginal solar gain.

When a dry air-cooler is applied the blower fan of the heat rejection system significantly contributes to the overall electric power demand. For this case a change of the system's capacity control promises increased system efficiency: Given sufficient solar driving heat, in case of decreasing chilled water demand the transfer characteristic of the heat rejection system can be reduced by manipulating the fan speed of the dry air-cooler. As a result the chilled water capacity is reduced at constant operation of the driving solar thermal system. In a second step the heat input into the desorber may be adjusted for matching the chilled water demand if required.

4.4 Solar Heating & Cooling

As stated earlier, the topology of the solar cooling system incorporates the solar collector system as driving heat source. With regard to the year-round operation the system has to serve for both heating and cooling. For this purpose the hydraulic system has to be prepared for a switch from solar cooling to solar heating. An example is shown in . Ideally the room-side appliances, e.g. radiative heating/cooling surfaces, are designed for both modes of operation. Heat for space heating and tap water preparation is provided by the solar collector system. Backup heating is accomplished by a backup heater.

Apart from solar heating supported by a backup heater the system may be operated in heat pump mode with the thermally driven chiller serving as a heat pump for the provision of low-grade heat for space heating (). A geothermal installation, e.g. a ground water well operating as backup cooler in cooling mode, can be utilized as heat source linked to the evaporator of the heat pump. During winter, only marginal contributions of the solar collector system are to be expected. Thus driving heat for the sorption heat pump is predominantly provided by the backup heater. Useful heat is extracted at absorber or adsorber and condenser of the sorption cycle and is distributed to the heating appliances.

Peak load heating demand can be covered by additional heat input from the high temperature sub-system comprising the solar collectors and the backup heater.

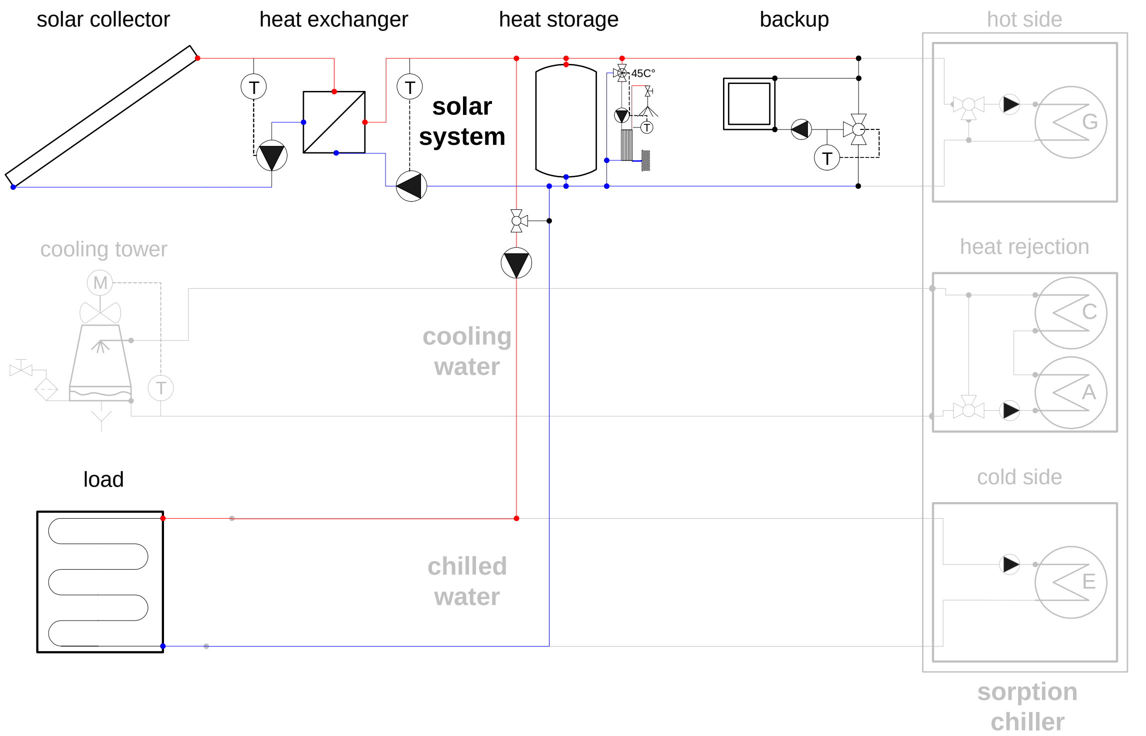


Figure 15: Operation of the solar heating and cooling system in solar heating mode.

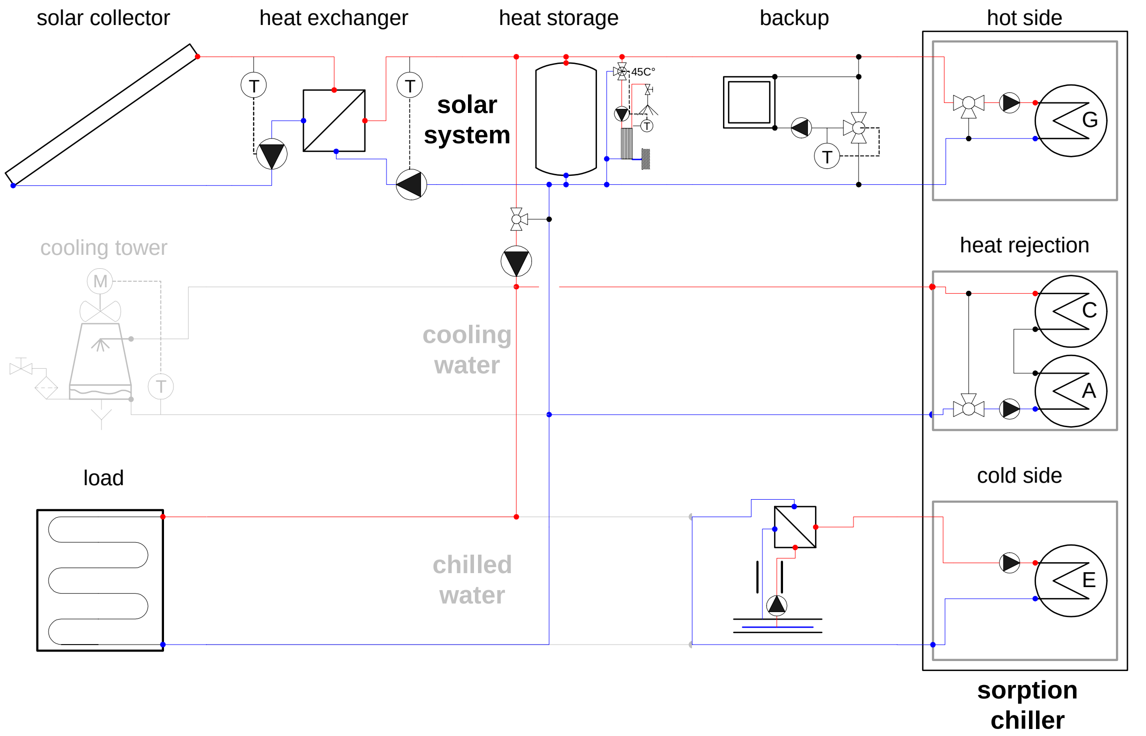


Figure 16: Operation of the solar heating and cooling system in heat pump mode.

5. Selection guide

In the following tables a comparative characterization is given, summarizing the technical discussion in the previous sections.

	function or main characteristic	additional information
solar thermal sub-system		
solar collector	accumulation of solar heat	
flat plate collector	cost efficient	robust, reliable
vacuum-tube collector	low thermal losses	high collector efficiency, low collector area required
concentrating collector	optimum efficiency for direct radiation	high collector efficiency, suitable for double-effect chiller
heat exchanger	separation of hydraulic loops	
with	anti-freeze liquid in solar loop	negative impact on thermal performance due to driving temperature difference
without	measures for freeze-protection required	pure water as solar heat carrier or drain back system or water-glycol mixture directly supplied to sorption chiller
heat storage	storage of surplus solar heat	
storage 1	direct link between collector and heat consumer	simple construction, reduced flow facilitates stratification
storage 2	no bypass, all hot water flows pass through storage	simple construction, decouples volume flows, no parasitic flows
storage 3	not applicable	
storage 4	charging and discharging section-wise	switching valves for selction of heat storage section
storage 5	exact stratification	gravity loading system
storage 6	charging and discharging section-wise	switching valve for selction of heat storage section
storage 7	charging and discharging section-wise	switching valve for activation of addional storage volume
storage 8	enhanced storage capacity	adjustment of storage volume by supplementary installation of storage tanks
storage 9	latent heat storage: maximum storage density	operation with characteristic temperatures for loading and unloading
DHW (domestic hot weater)	tap water heating	
DHW 1	not applicable	
DHW 2	not applicable	
DHW 3	external heat exchanger	instantaneous tap water preparation
DHW 4	external tap water storage	storage covers load peaks
backup	auxiliary heat source	
parallel	hot water boiler	parallel installation, loading of heat storage possible
serial	hot water boiler	serial installation, boosts hot water supply temperature, no loading of heat storage
without		

	function or main characteristic	additional information
heat rejection sub-system		
main cooler	guarantees transfer of reject heat under all ambient conditions	
wet cooling open	best cooling capability, theoretic limit: wet bulb temperature	low cooling water temperature, consequently low driving hot water temperature for sorption chiller. water make-up required, risk of legionella infection and fog formation.
wet cooling closed	good cooling capability, increased cooling water temperature due to additional heat transfer	water make-up required, risk of legionella infection and fog formation.
dry cooling	reliable, trouble-free operation. only sensible cooling: elevated cooling water temperature	requires higher driving hot water temperature for sorption chiller. For water/LiBr chiller not applicable in hot climates.
swimming pool	re-utilization of the reject heat	increasing cooling water temperature due to limited storage capacity
ground heat exchanger	heat transfer to surface ground layer	high installation effort, low parasitc power demand
borehole	heat transfer to deep ground	high installation effort, low parasitc power demand
well	heat transfer to ground water	high installation effort, low parasitc power demand
heat exchanger	separation of hydraulic loops	
with	avoids intake of pollutants from open cooling loop	
without		
auxiliary cooler	re-utilization of the reject heat	
without		
DHW preheating	tap water heating	substitutes fresh heat (solar heat or fossil fuel)
AHU	heat transfer to exhaust air of AHU	replaces dry air-cooler
latent heat storage (PCM)	storage of waste heat	facilitates application of dry air-cooler even in hot climates
swimming pool	heating of pool water via heat exchanger	substitutes fresh heat (solar heat or fossil fuel)
heat exchanger	separation of hydraulic loops	
with	separation from water/glycol loop	
without		

	function or main characteristic	additional information
chilled water sub-system		
load		
radiative heating/cooling	only sensible cooling, no dehumidification, heating in winter	moderate temperature level for heating and cooling , chilled water temperature above dewpoint
AHU	heating, cooling and dehumidifcation	low chilled water temperature required for dehumidifcation
fan coil	heating, cooling and dehumidifcation	condensate handling required for dehumidification
distribution		
single	single cooling appliance (see options "load")	pump and mixing valve for control of entering chilled water temperature. No chilled water pump at chiller port required.
serial	cascade of cooling appliances operating at different temperature levels	chilled water return flow from first cooler is used in following cooler, supply temperatures are controlled indiviually
parallel 1	parallel installation of cooling appliances	pump and mixing valve for individual setting of supply temperatures, chilled water supply to distributor by chilled water pump at chiller port
parallel 2	parallel installation of cooling appliances	pump and flow control for individual setting of supply temperatures, chilled water supply to distributor by chilled water pump at chiller port
cold storage		
without		
buffer 1	direct link between chilled water generation and load	simple construction, reduced flow facilitates stratification
buffer 2	no bypass, all chilled water flows pass through storage	simple construction, decouples volume flows, no parasitic flows
buffer 3	storage loaded by internal heat exchanger, ice storage	chiller operates at lower temperature, negative impact on chiller efficiency and/or higher driving temperature required
buffer 4	latent cold storage	hydraulic concept to be defined
backup		
without		
parallel	compression chiller, parallel integration	split flow to different chillers, sorption cooling covers base load only
serial	compression chiller, serial integration	sorption chiller designed for full load, backup chiller provides final cooling when solar driving heat not sufficient
geothermal	free cooling by heat transfer to geothermal source	optimum energy saving

6. System examples

The following figures show a selection of system configurations and their representation on basis of the “generic system” scheme discussed in this report. The first three examples have been suggested by manufacturers. The last two examples represent pilot installations.

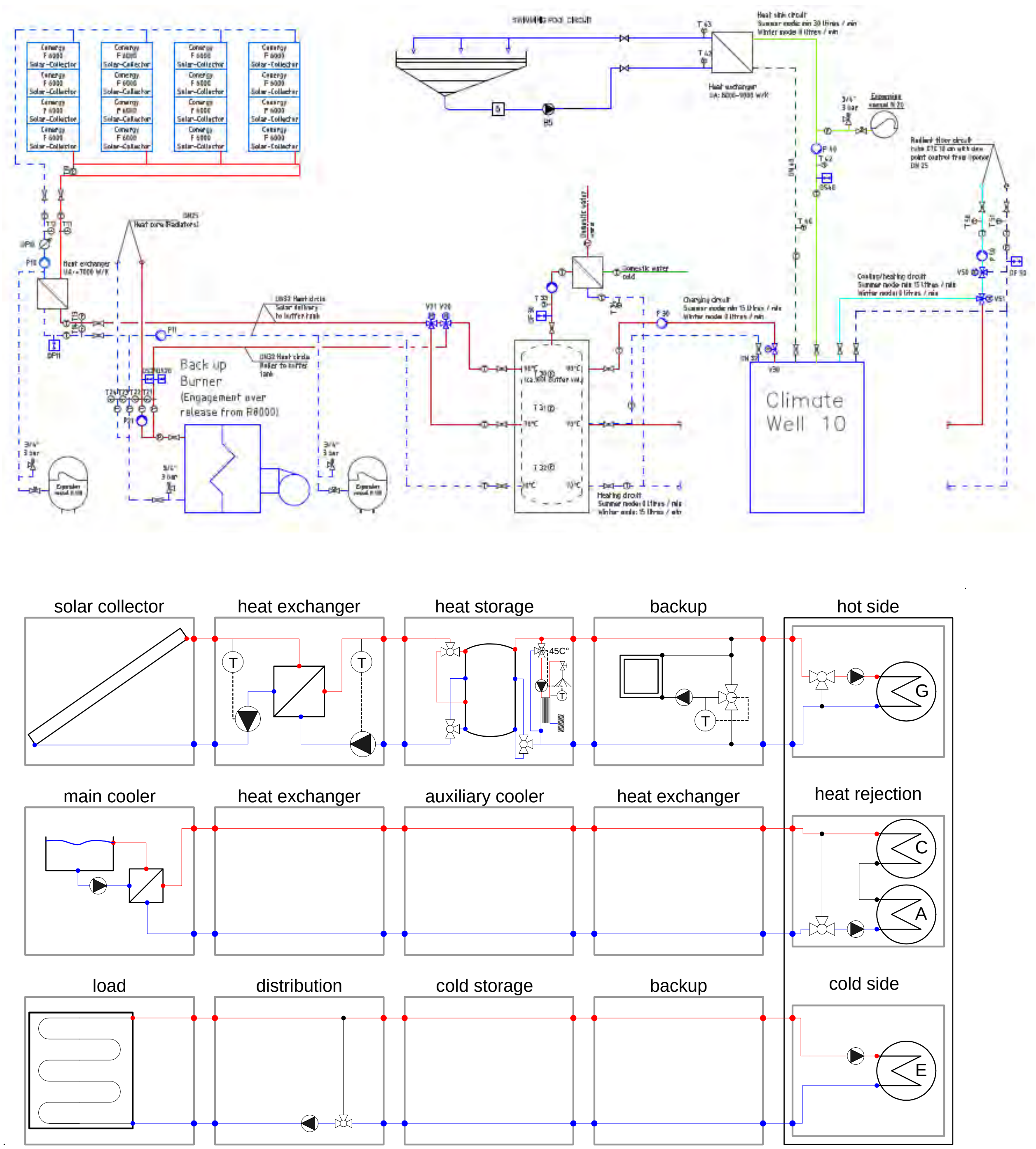


Figure 17: Solar heating and cooling system, Example 1 (source: Climatewell).

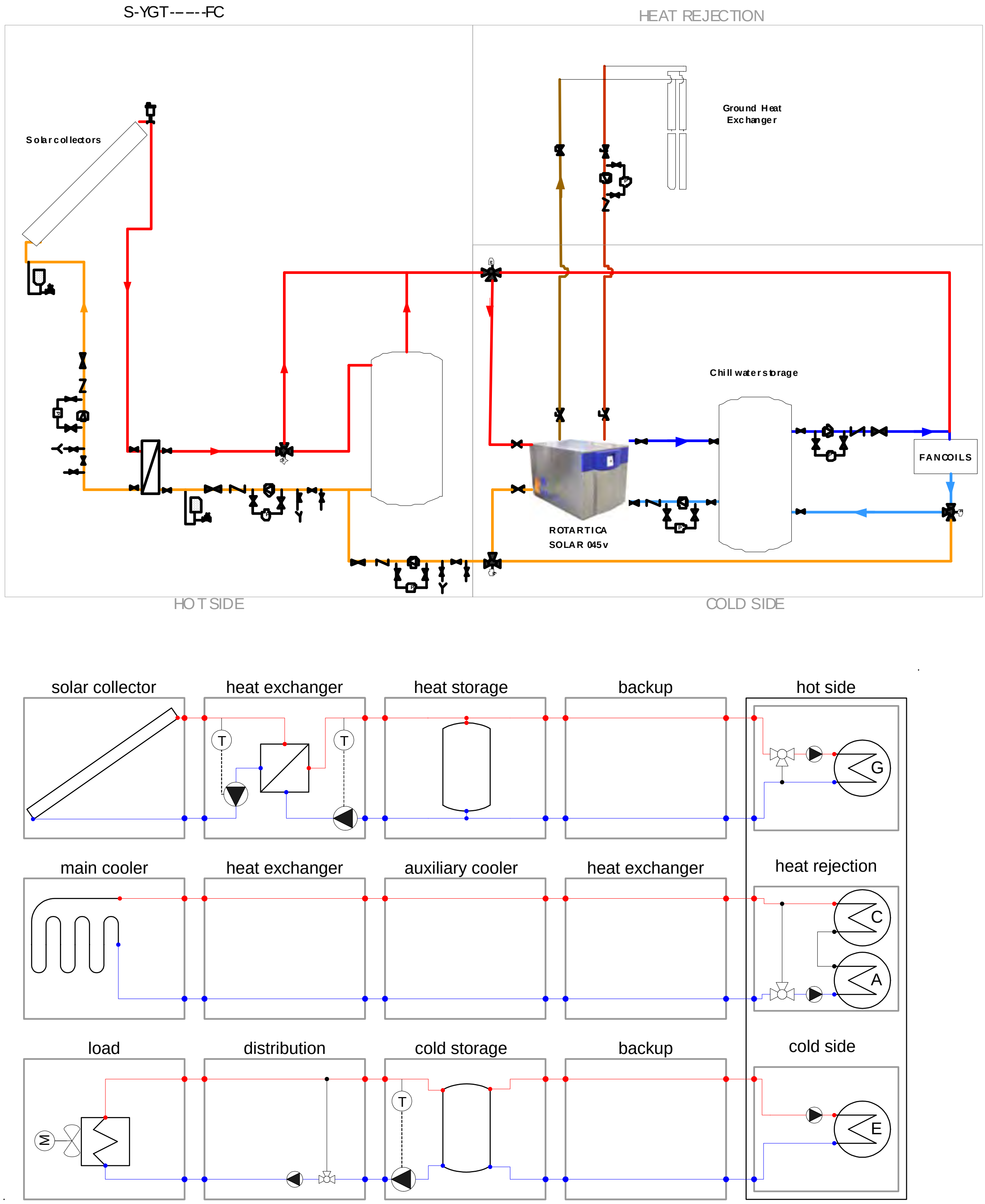


Figure 18: Solar heating and cooling system, Example 2 (source: Rotartica).

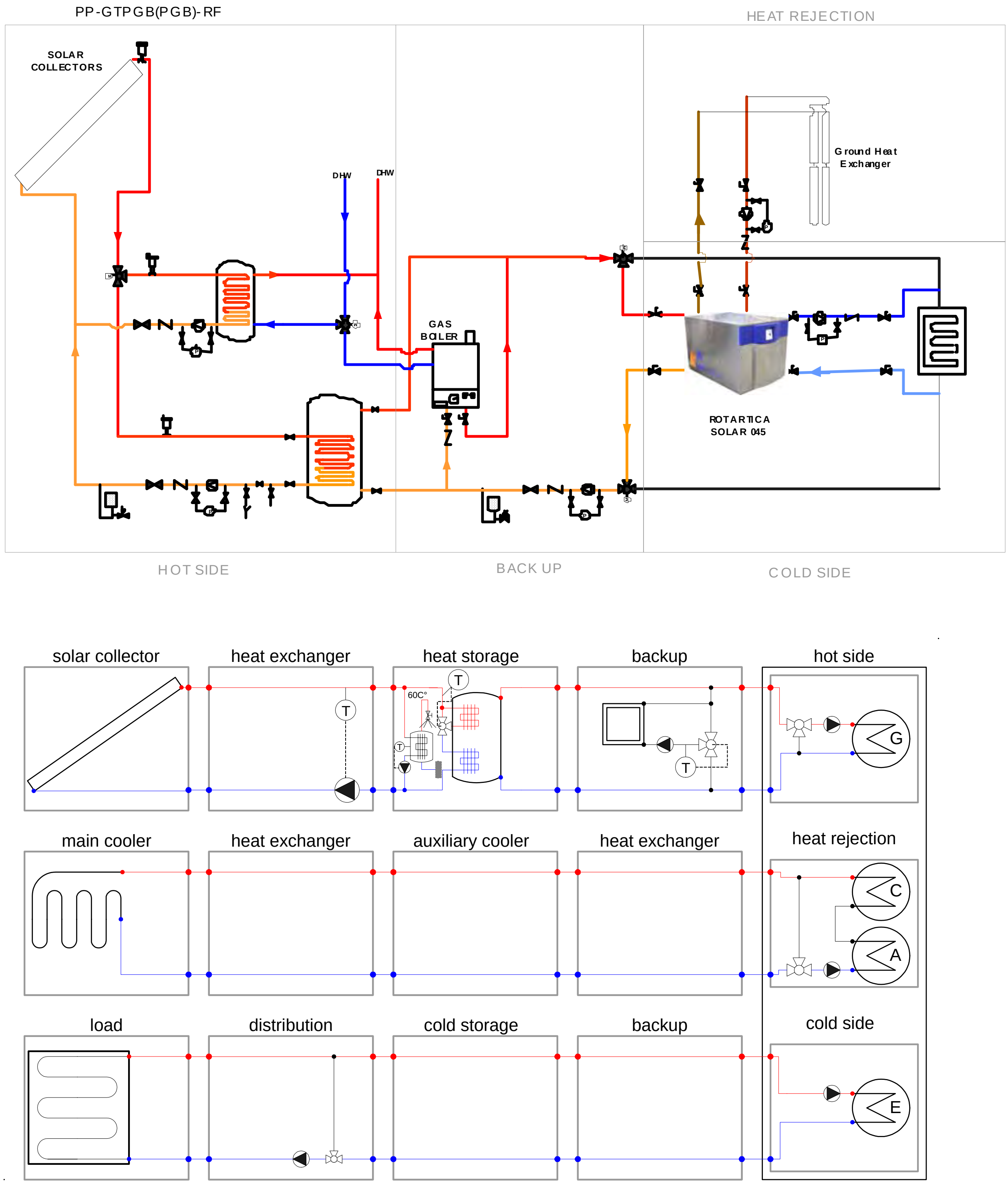


Figure 19: Solar heating and cooling system, Example 3 (source: Rotartica).

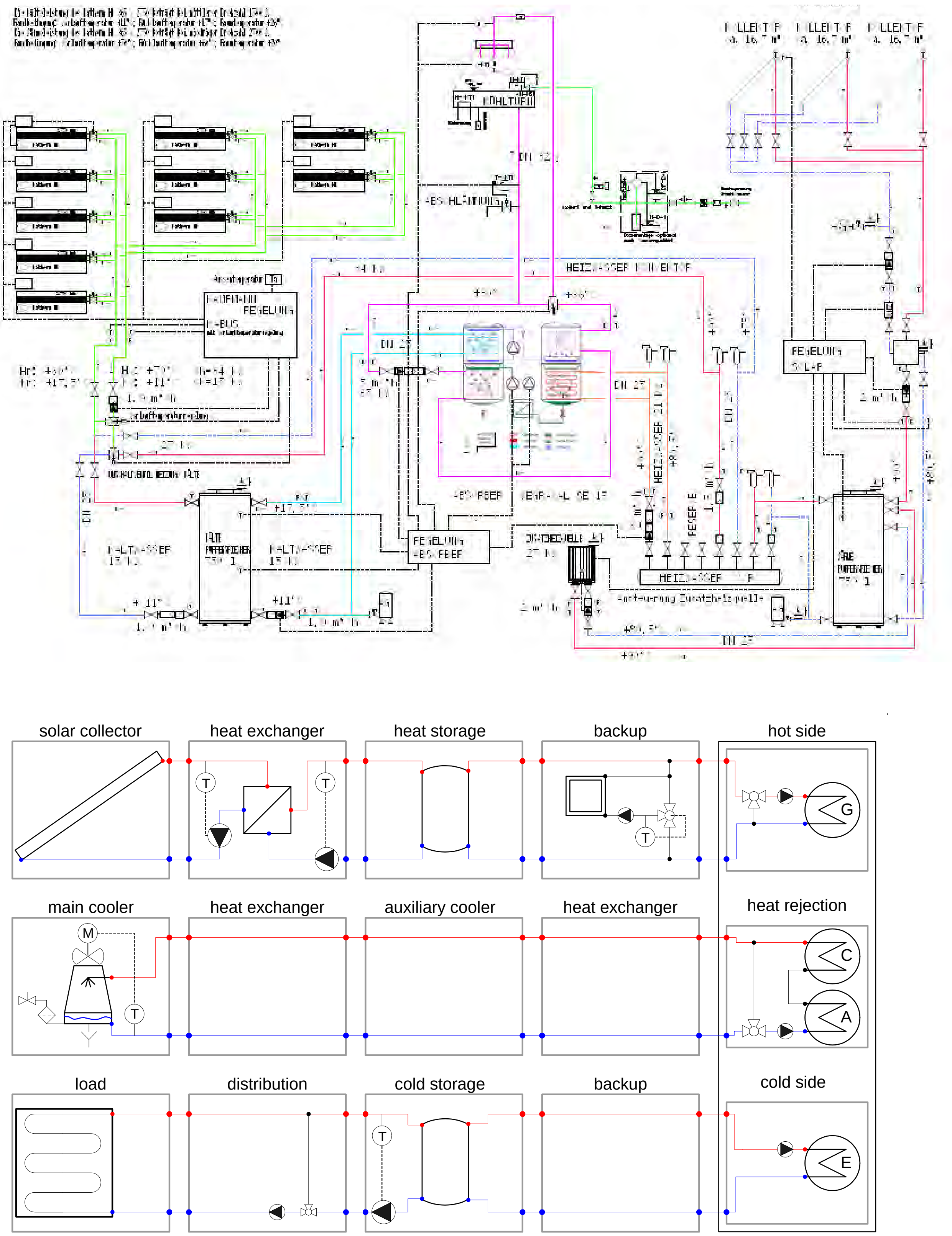


Figure 20: Solar heating and cooling system, Example 4: Pilot installation at company Festo, Esslingen, Germany (source: FH Offenburg).

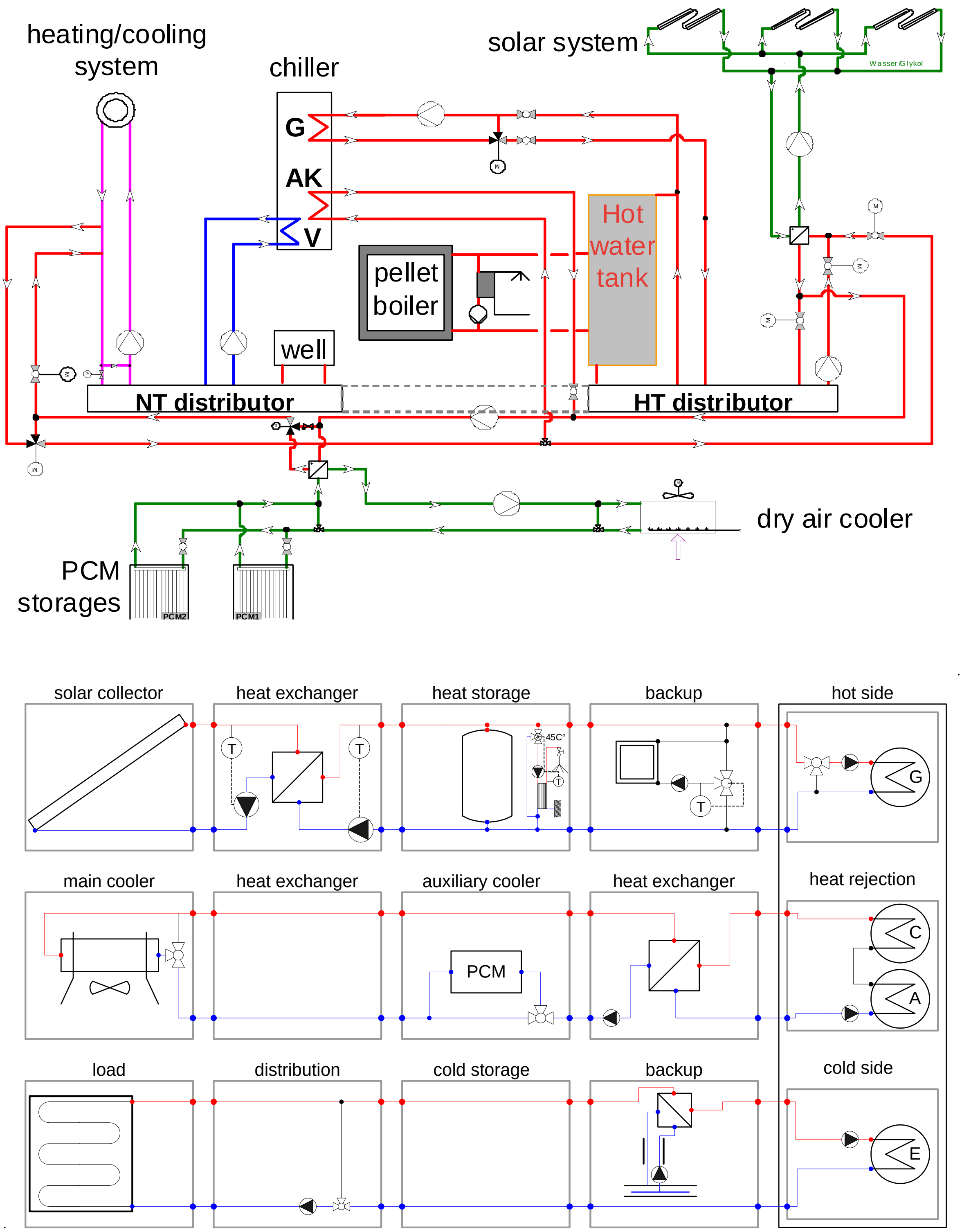


Figure 21: Solar heating and cooling system, Example 5: Pilot installation at ZAE Bayern, Garching, Germany (source: ZAE Bayern).

Task 38
Solar Air-Conditioning
and Refrigeration

Monitoring Procedure for Solar Cooling Systems

A joint technical report of subtask A and B (D-A3a / D-B3b)

Date: 2011, August 1st 99% draft Version

Relevant version of the monitoring procedure: 110801_T38_MonProc_V6-0.xls

Download: http://www.iea-shc.org/publications/task.aspx?Task=38

Authors: Assunta Napolitano[1], Wolfram Sparber[1], Alexander Thür[2], Pietro Finocchiaro[3], Bettina Nocke[3]

[1] Institution	EURAC research
Address	Viale Druso 1, 39100 Bolzano, Italy
Phone	0039 0471 055 600
Fax	0039 0471 055 699
e-mail	assunta.napolitano@eurac.edu wolfram.sparber@eurac.edu

[2] Institution	AEE INTEC
Address	A-8200 Gleisdorf, Feldgasse 19
Phone	+43 (0)3112 5886-26
Fax	+43 (0)3112 5886-18
e-mail	a.thuer@aee.at

[3] Institution	Dipartimento per le Ricerche Energetiche ed Ambientali (DREAM), Università di Palermo
Address	Viale delle Scienze ed. 9, 90128 Palermo, Italy
Phone	0039 091 238 61908
Fax	0039 091 484 425
e-mail	bettina@dream.unipa.it finocchiaro@dream.unipa.it

Table of contents

Table of Figures

Table of tables

1. Introduction

Monitoring of installed solar assisted cooling systems represents a fundamental tool in order not only to optimize the monitored system itself, but as well to draw conclusions for a suited selection of design and control for future installations.

This is especially true for a technology in an early stage of market penetration as it is the case for Solar Heating and Cooling (SHC) systems. In fact today, according to the investigations carried out in the Task 38 only around 300 SHC are documented worldwide.

Within the documented plants, 14 small scale (< 20 kW cooling capacity) and 12 large scale systems result to be monitored and they show large different designs and surrounding conditions.

In order to enable a structured collection of monitoring data and to define a common evaluation methodology of the energy performance of SHC plants, a unified monitoring procedure has been developed.

In practice the procedure aims at:

- evaluating the performance of the existing systems;
- comparing the energy performance of the existing system with the performance of a conventional system;
- enabling the comparison between different existing systems.

On the basis of such output, the procedure could lead to the identification of best practices hence of best design solutions in relationship with the climate, the building features and use, occupation conditions and so on.

Furthermore the procedure defines the minimum monitoring equipment required to evaluate the energy performance of the existing systems. So it is not restricted to the IEA activities and the mentioned systems but can support future monitoring activities in general.

Finally, the procedure should permit as well to drawing (with the results) a learning curve over the coming years on the cost development of installed solar assisted heating and cooling systems.

The procedure can be applied for both Solar Heat Driven Chiller (SHDC) and Desiccant Evaporative Cooling (DEC) systems even if for the latter case an additional special tool has been developed

This report describes the developed monitoring procedure which includes:

- one standard tool for the graphical representation of monitored systems (ppt-file);
- one standard tool for the visual representation of a conventional system as term of comparison with SHC systems (ppt-file);
- one excel file which enables the calculation of key figures such as primary energy ratio, electrical coefficient of performance, solar heat management efficiency and fractional savings with respect to a conventional (non solar) heating and cooling system
- one more excel file to be applied in case of DEC systems with the aim at collecting very detailed monitoring data and derive intermediate figures useful for the evaluation of relevant energy performance with respect to a conventional system.

2. Selection of graphical representations

The monitoring procedure is based on the evaluation of monitored systems and their comparison with a selected conventional system.

To enable a clear and homogeneous representation of monitored systems and of the conventional system to be taken as reference for comparisons, two standard diagrams have been drawn which are below described.

Note that depending on whether the monitored system is based on SHDC or DEC, the conventional system to be taken as reference is different.

On the other hand, to determine if the solar heating and cooling system performs better than a conventional system that covers the same loads, a reference system was defined as well. Both the schemes are below presented.

2.1 Selection of a standard graphical scheme for the representation of real monitored installations

A maximum diagram (Figure 2-1) for the representation of monitored systems has been developed on the basis of commonly applied layouts for SHC systems and it enables to represent largely different systems. This scheme is available as a power point file (see Appendix) which can be adapted mainly by deleting those parts which do not exist in the system under investigation.

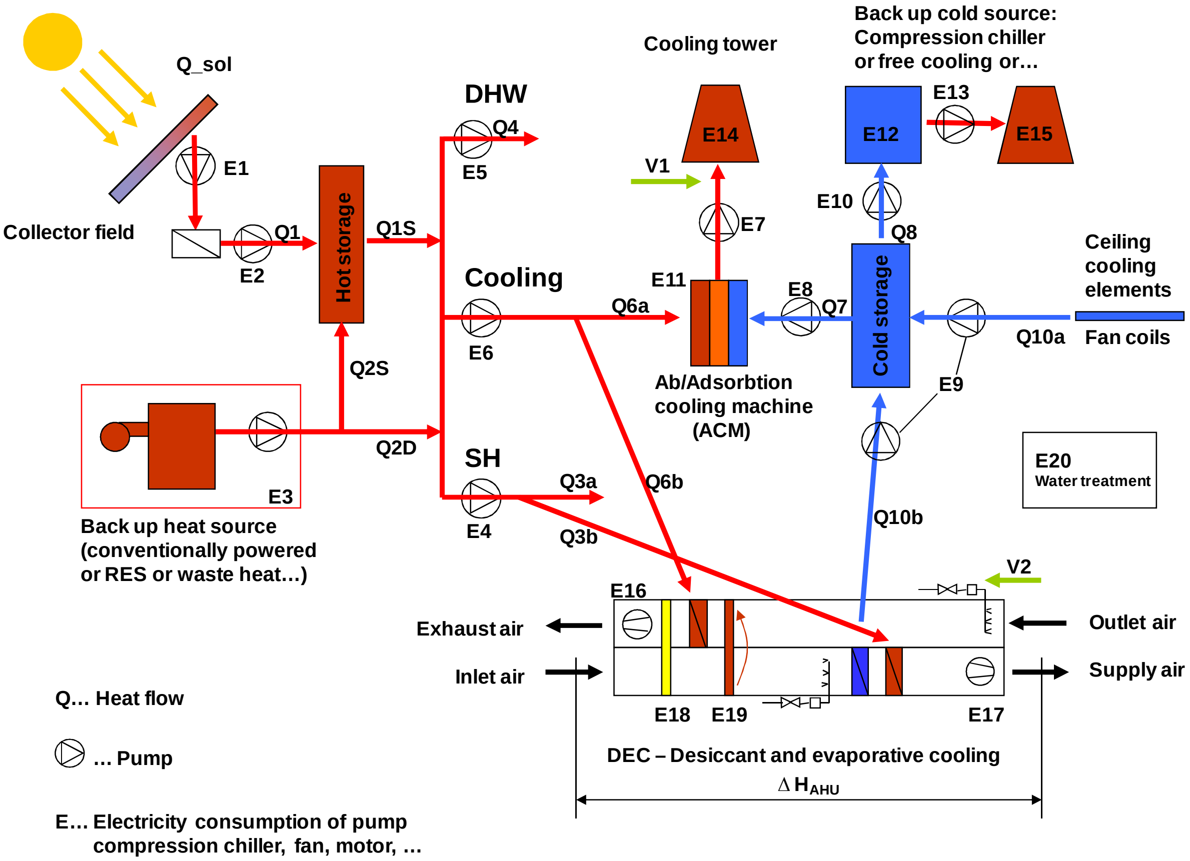


Figure 2-1 Reference solar heating and cooling system including the single energy fluxes (SHC Max System)

Electricity and thermal flows involved in a SHC system are shown in the figure as well. For a detailed explanation of each flow, refer to Table 1, Table 2 and Table 3.

Table 1 List of electrical flows involved in a SHC system

Electricity consumer [kWh]	Label
Heating System	
pump collector field (primary loop)	E1
pump collector field (secondary loop)	E2
pump boiler hot-storage (including internal boiler consumption)	E3
pump hot-storage to space heating (SH)	E4
pump hot-storage to domestic hot water (DHW)	E5
Cooling System	
pump hot-storage to cooling machine	E6
pump cooling machine (ACM) to cooling tower	E7
pump cooling machine (ACM) to cold-storage	E8
pump cold storage to cold distribution	E9
pump back up source - cold storage	E10
absorption/adsorption cooling machine (ACM)	E11
compression chiller (back-up system)	E12
pump compression chiller to fan (back-up system)	E13
fan; cooling tower	E14
fan of compression chiller (back-up system)	E15
Desiccant cooling/ dehumidification System	
fan exhaust air	E16
fan supply air	E17
motor for desiccant wheel	E18
motor for heat recovery wheel	E19
Water treatment System	
water treatment for wet cooling tower and humidifier for DEC	E20

Table 2 List of thermal flows involved in a SHC system

Thermal Energies [kWh]	Label
solar irradiation on total collector aperture area	Q_sol
solar thermal output to hot storage	Q1
heat output from hot storage	Q1S
boiler thermal output (fossil) into storage	Q2S_fossil
renewable energy source (RES) thermal output into storage	Q2S_RES
fossil boiler thermal input (fossil) bypassing hot storage (directly used)	Q2D_fossil
renewable heat source (RES) thermal input bypassing hot storage (directly used)	Q2D_RES
space heating (SH) consumption (conventional)	Q3a
space heating (SH) consumption (ventilation system)	Q3b
domestic hot water (DHW) consumption	Q4
hot storage input to cooling machine (ACM)	Q6a
hot storage input to DEC-system (sorption regeneration)	Q6b
cold output ACM to cold-storage	Q7
cold output back-up chiller or free cooling to cold-storage	Q8
cold storage output to cold-distribution	Q10a
cold storage to Air Handling Unit (AHU)	Q10b
Enthalpy difference - Air Handling Unit (Inlet Air => Supply Air)	ΔH_{AHU}

Table 3 List of water consumptions involved in a SHC system

Water Consumption [Liter]	Label
water consumption for wet cooling tower	V1
water consumption for DEC humidifier	V2

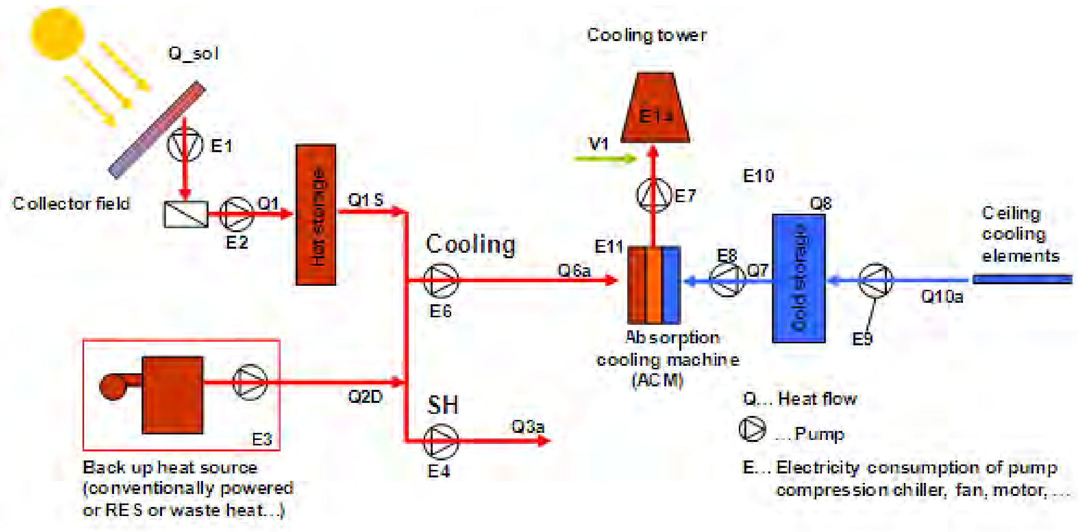


Figure 2-2 Example of a solar heating and cooling system of a specific installation based on the "SHC Max System" shown in Figure 2-1.

2.2 Selection of a standard graphical scheme for the representation of a reference system

The selected conventional system to be taken as reference for comparisons consists of a condensing natural gas boiler to match the space heating (SH) load and the domestic hot water (DHW). A small storage tank, with typical heat losses based on the measured domestic hot water consumption, has been assumed to be also heated by the natural gas boiler. Finally, the cooling load is supposed to be matched by a compression chiller.

The conventional system has to include an AHU when the monitored system is based on a DEC. In case an AHU is present in the monitored system, the conventional include it as well but it is not significant for the comparison as the distribution systems are supposed to be the same. Figure 2-3 shows the selected conventional system and the relevant energy flows. A description of each flow is shown in the Table 1, Table 2 and Table 3.

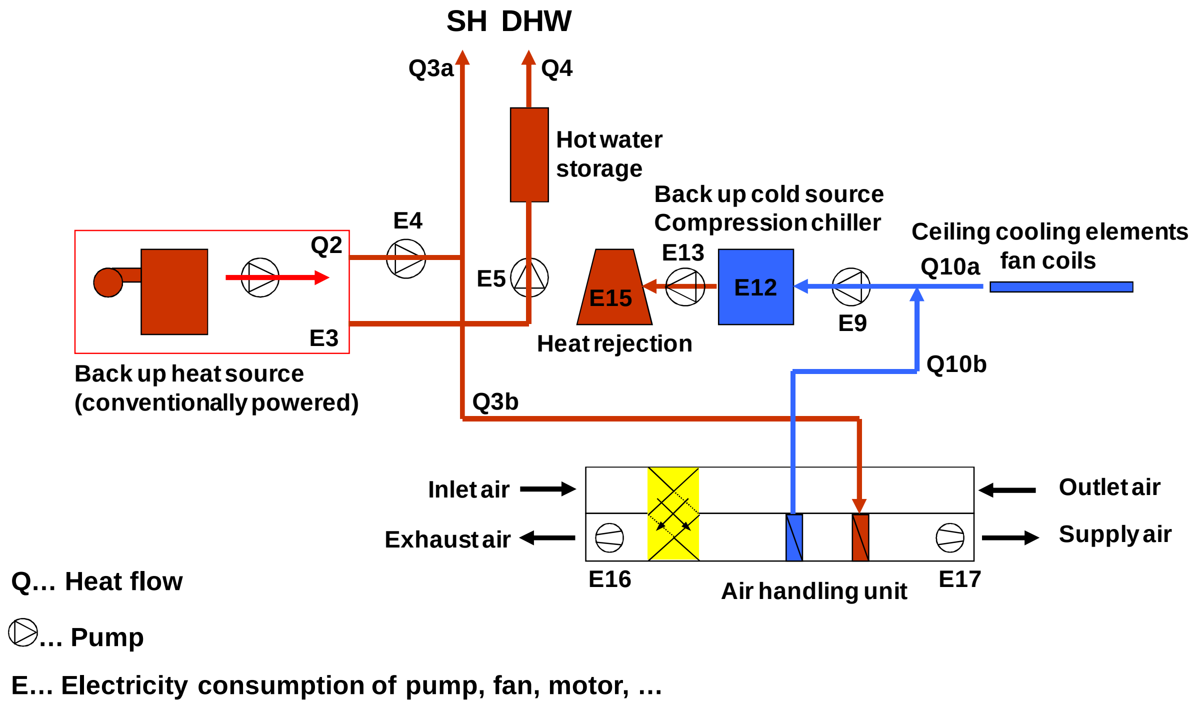


Figure 2-3 Diagram of the selected conventional reference systems including energy flows

3. Monitoring procedure

The monitoring procedure consists of an excel file where monitored data are collected and elaborated on monthly and yearly basis for both SHDC and DEC systems. For DEC systems one more excel file has been developed to collect and elaborate special data which are then integrated in the main file for a complete and correct performance evaluation.

The main excel file of the monitoring procedure consists of 3 levels:

1. First level: Basic Information on COP, Primary Energy Ratio and Costs;
2. Second level: Basic monitoring procedure mainly evaluating the solar heat management efficiency (kept simple in sense of calculation and necessary monitoring hardware);
3. Third level: Advanced monitoring procedure evaluating primary energy savings and specific COP`s of components and groups (more complex in sense of calculation and necessary monitoring hardware).

Several key figures are defined in each of the three levels and can be calculated depending on the sensors installed. In practice, each completed level of the procedure indicates a certain level of detail of the monitoring system.

3.1 Assumptions

Before going in detail with the monitoring procedure, it is necessary to list the major assumptions at the base of the monitoring data collection and elaboration.

3.1.1 Accuracy of the sensors

In order to enable a correct evaluation of the performance of a monitored solar heating and cooling system and a fair comparison between different systems, accuracy of measurements should be checked. Nevertheless, an estimation of the accuracy for the entire monitoring system installed is out of scope of the developed procedure. For this reason the tool only includes an estimation of the accuracy of the chilling power of the heat driven chiller (according to Eq. 1) and allows users to enter accuracies values for specific devices, such as signal conditioning devices, electricity energy counters and pyranometers.

It is highlighted that the tool does not set any benchmark for the accuracy values but only show some accuracies to inform about the reliability of the output data.

$$\frac{\Delta\dot{Q}}{\dot{Q}} = \sqrt{\left(\frac{\Delta\rho}{\rho}\right)^2 + \left(\frac{\Delta\dot{V}}{\dot{V}}\right)^2 + \left(\frac{\Delta c_p}{c_p}\right)^2 + 2\cdot\left(\frac{\Delta\Delta T}{\Delta T}\right)} \qquad \text{Eq. 1}$$

3.1.2 Efficiency of the conventional system

Most key figures defined in the procedure for the evaluation of the performance of the monitored system require data on energy used for heating, cooling and DHW to be converted into primary energy.

The calculation of the energy use needs the definition of the efficiency or the coefficient of performance of the applied heating, cooling and DHW systems. In practice such efficiencies have to be defined for the heat and cold backup systems and depend of course on the technology used. Typical backup systems applied in the monitored installations are natural gas boilers and compression chillers. In these cases the efficiencies considered for the backup system in the monitored installations are the same as in the conventional system.

The use of common values for all the monitored installations enables the comparison between their performance results. For this reason some reference values have been defined, such as:

- Natural gas boiler efficiency

$$\eta_{boiler} = 0.95 \quad \text{Eq. 2}$$

- Seasonal Performance Factor of vapor compression chiller

$$SPF_{ref} = 2.8 \left[\frac{kWh_{th}}{kWh_{elec}} \right] \quad \text{Eq. 3}$$

In case a boiler fired by renewable energy source (e.g. biomass) is applied, a special efficiency value can be used, labeled as:

$$\eta_{boiler\_RES} = 0.90 \quad \text{Eq. 4}$$

All the default values listed efficiencies can also be substituted with special values in relationship with a different technology applied or according to the knowledge of the users (Refer to “3.2.1 General input data”).

3.1.3 Primary energy conversion factors

On the other hand it has been necessary to fix the primary energy factors for the main energy carriers used in the installations, i.e. electricity and fossil fuels like oil and natural gas[1]. In fact it is the “fossil” primary energy factor, which means how much fossil energy was used in order to produce useful energy like thermal heat or electricity.

[1] Please note that the values selected are actually the reciprocal of the so called primary energy factors defined according to the EN 15603

$$\varepsilon_{elec} = 0.40 \left[\frac{kWh_{elec}}{kWh_{pe}}\right]$$ Eq. 5

$$\varepsilon_{fossil} = 0.90 \left[\frac{kWh_{th}}{kWh_{pe}}\right]$$ Eq. 6

Primary energy factors depend on the process beyond the delivery of the energy carriers to the building. To enable comparisons between the different monitored installations common values have been selected. However if the user are interested in making calculations according to the special factors (national, local and so on), such values can also be defined as input.(Read “3.2.1 General input data”).

The procedure also enables the definition of primary energy factors for renewable energy carriers, e.g. biomass. As base values the following can be considered:

$$\varepsilon\_RES = 10 \left[\frac{kWh_{th}}{kWh_{pe}}\right]$$ Eq. 7

One special case for the calculation of the primary energy is the utilization of cogeneration units as heat backup system.

In these cases, the calculation of the primary energy factor has to account for the electricity produced with the same fuel consumption. In practice the primary energy related with the electricity output by the prime motor is discounted from the primary energy entering the cogeneration unit, according to the formula below:

$$\varepsilon_{COG} = \frac{Q_{CHP}}{\frac{Q_{fuel}}{\varepsilon_{fuel}} - \frac{W_{el}}{\varepsilon_{elec}}} \left[\frac{kWh_{th}}{kWh_{pe}}\right]$$ Eq. 8

Where:

Q_{CHP} is the heat recovered from the engine or turbine

Q_{fuel} is the monitored input energy of the driving fuel

ε_{fuel} is the primary energy conversion factor for the driving fuel

W_{el} is the electricity output by the engine or turbine

ε_{elec} is the primary energy conversion factor for the electricity

Note that the calculated value represents the efficiency of the cogeneration unit but already includes the primary energy factor for the driving fuel.

3.2 Input data

3.2.1 General input data

General data on the system are required: name, location and final use of the installation together with sizes, technologies and units of the major SHC components (i.e. solar collectors, HDC, tanks, back-up systems).

It is needed to adapt the standard scheme shown in Figure 2-1 to the monitored system and list the energy flows which are monitored according to Table 1. On the basis of the sensors available, different level of information can be derived which are represented by the three different levels of the procedure. Then the major input data are basically the monthly values measured by each listed sensor.

As already mentioned in the previous subchapter, the efficiencies of the machines and the primary energy factors can be also entered by the users according to special needs. So additionally to the above Tables, the parameters listed in Table 4 are added.

For instance if the users know the monthly efficiency of the installed boiler, this value can be entered and used for the calculation of the monthly and yearly energy use. On the other hand if the primary energy factor of the location where the system is installed differs from the default value, it can also be entered.

Table 4: Parameters which can be entered by the users

Parameters	Label
annual electricity generation efficiency	ε_{elec}
seasonal performance factor of the reference compression chiller	SPF_{ref}
mean annual / monthly efficiency of the auxiliary boiler	η_{boiler}
primary energy factor for fossil fuel	ε_{fossil}
mean annual / monthly efficiency of the RES device	η_{RES}
primary energy factor for RES fuel	ε_{RES}
mean annual / monthly efficiency of the reference boiler	$\eta_{boiler,ref}$

3.2.2 Special input data required for DEC systems

In the case of DEC systems further data are needed, especially for the 1° and the 3° level. Inputs for the 3° level partly have to be calculated in an extra excel file (110801_T38_MonProc_V6-0_DH-calc.xls) which has been developed to calculate monthly values based on high resolution monitoring data (e.g. 5 minutes):

- reference electricity consumption of the AHU (necessary for the 1° and the 3° level);
- reference enthalpy difference of the AHU supply air mass flow (3° level);

- reference enthalpy difference obtained from single components indoors the conventional AHU, such as dehumidification coil, post-cooling coil, post-heating coil (3° level).

For this reason the following monitoring data in reasonable time steps (5 to 10 minutes are recommended) from the DEC AHU are required to be able to do the calculations in this excel tool:

1. Heating is ON or OFF [1 or 0]
2. Cooling is ON or OFF [1 or 0]
3. Inlet Air Temperature [°C]
4. Inlet Air relative Humidity [%]
5. Supply Air Temperature [°C]
6. Supply Air relative Humidity [%]
7. Outlet Air Temperature [°C]
8. Supply Air Flow [kg/h]

Additionally the following parameters need to be defined:

1. The supply air temperature of the reference AHU: "t supply max conv." [°C]; as standard 20°C is set.
2. The heat recovery ratio of the reference heat recovery wheel: "Reference Heat Recovery"; as standard 75% is set.
3. Minimum supply air flow rate [kg/h] which indicates if the AHU is in operation or not. This value is only used if in the monitoring data no status value for "Heating is ON or OFF" and "Cooling is ON or OFF" is given.

The calculated values are finally presented in the worksheet "introduction" in a light green marked field which can directly be transferred to the main excel tool by copy and paste as explained in the tool.

3.3 Monitoring data elaboration and assessment

3.3.1 General key energy performance figures in the 1st monitoring level

In the first level, an overall evaluation for the entire system is done by calculating coefficients of performances (COP) and primary energy ratios (PER) on both monthly and yearly basis.

The inputs necessary for the achievement of this level are:

Table 5 Monitored data necessary for the completion of the 1st monitoring level

Electricity consumer [kWh]	Label
Overall electricity consumption of the energy facility (the overall and only sensors listed in Table 1 are included)	Eelec,overall

Heating System	
pump hot-storage to space heating (SH)	E4
pump hot-storage to domestic hot water (DHW)	E5
Cooling System	
pump cold storage to cold distribution	E9
Desiccant cooling/ dehumidification System	
fan exhaust air (eventually fan regeneration air)	E16
fan supply air	E17

Thermal Energies [kWh]	Label
boiler thermal output (fossil) into storage	Q2S_fossil
renewable energy source (RES) thermal output into storage	Q2S_RES
fossil boiler thermal input (fossil) bypassing hot storage (directly used)	Q2D_fossil
renewable heat source (RES) thermal input bypassing hot storage (directly used)	Q2D_RES
space heating (SH) consumption	Q3a
domestic hot water (DHW) consumption	Q4
cold storage output to cold-distribution	Q10a
Enthalpy difference - Air Handling Unit (Inlet Air => Supply Air)	ΔH_{AHU}

The *total electrical COP* ($COP_{el,tot}$) calculates the ratio of useful heat and/or cold in relation to the electricity consumption needed but excludes the electrical consumption of pumps and fans which are used to distribute heat and/or cold in the building by pumping water or blowing air (i.e. the so labelled Eelec,tot = Eelec,overall–E4-E5-E9-E16-E17).

$$COP_{el,tot} = \frac{Q3a + Q10a + Q4 + \left|\Delta H_{AHU}\right|}{Eelec,tot} \quad \left[\frac{kWh_{th}}{kWh_{elec}}\right] \qquad \text{Eq. 9}$$

The *overall electrical COP* (COPel,overall) includes also all the electrical consumers for distribution (i.e. the so labelled Eelec,overall)

$$COP_{el,overall} = \frac{Q3a + Q10a + Q4 + \left|\Delta H_{AHU}\right|}{Eelec, overall} \quad \left[\frac{kWh_{th}}{kWh_{elec}}\right] \qquad \text{Eq. 10}$$

At this monitoring level, to calculate the Eelec,overall is not needed to have all the relevant electricity meters installed, but just one meter on the overall consumption of the facility. To calculate the Eelec,tot is then needed at least one overall meter and the measurement of the electrical consumption of the distribution pumps (E4, E5 and E9). In case the monitored system includes a conventional AHU, this is considered as a distribution system and the electricity consumed for fans (E16 and E17) should be subtracted from Eelec,overall too. If the monitored system includes a DEC AHU, only the additional electricity due to special DEC components (see chapter 3.3.2) should be considered in Eelec_overall. So further measurements are needed, or at least pressure drops of the single AHU components must be known to be able to calculate the nominal pressure drops of the DEC AHU and a Reference AHU resulting in a correction factor which is used to estimate the additional electricity consumption of the fans due to the DEC components.

Primary energy ratio is calculated for the monitored installation as the ratio of useful heat and/or cold in relation to the primary energy demand.

$$PER\_res = \frac{Q3a + Q10a + Q4 + \left|\Delta H_{AHU}\right|}{\frac{Q2\_fossil\_tot}{\varepsilon_{fossil} * \eta_{boiler}} + \frac{Q2\_RES\_tot}{\varepsilon\_RES * \eta_{boiler\_RES}} + \frac{Eelec, tot}{\varepsilon_{elec}}} \quad \left[\frac{kWh_{th}}{kWh_{pe}}\right] \qquad \text{Eq. 11}$$

Where:

$$Q2\_fossil\_tot = Q2S\_fossil + Q2D\_fossil \qquad \text{Eq. 12}$$

$$Q2\_RES\_tot = Q2S\_RES + Q2D\_RES \qquad \text{Eq. 13}$$

Note that ΔH_{AHU} has only to be included when the monitored system is based on DEC. In case the monitored system includes a conventional AHU, this is considered as a distribution system and Q3a actually includes the heat which is then transferred to all the distribution system, AHU included.

In case back up systems based on renewable energy are installed, the procedure can also evaluate the impact of such RES compared to a conventional system according to:

$$PER\_fossil = \frac{Q3a + Q10a + Q4 + \left|\Delta AHU\right|}{\frac{Q2\_fossil\_tot}{\varepsilon_{fossil} * \eta_{boiler}} + \frac{Q2\_RES\_tot}{\varepsilon_{fossil} * \eta_{boiler\_RES}} + \frac{Eelec,tot}{\varepsilon_{elec}}} \quad \left[\frac{kWh_{th}}{kWh_{pe}}\right] \quad \text{Eq. 14}$$

In comparison with Eq. 11, Eq. 14 considers Q2 produced from RES as produced by fossil fuel but with the same efficiency of RES technology used. In this way the influence of the much more advantageous primary energy factor for RES is eliminated and the RES-system can be compared with a fossil-system based on the same fuel type (but of course different boiler efficiency is still a weak point which should be kept in mind).

Additionally a PER is calculated also for the supposed conventional system according to:

$$\mathrm{PERref} = \frac{\mathrm{Q3a} + \mathrm{Q10a} + \mathrm{Q4} + \left|\Delta \mathrm{AHU}\right|}{\frac{\mathrm{Q_{boiler,ref}}}{\varepsilon_{\mathrm{fossil}} * \eta_{\mathrm{boiler}}} + \frac{\mathrm{Q_{cooling,ref}}}{\mathrm{SPF_{ref}} * \varepsilon_{\mathrm{elec}}} + \frac{\mathrm{E_{el,ref}}}{\varepsilon_{\mathrm{elec}}}} \quad \left[\frac{\mathrm{kWh_{th}}}{\mathrm{kWh_{pe}}}\right] \quad \text{Eq. 15}$$

The so called Qboiler,ref not only takes into account the heat load of the monitored installation for heating and DHW purposes but of course includes also the possible heat losses of the storage for DHW in the reference system according to IEA SHC Task26: ENV 12977-1 (2000); (Weiss, Ed. 2003). So, it is defined as:

Eq. 16

$$\mathrm{Q_{boiler,ref}} = \mathrm{Q3a} + \mathrm{Q3b} + \mathrm{Q4} + \mathrm{Q_{loss\_ref}} \quad [\mathrm{kWh_{th}}]$$

Where:

Eq. 17

$$\mathrm{Q_{loss\_ref}} = 0.00016 * \sqrt{0.75 * \mathrm{V_D}} * (\mathrm{T_T} \_ \mathrm{Ta}) * 8760 \quad [\mathrm{kWh_{th}}]$$

And:

V_D : average daily hot water consumption (liter / day)

T_T : set point temperature of the hot water tank (default 52.5 °C)

T_a : ambient temperature around the hot water tank (default 15 °C) Note that Q3b has to be entered only when the monitored system is based on DEC. In case the monitored system includes a conventional AHU, this is considered as a distribution system and Q3a actually includes the heat which is then transferred to all the distribution system, AHU included.

The so called Eel,ref, is the electrical consumption of the conventional system which includes the consumption for operating the boiler and the pump between the boiler and the storage. This is the only item which differs from the electrical consumption of a SHDC in the heating mode. The electricity consumed by a conventional system in the cooling mode is taken into account in the SPF_{ref}.

In case the monitored system is based on a DEC AHU, the Eel,ref includes the electrical consumption of the corresponding AHU as described in subchapter 3.3.4.

For conventional systems to be compared to SHDC systems:

Eq. 18

$$E_{el\_ref} = 0.02 * (Q3a + Q3b + Q4 + Q_{loss\_ref}) \quad [kWh_{elec}]$$

The value 0.02 kWhel/kWhth is an assumption to be taken into consideration for small boilers and in case no real value is available (low level of detail of monitoring). In a very detailed monitoring, one more calculation procedure has been defined as described later.

The so-labeled Qcooling,ref is the cooling supplied to the load, (i.e. Q10a) and in a conventional system is entirely supplied by a compression chiller. In the 1st monitoring level it is enough to monitor Q10a and assume a proper SPF which takes into account all the relevant electrical consumption, i.e., with reference to Figure 2-3, not only E12 but also E13 and E15, e.g. the default value of SPF_{ref}. In a very detailed monitoring, one more calculation procedure has been defined to derive Qcooling,ref which is described later.

At the end of the 1st monitoring level one economic figure, the Cost Per kW, is calculated according to:

Eq. 19

$$CPK = \frac{\text{Cost of total cooling system}}{\text{Cooling capacity installed}} \quad \left[\frac{k€}{kW}\right]$$

3.3.2 General key energy performance figures in the 2nd monitoring level

In the second level mainly the quality of the subsystems solar thermal heat production and heat management within the system are evaluated.

The necessary measurements only concern thermal flows (Table 6).

Table 6 Necessary measurements for the completion of the second monitoring level

Thermal Energies [kWh]	Label
solar irradiation on total collector aperture area	Q_sol
solar thermal output to hot storage	Q1
heat output from hot storage	Q1S
boiler thermal output (fossil) into storage	Q2S_fossil
renewable energy source (RES) thermal output into storage	Q2S_RES
fossil boiler thermal input (fossil) bypassing hot storage (directly used)	Q2D_fossil
renewable heat source (RES) thermal input bypassing hot storage (directly used)	Q2D_RES
space heating (SH) consumption	Q3a
space heating (SH) consumption (ventilation system)	Q3b
domestic hot water (DHW) consumption	Q4
hot storage input to cooling machine (ACM)	Q6a
hot storage input to DEC-system (sorption regeneration)	Q6b

First, the *efficiency of solar collectors* is estimated by means of:

$$\eta\text{coll, util} = \frac{\text{Q1}}{\text{Qsol}} \qquad \text{Eq. 20}$$

Hence losses in the collectors are given by:

$$\Delta\text{Qsol} = \text{Qsol} - \text{Q1} \quad [\text{kWh}_{\text{th}}] \qquad \text{Eq. 21}$$

The solar energy collected per square meter of the collector:

$$Qcoll\_yield = \frac{Q1}{Collector\ \ Aperture\ Area} \quad \left[\frac{kWh}{m^2}\right] \qquad \text{Eq. 22}$$

Usually the heat acquired by the solar collectors is stored in buffer tanks before being used.

In some cases the hot buffer is shared with the heat backup system and it represents the only connection point between the heat sources and the load (Figure 3-1).

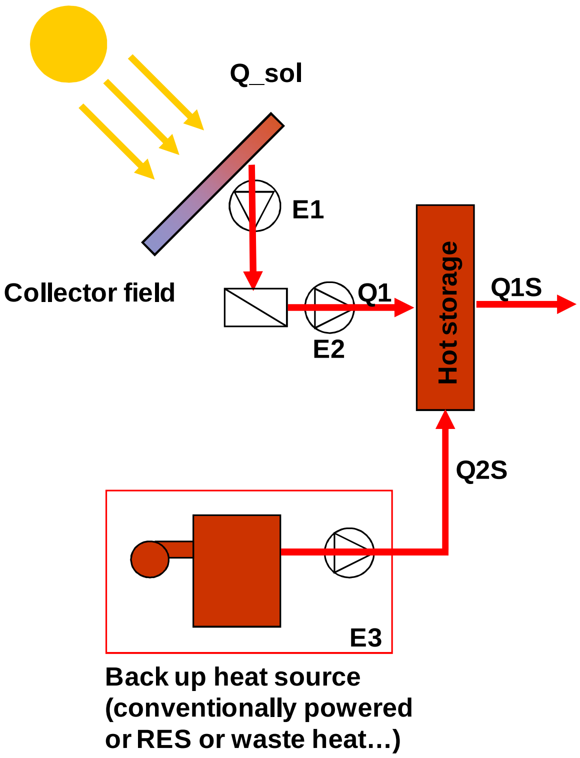


Figure 3-1 Connection between the heat back-up system and the solar collectors: shared hot storage

This means that:

$$Q2D\_fossil = 0 \quad [kWh_{th}]$$ Eq. 23

$$Q2D\_RES = 0 \quad [kWh_{th}]$$ Eq. 24

$$\begin{aligned} Q2\_total\,sum &= Q2S\_fossil + Q2D\_fossil + Q2S\_RES + Q2D\_RES \\ &= Q2S\_fossil + Q2S\_RES = Q2S \end{aligned}$$ Eq. 25

In these cases, as the output of the storage includes both the input from solar and back-up system, the *storage solar fraction*, defined as the ratio between the heat collected by the solar collectors and the heat produced by the backup system, is calculated according to:

$$SF_{hotstorage} = \frac{Q1}{Q1 + Q2S} \qquad \text{Eq. 26}$$

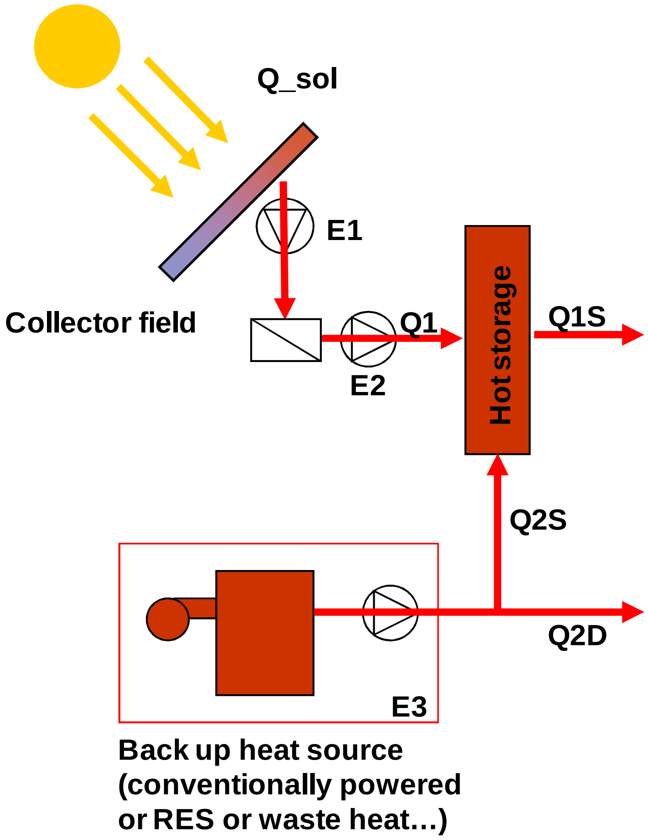


Figure 3-2 Connection between the heat back-up system and the solar collectors: direct use of the heat supplied by the back-up system

If the heat delivered by the back-up system is also directly used (Figure 3-2), the output from the storage due to the solar collectors is given by:

$$Q1s * \left(\frac{Q1}{Q1 + Q2S} \right) \ [kWh_{th}] \qquad \text{Eq. 27}$$

Hence the storage solar fraction amounts to:

Fehler! Es ist nicht möglich, durch die Bearbeitung von Feldfunktionen Objekte zu erstellen. Eq. 28

Note that in cases such as Figure 3-2 :

$$Q2\_total\,\mathrm{sum} = Q2S\_\,fossil + Q2D\_\,fossil + Q2S\_RES + Q2D\_RES$$
$$= Q2\_\,fossil + Q2\_RES = Q2S + Q2D \qquad \text{Eq. 29}$$

One special application of the Eq. 28 is when the heat from the back up source is only directly used, i.e.

$$\mathrm{Q2S\_fossil + Q2S\_RES = Q2S = 0}\quad [\mathrm{kWh_{th}}] \qquad \text{Eq. 30}$$

Under Eq. 30, Eq. 28 becomes:

$$\mathrm{SF_{hotstorage}} = \frac{\mathrm{Q1s}}{\mathrm{Q1 + Q2D}} \qquad \text{Eq. 31}$$

The losses through the buffer tank are calculated as the difference between the input and the output, if the latter is known.

$$\mathrm{Q\_loss\_stge = (Q1 + Q2S\_fossil + Q2S\_RES) - (Q1S)}\quad [\mathrm{kWh_{th}}] \qquad \text{Eq. 32}$$

In case, the output of the storage tanks is not known and the buffer tank is shared with the heat back-up system, the <u>*efficiency of the storage*</u> can be estimated as follows:

$$\mathrm{Q\_loss\_stge = (Q1 + Q2S\_fossil + Q2S\_RES) - (Q3a + Q4 + Q3b + Q6)}\quad [\mathrm{kWh_{th}}] \qquad \text{Eq. 33}$$

In the considered case, Q_loss_stge coincides with the overall losses in the system, which are given by:

Eq. 34

$$Q\_loss\_sys = (Q1 + Q2\_total\,sum) - (Q3a + Q4 + Q3b + Q6) \quad [kWh_{th}]$$

On the basis of the losses calculated in Eq. 33 and Eq. 34, the efficiency of the storage and the *efficiency of the system* are calculated according to:

$$\eta stge = \frac{(Q1 + Q2S\_fossil + Q2S\_RES) - Q\_loss\_stge}{(Q1 + Q2S\_fossil + Q2S\_RES)}$$

Eq. 35

$$\eta sys = \frac{(Q1 + Q2\_total\,sum) - Q\_loss\_sys}{(Q1 + Q2S\_fossil + Q2S\_RES)}$$

Eq. 36

The *solar heat management efficiency* is defined as the rate between the "solar load contribution" and the solar availability (Eq. 41).

The *solar load contribution* refers to the amount of load energy which is fully covered by solar energy. I.e. the *solar space heating contribution* is given by:

Eq. 37

$$Q3^{*} = SF_{hotstorage} * (Q3a + Q3b) \quad [kWh_{th}]$$

The *solar DHW contribution* is:

Eq. 38

$$Q4^{*} = SF_{hotstorage} * Q4 \quad [kWh_{th}]$$

The *solar cooling contribution* is:

$$Q6^* = SF_{hotstorage} * Q6 \quad [kWh_{th}]$$ Eq. 39

On the basis of Eq. 37, Eq. 38 Eq. 39, the *total solar load contribution* is given by:

$$Qtot^* = Q3^* + Q4^* + Q6^* \quad [kWh_{th}]$$ Eq. 40

Hence the *solar heat management efficiency* is calculated according to:

$$\eta heat\_solrad = \frac{Qtot^*}{Qsol}$$ Eq. 41

This means that the *solar energy unexploited* is:

$$Q\_solunex = Qsol - Qtot^*$$ Eq. 42

3.3.3 General key energy performance figures in the 3rd monitoring level

In the third level all the flows listed in Table 1, Table 2 and Table 3 are required to carry out deep analysis.

First of all, the *fractional energy saving* (Fsav) is calculated according to the method which was elaborated in the IEA SHC Task 26 for solar combisystems and extended in the IEA SHC Task 32 for solar heating and cooling systems.

This method requires the PER of a conventional system to be calculated according to the formula already presented in the 1st level:

$$PERref = \frac{Q3a + Q10a + Q4 + |\Delta AHU|}{\frac{Q_{boiler,ref}}{\varepsilon_{fossil} * \eta_{boiler}} + \frac{Q_{cooling,ref}}{SPF_{ref} * \varepsilon_{elec}} + \frac{E_{el,ref}}{\varepsilon_{elec}}} \quad \left[\frac{kWh_{th}}{kWh_{pe}}\right]$$ Eq. I

In comparison with the 1st level, in the 3rd level the calculation of some items included in the PERref is enabled in different ways thanks to the presence of many sensors.

For instance, the calculation of Eel,ref can be more accurate thanks to the presence of the sensors E3. In fact, with reference to the Eq. 18 below repeated, the assumed 0.02 kWhel/kWhth can be substituted with a real measurements of the electrical consumption for operating the boiler and the pump between the boiler and the storage. Such real value is given by the ratio between E3 and the overall production of the boiler.

Eq. 43

$$E_{el\_ref} = 0.02 * (Q3a + Q3b + Q4 + Q_{loss\_ref}) \quad [kWh_{elec}]$$

In case of DEC-AHU, it must be considered the additional pressure drops of DEC-components compared to a conventional AHU.

$$E_{ref}^{vent,el} = E_{SHC}^{DEC,el} \cdot \frac{\left(\Delta P_{supply}^{REF} \cdot \dot{V}_{supply} + \Delta P_{return}^{REF} \cdot \dot{V}_{return}\right)}{\left(\Delta P_{supply}^{DEC} \cdot \dot{V}_{supply} + \Delta P_{return}^{DEC} \cdot \dot{V}_{return}\right)} = E_{SHC}^{DEC,el} \cdot f\left(\Delta P^{REF}, \Delta P^{DEC}\right)[kWh_{elec}]$$

One more instance is the calculation of the so-labeled Qcooling,ref. It can be calculated in a as follows:

Eq. 44

$$Q_{cooling,ref} = Qcooling, missed + Q7 \quad [kWh_{th}]$$

"Qcooling,missed" refers to the cold production of the back-up system of the monitored system, i.e. Q8. This procedure enables to consider in the conventional system the real cold produced by the monitored installation, including losses in the cold tank. If the electricity consumotion of the complete back-up system is not measured, the reference SPF (=2.8) is used to calculate the electricity consumption accordingly.

On the other hand, the electrical consumption due to Qcooling,missed can be directly calculated by summing up all the electrical consumptions of the back-up system (pumps and fan included), i.e. E10, E12, E13 and E15 if monitored directly. This enables to have the real electrical consumption of the back up system without assuming for it any SPF.

For DEC systems the calculation of Qcooling,missed is based on a different concept, that will be explained in the paragraph 3.3.4.

Once the PER,ref is calculated, the fractional saving which evaluates the consumption of the monitored system compared to a conventional system is given by:

$$\text{fsav.shc} = 1 - \frac{\dfrac{Q_{\text{boiler}}}{\varepsilon_{\text{fossil}} \cdot \eta_{\text{boiler}}} + \dfrac{Q_{\text{RES}}}{\varepsilon_{\text{RES}} \cdot \eta_{\text{RES}}} + \dfrac{E_{\text{el}}}{\varepsilon_{\text{elec}}} + \dfrac{Q_{\text{cooling,missed}}}{\text{SPF} \cdot \varepsilon_{\text{elec}}}}{\dfrac{Q_{\text{boiler,ref}}}{\varepsilon_{\text{fossil}} \cdot \eta_{\text{boiler,ref}}} + \dfrac{E_{\text{el},\text{ref}}}{\varepsilon_{\text{elec}}} + \dfrac{Q_{\text{cooling,ref}}}{\text{SPF}_{\text{ref}} \cdot \varepsilon_{\text{elec}}}} \quad \text{Eq. 45}$$

Which is equivalent to:

$$\text{fsav.shc} = 1 - \frac{\text{PER}_{\text{ref}}}{\text{PER}_{\text{RES}}} \quad \text{Eq. 46}$$

If it is meant to exclude the effect of the RES, the fractional saving can be also calculated considering the PERfossil of the monitored installation.

$$\text{fsav.shc} = 1 - \frac{\dfrac{Q_{\text{boiler}}}{\varepsilon_{\text{fossil}} \cdot \eta_{\text{boiler}}} + \dfrac{Q_{\text{RES}}}{\varepsilon_{\text{fossil}} \cdot \eta_{\text{RES}}} + \dfrac{E_{\text{el}}}{\varepsilon_{\text{elec}}} + \dfrac{Q_{\text{cooling,missed}}}{\text{SPF} \cdot \varepsilon_{\text{elec}}}}{\dfrac{Q_{\text{boiler,ref}}}{\varepsilon_{\text{fossil}} \cdot \eta_{\text{boiler,ref}}} + \dfrac{E_{\text{el},\text{ref}}}{\varepsilon_{\text{elec}}} + \dfrac{Q_{\text{cooling,ref}}}{\text{SPF}_{\text{ref}} \cdot \varepsilon_{\text{elec}}}} = 1 - \frac{\text{PER}_{\text{ref}}}{\text{PER}_{\text{fossil}}} \quad \text{Eq. 47}$$

The ***fractional solar consumption*** is defined as the ratio between the available and useful solar radiation (I [kW/m^2]) on the aperture area (A [m^2]) of the collector under investigation (Qsol = I x A [kWh]) and the reference heat necessary to match the space heating, DHW and cooling demand on a monthly basis and summed up for a year:

$$FSC\_SHC = \sum_{m=1}^{12} \left(\frac{Min(Q_{ref}, Q_{sol})}{Q_{ref}} \right)_m \quad \text{Eq. 48}$$

Where Q_{ref} the monthly sum of the heat needed for different purposes. Please note that to calculate the heat needed by a thermally driven chiller is calculated with a $COP_{th,ref}$ of 0.6 as default.

$$Q_{ref} = \frac{Q_{SH} + Q_{DHW} + Q_{loss,ref}}{\eta_{boiler,ref}} + \frac{\mathrm{Q_{cooling,ref}}}{COP_{th,ref}} \quad [\mathrm{kWh_{th}}]$$ Eq. 49

The 3rd level is also focusing on the water consumption for cold production. The water consumption of the wet cooling tower and the DEC system is evaluated in terms of:

- Cooling tower water consumption per kWh dissipated heat

$$\mathrm{ACM\_water} = \frac{\mathrm{V1}}{\mathrm{Q6a + Q7}} \quad \left[\frac{\mathrm{l}}{\mathrm{kWh_{th}}}\right]$$ Eq. 50

- DEC humidifier water consumption per kWh COLD production

$$\mathrm{DEC\_water} = \frac{\mathrm{V2}}{\Delta\mathrm{H_{AHU\_cool}}} \quad \left[\frac{\mathrm{l}}{\mathrm{kWh_{th}}}\right]$$ Eq. 51

On the other hand the electrical consumption related to the treatment of the water is calculated in terms of:

- Electricity consumption for water treatment per liter treated water in SHDC systems

$$\mathrm{El\_treat\_water} = \frac{\mathrm{E20}}{\mathrm{V1 + V2}} \quad \left[\frac{\mathrm{kWh_{el}}}{\mathrm{l}}\right]$$ Eq. 52

- Electricity consumption for water treatment per kWh cold production in SHDC and DEC systems:

$$El\_treat\_water = \frac{E20}{\Delta H_{AHU\_cool} + Q7} \left[\frac{\mathrm{kWh_{el}}}{\mathrm{kWh_{cold}}}\right]$$ Eq. 53

Several specific defined COP's evaluating the thermal and the electric performance of the subsystems sorption chiller and DEC unit are calculated.

The *thermal and electrical COP* of the chiller itself is evaluated according to:

$$COP\_el\_chill = \frac{Q7}{E11} \left[\frac{\mathrm{kWh_{th}}}{\mathrm{kWh_{elec}}}\right]$$ Eq. 54

$$COP\_th\_chill = \frac{Q7}{Q6a}$$ Eq. 55

Then the cold production is rated to the sum of all the electricity consumptions relevant with the heat driven chiller (HDC) (Eq. 56), i.e. pump hot-storage to cooling machine, pump cooling machine (ACM) to cooling tower, pump cooling machine (ACM) to cold-storage, absorption/adsorption cooling machine (ACM) and cooling tower:

$$\mathrm{COP\_el\_cold\_chill} = \frac{\mathrm{Q7}}{\mathrm{E6+E7+E8+E11+E14}} \left[\frac{\mathrm{kWh_{th}}}{\mathrm{kWh_{elec}}}\right]$$ Eq. 56

One more COP includes also the electricity to run the solar pumps according to:

$$COP\_el\_cold\_chill\_Sol = \frac{Q7}{E1+E2+E6+E7+E8+E11+E14} \left[\frac{\mathrm{kWh_{th}}}{\mathrm{kWh_{elec}}}\right]$$ Eq. 57

For DEC systems following COP-values are interesting during cooling operation months:

$$COP\_el\_cold\_DEC = \frac{\Delta H_{AHU\_cool}}{(E16+E17)\cdot(1-f(\Delta P^{REF};\Delta P^{DEC})+E18+E19} \left[\frac{\mathrm{kWh_{th}}}{\mathrm{kWh_{elec}}}\right]$$ Eq. 58

Where:

- ΔH_{AHU_cool} is the Cooling Energy from AHU, without auxiliary (conventional) contribution
- $(E16+E17)\cdot(1-f(\Delta P^{REF};\Delta P^{DEC})$ is the Electricity Consumption for fans (only the additional part for DEC relevant components, like: exhaust air humidifier, sorption wheel, exhaust air regeneration heat exchanger,…)
- $E18+E19$ is the electricity consumption for motors of desiccant wheel and other additional DEC-components

To take in consideration also additional PE-consumption for solar components, further is defined:

$$COP\_el\_cold\_DEC\_Sol =$$

Eq. 59

$$= \frac{\Delta H_{AHU\_cool}}{(E16+E17)\cdot(1-f(\Delta P^{REF};\Delta P^{DEC})+E18+E19+E1+E2} \left[\frac{\mathrm{kWh_{th}}}{\mathrm{kWh_{elec}}}\right]$$

With:

- E1 is the electricity consumption of the pump in the primary loop of the collector field.
- E2 is the electricity consumption of the pump in the secondary loop of the collector field.

Finally the thermal COP of the DEC-AHU has to be calculated:

$$COP\_th\_DEC = \frac{\Delta H_{AHU\_DEC}}{Q6a}$$ Eq. 60

Where Q6a is the regeneration heat for the sorption wheel coming from the solar tank.

3.3.4 Special energy performance figures for DEC systems in the 3rd monitoring level

In case the monitored system includes a DEC-AHU, the comparison to a conventional AHU requires an in-depth consideration of both the air-treatment processes.

Savings in terms of Primary Energy are obtained in a DEC-Process through:

- Avoiding the cooling of the air until below the dew-point for dehumidification
- Avoiding the necessary post-heating of the air to reach required inlet-temperature

On the other hand, inlet-temperature of supply air from the DEC-AHU can be higher than in a conventional AHU, it means the DEC-AHU delivers less cooling energy than conventional AHU

Further important advantage of a DEC AHU can be the possibility of humidity recovery during winter time. This can lead to quite high energy savings compared to conventional AHU, of course strong depending on the specific climatic zone.

In order to make easier and more comparable the results of single plants, an excel-tool has been created, where measured data can be directly inserted (in same time-step as measured), and the enthalpy differences reached in DEC-AHU and reference AHU will be calculated for a month.

For a better validation of possible scenarios, 4 cases are considered:

1. with post-heating of supply air in the Reference AHU after the cooling to the dew-point for dehumidification up to a FIXED supply temperature (to be chosen individually for every plant) (figure 3-3)

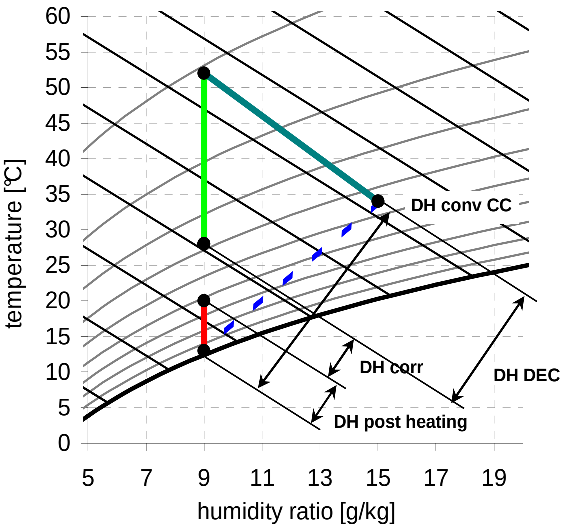


Figure 3-3 Air-treatment in DEC-AHU (green line) and in conventional AHU (blue-red line); e.g.: reference inlet temperature = 20°C

2. with post-cooling of supply air in the DEC AHU to the IDENTICAL supply temperature as FIXED for the reference AHU. (light green line down to the top end of the red line in Figure 3-3)

3. with post-heating of supply air in the Reference AHU after the cooling to the dew-point for dehumidification up to the IDENTICAL supply temperature as measured in the DEC-AHU

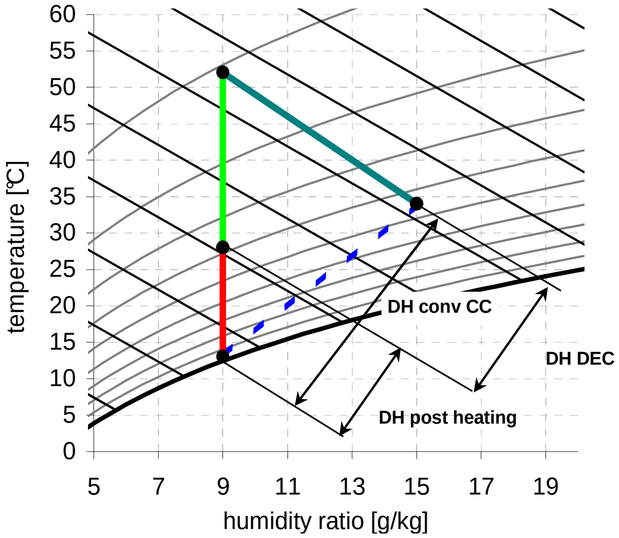


Figure 3-4 Air-treatment in DEC-AHU (green line) and in conventional AHU (blue-red line); post heating up to the identical supply temperature as measured in the DEC-AHU

4. without post-heating. It is assumed that in the conventional AHU the heat demand for post heating is for free because any kind of waste heat is used (e.g. from CHP plant or using an additional heat recovery wheel or using the waste heat of the vapor compression chiller).

Necessary Input-data for every time-step are shown in Table 7.

Table 7 Necessary measurements for the calculation of the enthalpy differences (DH)

Description of data input	**Label**
Heating mode	Heating ON/OFF
Cooling mode	Cooling ON/OFF
Temperature of inlet air in the AHU (external air)	T inlet
Relative humidity of inlet air in the AHU (external air)	rH inlet
Temperature of supply air from the AHU	T supply
relative humidity of supply air from the AHU	rH supply
Supply air flow,	Supply air flow

Besides the inputs shown in Table 7, one parameter has to be set: the supply temperature of reference AHU if it is considered as fixed.

The values output by this spreadsheet are listed inTable 8.

Table 8: Output of the spreadsheet for the calculation of enthalpy differences (DH)

Description of data output	**Label**
Difference between cooling energy delivered from DEC AHU and from conventional AHU with fixed supply temperature, m*DH AHU corr	ΔH_{corr}
Difference of enthalpy (cooling energy) achieved from cooling coil in conventional AHU, m*DH AHU CC	ΔH_{conv_CC}
Difference of enthalpy (heating energy) achieved from heating coil in conventional AHU, m*DH postheat	$\Delta H_{postheat}$
Difference of enthalpy between inlet (external) air and outlet (supply) air of conventional AHU, m*DH AHU conv	ΔH_{conv_AHU}
m*DH AHU DEC cooling	ΔH_{AHU_cool}
m*DH AHU DEC heating	ΔH_{AHU_heat}

These outputs are the special input on the monitoring procedure for DEC systems.

On their basis the *Primary Energy Ratio for conventional AHU (PER REF)* can be calculated as follows:

$$PER_{ref\_DEC} = \frac{\Delta H_{conv\_AHU}}{\frac{\Delta H_{conv\_CC}}{SPF \cdot \varepsilon_{el}} + \frac{\Delta H_{postheat}}{\eta_{boiler} \cdot \varepsilon_{fossil}} + \frac{E_{elec,conv\_AHU}}{\varepsilon_{elec}}} \left[\frac{kWh_{th}}{kWh_{elec}}\right] \qquad \text{Eq. 61}$$

The Primary Energy Saving factor f_{save} is calculated as for all Solar Heating and Cooling systems with PER_{RES} or PER_{fossil}:

$$f\,sav,shc = 1 - \frac{PER_{ref}}{PER_{RES\,/\,fossil}} \qquad \text{Eq. 62}$$

3.4 Summary of the output of the procedure

The monitoring procedure supports the evaluation of the performance of the monitored system and its comparison with a conventional system. To this purposes, three level of data elaboration have been defined depending on the kind of monitored energy flows. The major output of each level is summarized in the following tables.

Table 9 Summary of the output of the 1st monitoring level

<table>
<tr><th colspan="2">First monitoring level</th><th rowspan="2">Description</th></tr>
<tr><th>SHDC</th><th>DEC</th></tr>
<tr><td colspan="2">COPel, tot</td><td>Ratio between load and the electrical consumption excluding distribution</td></tr>
<tr><td colspan="2">COPel, overall</td><td>Ratio between load and the electrical consumption including distribution</td></tr>
<tr><td>PER_res</td><td>PER_res with DEC postcooling, Tsupply fixed to: xx°C</td><td>Primary energy ratio for the monitored system accounting for RES contribution</td></tr>
<tr><td colspan="2">PER_fossil</td><td>Primary energy ratio for the monitored system accounting RES contribution as fossil</td></tr>
<tr><td rowspan="3">PER_ref</td><td>PER_ref with ref_AHU postheating, Tsupply fixed to: xx°C</td><td rowspan="3">Primary energy ratio for the conventional system set as reference for comparisons</td></tr>
<tr><td>PER_ref_AHU postheating, Tsupply = measured in DEC</td></tr>
<tr><td>PER_ref without ref_AHU postheating</td></tr>
</table>

Table 10 Summary of the output of the 2nd monitoring level

Second monitoring level	Description
ηcoll,util	Collectors efficiency
ηstge	Storage efficiency
ηsys	System efficiency (taking into account the heat losses in all the system)
ηheat_solrad	Solar heat management efficiency
Q_solunex	Solar energy unexploited
Q_tot, heat	Total heat load

Table 11 Summary of the output of the 3rd monitoring level

Third monitoring level		Description
SHDC	DEC	
Fsav,fossil or Fsav,res		Fractional primary energy saving of the monitored system accounting for RES or fossil as it is existing compared with a conventional system For DEC systems 5 different versions are calculated.
Fsav,fossil		Fractional primary energy saving of the monitored system accounting RES as fossil compared with a conventional system; (artificial number for comparison reasons) For DEC systems 5 different versions are calculated as described before.
FSC_SHC		Fractional solar consumption of the available solar energy compared to the overall heat load
ACM_water, y	DEC_water, y	Water consumption
Water_treat, y		Electricity consumption for water treatment
COP_el_chill		Electrical COP of the chiller
COP_el_coldSol,y	COP_el_DEC-Sol,y	Electrical COP of the heat driven cooling system
COP_el_cold,y	COP_el_DEC,y	Electrical COP of the heat driven cooling system plus solar collectors
COP_th_chill,y	COP_th_DEC,y	Thermal COP of the heat driven cooling system

For a visual representation of the results the following kind of graph has been used:

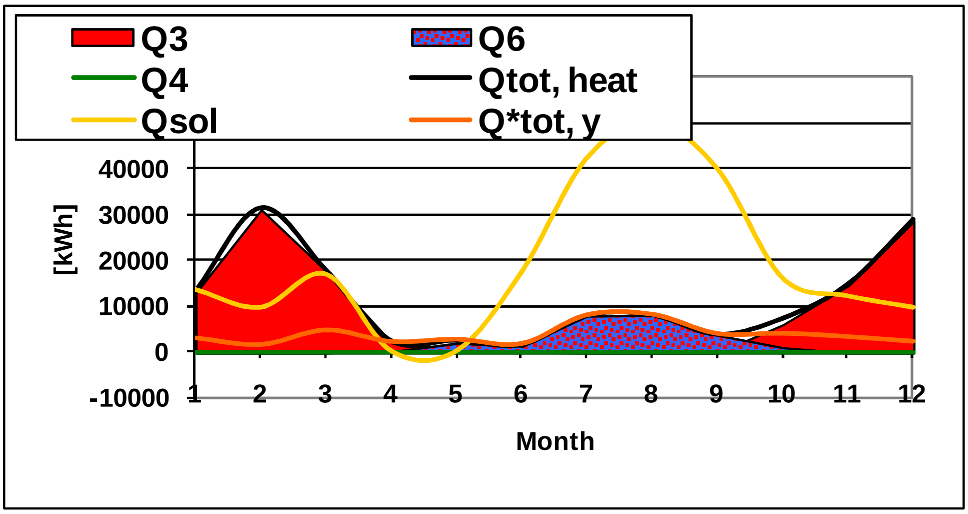


Figure 3-5 Example for the solar energy source management in a SHDC system

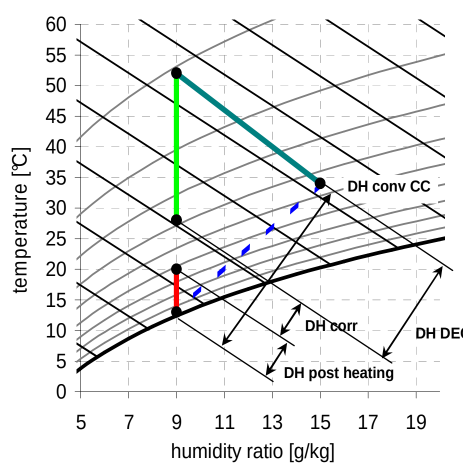
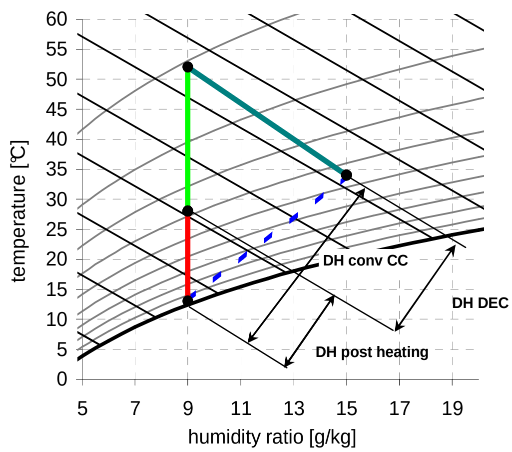
	
postheat fix: post heating up to a fixed supply temperature (to be chosen individually for every plant)	**postheat meas**: post heating up to the identical supply temperature as measured in the DEC-AHU

Figure 3-6 Representation of an example for DEC process

4. How to fill in the monitoring procedure

The excel tool for the evaluation of monitored data is composed of six sheets.

The first sheet, named “Introduction”, shortly presents the tool and its authors.

Users are required to enter data:

- in the second sheet, named “Data”, entering pieces of information on the installed SHC and monitoring system;
- in the fifth sheet, named “3rdLvl”, entering the required monitoring data.
- In the sixth sheet, named “Summary” entering main key data of the installed SHC and inserting the energy flow diagram as shown in the sheet “Introduction” (see also Figure 2-1) adapted to the installed SHC.

Yellow cells require data to be entered whereas.

The sheets automatically calculate all the performance figures described in this report. Orange cells output results. All the results are summarized in the sixth sheet, named “Summary”. In this way the sheet is able to present the overall system and its performance.

5. Conclusions

This technical report describes a tool which has been developed within IEA SHC Task 38 in order to evaluate, with a common procedure, the performance of monitored solar heating and cooling installations. The tool also enables the relative evaluation of a monitored system with respect to a conventional system. Nevertheless, such comparison is based on many assumptions of the conventional system which can lead to inaccurate results. The development of a standard procedure was expected to enable the comparison between the results of several systems as well. Actually, at the current state of the art, systems differ so much that a comparison between many of the energy performance figures defined in the tool sometimes is not fair. Further improvement of this tool is planned within IEA SHC Task 48, starting in autumn 2011.

The tool is usually referred to as a "monitoring procedure". In fact, even if it is not explicitly giving any guidelines for carrying out monitoring campaigns for solar heating and cooling systems, it suggests the necessary measurements for the evaluation of selected energy performance figures. Depending on the kind and the number of measurements needed, three level of detail of monitoring have then been defined, which correspond to different scopes, level of efforts and thus costs of monitoring.

The procedure has been applied to more than twenty five installations, including large and small systems. Results are presented in the following documents of Task 38:

- Jähnig D. and Thür A., "Monitoring Results of fourteen small scale systems - A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)", December 2010;
- Sparber W. and Napolitano A., "Monitored installations and results – A technical report of subtask B (large scale applications)", December 2010.

The complete package of this IEA SHC Task 38 Monitoring Procedure can be downloaded from IEA SHC Task 38 Webpage:

http://www.iea-shc.org/publications/task.aspx?Task=38

and it consists of 4 files:

1. This joint monitoring procedure report of Subtask B (D-B3b) and Subtask A (D-A3a):

   Task38-SubtaskA+B_report_D-B3b_D-A3a_monitoring procedure V6-0.doc
2. XLS-file as the master file for the monitoring procedure:

   110801_T38_MonProc_V6-0.xls
3. PPT-file with reference energy flow schemes:

   110801_T38_MonProc_V6-0_Schemes.ppt
4. XLS-file for extra calculations for DEC systems:

   110801_T38_MonProc_V6-0_DH-calc.xls

Bibliography

Letz, T. (2002). *Validation and background information on the FSC procedure.* www.iea-shc.org/output/task26/A_Letz_FSC_method.pdf.

Letz, T. (2007). „The Extended FSC Procedure for Larger Storage Sizes - Report A1 of Subtask A http://www.iea-shc.org/publications/downloads/task32-Extended_FSC_Procedure.pdf

Sparber, W., & Napolitano, A. (2009). *State of the art on existing solar heating and cooling systems - A technical report of Task 38 - Subtask B1.* http://www.iea-shc.org/task38/work.htm.

Sparber, W., Napolitano, A., & Melograno, P. (2007). Overview on worldwide installed solar cooling systems. *2nd International conference on Solar Air Conditioning.* Tarragona.

Thür, A., Jähnig, D., Nunez, T., Wiemken, E., Helm, M., Mugnier, D., Finocchiaro, P., Nocke, B., (2009). "Monitoring Program of Small-Scale Solar Heating and Cooling Systems within IEA-SHC Task 38 – Procedure and First Results", proceedings No161, Eurosun Conference 2010, Graz, Austria

Thür, A., Vukits, M., Jähnig, D., (2011), „IEA SHC Task38 – Monitoring Prozedur – Vorstellung und Ergebnisse", OTTI - 21. Symposium Thermische Solarenergie, Bad Staffelstein, Germany

Weiss, Ed. (2003), IEA SHC Task26, "Solar Heating Systems for Houses – A Design Handbook for Solar Combisystems", James & James

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b:
Monitoring Results

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: Third Draft December 22, 2010

By Dagmar Jaehnig and Alexander Thuer[1]

[1]Institution	**AEE INTEC**
Address	**Feldgasse 19**
Phone	**+43-3112-5886-26**
Fax	**+43-3112-5886-18**
e-mail	**d.jaehnig@aee.at, a.thuer@aee.at**

Contributions from Tim Selke and Anita Preisler[2], Daniel Neyer[3], Moritz Schubert[4], Daniel Mugnier and Romain Siré[5], Edo Wiemken and Tomas Núñez[6], Martin Helm[7], Fernando Palacín[8], Carlos Monné[9]

[2]Institution	**Austrian Institute of Technology (AIT)**
Address	**Giefinggasse 2, A-1210 Vienna, Austria**
Phone	**+43(0) 50550-6634**
Fax	**+43(0) 50550-6613**
e-mail	**anita.preisler@ait.ac.at / tim.selke@ait.ac.at**

[3]Institution University of Innsbruck, AB Energieeffizientes Bauen, Institut für Konstruktion und Materialwissenschaften
Address Techniker Str. 13, A-6020 Innsbruck, Austria

Phone +43 512 507-6618
Fax +43 512 507-36092
e-mail daniel.neyer@uibk.ac.at

[4]Institution S.O.L.I.D. Solarinstallation und Design GmbH
Address Puchstrasse 85, 8020 Graz, Austria

Phone +43 316 292840-81
Fax +43 316 292840-28
e-mail m.schubert@solid.at

[5]Institution TECSOL
Address 105 avenue Alfred Kastler
Tecnosud
66004 Perpignan CEDEX, France

Phone +33 4 68 68 16 40
Fax +33 4 68 68 16 41
e-mail romain@tecsol.fr

[6]Institution Fraunhofer-Institut für Solare Energiesysteme ISE
Dept. Thermal Systems and Buildings
Address Heidenhofstraße 2, 79110 Freiburg, Germany

Phone +49 (0) 761/4588-5533 / -5412
Fax +49 (0) 761/4588-9533 / -9000
e-mail tomas.nunez@ise.fraunhofer.de /
edo.wiemken@ise.fraunhofer.de

[7]Institution The Bavarian Center for Applied Energy Research (ZAE Bayern)
Division 1: Technology for Energy Systems and Renewable Energy
Address Walther-Meißner-Str. 6
D-85748 Garching, Germany

Phone +49(0)89/329442-33
Fax +49(0)89/329442-12
e-mail helm@muc.zae-bayern.de

[8]Institution National Renewable Energy Centre

Address	**C/Ciudad de la Innovación 7, 31621 Sarriguren, Navarra (Spain)**
Phone	**+34 948 252 800**
Fax	**+34 948 270 774**
e-mail	**fpalacin@cener.com**
[9]Institution	**Group of Thermal Engineering and Energy Systems (GITSE), University of Zaragoza (Spain)**
Address	**C/María de Luna s/n, 50018, Zaragoza, Spain.**
Phone	**+34 976 762 042**
Fax	**+34 976 762 616**
e-mail	**cmmb@unizar.es**

Contents

1 Introduction

This report summarizes the monitoring activities on small-scale solar heating and cooling systems carried out within subtask A of IEA-SHC Task 38. All systems were monitored according to the monitoring procedure developed with Task 38. However, the level of detail varies from system to system. The monthly monitoring data has been filled into the Monitoring Excel Tool of Task 38.

In the first part of the report, a short description of all 13 systems that were monitored is given including the characteristic data of the application and the installation, a few pictures and a short written summary of the monitoring results.

In the second part, an overview of the key performance figures of 11 of the systems is given and the results are compared with each other (For 2 systems there were not enough reliable data available to include them in the comparison). The shown performance figures include different coefficients of performance of the chillers and the whole systems as well as the reached fractional primary energy savings, the collector yields and the water consumption of the cooling towers.

Finally, for most systems, there is a more detailed report on the monitoring results available in the appendix.

2 General Description of Monitored Systems

2.1 Austria: SOLution Headquaters, Sattledt

Description of the application

Type of building office building

Location Sattledt, Austria

In operation since 2005

System operated by SOLution Solartechnik GmbH

Air-conditioned area 150 m^2

System used for space heating? Yes

System used for DHW preparation? No

General description of the system

The solar thermally driven absorption cooling system is designed to control the air temperature of the SOLution Solartechnik GmbH headquarter office (150 m² useful area) in summer and winter. The system configuration allows a solar-autonomous cooling operation, e.g. the driving heat for the absorption chiller is only generated and provided by the solar system. A gas driven boiler allows test routines and as well a significant heating support in winter.

Since 2008 the company of SOLution Solartechnik GmbH has moved to a new headquarter building. Therefore the solar absorption cooling system has been disassembled and the system is not anymore in operation.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	15 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	EAW WEGRACAL SE 15
Chilled water application	Chilled ceiling
Dehumidification	no
Heat rejection system	Wet open (EWK 036/06 Axima)

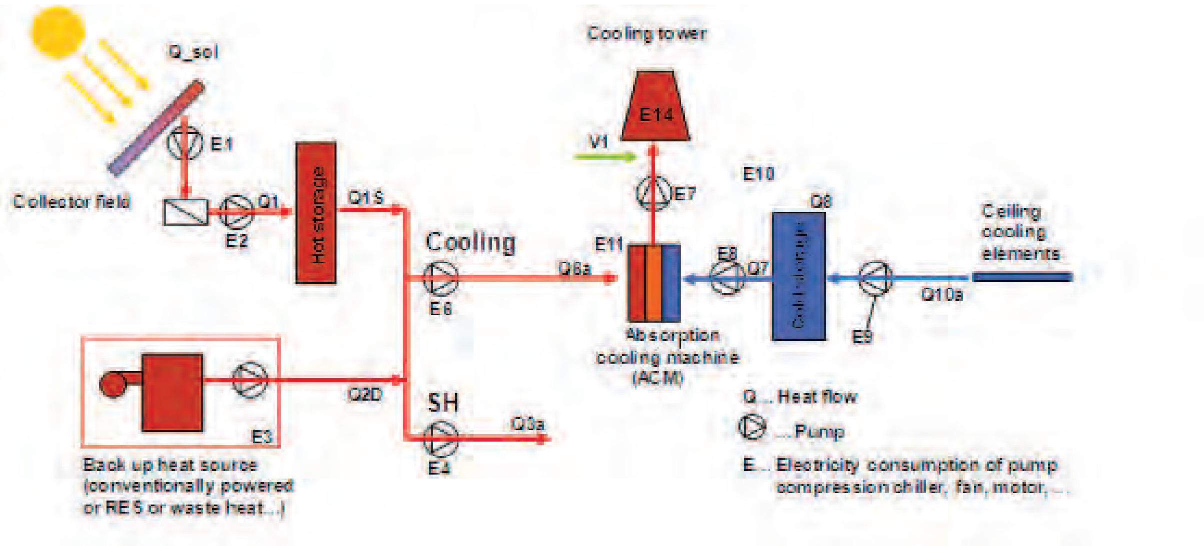
Solar thermal

Collector type	flat-plate
Brand of collector	SOLution
Collector area	40,50 m^2 gross
Tilt angle, orientation	21°, South
Collector fluid	water-glycol
Typical operation temperature	85°C driving temperature for chiller operation

Configuration

Heat storage	2 m^3 water
Cold storage	0.8 m^3 water
Auxiliary heater	Gas boiler, 9 kW
Use of auxiliary heating system	Space heating in winter
Auxiliary chiller	none

System scheme



System performance

The small-scale solar heat driven cooling (SHDC) system has been put into operation in 2005. The monitoring campaign carried out by Austrian Institute of Technology allows the assessment of the energy system performance from September 2006 till September 2008. Based on the measurement data following conclusion regarding the energy system performance can be stated:

- Around 1,026 kWh chilled water has been produced by the SHDC system during the entire monitoring period. This corresponds to around 70 full load hours. The solar system including flat-plate collector and hot water storage did provide around 2,356 kWh heat to the adsorption chiller which leads to a seasonal performance factor of SPF = 0.435. The annual electricity generation efficiency with reflects the ratio of produced energy of chilled water and the consumed electricity of the overall SHDC system is calculated to 3.39.

- The SHDC system energy performance can be distinguished between two different categories:

    - ***High chilling capacity and energy efficient system performance:***
      The solar thermally driven absorption system operates with coefficient of performance COP factors corresponding to manufacturer data. Cold water temperatures are in the range of 12°C and the absorption chiller operates in the capacity range of 5 to 15 kW_{cold}. In all three hydraulic circuits of the chiller nominal temperature level are achieved.

    - ***Low chilling capacity and inefficient system performance:***
      The absorption chiller operates with COPs significantly lower than nominal COP according to manufacturer data. Especially during the starting phase the nominal chilling capacity of 15 kWcold is not achieved. The required temperature levels in all three water circuits around the chiller are insufficient and the chiller operates mainly under low part load conditions.

- Heat rejection – the applied wet open cooling tower operates efficiently und supplies sufficient cooling water temperatures to the absorption chiller. Till September 1,700 liter fresh water have been consumed this corresponds to 70 l fresh water per hour of full load operation. The open wet cooling tower did supply cooling water temperatures below 20°C a couple of days, e.g. the direct use of this cooling water for the chilled ceiling circuit is technically reasonable.

- The solar thermal system - composed of 40.50 m² flat-plate collectors and 2000 liter stratifying hot water tank - generates sufficient driving temperatures for the absorption chiller. Typically the driving temperature starts with 75°C and decreases to 55°C. The low flow operation strategy for the flat-plate collectors in the rage below 20 l/h m² supplies sufficient driving temperatures.

System reliability and overall success of the installation

The monitoring system did operate reliably. Some measurement data logging gaps appeared because of reboot troubles of the data logging system in case of temporary grid blackouts. The measurement signals of the pyranometer did not deliver reasonable values and the technical problem could not be solved on site.

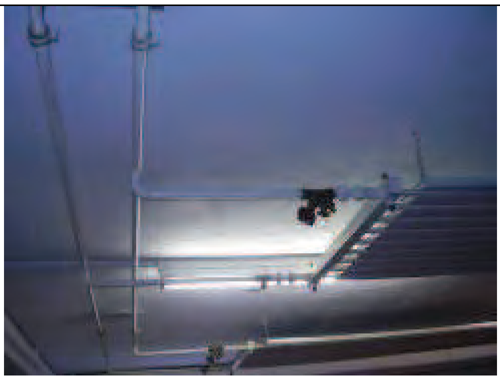
During the monitoring period once the absorption chiller produced too low cold water temperatures which led to crystallization phenomena of the working fluid. The manufacturer did maintenance work which means mainly refilling of lithium-bromide and recreation of vacuum.

Office room air temperatures were measured as well and during the monitoring period and only some few hot days were identified with office room air temperatures exceeding 26°C. Consequently the thermal user comfort lead to a high acceptance of the office users.

Photos

Office building of SOLution headquarters Sattledt/ Austria

Technical room equipped with EAW WEGRACAL SE 15 absorption chiller, hot and cold water storage tank



Chilled ceiling device

Wet open cooling tower

	(EWK 036/06 Axima)

Monitoring Data

Measured period	09/2006 till 09/2008
Monitoring level (according to Task 38 procedure)	2
Person responsible for monitoring	Tim Selke (Austrian Institute of Technology, AIT) Phone: +43 (0)50550 6651 Email: Tim.Selke@ait.ac.at

2.2 Austria: Bachler, Gröbming

Description of the application

Absorption chiller with open wet cooling tower to cool 700 m² office building with a peak cooling load of 8 kW.

Type of building office building

Location Gröbming, Austria

In operation since 2007

System operated by Bachler HVAC company (own office building)

Air-conditioned area 700 m^2

System used for space heating? Yes

System used for DHW preparation? Yes

General description of the system

Closed cycle absorption with ammonia/water as working pair. The system has biomass as auxiliary heating for heating and for the absorption machine.
The building is an office building for a HVAC company with 700 m² cooled office area. The peak cooling load of the building is 9 kW and the building is cooled via concrete core activation.
The building has mechanical building ventilation with heat recovery.
The heat rejection is wet, open EWK Axima refrigeratoin with 25 kW.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	12 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	PINK chillii PSC 12
Chilled water application	Chilled ceilings
Dehumidification	no
Heat rejection system	Wet open

Solar thermal

Collector type	flat-plate
Brand of collector	"Neuma-Solar" in Kremsmünster, Austria
Collector area	46 m^2 aperture area
Tilt angle, orientation	45°, south
Collector fluid	water-glycol
Typical operation temperature	80 °C (driving heat for cooling application)

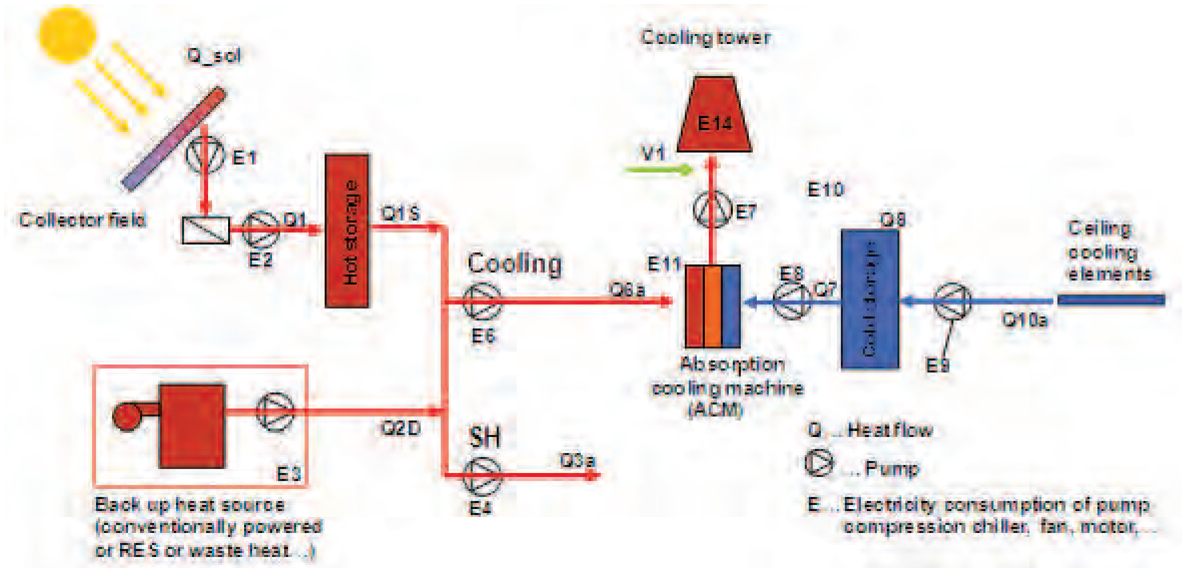
The solar collectors are partly mounted on the ground, partly installed on the façade of the building. The system is operated with a variable flow control.

Configuration

Heat storage	3*1.5m³ water
Cold storage	none
Auxiliary heater	Biomass, 150 kW

Use of auxiliary heating system	Space heating in winter, DHW and cooling
Auxiliary chiller	no

System scheme



System performance

An average thermal COP for the period between the 21st of August and the end of September of 0.565 could be reached. Electrical COPs range in daily values from 0.5 to 5 and achieve an average value of 3.1. In the period between the 11th and 18th of September no results were monitored due to a data processing problem of the computer system.

System reliability and overall success of the installation

In the time where the plant was running, it worked quite well. Nevertheless the average values over a longer time period are still improvable. In the beginning of the cooling season problems did occur due to the low volume flow of the solar cycle, therefore the mode was changed to maximum speed. Now the solar pumps are working in summer times without rpm-regulation.
During the monitoring period some problems did occur including monitoring and control system problems. The wet cooling tower works very well in the Gröbming climate. At a glance the PSC12 Pink chiller fulfilled the promised thermal COPs at the different driving, cooling and recooling temperatures. Considering the other parts of the system, such as the heating of the swimming pool, the hot storages or the district heat the complete hydraulic scheme gets overloaded and not clearly arranged at the first look.

Summarizing five main practical suggestions for improvement have been found:

- adapting the control system in order to make clear division between heating and cooling periods
- including the cooling cycle of the DHW priority rule
- excluding the logical test room temperature >24°C of the control strategy in order to use the concrete core activation as a storage and make the control system easier

- changing the switching operations between summer and winter to automatically by including the valves V3 and V4 in the controller
- possibly rising the starting driving temperature in the control strategy if there are still problems with a clocking behavior of the chiller

Photos

	
Office building with solar collector field	Technical premises
	
Wet cooling tower and Pink chillii PSC12 in front of the building	Pink chillii PSC 12

Monitoring Data

Measured period	Aug 2009 – ongoing
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Neyer (University of Innsbruck UIBK, Austria) Phone: +43 512 507-6618 Email: daniel.neyer@uibk.ac.at

2.3 Austria: SOLID Office Building, Graz

Description of the application

The office building of SOLID was renovated in 2004 and the solar cooling device was installed in the year 2008. The façade of the office rooms are south and west oriented. To reduce the solar gains external shading devices are installed at each glazing. Because of internal gains and ventilation via windows, active cooling is indispensable. The solar cooling equipment is installed in a so-called “Cooling Cabin” placed under a pergola which is situated in front of the office building as a nice main entrance. The solar collectors are installed on the roof of this pergola. The hybrid cooling tower is placed on the flat roof of the office building. The cooling load of the office rooms is taken out via ceiling cooling elements.

Type of building office building

Location Graz, Austria

In operation since 2008

System operated by SOLID

Air-conditioned area 435 m^2

System used for space heating? Yes

System used for DHW preparation? No

General description of the system

A closed absorption cycle for generating cooling energy is employed. Autonomously solar thermal generated heat by high temperature flat plate collectors is used to regenerate the process. Within the absorption cooling machine water is used as refrigerant and lithium bromide is used as solvent. The cold water generated by the absorption cooling machine is used to cover internal and external heat gains and also the heat income caused by window-ventilation in the office rooms. In winter the solar collectors are in assistance to the space heating. In summer and winter the solar generated heat is stored in one buffer storage and all energy demand is taken out of it. The additional heat from the district heating is not stored in the tank but directly carried to the space heating system. A special application is the usage of the hybrid cooling tower for direct cooling via the ceiling cooling elements.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	17.5 kWcold, 21 kW peak
Type of closed system	Absorption
Brand of chiller unit	Yazaki WFC SC5
Chilled water application	Chilled ceilings
Dehumidification	no
Heat rejection system	Hybrid cooling tower

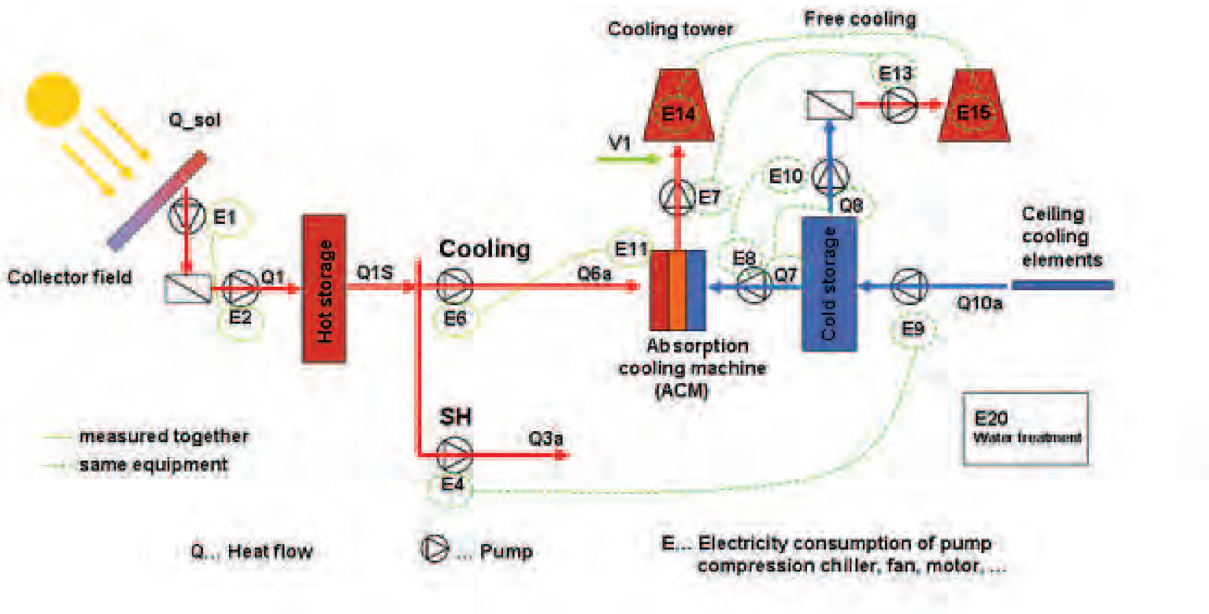
Solar thermal

Collector type	High temperature flat-plate
Brand of collector	ÖkoTech Gluatmugl HT
Collector area	60 m^2 gross area
Tilt angle, orientation	11°, south
Collector fluid	water-glycol
Typical operation temperature	88 °C driving temperature for chiller operation

Configuration

Heat storage	2 m^3 water
Cold storage	0.2 m^3 water
Auxiliary heater	District heat
Use of auxiliary heating system	Space heating in winter
Auxiliary chiller	no

System scheme



System performance

Over the summer period 2009 (July-Sept) a thermal coefficient of performance of 0.6 was observed. About 1/3 of the cooling demand was delivered by free cooling, 2/3 by solar cooling. The electrical coefficient of performance for free cooling was COPel = 1.8, for solar cooling a COP_{el} = 1.6 was measured. The overall coefficient of performance for the summer period was approximately COP_{el} = 1.7.
In the winter period from October 2009 to April 2010 the average coefficient of performance for free cooling was about COP_{el}. = 8.2, no solar cooling was observed.

System reliability and overall success of the installation

The whole system is running reliably but with a poor overall performance, especially the electrical consumption is very high. The chiller was running very reliably throughout the whole summer. It also brought the expected power level and reached a feasible COP as well. Free cooling mode is activated mainly in night times and running on a poor power level. The two largest electrical loads are the cooling tower and the pump towards the cooling tower. These two consumers are responsible for more than 75% of the electrical consumption. The reason for the high electrical consumption of the cooling tower is its high fan power which is not speed-regulated. Due to this a high potential in the improvement of the control system can be observed, mainly through an adequate control of the cooling tower. Small changes in hydraulic design can also decrease the electrical consumption, especially for free cooling.

Photos

		
“CoolCabin” placed under a pergola	Technical premises in the “CoolCabin”	

Monitoring Data

Measured period	Jun 2009 – ongoing
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Neyer (University of Innsbruck, UIBK, Austria) Phone: +43 512 507-6618 Email: daniel.neyer@uibk.ac.at

2.4 Austria: Municipal Administration, Vienna

Description of the application

The MA34 solar cooling system was put up in April 2009. It consists of an Adsorption chiller with dry cooling tower to cool the office building with a peak cooling load of 7.5 kW. The hybrid cooling tower and the solar collectors are placed on the flat roof of the building.

Type of building office building	
Location Vienna, Austria	
In operation since 2009	
System operated by	
Air-conditioned area	
System used for space heating? Yes	
System used for DHW preparation? No	

General description of the system

Main item of the plant is an adsorption chiller with a nominal cooling capacity of 7.5 KW, type SOL ACS 08. The heat rejection is done by a dry cooling tower, type RCS 08, with EC-fan technology and an additional fresh water spraying system. The drive heat for the chiller is generated by 12 universal panel collectors with a total gross surface of 32.40 m ². The flat-plate collectors are installed in approximately 40 ° inclination and south orientation on an existing roof. In the concept no reheating is intended by other boilers. The solar heat is stored in a 2000 l solar buffer - the heat transfer from the primary to the secondary solar cycle is made by a layer loading unit. In the cooling mode the chilled water is stored in a water buffer with the capacity of 800 l. Depending on the cooling requirement of the different rooms the available fan coils are supplied by the water tank. In winter mode the hydraulic interconnecting makes a solar thermal support for heating purposes of the areas possible.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	7.5 kW_{cold}
Type of closed system	Adsorption
Brand of chiller unit	Sortech AG / SOL ACS 08
Chilled water application	Fan coil
Dehumidification	no
Heat rejection system	Dry cooling tower with water straying system

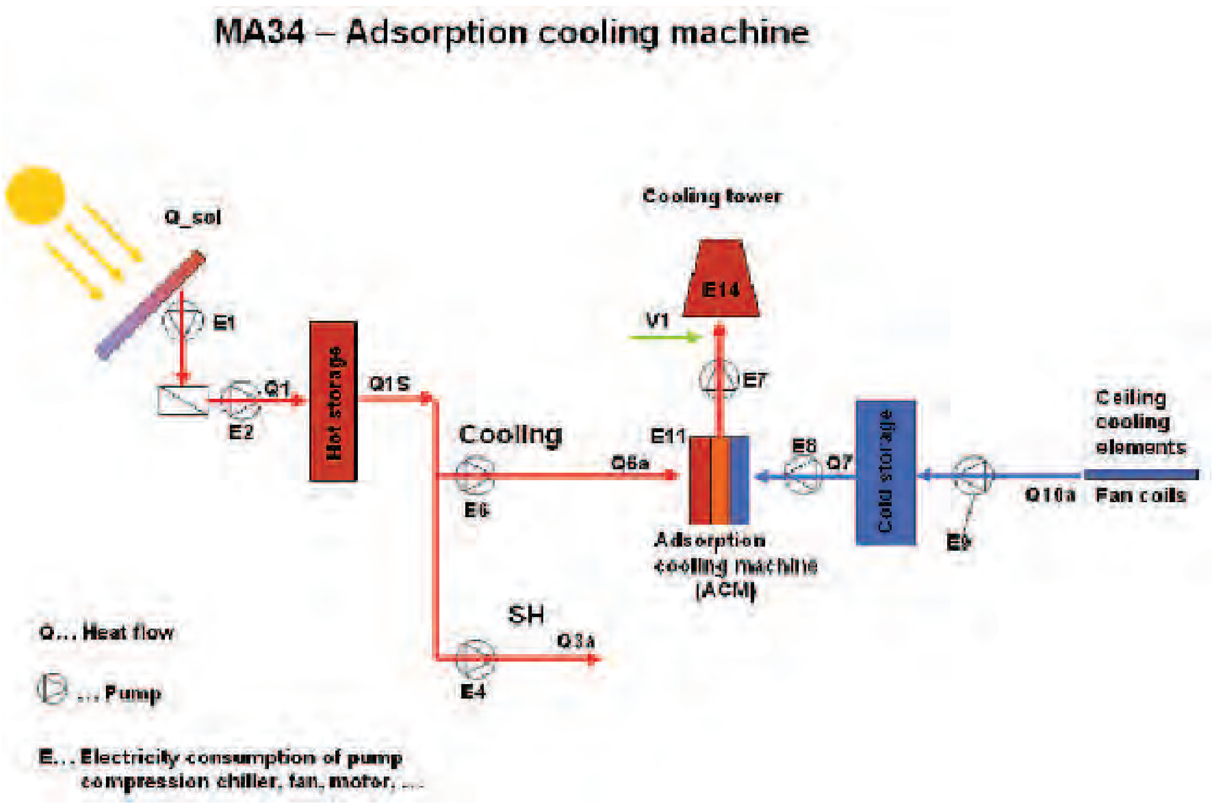
Solar thermal

Collector type	flat-plate
Brand of collector	SOLution
Collector area	62.4 m^2 gross area
Tilt angle, orientation	40°, South
Collector fluid	water-glycol
Typical operation temperature	65°C – 70°C driving temperature for chiller operation

Configuration

Heat storage	2 m^3 water
Cold storage	0.8 m^3 water
Auxiliary heater	None
Use of auxiliary heating system	--
Auxiliary chiller	None

System scheme



System performance

The solar cooling system worked steadily in summer 2009 and 2010 without major failures concerning the whole system or individual components. It was possible to keep the office room temperature below 24 °C. The average daily thermal COP kept with 0,3 in the same range in summer 2009 and 2010, only when the heat rejection temperature going into the adsorption cooling chiller was below 27°C the nominal thermal COP of 0,56 was reached. The average daily electrical COP increased from 2,3 in August 2009 to 4,00 in August 2010. The highest part of the electricity demand (50-60 %) was caused by the fans of the heat rejection and the heat rejection pump. As the heat rejection pump was changed to a variable speed high efficient pump in September 2010 and an adaptation of the whole control system will be implemented before summer 2011 the electrical COP should be better in the next cooling season. The solar thermal system worked very efficiently in both summers using only around 3 % of the electricity demand. The cooling capacity used from the Fan-Coils varied between 2,2 and 3,2 kW which caused a continuous part load behavior in the adsorption cooling system.

System reliability and overall success of the installation

In summer 2009 the monitoring system didn't work properly which led to many gaps in the monitoring recording. This problem was solved between summer 2009 and summer 2010. It was already possible to implement improvements between the first and the second summer by variation of the set points for the desorption temperature and the cold water temperature as well as by inventing the water spraying mode in the heat rejection device. The biggest electricity consumers are still the fans in the heat rejection which couldn't be reduced essentially by using variable speed control of the fans in summer 2010. An adaptation of the whole control system also taking load dependent operation into account will be the next step for improvements. Overall a know-how gain concerning the interrelation between the several components of a small scale adsorption system during operation to identify improvement strategies was given.

Photos

Monitoring Data

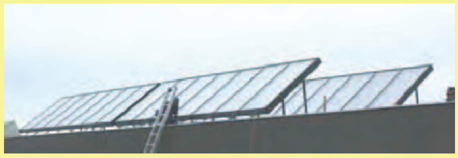
Measured period	April 2009 - ongoing
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Anita Preisler (AIT, Austria) Phone: Email: anita.preisler@ait.ac.at

2.5 Austria: Office Building Feistritzwerke, Gleisdorf

Description of the application

The office building of the Feistritzwerke STEWEAG GmbH, a municipal energy and water supplier, was renovated in 1995 and was equipped with a solar heating and cooling device in June 2010 for air conditioning both floors of the office building. Before that, no air conditioning system existed except a conventional split unit for a special server room. Air ventilation was and is done via manual window openings. The long back side of the office building is oriented +236° (south is 0°) and several windows are equipped with external, manually operated shading devices. Cold water is generated by an ammonia-water absorption chiller and is distributed via chilled ceilings. Space heating occurs with those same new installed ceiling elements and with existing radiators. Some technical installations already existed in this building and were used in the new heating and cooling concept. Five heat storages with 2 m³ volume each and heat generation devices like a condensing natural gas boiler, three combined heat and power plants (CHP) powered by vegetable oil and natural gas as well as a high temperature ground source heat pump were integrated in the system.

Type of building	Office building
Location	Gleisdorf, Austria
In operation since	15.06.2010
System operated by	Feistritzwerke
Air-conditioned area	1,000 m²
System used for space heating?	Yes
System used for DHW preparation?	Yes



General description of the system

Solar thermal energy is produced by a 64 m² collector field and is stored in five 2 m³ heat storage tanks. One of these tanks is switched in series to the other four tanks and is working as a high temperature tank, equipped with a stratifier unit and can be operated independent of the other four tanks. They are switched in parallel to each other and are used to store surplus heat generated by the solar collectors or the CHPs.

During the heating season the three CHP's, the condensing natural gas boiler and the heat pump serve as backup. Except the Feistritzwerke office building also a local district heating net has to be supplied with heat.

The absorption chiller's generator heat is taken out of the high temperature tank. The cold water generated by the chiller is transported directly to the cold distribution system without any cold water storage in order to avoid an extra cold water pump to reduce the electricity effort. A dynamic cooling power control configuration is achieved for the chiller. The mass flows and relevant temperatures of the particular hydraulic cycles of the chiller (generator-, recooling- and cold water cycle) are varied in that way, that an infinitely variable control of the produced cooling power is possible. Heat rejection of the absorption cooling process is realised with an open, wet cooling tower combined with an electrolytic water preparation device with the advantage of reduced electricity consumption.

Central air-conditioning unit

Technology	Closed cycle
Nominal capacity	19 kW_{cold}
Type of closed system	Absorption (Ammonia-water)
Brand of chiller unit	PINK
Chilled water application	Chilled ceiling
Dehumidification	No
Heat rejection system	Wet, open

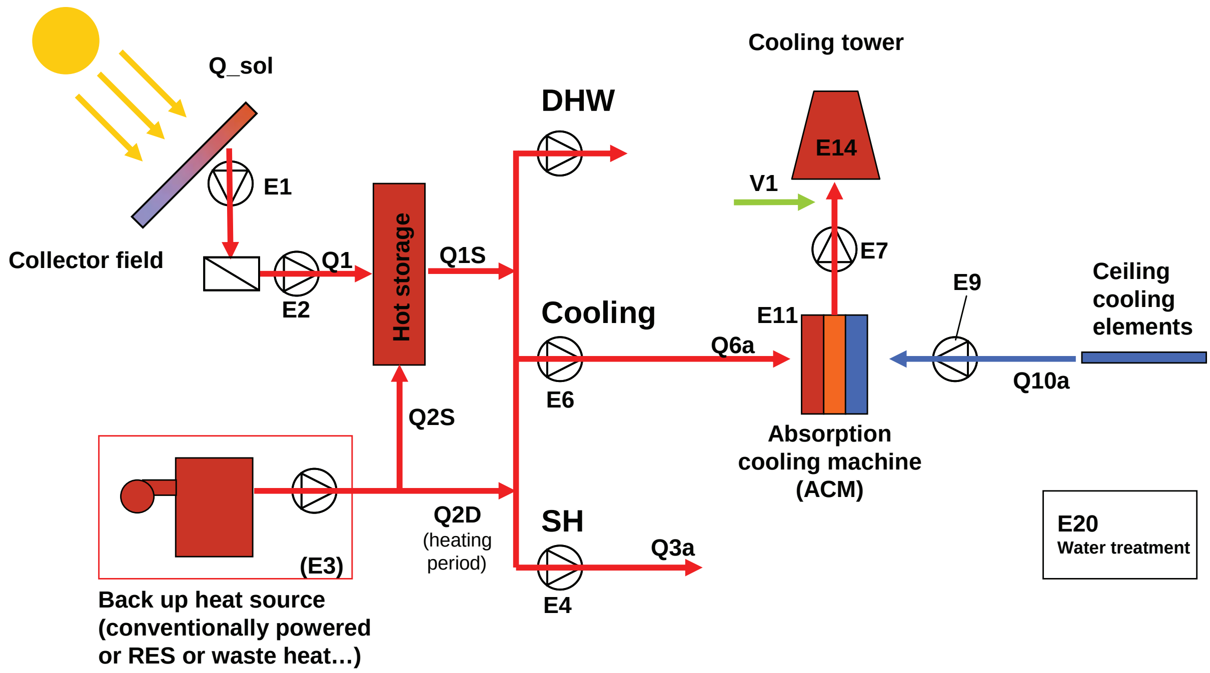
Solar thermal

Collector type	Double glazed flat plate collectors Gluatmugl HT
Brand of collector	ÖkoTech
Collector area	64 m²
Tilt angle, orientation	40°, 34° east
Collector fluid	Water-glycol
Typical operation temperature	85°C

Configuration

Heat storage	5 x 2 m³
Cold storage	None
Auxiliary heating support	If insufficient solar energy is available (into heat storage)
Auxiliary heater	3xCHP (35+18+16 kW_{th}), natural gas boiler, heat pump
Auxiliary chiller	Split units in individual rooms like the control room for the local electrical grid

System scheme



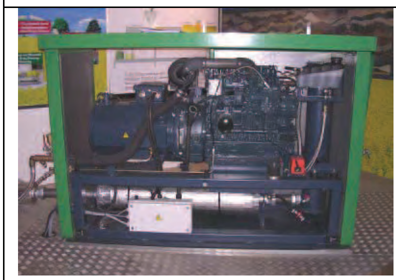
System performance

The monitoring data showed thermal coefficient of performances (COP_th_chill,m) of 0.55 and 0.53 as well as electrical coefficient of performances (COP_el_coldSol,m) of 5.75 and 5.23 in July and August 2010.

System reliability and overall success of the installation

Due to the fact that detailed monitoring data was not available for the whole first cooling period, no concrete statement to the system reliability can be given. But the fragmented measurement data showed good operation behaviors of the solar thermal system and the cold distribution system. Additionally the feedback of the Feistritzwerke's employees was pretty positive in view of room comfort conditions. A huge optimization potential could be detected in the storage charging control strategy. Currently, all five heat storage tanks are charged (Solar, CHP) - with resulting storage losses of 65% respectively 59%! Improvements will be reached by operating only the one high temperature tank. Nevertheless, for the circumstance that this system was installed and put in operation with no carried out optimization measures, the monitoring data showed quite acceptable values.

Photos

		
Absorption chiller	Chilled ceiling, water treatment	Cooling tower
		
Vegatable oil driven CHP	High temperature storage tank	External plate heat exchanger of the solar thermal system

Monitoring Data

Measured period	July 2010 - ongoing
Monitoring level (according to Task 38 procedure)	Level 3
Person responsible for monitoring	Alexander Thür, AEE INTEC Phone: +43-3112-5886-26 e-mail: a.thuer@aee.at

2.6 France: Résidence du Lac, Maclas

Description of the application

The targeted building welcoming the solar cooling application is the Résidence du Lac, a building dedicated to retired people. This building is located in the small town of Maclas in the Rhône Alpes area, close to Lyon. The town is in altitude, nearly 450 m high. The building was created in the seventies and has an average quality of energy efficiency. Only one small part of the building is cooled, the leisure space/restaurant which is compulsory since summer 2003 in retired buildings. This area is of 210 m² and includes a veranda oriented in the Southern direction. Efforts were made to increase the solar protection level in the veranda by adding dark thin protection films. Till 2007, the building owner used electric compression chillers (3 monosplits). Two of them were out of order in 2007 and the management took the decision helped by the SIEL (Syndicat Intercommunal d'Energie de la Loire), to go for a solar cooling system. The owner of the system is the SIEL itself.

Type of building Retired people residence	
Location Maclas, France	
In operation since 2007	
System operated by SIEL	
Air-conditioned area 210 m^2	
System used for space heating? Yes	
System used for DHW preparation? No	

General description of the system

The system is based on an absorption chiller of 10 kW coupled with evacuated tube collectors. The system works as a quasi solar autonomous cooling system because only a small electric compression chiller (split type) is used in case of failure of the solar system. The load of a part of the building is based on the following scenario: cooling demand from June to mid September and heating demand from mid October to end of May. The solar system uses fan coils for the cooling and heating modes. In addition, thanks to buffer storage, the energy is valorized in the heating mode through the central heating network of the Résidence du Lac. The heat rejection system is done by a drycooler located in the northern facade of the building.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	10 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	Sonnenklima
Chilled water application	Fan coils
Dehumidification	no
Heat rejection system	dry

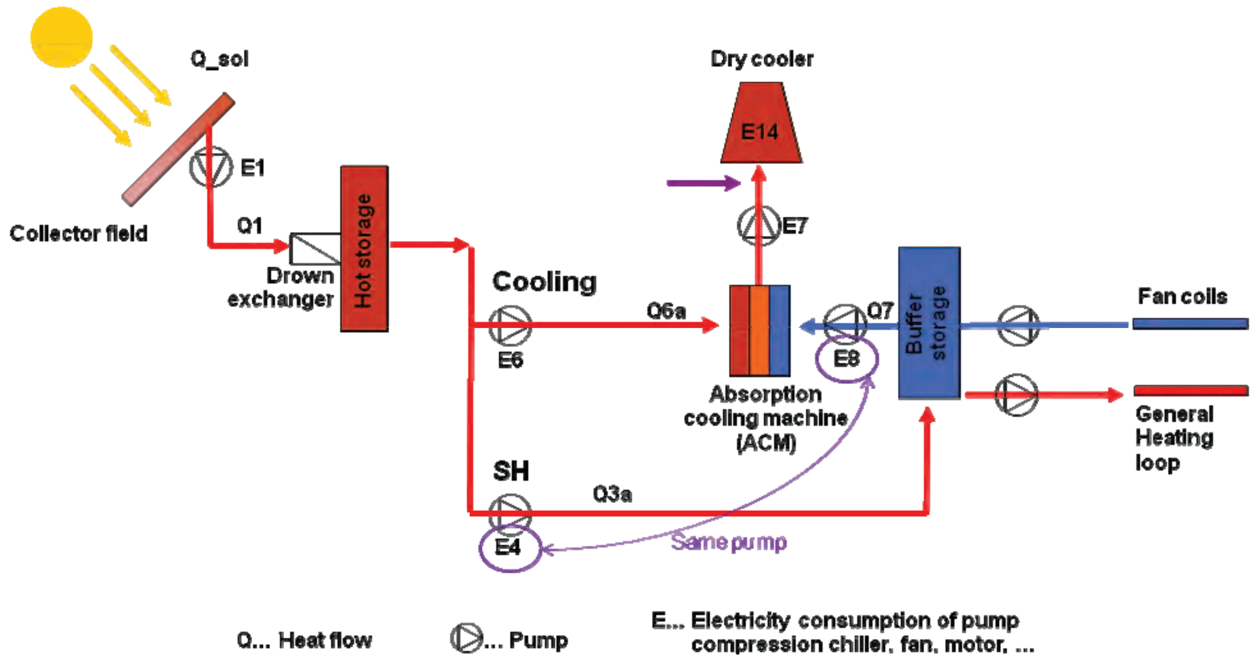
Solar thermal

Collector type	Evacuated tube
Brand of collector	Thermomax Mazdon 20
Collector area	24 m^2 absorber
Tilt angle, orientation	30°, 15° west
Collector fluid	water-glycol
Typical operation temperature	75 °C driving temperature for chiller operation

Configuration

Heat storage	0.5 m^3 water
Cold storage	Buffer water (80 liters)
Auxiliary heater	none
Use of auxiliary heating system	-
Auxiliary chiller	yes
- type	el. Compression chiller
- capacity	3 kW_{cold}

System scheme



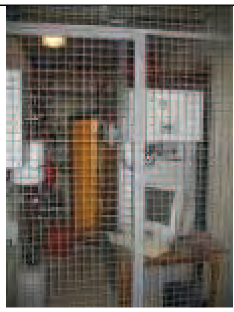
System performance

For one cooling season monitored in summer, an average thermal coefficient of performance of COPth = 0.54 was observed while the collector yield is about 40%. And the electrical coefficient of performance for the same period is about COPel = 2.5. Focusing on a day with especially good climatic conditions, the COPth can reach 0.63, and the COPel reached 3.08.

System reliability and overall success of the installation

The main issue for the Maclas installation was actually an “administrative” one. Indeed, the data transfers were a bit chaotic since they were transferred first by the chiller manufacturer, and then sent to the institution in charge of the monitoring. And because of the solvability situation of the chiller manufacturer, it was not possible to access the data to calculate performances or to check if the installation is working properly. However, this partial monitoring led to several conclusions. The strengths of the installation were: the ability to cool down the building from 12pm to 18pm (when it was needed), the compatibility with a dry cooler which avoid legionella problems which is crucial for a retired people residence, the installation had also good media coverage. This monitoring permits also to enlighten the points which have to be improved: the electric performances (COPel) and of course the monitoring system reliability.

Photos

		
Solar collector field	Technical premises	Dry cooler
		
Cooled space	Outside view of building	Fan coils

Monitoring Data

Measured period	June – September 2009
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Mugnier ; Romain Siré (TECSOL, France) Phone: +33468681640 Email: daniel.mugnier@tecsol.fr ; romain@tecsol.fr

2.7 France: CNRS PROMES Research Center Office, Perpignan

Description of the application

The targeted building welcoming the solar cooling application is the CNRS PROMES research center office. It is dedicated to research works and offices in the technical area TECNOSUD of Perpignan located in Languedoc Roussillon area (South of France). The building is a large building of more than 5,000 m² made of 3 levels and the solar cooling system is located on the ground floor and producing energy only for a small proportion of the building. The general orientation of the building is North/South (30° tilt, 45° south east) and the collector field is oriented in the same direction on the roof. The building was created in 2000 and is of good quality level for the energy efficiency.

Type of building Office building

Location Perpignan, France

In operation since July 2008

System operated by Neotec

Air-conditioned area 180 m^2

System used for space heating? Yes

System used for DHW preparation? No

General description of the system

The system is based on an adsorption chiller of 7.5 kW coupled with 24 m² double glazed flat plate collectors. The system is producing independently energy in parallel of a general multi split compression chiller system. The distribution system for the solar cooling system is an independent chilled/hot water network using fan coils working at 14/18 °C temperature level. The heat rejection is carried out by a drycooler assisted by a spring water spraying device, only used in case of very hot days.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	7.5 kW_{cold}
Type of closed system	Adsorption
Brand of chiller unit	SORTECH
Chilled water application	Fan coils
Dehumidification	no
Heat rejection system	dry cooling tower with adiabatic spaying

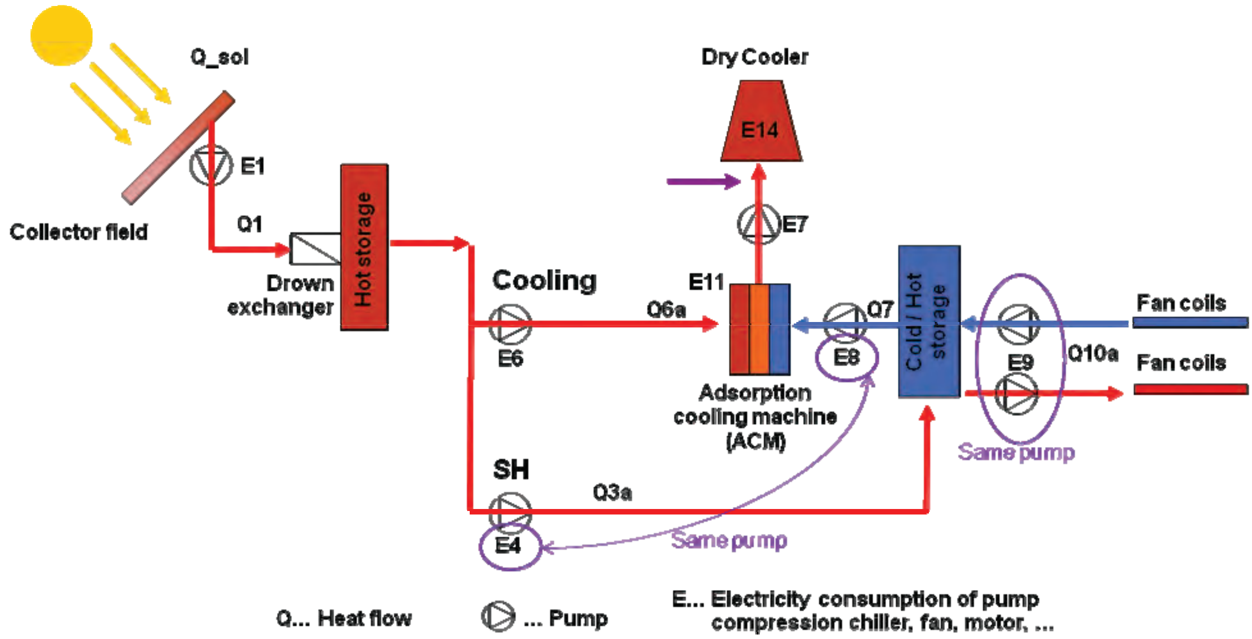
Solar thermal

Collector type	Double glazed flat plate collectors
Brand of collector	Schüco
Collector area	25 m^2 absorber
Tilt angle, orientation	30° tilt, 45° east
Collector fluid	water
Typical operation temperature	75 °C driving temperature for chiller operation

Configuration

Heat storage	0.3 m^3 water
Cold storage	0.3 m^3 water
Auxiliary heater	none
Use of auxiliary heating system	-
Auxiliary chiller	Yes, el. Compression chiller

System scheme



System performance

The energy production for the monitored season 2008-2009 was:

– Cooling: 2 500 kWh/year (5 months on 12)

– Heating: 4 000 kWh/year (6 months on 12)

Energy savings:

– Cooling & heating: 10 ç€/kWh (ESEER = 2; average quality multisplit split)

– Electricity consumption: 580 kWh = 58 €/year

TOTAL= 540 €/year (on the basis of an average increase of energy price of 5%/year)

System reliability and overall success of the installation

The system has been working properly for more than 1.5 years on cooling and heating mode. The overall electrical COP (electrical efficiency of the solar system) has reached an average of 10 on a yearly monitoring duration. The building owner is satisfied by the solar cooling and heating system. However, nearly 10% of the annual solar resource cannot be valorized because the targeted building has no need of heating nor cooling in April (example in 2009) however, the drainback system on the collectors' field is protecting the system from overheating risks.

Photos

		
Solar collector field	Building	

Monitoring Data

Measured period	2008-2009
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Mugnier ; Romain Siré (TECSOL, France) Phone: +33468681640 Email: daniel.mugnier@tecsol.fr ; romain@tecsol.fr

2.8 France: INES Research Center Offices, Chambéry

Description of the application

The targeted building welcoming the solar cooling application is the PUMA3's INES research office. The INES (National Institute of Solar Energy) was created in 2006 by the public institutions to promote and develop solar technologies in France. To reach these objectives, the INES is divided into two centers: research, development and innovation on the one hand, and training and education on the other hand. The INES is located in the "Savoie Technolac" area which is very close to Chambéry in Rhônes-Alpes area (close to the French Alps and Lyon). The PUMA3 building is large, but only 3 mezzanine offices are cooled down by the solar system. The building was created recently so it has a good level of energy efficiency.

Type of building Office building	
Location Chambéry, France	
In operation since April 2009	
System operated by INES	
Air-conditioned area 21m²	
System used for space heating? Yes	
System used for DHW preparation? Yes	

General description of the system

The system is based on an absorption chiller of 4.5 kW coupled with 30 m² flat plate collectors. The installation cools the building in summer, heats it in winter, and it is also able to produce a small amount of domestic hot water all year long. The 400 liters heat storage tank is included in a packaged device supplied by Clipsol (SSC BlocSol RSD 120). Included in this device there is also an electric backup and all the security devices related to the heat storage. Another tank is installed in the system which is ensuring the cold storage in summer and a second heat storage in winter. The heat rejection is carried out by a horizontal geothermal field.

Central air-conditioning unit

Technology	Closed cycle
Nominal capacity	4.5 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	ROTARTICA
Chilled water application	Fan coil
Dehumidification	No
Heat rejection system	Geothermal field, exchange area about 138m²

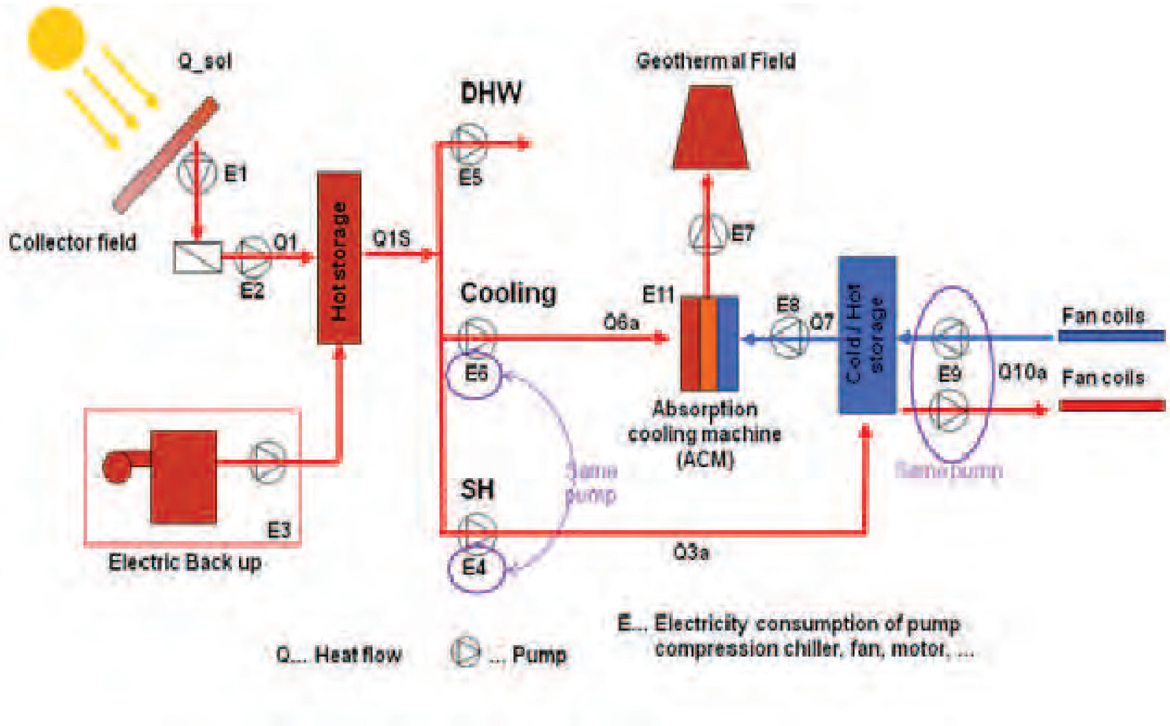
Solar thermal

Collector type	Flat plate collectors
Brand of collector	CLIPSOL
Collector area	30m²
Tilt angle, orientation	30°, 10°
Collector fluid	Water glycol
Typical operation temperature	80°C

Configuration

Heat storage	0.4 m^3 water (part of:SSC BlocSol RSD 120 CLIPSOL)
Cold storage	0.3 m^3 water
Auxiliary heater	Electric heater (part of:SSC BlocSol RSD 120 CLIPSOL)
Use of auxiliary heating system	Hot backup (heating the heat storage tank)
Auxiliary chiller	none

System scheme



System performance

For one year of monitoring, from October 2009 to September 2010, the average Electrical COP is about 5,37 which is a good value. Meanwhile, the collector efficiency was about 22%. In summer of this year, the thermal COP of the chiller was very good (high and stable) and was about 0,74 in average for the entire cooling period. Focusing on the performances of one very good and sunny day in cooling period (28th June 2010), the daily solar collector efficiency reached 32%, the thermal COP was 0,75, and the electrical COP reached 5,68. And for the heating period, the daily electrical COP can go up to 15 for very sunny winter days.

System reliability and overall success of the installation

The system has been working properly for more than 1.5 years on cooling and heating mode. The overall electrical COP (electrical efficiency of the solar system) has reached an average of 5.37 on a yearly monitoring duration. The building owner is satisfied by the solar cooling and heating system.
The main conclusions obtained regarding the performances analysis carried on are:
- Excellent performances of the chiller.
- Good behavior of the geothermal horizontal probes.
- Good performances of the collector field considering they are flat plate collectors, but a lot of days are not sunny enough to start the solar system.
- Difficulties to reach a high electrical COP due to the “experimental state” of the installation: explained by the presence of numerous pumps, of devices consuming more electricity than usually (hot tank, chiller), and of a large set of sensors (more numerous than in a “basic”

solar cooling installation).
- The average Primary Energy Ratio is higher than 1, but it's still a bit low. It's explained by the low electrical COP, and by the use of an electric boiler as a backup which is disadvantageous when the calculations are based on primary energy.

Photos

		
Solar collectors	Technical premises	Geothermal field

Monitoring Data

Measured period	May 2009 – ongoing
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Mugnier ; Romain Siré (TECSOL, France) Phone: +33468681640 Email: daniel.mugnier@tecsol.fr ; romain@tecsol.fr

2.9 Germany: Technical College Butzbach

Description of the application

In the low-energy building of the Technical College, demand for summer air conditioning arises due to high occupation rates and frequent use of computer equipment. The building is used throughout the summer season. Two existing ventilation systems (supply/return air with heat recovery, 1,250 m³/h air volume flow rate each) are not sufficient to remove the sensible and latent cooling loads in summer. For this reason, a solar thermally driven chiller plant was added and supported in the frame of the German Solarthermie 2000plus funding programme. The ventilation units were extended by cooling coils, and additionally chilled ceilings and cooling panels were installed.

Type of building College building, low energy building	
Location Butzbach, Germany	
In operation since 2008	
System operated by Technical College Butzbach	
Air-conditioned area 335 m^2	
System used for space heating? Yes	
System used for DHW preparation? No	

General description of the system

Two absorption chillers are installed and thermally driven with solar heat alone. The condensing boiler is used for space heating in winter only. Different operation modes are possible with the system:

- only one chilled water temperature level, e.g. for the removal of sensible cooling loads, is required. In case both chillers are in operation, the driving circuits of the chillers are usually connected in parallel;
- one chiller operates at low chilled water temperature level for supply air dehumidification and cooling, while the other provides chilled water at a higher temperature level for the chilled ceilings. In this mode, it is possible to connect the driving circuits of the chillers in series with the effect of an increased temperature difference in this circuit, resulting in a more favourable collector system operation.

The evacuated tube collector system contains pure water as collector fluid. A special freezing protection control runs the collector loop in short intervals at low ambient temperatures. No heat exchanger is installed in the total heat supply system.
System concept and overall system control: Hindenburg Consulting (www.hindenburg-consulting.com)
Detailed planning: IGT GmbH

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	2 x 10 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	SK Sonnenklima: Suninverse
Chilled water application	Supply air cooling and dehumidification, chilled ceilings

	and a cooling panel
Dehumidification	in supply air system and in the cooling panel
Heat rejection system	Wet open

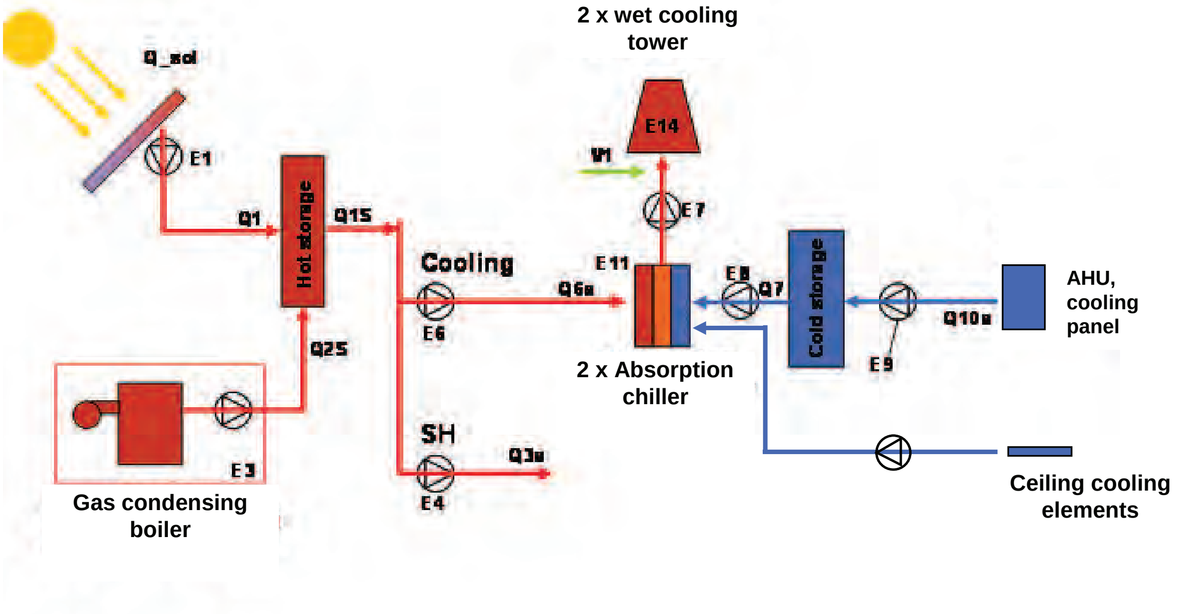
Solar thermal

Collector type	evacuated tube with CPC-mirror
Brand of collector	CPC Star azzurro
Collector area	60 m^2 aperture area
Tilt angle, orientation	30°, south
Collector fluid	water
Typical operation temperature	70 °C driving temperature for chiller operation

Configuration

Heat storage	3 m^3 water
Cold storage	1 m^3 water
Auxiliary heater	gas condensing boiler, 28 kW
Auxiliary heating support	Space heating in winter
Auxiliary chiller	none

System scheme



System performance

Data for complete monitoring period 2009:

Radiation sum horizontal	1080	kWh/m²	
Radiation sum in collector plane	1216	kWh/m²	
Specific collector yield *	337	kWh/m²	(collector output, incl. expenses for freezing protection)
Collector efficiency *	21	%	

* on base of collector output; diminished through heat return to the collector for freezing protection

Annual thermal COP of Absorption chiller 1: 0.52

Annual electric COP of Absorption chiller system 1: 4.7
(chiller, heat rejection, collector and hot water driving pump)

Annual electricity demand collector system only: 75 kWh thermal per kWh el.

Solar fraction on total annual heat supply (heating and cooling: approx. 37 %

System reliability and overall success of the installation

One of the two absorptions chillers was in the complete year 2009 not in operation, caused by a vacuum leakage. The problem could not be solved in the first year due to the insolvency procedure of SK Sonnenklima. The chiller was repaired in 2010 and since end of July 2010, both chillers are in operation.

The solar system and chiller no. 1 was working with high reliability in 2009. Some stagnation periods in the collector system were passed without difficulty. The collector yields are a little below expectations, mainly caused through shading (trees and a scaffold in 2009 in front of the building).

The high temperature flexibility in the driving circuit of the chiller (operation between 90°C and 60°C driving temperature) allowed often a continuous operation of the chiller for 6 hours or more. The chiller thus meets well the requirements for a solar thermal autonomous operation.

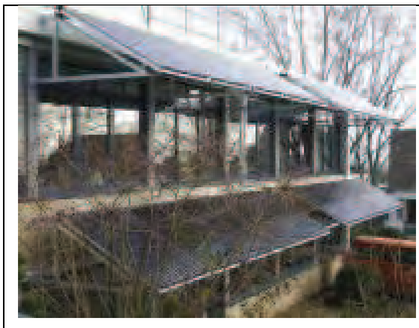
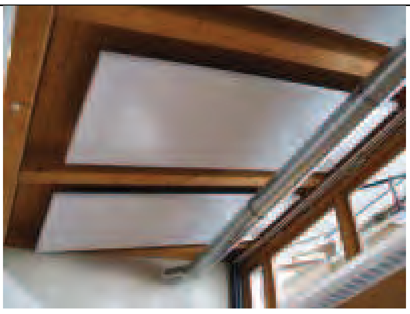
Problems occurred:

- Pollution in the open heat rejection circuit, caused by dust, pollen etc. This lead often to decreased mass flow and thus high heat rejection temperatures. Consequently, for several operation periods the performance of the chiller was low. The problem was solved through the installation of an automated cleaning procedure in the heat rejection circuit;
- A measurement error occurred in the heating season in the mass flow of the gas boiler heating circuit. Thus, the heat input of the boiler could not be calculated accurately, but estimated through assumptions of storage heat losses.

Overall success: although in 2009 only one of the two chillers was in operation, this (part) system worked with high reliability. The comfort in the seminar rooms has clearly increased, the users (students and teaches) are very satisfied in general. The solar autonomous operation mode matches well the occupation profile and cooling demand profile.

Since 2010, both chillers are in operation and during the first operation period, their driving circuits are connected in series (different chilled water temperatures for supply air dehumidification and chilled ceilings). This mode works reliable, the temperature difference in the solar loop is increased by this measure.

Photos

		
Collectors system, installed at the main college building	Chilled ceilings	One of the supply air systems

Monitoring Data

Measured period	Jan 2009 – Aug 2010
Monitoring level (according to Task 38 procedure)	3
Monitoring by	ZfS Rationelle Energietechnik GmbH, Hilden
Person responsible for evaluation	Edo Wiemken (Fraunhofer ISE, Germany) Phone: +49 (0)761 4588 5412 Email: edo.wiemken@ise.fraunhofer.de

2.10 Germany: Fraunhofer ISE, Freiburg

Solar air-conditioning with adsorption chiller of the canteen kitchen area at Fraunhofer ISE, Freiburg, Germany

Description of the application

The Fraunhofer Institute for Solar Energy Systems (ISE) institute building is an energy efficient building with passive cooling measures. An exception is the canteen kitchen area, where due to high internal loads active cooling of the supply air is appropriate. This is done by means of a small size thermally driven chiller.

Type of building kitchen area of institute

Location Freiburg, Germany

In operation since 2007

System operated by Fraunhofer ISE

Air-conditioned area 42 m^2

System used for space heating? Yes

System used for DHW preparation? No

General description of the system

The system technology is a closed cycle chilled water system with an adsorption chiller. Heat is provided by a solar thermal system and by the heat network of the institute which is fed by a CHP unit. During summer, the system runs in cooling mode. The medium temperature heat of the chiller is rejected by three ground boreholes of 80 m each. In winter, the heat pump function of the machine is activated and the ground tubes act as low-temperature energy source. The system thus cools the supply air into the kitchen (max. flow rate 3000m^3/h) and pre-heats the air in the main channel of the air handling unit (max. flow rate 9000m^3/h).

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	5.5 kW$_{cold}$ (08/2008 – 07/2009) 7.5kW (since 11/2009)
Type of closed system	Adsorption
Brand of chiller unit	SorTech ACS 05 (till 08/2009) ACS08 (since 11/2009)
Chilled water application	supply air cooling
Dehumidification	occasionally
Heat rejection system	dry, ground coupled boreholes

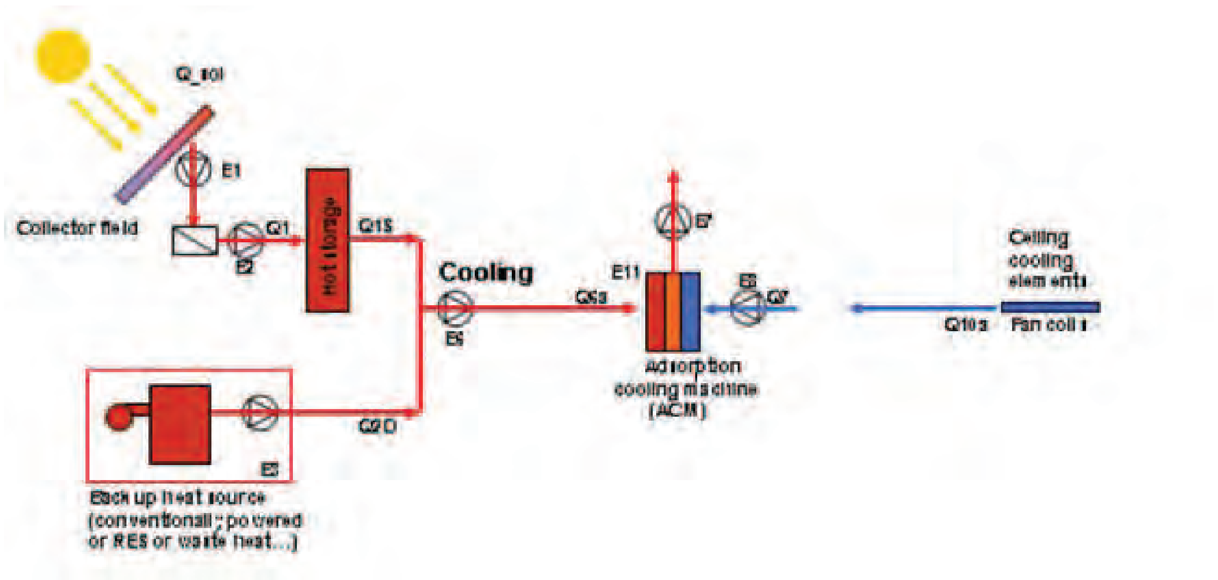
Solar thermal

Collector type	flat-plate
Brand of collector	Solvis FF 35s 3/2 FKY
Collector area	22 m^2 aperture
Tilt angle, orientation	30°, south
Collector fluid	water-glycol
Typical operation temperature	75 °C driving temperature for chiller operation

Configuration

Heat storage	2 m^3 water
Cold storage	none
Auxiliary heater	Institute heat network, operated by CHP and gas boiler
Use of auxiliary heating system	Auxiliary driving source for chiller, auxiliary driving source for heat pump operation in winter
Auxiliary chiller	no

System scheme



System performance

For one year of operation (August 2008 until July 2009) an average seasonal Coefficient of Performance of COP = 0.43 was observed. The solar thermal coverage of the total heat input for cooling was around 45% while for heating it was 6.7%. It has to be remarked that the solar energy in winter is only used to drive the adsorption heat pump if it reaches 70°C and is not used directly for heating. The chiller was newly evacuated before the monitoring period. During winter, the machine runs in heat pump mode; the resulting COP in this operation mode for the whole heating season was 1.25.

In summer 2009 the adsorption chiller ACS05 was exchanged by the model ACS08, which started its operation in November 2009. In the heating season 2009/2010 (Sept. 2009-May 2010) a thermal COP of 1.27 was achieved while the solar fraction in the driving heat was 3.2%. This means that 3.2% of the driving heat was provided by the solar system. The summer period 2010 (April – Sept. 2010) a cooling COP of 0.58 and a solar fraction of 65.2% was obtained.

It has to be mentioned that the systems was operated every day from around 8 to 17 o'clock whenever the temperatures in the supply air were above 20°C (Cooling mode) and below 14.5°C (heating mode). Solar energy was used if the temperature in the upper part of the storage was above 73°C. Further, it has to be mentioned that the driving temperatures from the CHP driven network were very irregular and often in the range between 55 and 70°C.

System reliability and overall success of the installation

After a decrease in performance during the first operation months (inert gasses), the chiller ACS05 was newly evacuated in August 2008. Apart from this, the operation was very reliable. The new chiller (ACS08) did not require any service in the operation period nor showed any degradation in its performance. The system concept is promising: no cooling tower is required, and a fraction of the rejected heat in summer in the borehole may be available in the low-temperature heat source during the heat pump operation mode in winter. Another advantage is the noiseless operation of the chiller. The use of the solar resource could be improved by using the solar heat in winter directly instead of driving the adsorption machine. This would be advantageous as lower temperatures could be used.

Photos

		
22m^2 of Solvis flat-plate collectors on the instute's rooftop	SorTech ACS 05 chiller unit	Fraunhofer ISE institute building. In the foreground: installation of ground tubes

Monitoring Data

Measured period	Aug 2008 – July 2009 (ACS05); Nov. 2009 – Sept. 2010 (ACS08)
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Tomas Nú□ez (Fraunhofer ISE, Germany) Phone: +49 761 4588 5533 Email: tomas.nunez@ise.fraunhofer.de

2.11 Germany: ZAE Bayern Office Building, Garching

Description of the application

Within the framework of the German “Solarthermie 2000plus” Programme a pilot installation of an innovative solar heating and cooling system has been erected at the office building of the ZAE Bayern in Garching, Germany in 2007. The installation simultaneously serves as a field test project for a compact water/LiBr absorption chiller with 10 kW nominal capacity. Further more the performance and durability of a low temperature latent heat storage based on salt hydrate is tested.

Type of building	Office building
Location	Garching, Germany
In operation since	2007
System operated by	ZAE Bayern
Air-conditioned area	400 m²
System used for space heating?	Yes
System used for DHW preparation?	Yes

General description of the system

In conventional absorption cooling installations, wet cooling towers designed for coolant supply/return temperature 27/35°C are applied. When a dry air-cooler is to be used, cooling water temperatures have to be increased. As a consequence of the increase of the cooling water temperature, the temperature level of the driving heat supplied to the regenerator of the absorption chiller has to be increased accordingly. By integration of a latent heat storage into the heat rejection system of the absorption chiller, a part of the reject heat of the chiller can be buffered during the operation of the solar cooling system, allowing for lower coolant temperatures during peak load operation of the chiller. The stored reject heat then can be discharged during off-peak operation or night time when more favourable ambient conditions, i.e. lower ambient temperatures or electricity tariff, are available. During heating operation the latent heat storage balances heat generation by the solar system and other heat sources and the supply to the consumer. Thus a low operating temperature of the solar thermal system is accomplished yielding efficient operation with optimum solar gain.

Central air-conditioning unit

Technology	Closed cycle
Nominal capacity	10 kW $_{cold}$ (basic load)
Type of closed system	Absorption
Brand of chiller unit	Sk Sonnenklima: Suninverse
Chilled water application	Ceiling panel
Dehumidification	No
Heat rejection system	Dry cooler supported by a latent heat storage

Solar thermal

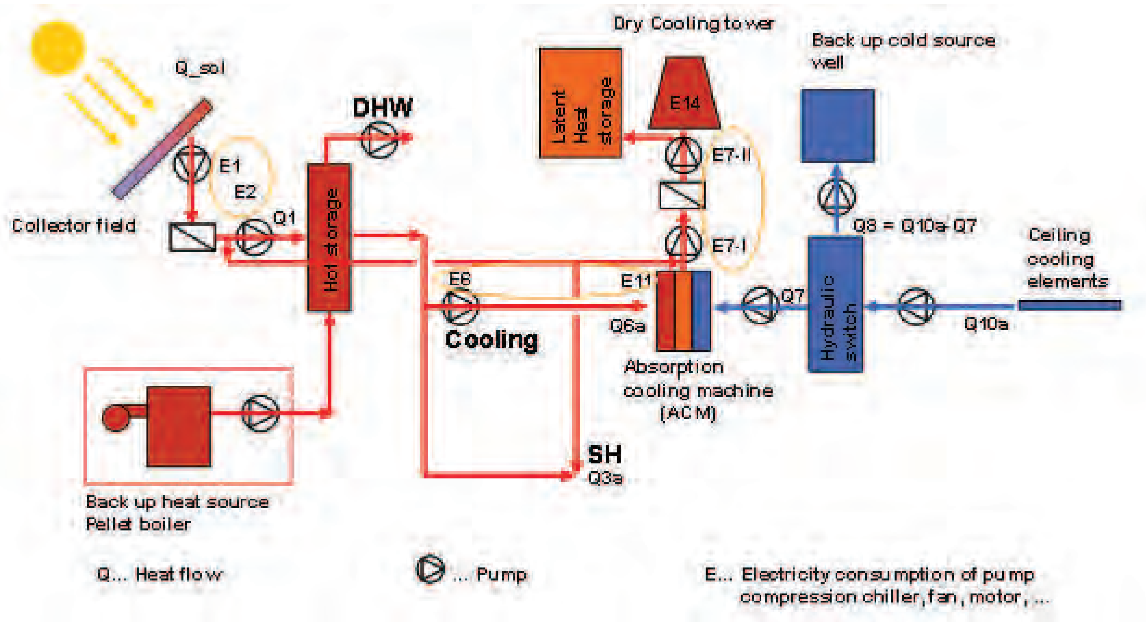
Collector type	Flat plate
Brand of collector	Wagner EURO C20 AR
Collector area	57.4 m²
Tilt angle, orientation	40°, south +10°
Collector fluid	Water-glycol

Typical operation temperature	92 °C driving temperature for chiller operation

Configuration

Heat storage	2x1 m³ water tank (serial) and 1,6 m³ latent heat storage
Cold storage	No
Auxiliary heater	Pellet boiler
Use of auxiliary heating system	Yes
Auxiliary chiller	Well

System scheme



System performance

The system has been in operation and monitored for three complete years (2008-2010). Due to the low system temperatures for heating and moderate chilled water temperatures supplied to the activated ceilings in summer a high collector yield of 400 kWh / (m² a) is obtained.

The absorption chiller itself has been operated for about 600 hours a year, producing cold at 15 °C by means of solar heat about 90 °C with an average thermal Coefficient of Performance of 0.69 up to 0.72. By replacing some ineffective pumps with high efficiency components in spring 2009 the overall electrical COP increased significantly (about 20 %). The average total electrical COP of the solar cooling system was 6.6 in 2009. Almost 60% of the primary energy has been saved compared to a conventional compression cooling system supplying the same amount of cold.

During the monitoring period the two latent heat storage modules have undergone over 800 loading and unloading cycles under real conditions. Due to the use of the latent heat storage the cooling water return temperature did not exceed 33.5 °C despite dry air cooling and ambient temperatures above 32 °C. In heating mode about 15 % of the overall heat demand has been provided by the latent heat storage.

System reliability and overall success of the installation

After installation in 2007 followed by an optimization phase of the control strategy, the solar cooling and heating system has been operated in automatic mode since January 2008.

- Solar thermal system and hot water buffer

The solar thermal system has been working properly. An overheating of the solar thermal panel system is avoided by using the dry air cooler as heat sink for surplus solar heat. The system performance can be improved by adding an irradiation related speed control of the primary loop pump and an enlarged heat exchanger.

- Absorption chiller

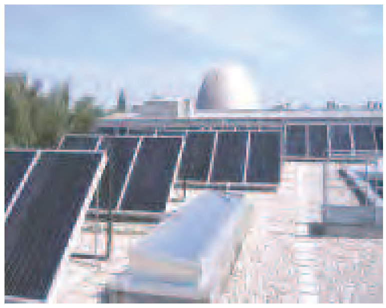
The thermally driven chiller worked quite well. Only once the automatic anti-crystallization routine has been activated due to inert gases caused by corrosion. Even after standstill during winter the chiller has been available for automatic operation without manual intervention. Yet, the chiller operated at reduced thermal COP due to the effect of inert gases accumulated during standstill. After a manual evacuation the chiller reached nominal values again.

- Reject heat loop with dry air cooler and latent heat storage modules

The dry air cooler has been cleaned once a year in spring to remove pollen of the heat exchanger surface. No further maintenance has taken place. By implementing a 3-way valve in the hydraulic loop of the latent heat storage a speed control of the fan is no longer required in further installations. The latent heat storage modules have proved their functionality and reliability. Up to now no degradation in performance or storage capacity has been detected. In summertime cooling water temperature has been reduced significantly and in wintertime the efficiency of the solar system has been increased due to low and constant temperatures of the latent heat storage.

In consequence of the good performance and reliability of the system the monitoring will be continued and further improvements are going to be implemented.

Photos

	 	 
Libr/water absorption chiller: Suninverse	Heat reject system: dry cooler +latent heat storage	Solar panels and office building, ZAE Bayern

Monitoring Data

Measured period	Jan. 2008 – ongoing
Monitoring level (according to Task 38 procedure)	1
Person responsible for monitoring	Martin Helm (ZAE Bayern, Germany) Phone: +49 89 329442 33 Email: helm@muc.zae-bayern.de

2.12 Germany: Radiological Practice, Berlin

Description of the application

In the Radiological Practice in Berlin cooling demand occurs throughout the day and throughout the year for permanent cooling of the tomografic equipment. Originally, the cooling demand is covered by a chilled water network, serving also other medical services in the building. An electrically driven compression chiller is operating the network.

Since summer of 2008, an additional small solar autonomous cooing system was installed for covering daily cooling peak loads of the practice. The approach made is interesting, since

- pre-installed roof-top system: the chiller, hot water storage and all hydraulic components are installed in a small size container and placed on the roof-top of the building, since no further technical area inside the building was available;
- dry heat rejection with an air-cooler, which is used in winter for direct chilled water production without chiller operation (free cooling mode at sufficient low ambient temperatures);
- beside a constant ground level of cooling demand (approx. 8 kW), daily peaks of cooling demand up to 30 kW matches very well the the operation time of the solar thermally driven chiller

A monitoring system was operated in 2009.

Type of building heritage protected commercial building	
Location Berlin, Germany	
In operation since 2008	
System operated by SK Sonnenklima, Berlin	
Air-conditioned area 120 m^2	
System used for space heating? No	
System used for DHW preparation? No	

General description of the system

An absorption chiller was installed and thermally driven with solar heat alone. The solar system is used only for the chiller; due to building related properties, it was not possible to connect the solar system to the building heating network. The solar thermal system was prepared to be operated with pure water and a corresponding control was installed. However, for test reasons, the system was filled with water-glycol.

The plant is pre-cooling the chilled water, returned from the practice. In case the temperature of the chilled water is not sufficient low (approx. 8°C) leaving the absorption chiller, it is subcooled passing a heat exchanger in the building chilled water network.

During winter, chilled water can be prepared directly through the air cooler. This mode was already widely used in 2008 and 2009. As a consequence, the heat rejection circuit as well as the cold water circuit are filled with water-glycol. Also the hot water circuit is using water-glycol (foreseen to be replaced with pure water later).

In order to simplify the installation and maintenance of the roof-top system, dry heat rejection was chosen.

The system is a pilot installation to gain experiences with

- solar pre-cooling in a process cooling network;
- dry heat rejection in combination with the installed absorption chiller;
- free cooling mode in winter
- pre-installed contained roof-top system

Detailed planning: SK Sonnenklima

Central process pre-cooling unit

Technology	closed cycle
Nominal capacity	10 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	SK Sonnenklima: Suninverse
Chilled water application	Process cooling (medical equipment); pre-cooling
Dehumidification	none
Heat rejection system	dry air-cooler

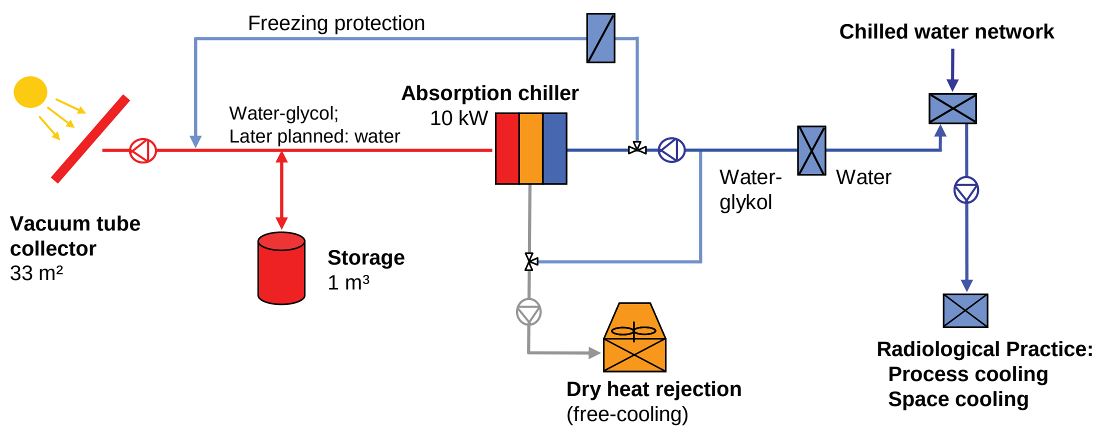
Solar thermal

Collector type	evacuated tube with CPC-mirror
Brand of collector	Phönix Sonnenwärme AG
Collector area	33 m^2 aperture
Tilt angle, orientation	45°, south
Collector fluid	water-glycol
Typical operation temperature	80°C

Configuration

Heat storage	1 m^3 water-glycol
Cold storage	none
Auxiliary heating support	none
Auxiliary heater	none
Auxiliary chiller	backbone chilled water network (el. compression chiller)

System scheme



System performance

For reasons given below, no performance data can be given.

System reliability and overall success of the installation

The system went into operation in late 2008, monitoring data were available since February 2009. In 2009, several adjustments and tests were made on the system. A continuous system operation with effective cooling was available in July and August 2009. However, the system was still not optimized and the monitored performance was below expectations (e.g., monthly thermal COP of below 0.3; chilling capacity too low). Some monitoring signals were not reliable; e.g., the electrical performance could not be calculated, further monitoring errors occurred in the heat rejection circuit. Before solving the remaining problems, the company SK Sonnenklima went into a insolvency procedure. The system operation stopped end of August 2009 and was not restarted since then (also due to the ongoing insolvency).

In the available operation period of the chiller July and August 2009, the system has shown apart from optimizing problems the general ability to reduce the peak-load cooling demand from the chilled water network; pre-cooling was done reliable. Thus, the concept is still promising. Furthermore, the free cooling mode via the cooling tower was also working reliable.

Since the insolvency procedure was not closed until August 2010, the further use of the system is not clear yet.

Photos

		
Collector; the container with the chiller is in the background	Absorption chiller in the container	Dry heat rejection, attached to the container

Monitoring Data

Measured period	Feb 2009 – Aug 2009 (commissioning phase)
Monitoring level (according to Task 38 procedure)	3 (planned)
Monitoring by	Technical University Chemnitz
Person responsible for evaluation	Edo Wiemken (Fraunhofer ISE, Germany) Phone: +49 (0)761 4588 5412 Email: edo.wiemken@ise.fraunhofer.de

2.13 Spain: Gymnasium of the University of Zaragoza, Zaragoza.

Description of the application

The installation is located in Zaragoza (Spain) at the indoor sports centre of the University of Zaragoza and it is used to cool a gymnasium. This installation was designed as a consequence of the overheating in the existing solar collectors. In summer, the solar field was oversized because solar power was higher than needed.

Type of building Gymnasium

Location Zaragoza, Spain

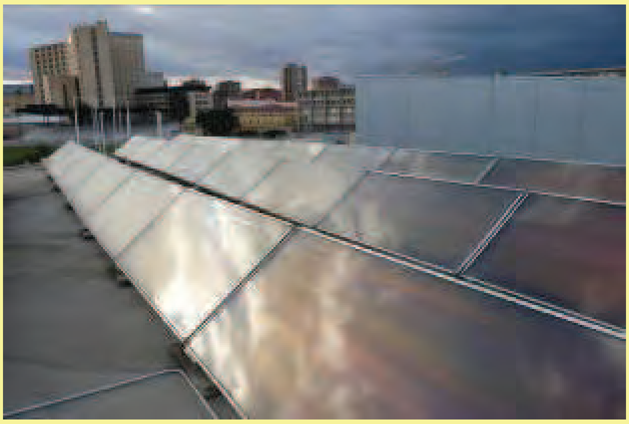
In operation since 2007

System operated by University of Zaragoza

Air-conditioned area 215 m^2

System used for space heating? No

System used for DHW preparation? No



General description of the system

This solar cooling installation was designed as a consequence of the overheating problems of the existing solar filed used to contribute to the domestic hot water supply of the building. In the summer, to solve this problem and to use this solar waste energy, the chosen solution was the installation of an absorption chiller. Therefore, the solar collector field of the solar air-conditioning system has 37.5 m^2 of useful area. Solar radiation is absorbed and transformed into thermal energy to feed a 4.5 kW absorption machine by Rotartica. The installation contains 700 liters of hot water storage and two gas boilers as an auxiliary system, but both have been never used. Two fan coils transfer the chilling power from the evaporator of the absorption machine to the gymnasium.

Initially, a dry cooling tower was installed to reject the produced heat of the absorption cycle to the outdoor air. After the analysis of the two first years, and due to the negative influence of the ambient temperature on the COP of the chiller, an open geothermal system was installed as a heat rejection system, to improve the COP and the performance of the chiller.

Central air-conditioning unit

Technology	Closed cycle
Nominal capacity	4.5 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	Rotartica 045
Chilled water application	Fan coil
Dehumidification	No
Heat rejection system	Dry cooling tower and an open geothermal system since 2009

Solar thermal

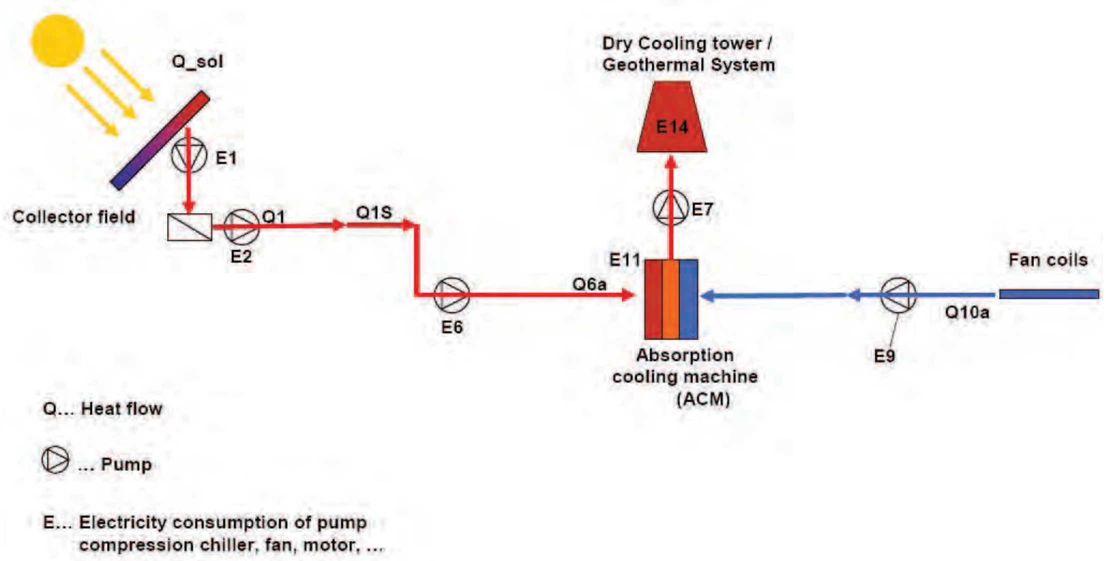
Collector type	Flat-plate
Brand of collector	Viessmann
Collector area	37.5 m^2 aperture area
Tilt angle, orientation	30°, South
Collector fluid	Water-glycol

Typical operation temperature	80 °C driving temperature for chiller operation

Configuration

Heat storage	0.7 m^3 water
Cold storage	no
Auxiliary heater	yes (not used)
Use of auxiliary heating system	no
Auxiliary chiller	none

System scheme



System performance

In the last three years, 2007, 2008, 2009 and 2010, the chiller has been operated under different scenarios of performance. In 2007 the chiller worked with two different solar collector areas (20 m^2 and 37.5 m^2) in order to analyze the influence on the chiller performance The average COP value operating with 20 m^2 was 0.51 whereas with the 37.5m^2 the COP value achieved 0.49, although in the first case the average chilling power 4.0 kW, in the second one, the chilling power increased to 5.3 kW. In both cases, the chiller rotary drum speed was 300 rpm. In the next scenario the chiller worked with the whole solar surface and its rotary speed was increased till 400 rpm to improve the heat and mass process of the absorption cycle. The performance of the system improved (COP: 0.57, W_{ch}: 5.78 kW), but the installation kept on having a strong influence of the outdoor temperature. In the year 2008, the average outdoor temperature was 12% higher than the year before, a fact that was displayed in the results of that year (COP: 0.51, W_{ch}: 4.4 kW). Finally, to eliminate the outdoor temperature influence on the heat rejection system and taken the opportunity of using a closed water well with a constant temperature of 17ºC, in 2009, an open geothermal cycle was installed. The obtained results presented constant values, such as the COP as well as the chilling and heat rejection powers, but they were lower than the expected (COP: 0.52, W_{ch}: 4.2 kW), due to the higher operational temperature of the water well (25ºC instead of 17 ºC used in the design process).

System reliability and overall success of the installation

In general, the operation of the installation works reliable. The principal objective of the solar cooling system has served the purpose of resolve the overheating problems of the solar field.

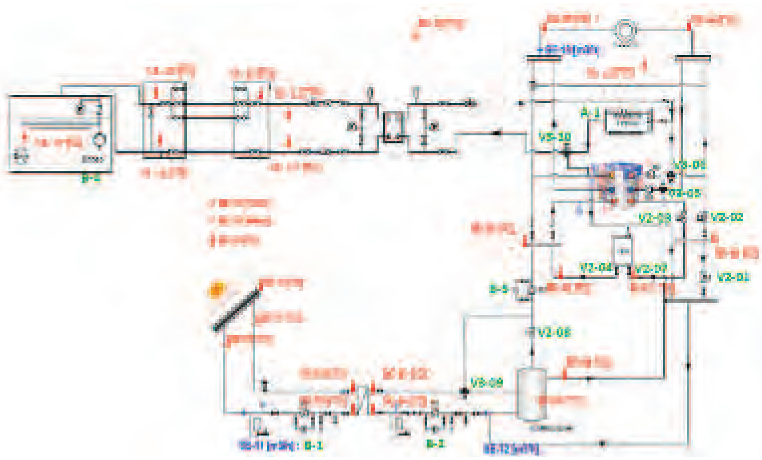
Besides that the users of the gymnasium are satisfactory because the ambient conditions in the gym have been improved.

Photos

		
Solar field	Machinery room	Gymnasium
		
Fan coil	Initial heat rejection system	Geothermal heat rejection system (2009)

Monitoring Data

Measured period	2007 - ongoing
Monitoring level (according to Task 38 procedure)	2
Person responsible for monitoring	Fernando Palacín (National Renewable Energy Centre (CENER)) Phone: +34 948 252 800 Email: fpalacin@cener.com
	Carlos Monné (University of Zaragoza) Phone: +34 976 762 042 Email: cmmb@unizar.es



Scheme of the solar cooling installation in Zaragoza

3 Results

In this chapter, the monitoring results of 11 of the 13 systems described above will be presented and compared with eachother. For two systems, no results can be presented because there were too many operational and/or monitoring problems during the monitoring period.

For 6 of the 11 systems, at least one year of data is available. For all the other systems, between two and six months of data have been recorded and will be presented here.

The 11 presented systems can be subdivided into different groups of systems:

Group 1: Systems that were only used for cooling (Zaragoza and Maclas, 4 months of data)

Group 2: Systems where winter backup was not monitored, the measured space heating consumption is therefore only the part that was produced by the solar thermal system (Perpignan).The system in Perpignan uses a compression chiller as cold backup.

Group 3: Systems that use the hot backup system only for winter operation. In summer, the system is operated either as solar autonomous system or with a cold backup system (Sattledt, Chaméry, Butzbach, Garching, Graz).

Group 4: Systems that use the hot backup system for both summer and winter operation.

3.1 Thermal COP of the Chiller

In all eleven monitored systems, the thermal coefficient of performance (COP) of the thermally driven chiller was determined by measuring the produced cold and the driving heat. Only two systems include an adsorption chiller (Perpignan and Freiburg). In all other systems, absorption chillers are used.

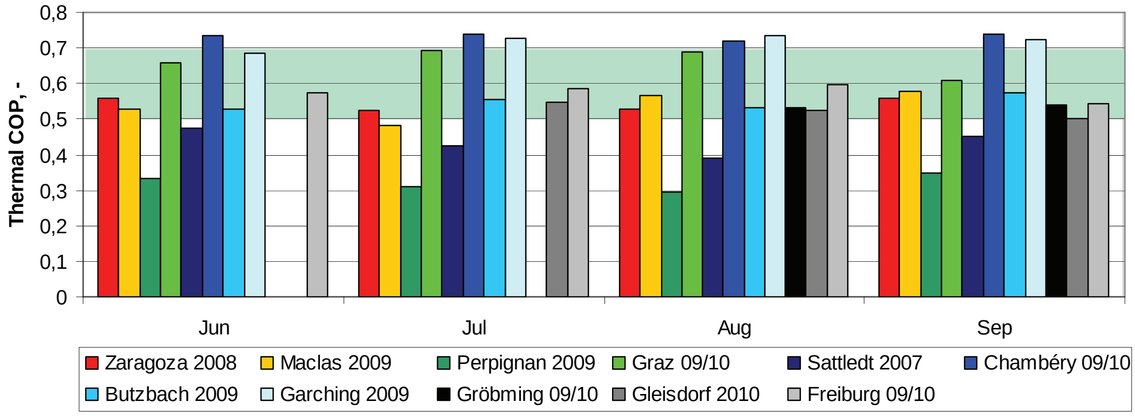


Figure 1: Measured thermal COP of 11 small-scale solar cooling systems.

The results show that almost all chillers have thermal COPs in a range between 0.5 and 0.7, i.e. close to the manufacturers' specifications but of course depending on the operating conditions in the specific system. Only two systems show significantly lower values. One is the adsorption chiller in Perpignan that has to operate under unfavorable heat rejection conditions (high ambient temperatures and most dry heat rejection). The other one is the system in Sattledt where the driving temperature was relatively low (55-75°C).

3.2 Electrical COP of the Chiller

The next step is to look at the electricity consumption of the chiller itself, i.e. the solution pump or any internal valves. This value was measured separately only for 4 of the analyzed systems. The electrical COP of the chiller is defined as the produced cold divided by the electricity consumption of the chiller by itself.

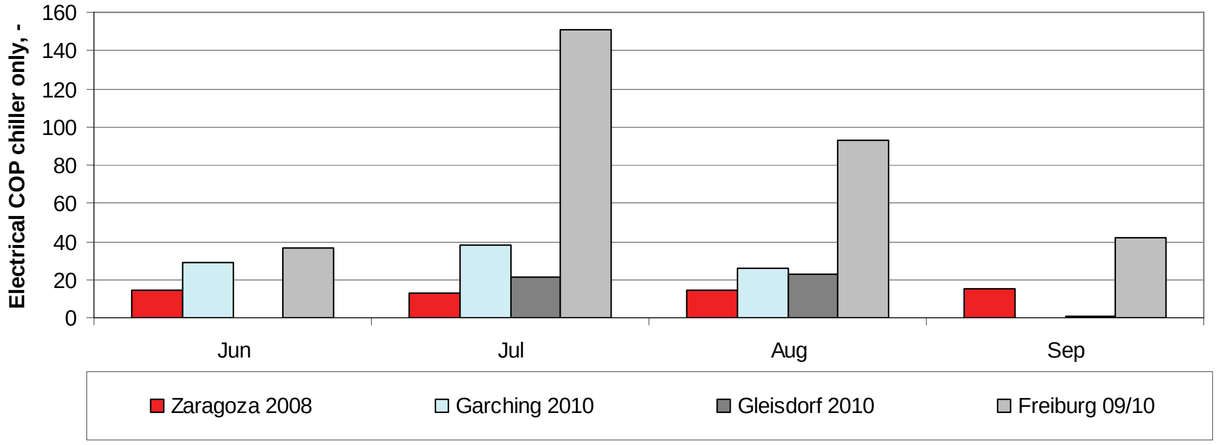


Figure 2: Electrical COP of the chiller only for 4 small-scale solar cooling systems.

The results in Figure 2 show that the electricity consumption of the adsorption chiller in Freiburg has very little electricity consumption. The reason is that there is no solution pump necessary. The electricity consumption of the chiller alone is almost independent of the cold production: it only depends on the switch-on time and this time is almost the same for all months. In July 2010 on the other hand a high amount of cold was produced, increasing significantly the electric COP of the chiller alone. The absorption chillers are all in the same order of magnitude. The best one is the one in Garching. The results shown here are values of 2010 because the electricity consumption of the chiller was not measured separately in 2009. In addition, only the electricity consumption of the solution pump was included and not the pump for the generator circuit which is also included in the chiller. The system in Zaragoza uses a Rotartica absorption chiller which consumes more electricity than other absorption chillers due to its rotating technology.

3.3 Electrical COP Cold Production

The electrical COP of the cold production includes in addition to the electricity consumption of the chiller itself, the electricity consumption of the heat rejection system (pump and fan) and the pump in the generator loop. If there is a cold storage tank, it includes also the electricity consumption of the pump between chiller and cold storage tank. The results for the seven systems where this value was measured are shown in Figure 3. Obviously, the values are significantly lower than for the electrical COP of the chiller by itself. The best system in this monitoring campaign reaches a value of 8. A number of systems lie in a range of 5 to 6 which are acceptable values but still leave room for improvements.

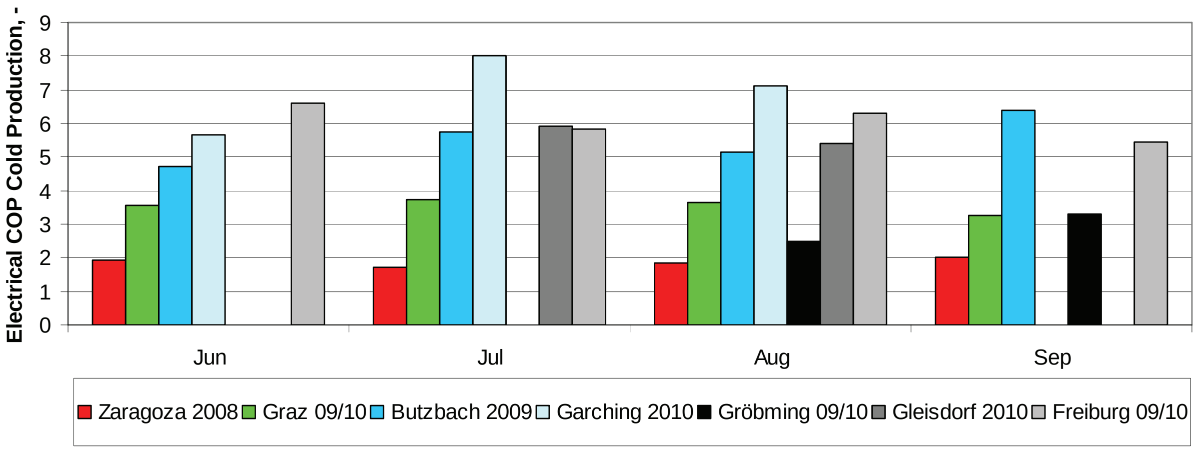


Figure 3: Electrical COP of the cold production

A few systems show values below 3 which means that the system probably consumes more electricity than a conventional compression chiller would.

A general conclusion is that in most systems the electricity consumption of the external components of the cold production (pumps, fans) has not been optimized and is therefore too high. In the best system in Figure 3 (Garching) the pump in the generator loop has been exchanged against a more energy efficient one. In addition it is planned to replace the fan of the dry cooler in the near future which would increase the electrical COP even further.

Another topic is the part load operation of the system. If a system is operated a lot in part load but the external pumps are still operated at full speed, this will lead to low electrical COPs. Optimized control strategies for this operation scenario are mandatory to ensure primary energy savings of the system.

3.4 Total electrical COP of the Solar Cooling System

As a next step, the total electrical COP of the system can be calculated. This includes in addition to the electrical consumers mentioned before, the electricity consumption of the pumps in the solar circuit(s) and the electricity consumption of the auxiliary heater and the associated pumps if applicable.

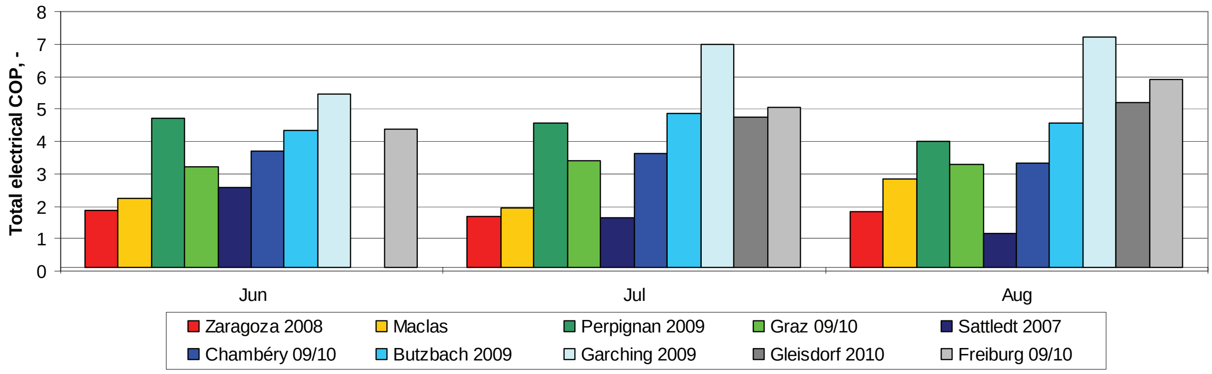


Figure 4: Total electrical COP of 10 solar cooling systems

Figure 4 compares the total electrical COP of 10 systems only for the summer months. The value for the system in Gröbming is not shown because the system was also used for DHW preparation and space heating. Therefore, the value is significantly higher but not comparable.

For the other 10 systems the trend is the same as described in the previous section. But the total electrical COPs of the systems are obviously somewhat lower than for the cold production itself, but the more critical factor seems to be the electricity consumption of the pumps in the chiller circuits and the heat rejection system. These are the components that need to be optimized in terms of component selection and control strategies.

3.5 Fractional Primary Energy Savings

Now, let's take a look at the probably most important monitoring result of a solar heating and cooling system: The primary energy saved by the system.

For this purpose, the primary energy ratio was calculated for both the monitored solar heating and cooling system and for a conventional reference system delivering the same amount of heating, cooling and domestic hot water.

The following assumptions were taken for the conventional reference system:

Space heating is done by a fossil fuel boiler without storage tank. The primary energy ratio for space heating is calculated as follows:

$$PER_{ref,space\_heating} = \frac{Q_{SH}}{\frac{Q_{SH}}{\eta_{boiler} \cdot \varepsilon_{fossil}} + \frac{0.02 \cdot Q_{SH}}{\varepsilon_{elec}}} = \frac{1}{\frac{1}{0.95 \cdot 0.9} + \frac{0.02}{0.4}} = 81.99\,\%$$

with

η_{boiler}	Mean annual efficiency of fossil fuel boiler:	0.95
ε_{fossil}	Primary energy factor for fossil fuel:	0.9 kWh_{final}/kWh_{PE}
ε_{elec}	Annual electricity generation efficiency:	0.4 kWh_{el}/kWh_{PE}

The electricity consumption of the fossil fuel boiler was assumed to be 2% of the generated heat of the boiler. All these assumption lead to a reference primary energy ratio for space heating of 81.99%. The numbers listed above are mean European values and have been chosen in order to compare different systems. Local values may be used in order to evaluate the saving in a specific location.

Domestic hot water preparation is done using the same fossil fuel boiler than for space heating but assuming a storage tank containing 75% of the average daily hot water consumption in the monitored system. Average heat losses of such a tank were then added to the measured heat consumed for domestic hot water.

Cooling is done by a conventional compression chiller. The primary energy ratio for cooling is calculated as follows:

$$PER_{ref,cooling} = \frac{Q_{cold}}{\frac{Q_{cold}}{SPF_{ref} \cdot \varepsilon_{elec}}} = \frac{1}{\frac{1}{2.8 \cdot 0.4}} = 112\,\%$$

With

SPF_{ref}	Reference seasonal performance factor:	2.8

This leads to a reference primary energy ratio for cooling of 112%.

This reference primary energy consumption is then compared with the measured primary energy consumption of the system according to the equations below:

$$f_{sav,shc} = 1 - \frac{\frac{Q_{boiler}}{\varepsilon_{fossil} \cdot \eta_{boiler}} + \frac{Q_{RES}}{\varepsilon_{RES} \cdot \eta_{RES}} + \frac{E_{el}}{\varepsilon_{elec}} + \frac{Q_{cooling,missed}}{SPF \cdot \varepsilon_{elec}}}{\frac{Q_{boiler,ref}}{\varepsilon_{fossil} \cdot \eta_{boiler,ref}} + \frac{E_{el,ref}}{\varepsilon_{elec}} + \frac{Q_{cooling,ref}}{SPF_{ref} \cdot \varepsilon_{elec}}}$$

$$f_{sav,shc} = 1 - \frac{PER_{ref}}{PER}$$

Some monitored systems use fossil backup systems. In that case, the same primary energy factors and efficiencies as for the reference system were used. Also for the electricity consumption the same electricity conversion efficiency was used.

For other backup systems (in the equation called renewable energy sources Q_{RES}), individual primary energy factors and efficiencies were used. The primary energy factor ε_{RES} here is defined as the “non renewable primary energy factor”:

a) Wood pellets (only Gröbming): η_{RES}=0.9, ε_{RES}=10 kWh_{fuel}/kWh_{PE}

b) District heating system city of Graz: η_{RES}=1, ε_{RES}=0.96 kWh_{heat}/kWh_{PE}

c) Combined heat and power plants (CHP):

The primary energy for heat taken from a CHP plant is calculated according to the following equation:

$$\varepsilon_{RES} = \frac{Q_{CHP}}{H_{fuel}/\varepsilon_{fuel} - W_{el}/\varepsilon_{el}}$$

Where Q_{CHP} is the heat produced by the CHP plant, H_{fuel} is the fuel used (in kWh) and W_{el} is the electricity produced. The electricity produced by the CHP replaces electricity in the power grid, thus ε_{el} is the primary energy factor for the electricity in the grid. Expressed in CHP efficiencies this gives:

$$\varepsilon_{RES} = \frac{\eta_{th}}{1/\varepsilon_{fuel} - \eta_{el}/\varepsilon_{el}}$$

Where η_{th} is the thermal efficiency of the CHP and η_{el} is the electric efficiency of the CHP.

With this formalism, the efficiency of the heat source η_{RES} has to be considered as equal to 1.

In order to compare different systems in different locations the same average primary factors for electricity (ε_{elect}=0.4 kWh_{el}/kWh_{PE}) and fuel ($\varepsilon_{fuel}=\varepsilon_{fossil}$=0.9 kWh_{fuel}/kWh_{PE}) should be used. Local values should be used in order to evaluate the performance of the system in a specific surrounding.

For Freiburg (η_{th}=49.2%; η_{el}=26.2%), using average values for ε_{elec}=0.4 kWh_{el}/kWh_{PE} and ε_{fuel}=0.9 kWh_{fuel}/kWh_{PE} (natural gas) we obtain an average value for the CHP of ε_{RES}=1.08 kWh_{heat}/kWh_{PE}, with local values (ε_{fossil}=0.909 kWh_{fuel}/kWh_{PE}; ε_{elec}=0.36 kWh_{el}/kWh_{PE}) we obtain an average value for the CHP of ε_{RES}=1.32 kWh_{heat}/kWh_{PE}.

For Gleisdorf thermal and electrical efficiency of the two bio-fuel (rapeseed oil) CHP's needs to be estimated based on data sheets, because fuel consumption was not measured. The following values were assumed: η_{th}=45%; η_{el}=24%. Using average values for ε_{elec}=0.4 and ε_{fuel}= 2.35 kWh_{fuel}/kWh_{PE} (rapeseed oil) we obtain an average value for the CHP of: ε_{RES}= -2.53 kWh_{heat}/kWh_{PE}. This negative value shows that the produced electricity is replacing more non renewable energy from conventional (fossil) electricity production (η_{el} x ε_{fuel} = 0.24 x 2.35 = 0.564 kWh_{el}/kWh_{PE} instead of 0.4 kWh_{el}/kWh_{PE}) than non renewable energy remains in the produced heat by the rapeseed oil fired CHP.

To present the results more clearly, the graphs of the fractional primary energy savings are divided into summer and winter. This way, systems that were only operated or monitored in summer can be taken out from the winter graphs.

3.5.1 Summer

All 11 monitored systems could be analyzed for summer operation. In Figure 5 all systems that have no backup system in summer or a cold backup system are shown.

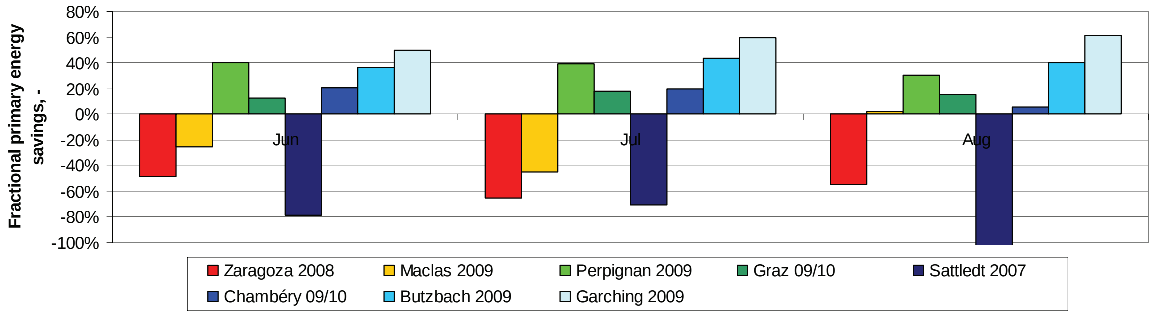


Figure 5: Fractional primary energy savings of the monitored systems without heat backup for the summer months

Five systems show positive fractional primary energy savings, in 3 cases these savings are negative.

Let's first look at the positive cases: The best is the system in Garching which already had the best total electrical COP. The system reaches approximately 60% in July and August. Four other systems (Perpignan, Butzbach, Graz and Chambéry) had reached total electrical COPs between 3 and 5 which translate into fractional primary energy savings between approximately 10 and 40%. While it is good that the primary energy savings of these systems are positive, as has been said before all of the systems can still be optimized (especially in terms of electricity consumption) which would further increase the fractional primary energy savings.

The systems with negative fractional primary energy savings are the ones in Zaragoza, Maclas and Sattledt which had already shown very low total electrical COPs. The fractional primary energy savings of these systems can very likely become positive by means of system optimization.

The three systems shown in Figure 6 use not only solar energy to operate the sorption chiller but a pellets boiler (Gröbming), a gas driven CHP unit (Freiburg) and a combination of a conventional gas boiler and different rapeseed oil fired CHP units (Gleisdorf) respectively.

For a differentiated analysis primary energy savings were calculated based on three different boundary conditions and assumptions respectively:

1) Primary energy factor for the fuel is the "non renewable primary energy factor" for the renewable heat sources (pellets in Gröbming: η_{RES}=0.9, ε_{RES}=10 kWh_{fuel}/kWh_{PE}; natural gas CHP in Freiburg: η_{RES}=1, ε_{RES}=1.08 $kWh_{heat}/kWh_{PE;}$ rapeseed oil CHP in Gleisdorf: η_{RES}=1, ε_{RES}=-2.53 kWh_{heat}/kWh_{PE} (in Figure 6: "res PE factor" or "CHP PE factor")
2) In case of CHP as auxiliary heater the heat is counted for free in terms of non renewable primary energy consumption based on the assumption, that the CHP is operated "electricity production controlled" and heat therefore is waste heat. (in Figure 6: "heat free")
3) For comparison reasons the auxiliary heater is assumed to be the reference natural gas boiler with boiler efficiency η_{boiler} = 0.95 and primary energy factor of natural gas ε_{fossil} = 0.9 kWh_{fuel}/kWh_{PE} (in Figure 6: "fossil PE factor")

The system in Gröbming was only measured in August. At first sight it seems to be the best system in this monitoring campaign (80% of fractional primary energy savings based on "res PE factor" are reached). However, there are two reasons for that: First, the backup system uses a wood pellets boiler with a primary energy factor of 10 kWh_{fuel}/kWh_{PE}. In addition, unlike the other 10 systems, the system was also used for domestic hot water preparation. Both factors are very positive for the performance of a solar heating and cooling system. Mainly due to the different fuel type it makes it difficult to compare the results with the other systems. Therefore, a comparison assuming the same (fossil) primary energy factor (0.9 kWh_{fuel}/kWh_{PE}) was performed. In Figure 6 this is shown in the three hatched columns on the right hand side. For Gröbming, the fractional primary energy savings are thus reduced to roughly 30%.

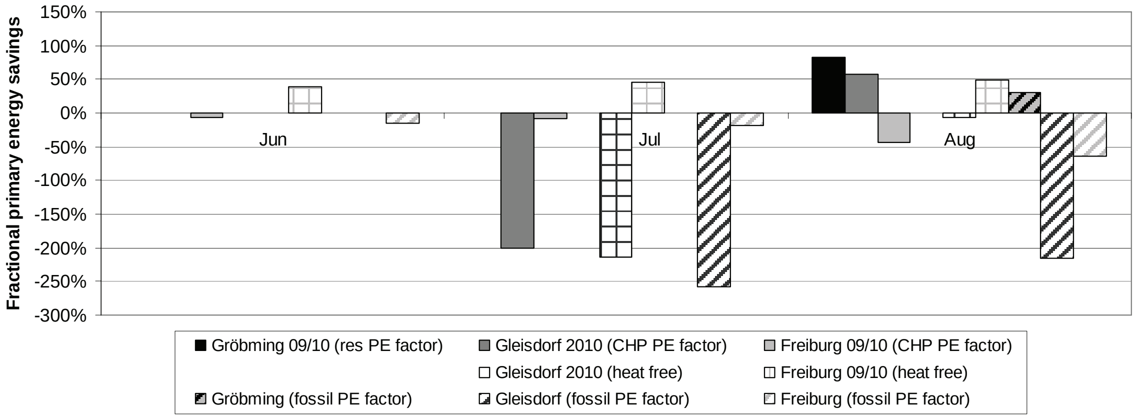


Figure 6: Fractional primary energy savings of the monitored systems with heat backup for the summer months, primary energy factors as RES (solid columns), primary energy factors with heat from CHP is for free (crossed columns) and fossil primary energy factor (hatched columns). In June only Freiburg, in August only Freiburg and Gleisdorf, in August all 3 systems.

Die beiden “heat free” Varianten sehen fast gleich aus, vielleicht sollte man es doch besser bunt machen!

Finally there are two systems with partly high negative fractional primary energy “savings” which had good or very good total electrical COPs: Freiburg and Gleisdorf. The reason for negative savings in this case is the use of a heat backup system rather than a cold backup system. In both cases, these are combined heat and power plants with primary energy factors ε_{RES} better than the reference fossil fuel ε_{fossil}. But the solar fractions of both plants are too low to compensate for the lower seasonal performance factor of the thermally driven chiller compared to the reference compression chiller.

In the case of Freiburg, the solar fraction is still relatively high, leading to only minus 5-45% (Jun-Aug) savings in the case “CHP PE factor” but real savings of plus 39-49% (Jun-Aug) in the case “heat free”.

In Gleisdorf, the system is mainly operated with non-solar but partly fossil and partly renewable heat sources, which translates in case of “CHP PE factor” into minus 200% “savings” in July (mainly natural gas boiler in operation) and 57% real savings in August (mainly the rapeseed fired CHP in operation). In case of “heat free” in July again due to natural gas boiler minus 213% “savings” are achieved, while in August the “savings” are only minus 7% due to mainly CHP operation.

In the case “fossil PE factor” the last three columns each month are comparable in terms of heat performance quality of the systems. They show clearly that both systems in Gleisdorf and Freiburg with only cold demand and high ratio of heat backup do not reach high enough solar fraction for “virtual” primary energy savings. The system in Gröbming is able to save primary energy mainly thanks to comparably high domestic hot water demand (893 kWh) and space heating demand (139 kWh) in comparison to only little cooling demand (175 kWh).

Concluding it can be stated that in solar cooling systems with heat backup based on CHP the primary energy savings are strongly depending on the type of fuel the CHP is fired (fossil or renewable) and how the boundary conditions for the CHP operation are defined (credit thanks to electricity generation or waste heat for free).

3.5.2 Winter

For the winter months only for 5 systems there are sufficient data to analyze the fractional primary energy savings. One important factor is again the used backup heat source. Therefore, just like for the summer case first a comparison using the real existing primary energy factors is shown and then another comparison assuming the same fossil fuel primary energy factor for all 5 systems.

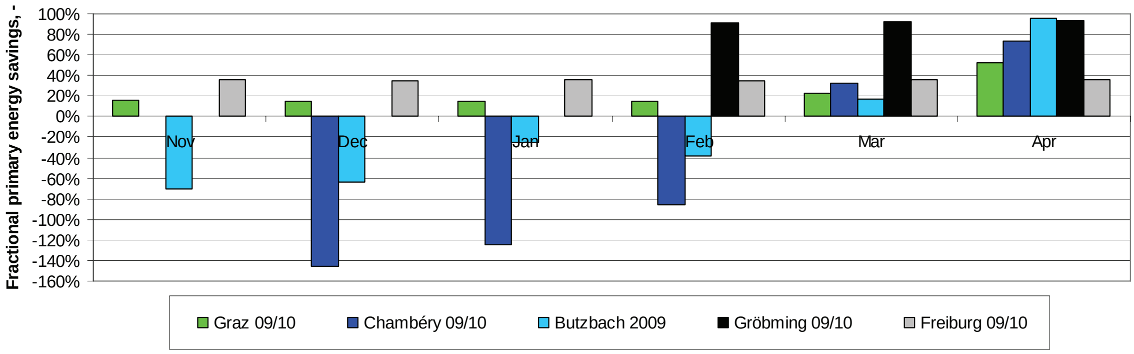


Figure 7: Fractional primary energy savings for the winter months for the 5 systems where sufficient monitoring data are available (real primary energy conversion factors)

The system in Graz uses the municipal district heating network as backup (Primary energy factor 0.96). The fractional primary energy savings are relatively small in November to February which is clear because of the weather conditions and the very small tilt angle of the collectors (11°, optimized for summer operation). However, the primary energy savings reached are still higher (because of the use of district heat instead of a purely fossil backup source). The primary energy savings assuming the primary energy factor of fossil fuels is shown in Figure 8. In March and April already much higher savings could be reached.

The system in Chambéry uses electricity as backup source. This is why the fractional primary energy savings are very negative in months where a lot of backup energy is needed. Figure 8 shows that the situation improves significantly if a gas boiler as backup is used. But in December and January, the savings are still negative because of high heat losses from the storage tank. Obviously, an electric heating element as backup source would not make sense in a real application, but has been used in this experimental installation for simplicity reasons.

In Butzbach, a natural gas boiler is used as backup heat source. But due to storage heat losses the primary fractional energy savings are negative in months without much solar gains.

The system in Gröbming was only monitored in February, March and April. Using 10 as primary energy factor for biomass, fractional primary energy savings of roughly 90% were reached. Assuming a fossil primary energy factor makes it more comparable to the other systems still. Even with that assumption siginifcant savings could be reached. Obviously, the values increase with increasing solar gains.

Finally, the system in Freiburg uses the adsorption chiller as heat pump in the winter months. This system concept reaches approximately 35% fractional primary energy savings for all winter months. These savings have two sources: the primary energy factor for the heat from the CHP unit and the heat pump effect of the chiller. Using a natural gas boiler as backup savings are still achieved, but in a range of only 10%. Savings could be increased if the solar energy is directly used for heating, which was not implemented in the installed system.

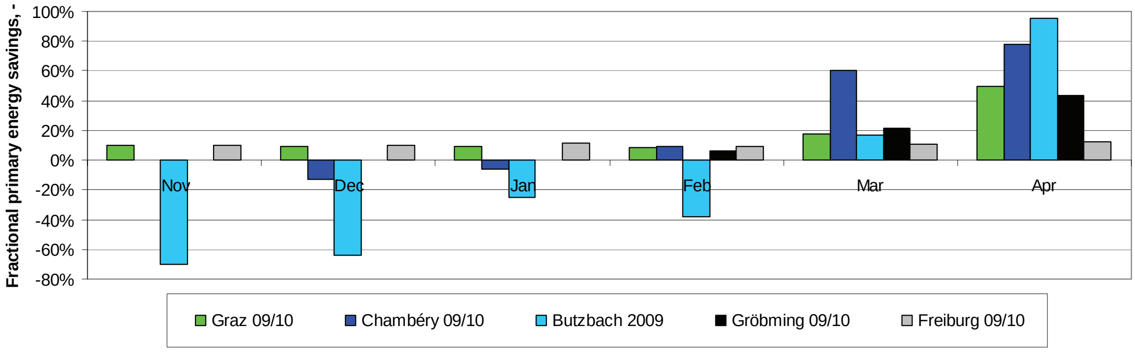


Figure 8: Fractional primary energy savings for the winter months for the 5 systems where sufficient monitoring data is available (assuming backup with gas boiler for all systems)

3.6 Collector Yield

Another interesting figure is the reached collector yield. Only 6 systems were monitored over a whole year of operation. Figure 9 shows the annual collector yield of these 6 systems. The reached values range from 250 kWh / (m² a) to slightly over 400 kWh / (m² a).

Of course the collector yield depends very much on the system concept and the energy management strategy. Monitoring shows that high values around 400 kWh / (m² a) are possible. The systems with significantly lower values very likely still have optimization potential. For example, the system in Freiburg uses solar energy only if it reaches the temperature required to drive the heat pump and not for preheating.

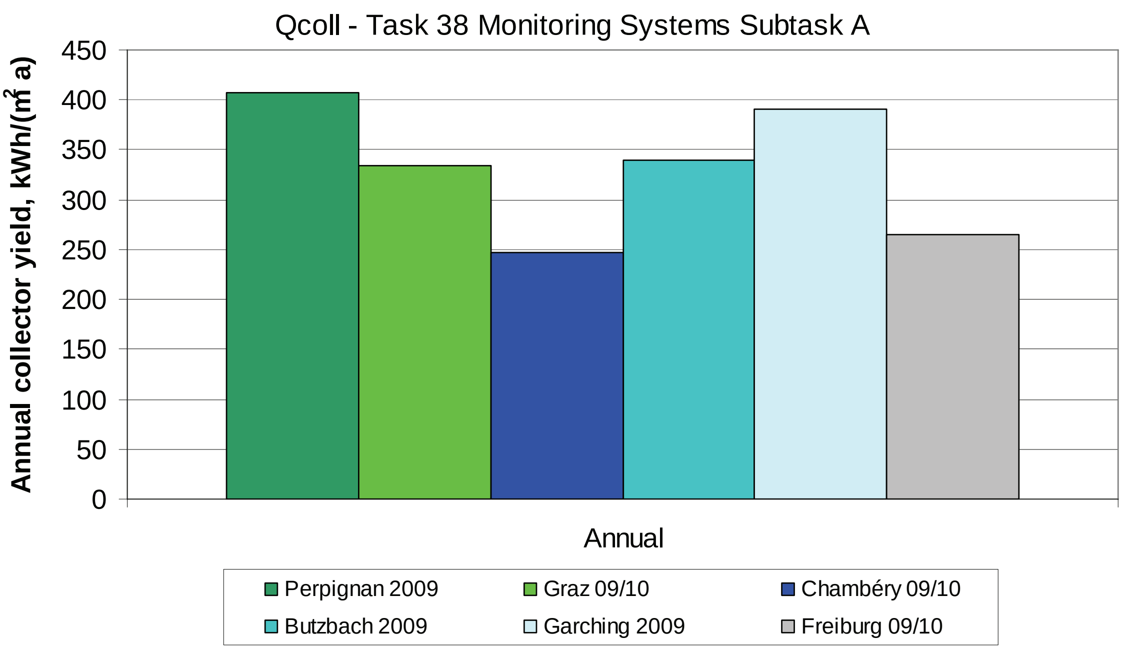


Figure 9: Annual collector yield of the 6 systems with a complete year of monitoring data

3.7 Water Consumption Cooling Towers

For 5 systems the water consumption of the wet or hybrid cooling towers was measured. A sixth system (Freiburg) had no water consumption at all because boreholes were used for rejecting heat.

The Figure 10 shows that the data scatter a lot for the different months and systems. There are some values of 0; this means that the value was not measured for this particular system and this particular month.

The reasons for scattering data are on one hand different weather conditions in the different locations and months as well as the different technologies. The systems in Perpignan and Graz use dry heat rejection systems with external spraying. The other 3 systems use wet cooling towers.

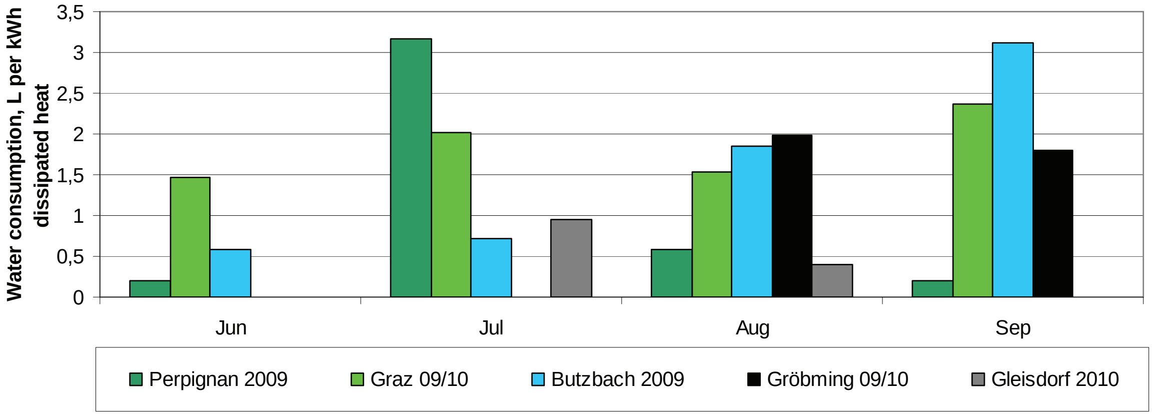


Figure 10: Water consumption of cooling towers per unit of rejected heat

On the other hand, the large differences between the systems also show that also in terms of water consumption, many systems can still be optimized.

4 Conclusions

Within this task, 11 small-scale solar heating and cooling systems have been monitored in great detail. All of them have operated and produced heat and cold reliably during most of the monitoring period.

However, the performance figures vary significantly. Some systems show very good results in terms of total electrical COPs as well as fractional primary energy savings. In some cases, this is due to the fact that during the monitoring campaign the system has been improved in a certain way.

Comparison of the performance figures revealed that all or most systems still have optimization potential. Even the systems that have bad performance figure could very likely be improved to reach good values.

The main optimization potential of most systems lies in the electricity consumption of certain components such as pumps or cooling tower fans. On one hand, the selection of these components is important. Energy efficient units should be chosen. On the other hand, the control strategy especially if operated in part load can reduce the electricity consumption significantly.

This monitoring campaign showed that

- A good system design is important in order to reach good performance figures.
- Monitoring of this kind of system is necessary to ensure proper operation because it still is a relatively new technology. It enables the system supplier to further improve the system during operation and to maximize primary energy savings of the system.

5 Bibliography

To this report related IEA SHC Task38 reports:

B3-B: Monitoring Procedure for Solar Cooling Systems - A joint technical report of subtask A and B

B3-A: Monitored Installations and Results - A technical report of subtask B (large scale applications)

Publications:

- W. Sparber, (2008). 240 - Unified Monitoring Procedure and Performance Assessment for Solar Assisted Heating and Cooling Systems, Eurosun 2008, Lisbon
- A. Thür, D. Jähnig, T. Núñez, E. Wiemken, M. Helm, D. Mugnier, P. Finocchiaro, B. Nocke; Monitoring Program of Small-Scale Solar Heating and Cooling Systems within IEA-SHC Task 38 – Procedure and First Results, Proceedings of EuroSun 2010, Graz, 2010.
- M. Helm, C. Keil, S. Hiebler, H. Mehling, C. Schweigler, Solar heating and cooling system with absorption chiller and low temperature latent heat storage: Energetic performance and operational experience, International Journal of Refrigeration, 32, 2009, 596–606.
- S. Hiebler, H. Mehling, M. Helm, C. Schweigler, Latent heat storage developed for solar heating and cooling with melting temperature 29 °C, Proceedings of Effstock 2009 – 11th International Conference on Thermal Energy Storage, Stockholm, 2009.
- Michael Himpel, Stefan Hiebler, Martin Helm, Christian Schweigler, Long term test results from a Latent Heat Storage developed for a Solar Heating and Cooling System, Proceedings of EuroSun 2010, Graz, 2010.
- Monné, C.; Alonso, S.; Palacín, F.; Serra, L., (2011). ***Monitoring and simulation of an existing solar powered absorption cooling system in Zaragoza (Spain).*** Applied Thermal Engineering, 31, 28-35. doi:10.1016/j.applthermaleng.2010.08.002.
- Monné, C.; Alonso, S.; Palacín, F., (2011). ***Experiencia en una instalación de refrigeración por absorción con energía solar***. Información tecnológica, 22 (3).
- Monné, C.; Palacín, F.; Alonso, S.; Muñoz, M.; Moreno, F., (2011). ***Analyses of a new dissipation system for a solar cooling Installation in Zaragoza (Spain***). Journal of thermodynamics. In Press Correction.
- Monné, C.; Alonso, S.; Guallar, J.; Palacín, F., (2011). ***Stationary analysis of a solar LiBr-H_2O absorption refrigeration system.*** International Journal of Refrigeration. Article in Press. doi: 10.1016/j.ijrefrig.2010.11.009.
- Tomas Núñez, Florian Mehling: "Heating and Cooling with a Small Scale Solar Driven Adsorption Chiller Combined with a Borehole System – Recent Results-" 3rd Int. Conference Solar Air-Conditioning, Sept. 30th to Oct. 2nd, 2009 Palermo, Italy.
- Tomas Núñez: "Small Capacity Solar Cooling: Two Systems Installed in the Frame of the Project SOLERA", Proceedings of EuroSun 2010, Graz, 2010.
- Palacín, F.; Monné, C.; Alonso, S., (2010***). Evaluation, diagnosis and improvement of a solar cooling plant by means of experimental analysis and dynamic simulation***. 8th International Congress on Solar Heating, Cooling and Buildings. Eurosun 2010. 28th September to 1st October 2010. Graz (Austria).

- Palacín, F.; Monné, C.; Alonso, S.; Izquierdo, J. M.; Heredero, J., (2010). ***Design and operational experiences of an alternative heat rejection sink for an existing absorption solar cooling system.*** 8th International Congress on Solar Heating, Cooling and Buildings. Eurosun 2010. 28th September to 1st October 2010. Graz (Austria).
- Palacín, F.; Monné, C.; Alonso, S., (2010). ***Experiences on heat rejection sinks of a solar absorption cooling system.*** 3rd International Conference on Passive and Low Energy Cooling for the Built Environment (PALENC 2010), 5th European Conference on Energy Performance & Indoor Climate in Buildings (EPIC 2010) and 1st Cool Roofs Conference. 30th September to 2nd October 2009. Rhodas Island (Greece).
- Palacín, F.; Monné, C.; Alonso, S., (2010). ***Comparing different sinks of heat rejection of an existing solar powered absorption cooling system.*** International conference on renewable energies and power quality (ICREPQ'10). 25th to 26th Marz 2010. Granada (España).
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Instalación experimental de refrigeración solar - primeros resultados.*** XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th to 21th June 2008. Vigo (España). SPE.
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Influencia de la velocidad de giro en las máquinas rotativas de absorción (parte I)***. XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th al 21th junio 2008. Vigo (España). SPE.
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Influencia de la velocidad de giro en las máquinas rotativas de absorción (parte II)***. XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th al 21th junio 2008. Vigo (España). SPE.

6 Appendix: Detailed Report for Each System

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 1
Monitoring Results of Bachler/Gröbming

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: November 2010

By Daniel Neyer[1] and Werner Pink[2]

[1]Institution	University of Innsbruck
Address	Technikerstr. 13, 6020 Innsbruck, Austria
Phone	+43 512 507-6618
Fax	+43 512 507-36092
e-mail	daniel.neyer@uibk.ac.at
[2]Institution	Pink GmbH Energie- und Speichertechnik
Address	Bahnhofstrasse 22, 8665 Langenwang, Austria
Phone	+43 3854 3666 -30
Fax	+43 3854 3666 -40
e-mail	w.pink@pink.co.at

Contents

1 Background

In the beginning the system was designed as a solar assisted heating system for two office rooms, two apartments and a spa area. Furthermore the swimming pool is heated with the solar energy. As a backup the hot water from a near biomass plant is used. The solar cooling concept was added to the system in a later planning stage. Therefore the absorption chiller as well as the wet cooling tower was placed outside the building in the garden.

The solar cooling plant at the Bachler GmbH training and office building was installed in spring 2007. It is delivering its cooling load to the office trough concrete core activation. This plant was built as an attachment to an existing solar combi system.

Figure 1: Exterior view of the office building in Gröbming, Styria [Bachler]

The system is installed without a conventional backup including a 12 kW Pink absorption chiller working with ammonia/water. The 46 m^2 solar panels are flat plate collectors (type Goliath from Neuma-Solar) integrated in the façade and also placed in front of the building. In the utility room three 1500 liter hot storages as well as all auxiliary hydraulics are placed. The absorption chiller together with the wet cooling tower is positioned outside the house. The hot water of the solar thermal collectors is used for domestic hot water production, for warming the water of the swimming pool and in the heating season it is also used to provide space heating combined with local district heating.

2 System Design

In Figure 2 the hydraulic scheme of the installed system in Gröbming is shown. There are two solar collector fields, one on the ground with 30.2 m^2 and the second is integrated in the façade with 15.8 m^2. From the solar collectors the heat flows through a heat exchanger to the hot water storage. Three 1500 liter water storages connected in series are used to store the solar energy. Depending on the return temperature from the heat exchanger the medium is stratified to the hot medium or cold storage. In winter the return flow from the distribution systems is delivered into storage 1 and in summer, when the cooling mode is running, into storage 3. Out of storage 3 the hot water can be distributed to different recipients, the domestic hot water station, the absorption chiller, the swimming pool as well as the space heating for the building. This can also be done alternatively from storage 1 or 2 if the temperature is sufficient.

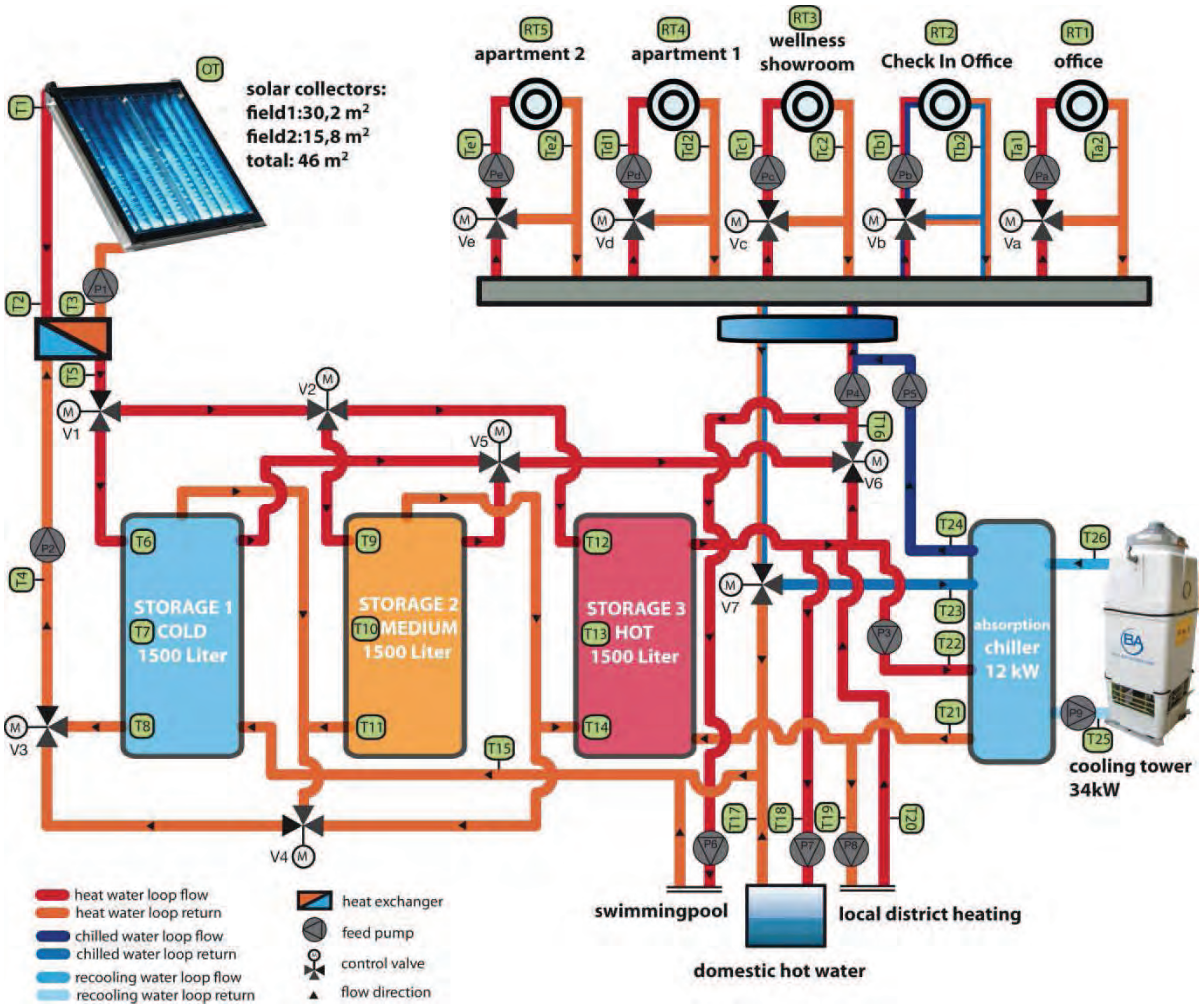


Figure 2: Hydraulic scheme of the solar cooling system in Gröbming

From the central distribution station the hot water is spread to the different recipients. In summer time, when the cooling mode is enabled, the chilled water is also distributed through the same central distributor station.

The recooling water loop connects the absorption chiller directly to the wet cooling tower. The absorption chiller PSC-12 is located outside the building. As a consequence of this location, twice a year a maintenance worker has to discharge and charge the hydraulic cycles of the absorption chiller as well as of the cooling tower before and after the winter period to avoid freezing damages.

All in all nine auxiliary pumps as well as five distribution pumps are installed in the system. To control the plant two Technische Alternative (TA) UVR1611 controllers are included. The monitoring task is done by separate devices installed by the company RKG.

This system design was chosen because of the easy and cheap integration feasibility in the solar combi system. The solar system and even the cold distribution system were there anyway, only the cold production had to be added.

The main problems encountered were caused by the solar system. It was designed originally as a low-flow-system. To drive the absorption chiller higher volume flows are necessary to reach the nominal power. Therefore the pumps of the solar pumping station had to be changed and the loading strategy for the buffer tanks had to be optimized.

Figure 3: View of the utility room [upper left, Bachler], the outside situation with the wet cooling tower and the absorption machine [upper right, Bachler] and the inside of the cooling machine [lower left / right, Pink]

3 Control Strategy

In Figure 4 the control strategy of the solar cooling plant in Gröbming is illustrated. All the temperatures and nomenclatures are linked to the hydraulic scheme (Figure 2). Only the cooling cycle and the management of the solar cycle are described here.

Certainly the combination of the whole system, including the space heating, demands special attention. Through the complex hydraulic layout of the system the control strategy of the solar cooling plant had to be harmonized with all other components such as the domestic hot water station and the pool heating. The main goal was to avoid a clocking behavior of the chiller.

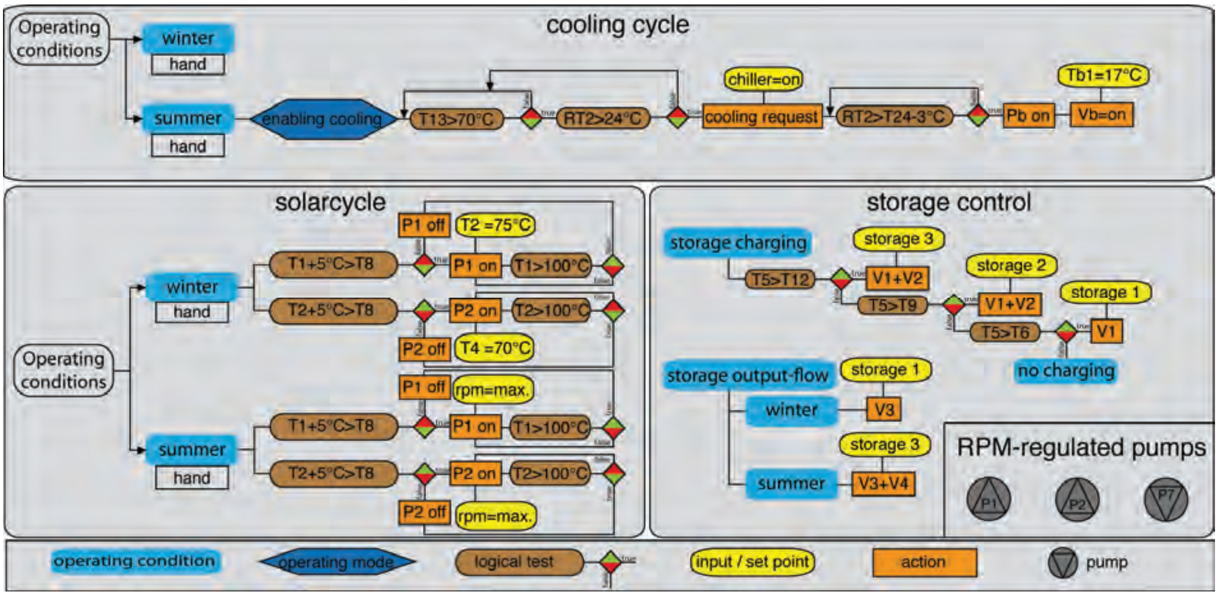


Figure 4: Control strategy of the solar cooling plant in Gröbming

Cooling cycle

To eliminate any control failure regarding the specific winter and summer behaviors the change between winter and summer operation is done by the maintenance technician. If summer mode is switched on cooling is basically enabled. The first continuous logical test is checking if temperature T12 is higher than 70℃. A hysteresis of 7 K is defined to secure that the chiller is not switched on and off too often. If the test turns out true another logical test for the office room is carried out. A cooling request is stated if the room temperature (RT2) exceeds 24℃ and the chiller is switched on. In the next stage the chilled water supply temperature has to be more than 3℃ below the room temperature, then the distributor pump is started and the control valve (Vb) is enabled to regulate the flow temperature at 17℃.

Solar cycle

Switching between summer and winter is done manually. Temperature T1 plus a margin of 5℃ has to be above the lowest temperature of stora ge 1 (T8) to start the primary solar pump. In summer this rpm-regulated pump (P1) is not controlled to any set temperature; it is always running at maximum speed. In winter the primary solar pump is regulated to a set temperature of 75℃. Because there are two solar co llector fields, T1 is averaged out of two collector temperatures measured in those two fields. Analogically the secondary solar cycle is turned on with T2. It is not regulated in summer but regulated to a set temperature of 70℃ in winter. To avoid too high pressure in the hot storage, the solar cycle is switched off if either T1 or T2 is above 100℃.

Storage management

As shown in Figure 2 the entire system has three 1500 liter hot water storages. To use this volume for storing the solar energy effectively, it is important to have a straight forward storage control strategy as it can be seen in Figure 4.

Depending on the temperature T5 the storages 1 to 3 can be charged from the solar collectors. This is especially useful in winter time and transition periods where solar energy should be stored as much as possible. In summer time, when the solar cooling plant is in operation, only the hottest storage (storage 3) is in use. The main reason for not using the entire storage size is the driving temperature of the absorption chiller. Following the manufacturing information of the chiller, a constant driving temperature of 75℃ to 80℃ is recommended. Nevertheless the machine is running down to 65℃ driving temperature, but with poor thermal COPs. To reach those heating temperatures for the chiller it would take too long for the solar plant to charge all three storages. A time offset of cooling demand and cooling distribution would occur. Using only storage 3 brings down the required temperature difference done by the solar panels and raises the volume flow through the collectors. This reduces the time in the morning until the chiller can be started.

4 Monitoring Equipment

4.1 Installed Equipment

Figure 5 shows the symbolic scheme of the system indicating all monitored energy fluxes.

The heat flow from the district heating Q2 is partly used direct and partly heating the hot water storage. This fraction cannot be calculated and therefore Q2 is handled as input to the storage. The heat back up is used for space heating, domestic hot water as well as an unmeant heat back up for the chiller. The heat source from the local district heat is additionally monitored from the district heat supplier. The small hydraulic switch linked to the central distribution station is not taken into account. Cold losses due to the switch are expected quite low.

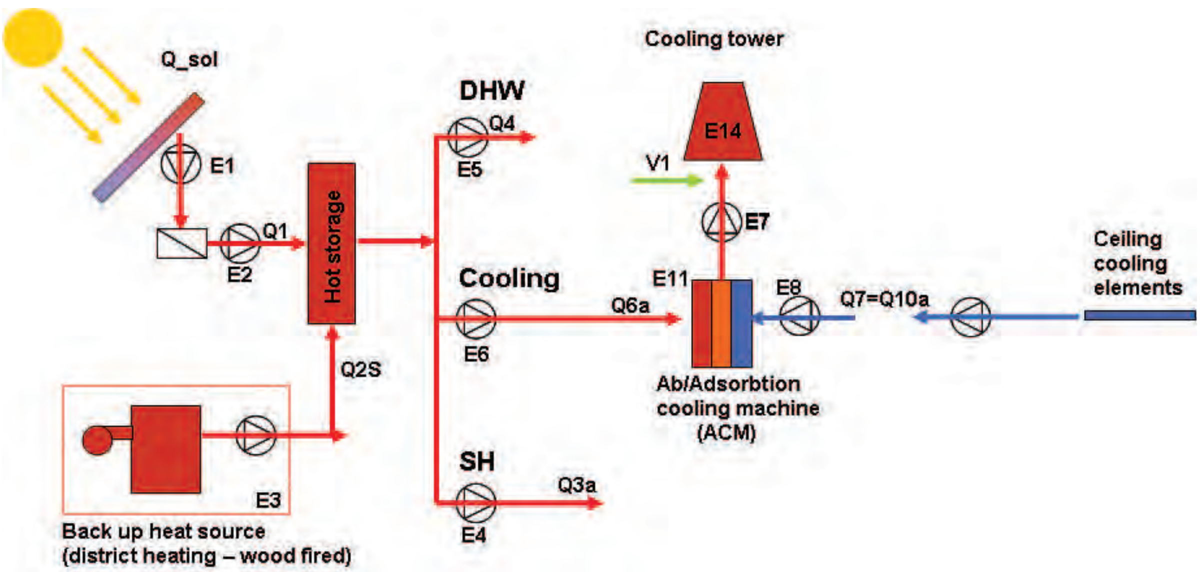


Figure 5: Monitoring scheme of the plant including electricity and heat measurement points

All in all seven heat flow meters and seven electricity meters are installed. The Kipp & Zonen SPlite pyranometer is used to measure solar radiation. The data points for monitoring only are logged separately. The two control units (Technische Alternative TA UVR1611) provide further information (temperatures, pump status,…), which is necessary for a detailed analysis of the plant.

The data is collected by two IQ3 controllers. These controllers are Building Management System controllers, which use Ethernet and TCP/IP networking technologies. Each controller incorporates a web server, which can deliver user-specific web pages to a PC or mobile device running internet browser software. If a system is set up with the correct connections, a user with the appropriate security codes can monitor or adjust the controller from any Internet access point. In the first cooling season (2009) the operator station was located externally at the office of the chiller manufacturer.

Due to a combination of monitoring problems during summer 2009 not enough valuable and comparable monthly data could be collected, but still some useful system operation experiences were made.

An overview of all monitored data points is given in Table 1.

Table 1: Monitoring data points in Gröbming

Quantity	No.	Type	Accuracy
monitoring (IQ3) / control inputs (TA UVR1611)			
temperature (UVR1611)	18	KTY 81-210	± 1% (25℃)
radiation	1	SP-Lite / Kipp&Zonen	temp. Dependence +0.15%/℃ directional error ±5% at 80°
temp./relative humidity	1		±0.5% ℃ / ±3%RH
heat meter energy/volume/power/flow rate/return-supply-temp./ΔT	7 (x7)	Kamstrup Multical 601	E_C= ±(0,15 +2/$\Delta\Theta$)% E_T= ±(0,4 +4/$\Delta\Theta$)%
electricity meter energy/power	7 (x2)	Kamstrup 162B	Class A
logged control outputs (TA UVR1611)			
Speed control	2		
on/off	7		
valve position automatic	10		
valve position hand	2		
…. **sum**	**148**		

4.2 Period of Measurement

The data logging of the control units (TA UVR1611) was started in January 2010.

The first monitoring equipment was installed and commissioned within the construction of the cooling plant in 2007. These measurements were mainly completed to observe the chiller performance. A detailed energy balance of the absorption chiller could be calculated.

The monitoring equipment for level III measurements was completed in July 2009. With this equipment only energy balances can be calculated. For a detailed analysis additional data logging of the control units was implemented in January 2010.

Due to problems in data logging (connection to operator, constant time intervals in logging,...), the operator station was moved to the plant. Now it’s located in the utility room and it’s logging the data from the monitoring and control devices.

5 Monitoring Results

5.1 Annually / Monthly Data

Figure 6 and Figure 7 illustrate the monitoring results for August and September 2009. The black line indicates the space cooling power for the office. The bars show the solar-, the district heat-, the domestic hot water- and the space heating power.

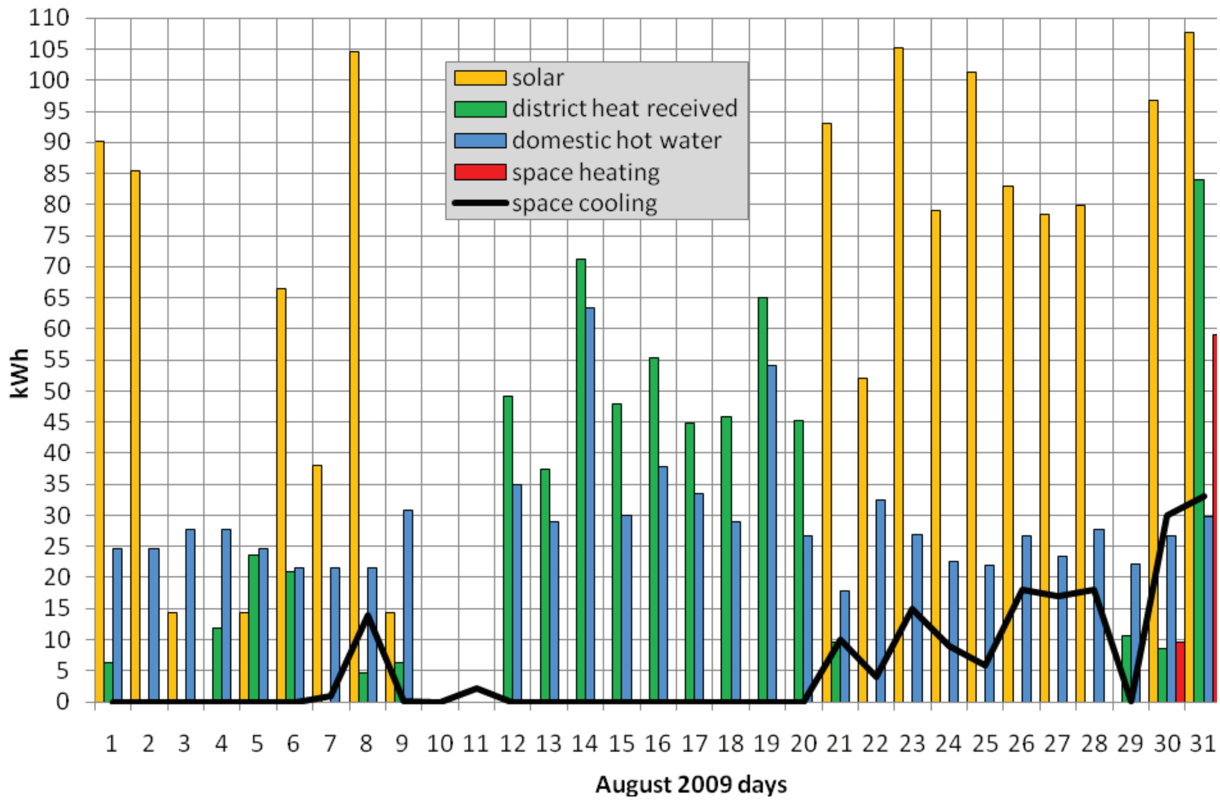


Figure 6: Daily energy fluxes for the plant in Gröbming in August 2009

Before August 2009 there were no cooling activities of the solar cooling plant. Reasons were the cold outside temperatures in May and June especially during night times and therefore no clearance through the control unit (Figure 4). A three day average value for outside temperature was calculated and used. Due to the low night temperatures in Gröbming this value never cut across the limiting temperature to enable the cooling mode.

In the beginning of August this logical test was skipped and the summer mode was turned on manually. On the 8th of August the chiller started for the first time in this summer. Unfortunately one day later one of the two controllers broke. The broken unit was controlling the solar- and the cooling cycle. After an exchange the chiller started every sunny summer day until the end of September. On the last day in August space heating was switched on for the first time. Gröbming is a small township in northern Styria located 776 m above sea level, where night temperatures fall easily below 10-15℃ even in summer. The outside temperature regulated heating system heated regularly during the nights in September.

To heat in the night and to cool during the day raises the costs of providing a pleasant room climate to the office and increases the primary energy consumption of the system as well.

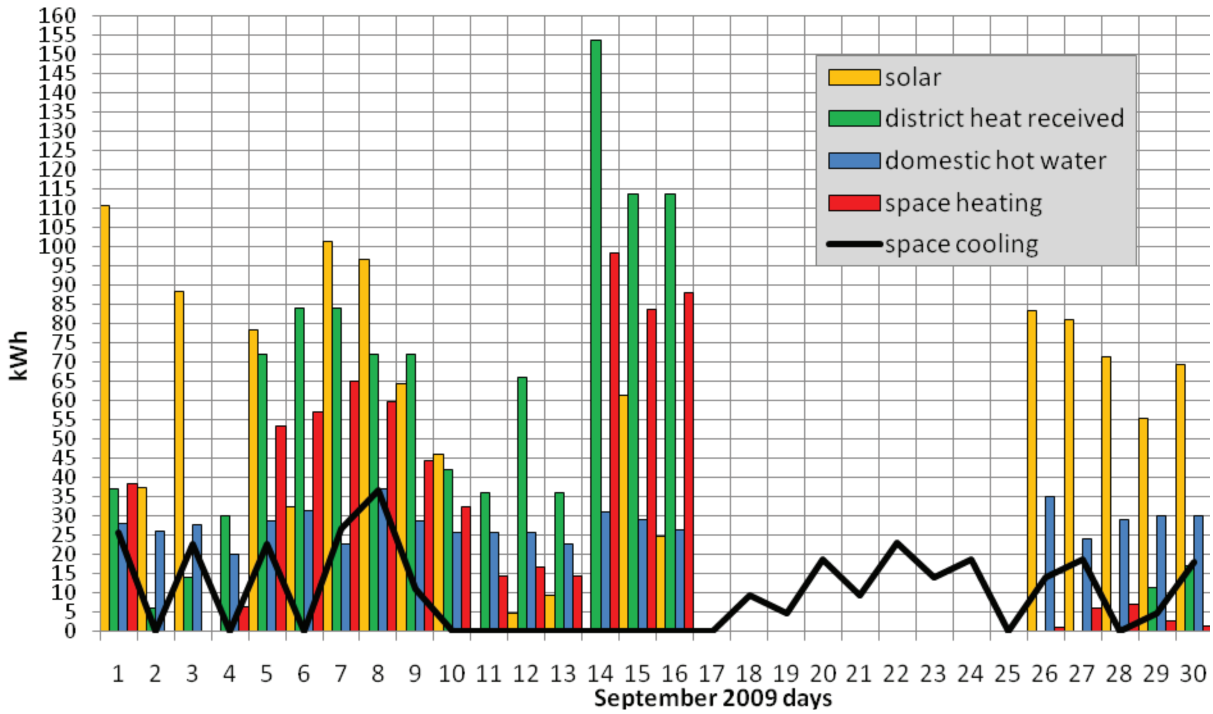


Figure 7: Daily results for the Bachler plant in Gröbming for September

The domestic hot water demand (blue bars in Figure 7) has a stable value of around 25 kWh per day. A nearby biomass heating plant provides district heat to the house shown with the green bars in the diagram. This heat source is planned as a backup system for the domestic hot water production in summer and also for the space heating in winter.

The energy balance of all incoming and outgoing heat flows outlines high overall system losses as shown in Figure 8. An average loss of 28.6 kWh per day is calculated for the period between August 21, 2009 and September 16, 2009. These losses are even higher than the average hot water consumption (25kWh).

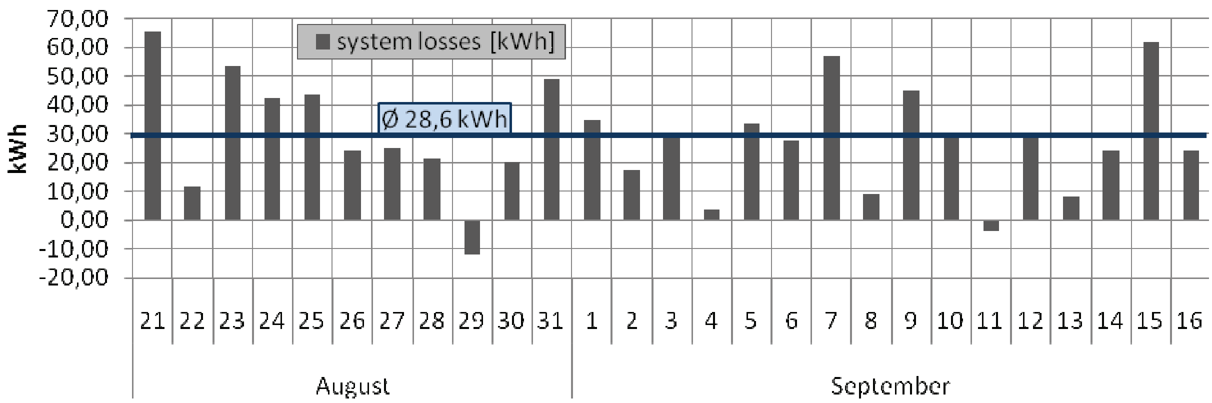


Figure 8: System losses of the solar heating and cooling plant in Gröbming

5.2 Analysis of Typical Days

Figure 9 shows data regarding the ammonia-water chiller on an average sunny day in September 2009.

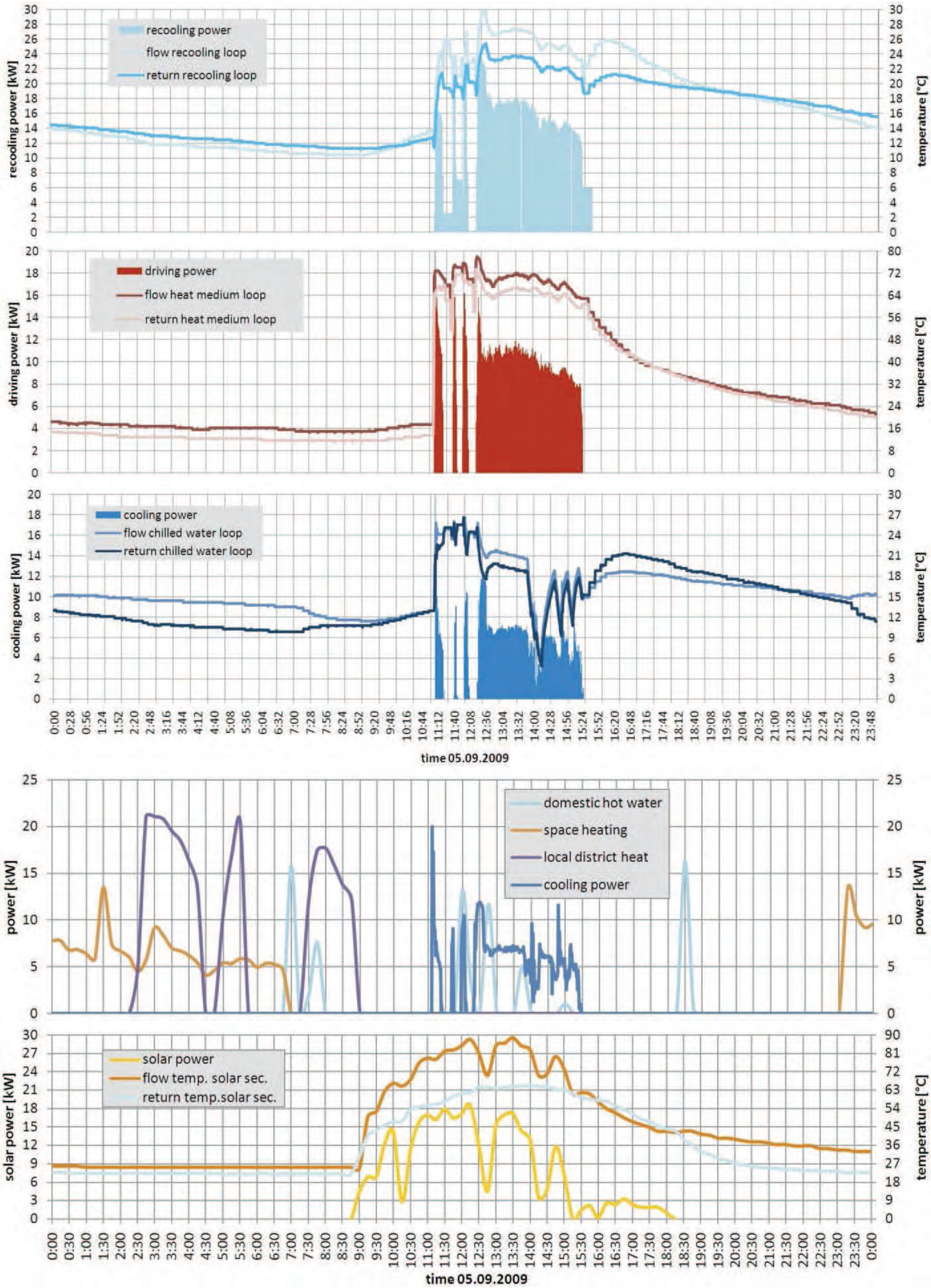


Figure 9: Absorption chiller and system analysis of the SHC plant on September 5, 2009

In the first three charts the cooling-, driving- and recooling power is drafted together with the linked supply- and return temperatures. In the beginning the chiller shows a clocking behavior, starting three times before running in a stable condition. The reason for switching on and off is the insufficient useable heat quantity (with a temperature base over 63°C) in the hot storage. At approximately 12:30 p.m. the incoming solar yield reaches sufficient values to keep the machine running. The solar plant is designed for a low flow configuration in order to assist the space heating of the house in winter times. This has certain disadvantages concerning the cooling mode. On this day the chiller has an average power share of around 6/10/16 kW (cooling/driving/recooling). For recooling the stable chiller inlet temperature values of approximately 22°C were reached. The heat rejection system is working well throughout the day. Furthermore the driving heat inlet temperature of the absorption chiller reaches an average value of 69°C for the day.

In the beginning the cooling outlet temperature towards the core activation shows stable values of approximately 19°C. At around 14:00 p.m. the flow and return temperature drops rapidly down to a value of 5°C. After a fast temperature rising up to 18°C the same characteristic temperature drop repeats two times. A reason for the temperature drop could be the closing of the regulation valve V_b correlating with a domestic hot water priority function. If the valve is closed the chiller cools down in its short closed cycle as far as possible until the valve opens again.

These low cooling temperatures affect thermal performance of the chiller. In Figure 10 the thermal COP during September 5, 2009 is shown. The bad performance can be identified exactly at the same time, when the cooling temperature drops down. Due to an internal ammonia storage inside the chiller there are some peaks in those periods, where the COP exceeds the value of 0.8. Generally the Pink absorption chiller handles the rough operating conditions quite well and balances some of the outside influences.

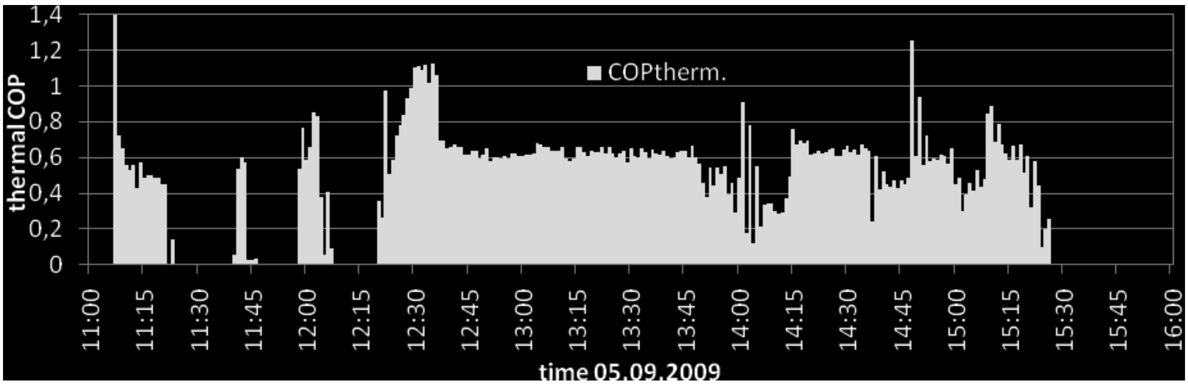


Figure 10: Thermal COP of the Pink chiller on September 5, 2009

Thermal and electrical COPs for the period between August 21, 2009 and the end of September are shown in Figure 11. An average thermal COP for that period of 0.57 could be reached. Electrical COPs range in daily values from 0.5 to 5 and achieve an average value of 3.1. In the period between the September 11 and 18, 2009 no results were monitored due to a data processing problem of the external operator station.

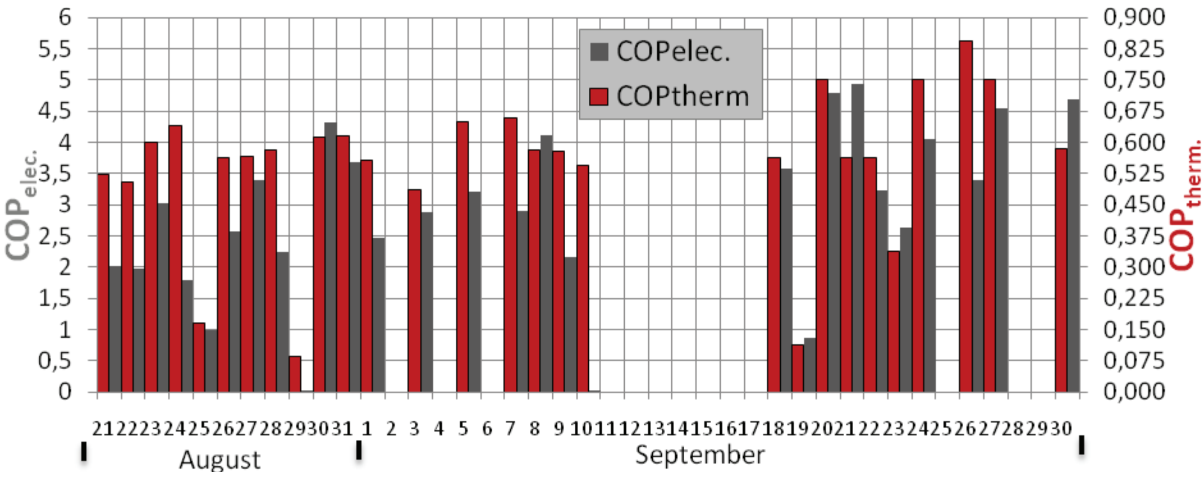


Figure 11: Daily thermal and electrical COPs for August and September

6 Outlook

Most of the occurred problems during summer 2009 happened in the field of the monitoring system. For that reason a redundant system including protection against energy blackouts is to prefer. Several changes of the system have already been succeeded.

Reengineering the monitoring system includes following five points:

- Room temperature and several other data points in the monitoring system
- switching the monitoring period of all data points towards minute intervals
- changing the logging function to an UPS protected local computer
- repositioning of the outside temperature- and humidity measurement sensor
- linking the data of the TA controller with the monitoring infrastructure of the IQ3

These suggestions of improvement regarding the monitoring system are expected to raise the reliability and accuracy of the monitored data. Furthermore a calibration of the temperature sensor is recommended in order to minimize the measurement errors.

Summarizing the five main practical suggestions for improvements:

- adapting the control system in order to make clear division between heating and cooling periods
- including the cooling cycle of the DHW priority rule
- excluding the logical test RT2>24℃ of the control strategy in order to use the concrete core activation as a storage and make the control system easier
- changing the switching operations between summer and winter to automatically by including the valves V3 and V4 in the controller
- rising the starting driving temperature in the control strategy, if there are still problems with a clocking behavior of the chiller

Expected improvements of these suggestions are reliable cooling and monitoring operations as well as higher thermal- and electrical COPs. In order to simulate other changes of the system in the course of the Solar Cool Monitoring project further computer simulations will be carried out.

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 2
Monitoring Results of SOLID CoolCabin

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: October 2010

By Daniel Neyer[1] and Moritz Schubert[2]

[1]Institution	University of Innsbruck
Address	Technikerstr. 13, 6020 Innsbruck, Austria
Phone	+43 512 507-6618
Fax	+43 512 507-36092
e-mail	daniel.neyer@uibk.ac.at
[2]Institution	S.O.L.I.D. Solarinstallation und Design GmbH
Address	Puchstrasse 85, 8020 Graz, Austria
Phone	+43 316 292840-81
Fax	+43 316 292840-28
e-mail	m.schubert@solid.at

Contents

1 Background

The office building was renovated in 2004 and the solar cooling device was installed in 2008. The office façade has a south and west orientation. To reduce the solar gains external shading devices are installed at each glazing. Because of internal gains and ventilation via windows, active cooling is indispensable. The solar cooling equipment is installed in a so called "CoolCabin" placed in front of the office. The solar collectors are installed on the roof. The hybrid cooling tower is placed on the flat roof of the office building. The cooling load of the office rooms is taken out via ceiling cooling elements.

Figure 1: View of the CoolCabin in front of the office building [SOLID]

A closed absorption cycle for generating cooling energy is employed. Autonomously solar thermal generated heat by high temperature flat plate collectors is used to regenerate the process. Within the absorption cooling machine water is used as refrigerant and lithium bromide is used as solvent. The cold water generated by the absorption cooling machine is used to cover internal and external heat gains. In winter the solar collectors are in assistance to the conventional space heating. In summer and winter the solar generated heat is stored in one buffer storage and all energy demand is taken out of it. The additional heat from the district heating is not stored in the tank but directly carried to the space heating system. A special application is the direct usage of the hybrid cooling tower for free cooling via chilled ceilings.

2 System Design

Figure 2 shows the functional scheme of the solar system. Hot water is produced in the 57.6 m^2 flat plate collectors, which are used as roof for the CoolCabin. The collector field is aligned south with a slope of 11°. The collector pump P1 drives the hot water to the heat exchanger. Pump P2 delivers the medium in the secondary solar cycle to the hot water storage tank, which has a volume of 2000 liter. Valve V1 controls the position (top or medium) where the hot medium is stratified into the storage.

In summer, when there is a cooling demand, the hot water flows to valve V4 through pump P3 to the absorption cooling machine (ACM). In winter, when space heating is needed, the hot water flows through valve V3 pumped by P6 directly to the distributor.

The ACM has a maximum cooling capacity of 17.6 kW, a rated COP of 0.65 and was manufactured by the Japanese company Yazaki. The hot water loop is piped back to the hot water storage or mixed in V4 with the flow to control the hot water inlet temperature.

The cooling tower can work in two different operation modes. Either the recooling water is going directly to the ACM through P5 or in free cooling mode to the heat exchanger by P7. Switching between chiller and free cooling is done with the valves V5 and V7. The chilled water produced by the ACM or from the heat exchanger (in case of free cooling) flows into a 200 liter hydraulic switch and further on to the distributor. Through the radiant ceiling system of the office building the chilled water is distributed.

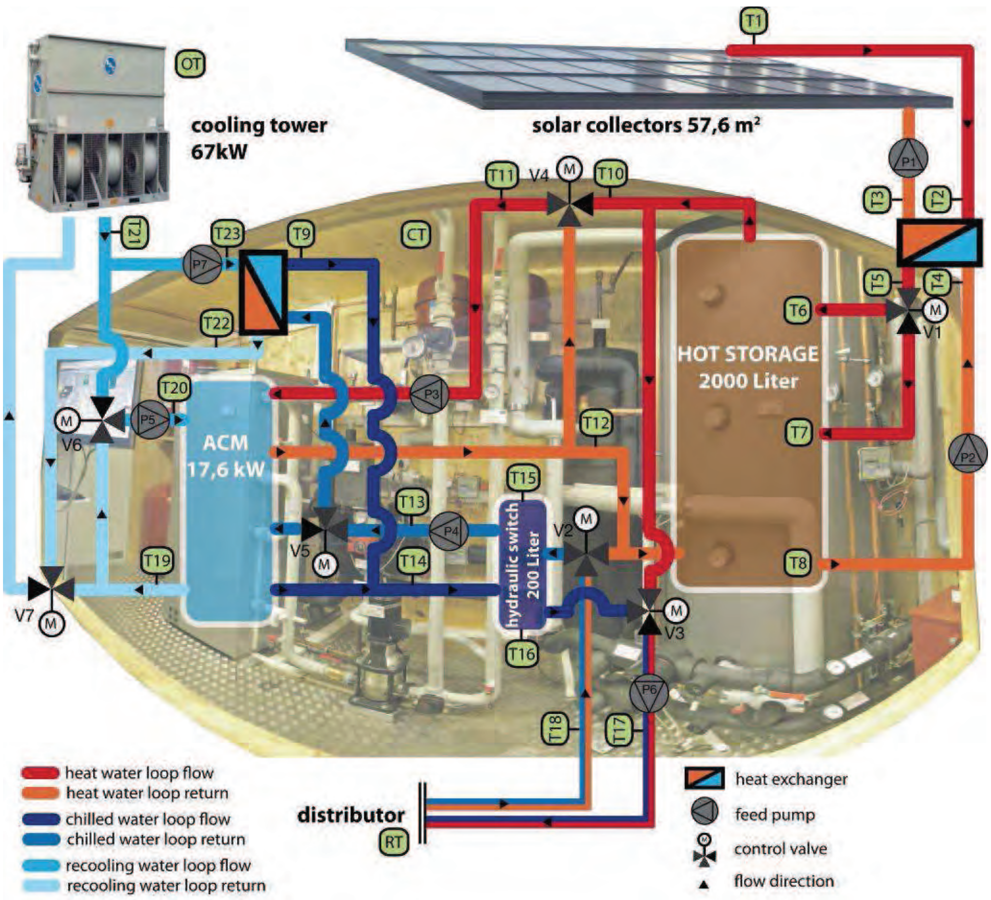


Figure 2: System scheme of the CoolCabin

Reasons for designing the system the way it has been designed [SOLID]

- The system is designed for 40-50 W/m² cooling load
- Restrictions in tilt angle for collectors (see Figure 1)
- No cold back-up for saving electricity and to see how this works in Austria
- Storage size 35 l/m² for heating support in shoulder season. Could be smaller for cooling
- Free cooling for research purposes
- Hybrid cooling tower mainly for free cooling. Over dimensioning of cooling tower for higher cooling output of cooling machine than nominal power.

Problems encountered [SOLID]

- Works well without backup. Some south-facing offices become too hot.
- Free cooling like it's done here is not efficient and useful for building cooling in summer. Could be more effective using free cooling in other ways and times.
- Sometimes it is too humid inside the building in summer time as only chilled ceilings are used and no air treatment is done.

Figure 3: View of the inside of the CoolCabin [SOLID]

3 Monitoring Equipment

3.1 Installed Equipment

The equipment is completed for IEA Task SHC Task 38 monitoring level III. To succeed simulations the relative humidity is measured additionally. The amount of heat and electricity meters is reduced to a minimum. To differentiate between the two modes (free and thermal cooling) the valve positions are logged and used for this allocation.

The conventional heating system is not included in the monitoring but is carried out through the clearing of the district heating. The district heating is directly carried to the distribution system.

The data is recorded each minute except V1 and E20. The water consumption of cooling tower (V1) is monitored by hand each week. E20 is logged each 10 minutes by a separate device - an electrical socket counter.

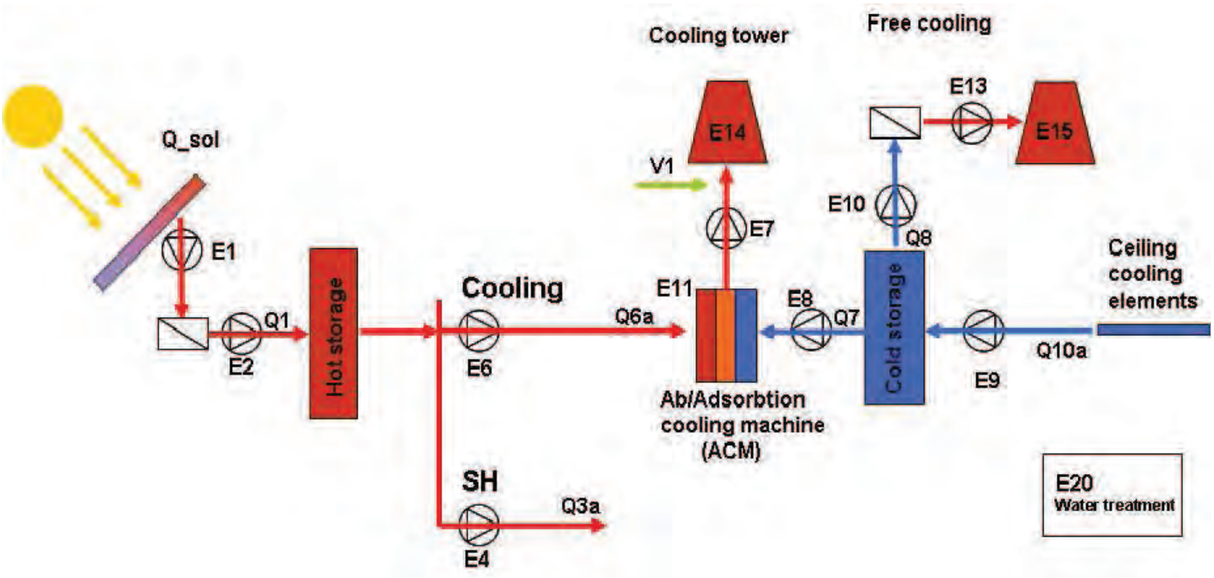


Figure 4: Monitoring scheme of the CoolCabin

Following data points are from the same device or measured together. The allocation between thermal and free cooling is done by the valve positions mentioned above. Nevertheless the important results for Task 38 calculations can be delivered.

E1 + E2	measured together
E6 + E11	measured together
E4 + E9	same pump, allocation by valve position V2, V4 (heating, cooling)
E7 + E13	same pump, allocation by valve position V5, V7 (free, thermal cooling)
E8 + E10	same pump, allocation by valve position V5, V7 (free, thermal cooling)
E14 + E15	same device, allocation by valve position V5, V7 (free, thermal cooling)
Q7 + Q8	same heat meter, allocation by valve position V5, V7 (free, thermal cooling)

An overview of all monitored data points is given in Table 1. The controlling and monitoring issues are done by three programmable logic controllers. Overall 111 points are logged on a local PC. The access to this data store is provided by internet and a remote control system.

Table 1: Data points of the CoolCabin

Quantity	No.	Type	Accuracy
monitoring / control inputs			
temperature	35	PT1000	class B + calibration
pressure	4	huba control, OEM relativ-Drucktransmitter	endvalue ±1.5% FS temp. dependence ±1.5%/10℃
radiation	1	SP-Lite / Kipp&Zonen	temp. Dependence +0.15%/℃ directional error ±5% at 80°
temp. / relative humidity	1		±0.5% ℃ / ±3%RH
heat meter (energy/volume/power/flow rate/return-supply-temp./ΔT)	4 (x7)	Landis + Gyr, UH50, ultraschall	class II; 2+0.02 q_p/q% (max 5%)
electricity meter (energy/power)	6 (x2)	Görlitz DCi 230V/400V, 65A	class I; IEC 62053-21 (±1%)
logged control outputs			
speed control	2		
on/off	3		
valve position	5		
failure signal / set temp. / …			
sum	**111**		

3.2 Period of Measurement

The monitoring system of the CoolCabin is in operation since the commissioning in September 2008. Additional monitoring equipment for level III was completed in June 2009. Monitoring is ongoing.

The patency of data logging (**Fehler! Verweisquelle konnte nicht gefunden werden.**) is very high (>96%). Nameable gaps crop up only when the CoolCabin is out of service. One problem occurred was a roof avalanche that got down on the radiation sensor. Consequently no solar radiation data was monitored from Jan10 to April10. The missing data was calculated out of Q1 using a monthly efficiency of the solar collectors including the heat exchanger.

4 Monitoring Results

4.1 Annual / Monthly Data

As an overview the solar energy source management is shown in Figure 5. Start of monitoring is July 2009 at the left side of the graph. The first cooling season lasts until October 09 including free cooling. Q3 represents the sum of solar heating and the heat from the conventional system. The solar fraction in summer is 100% (no heat backup) in winter the solar fraction for heating drops down from approximately 20% in Nov09 to 7% in Dec09. The highest fractions occur in spring (30% in Feb10; 60% in March10). The overall solar efficiency is approximately 30%, including efficiencies up to 40% in summer and 10% during winter.

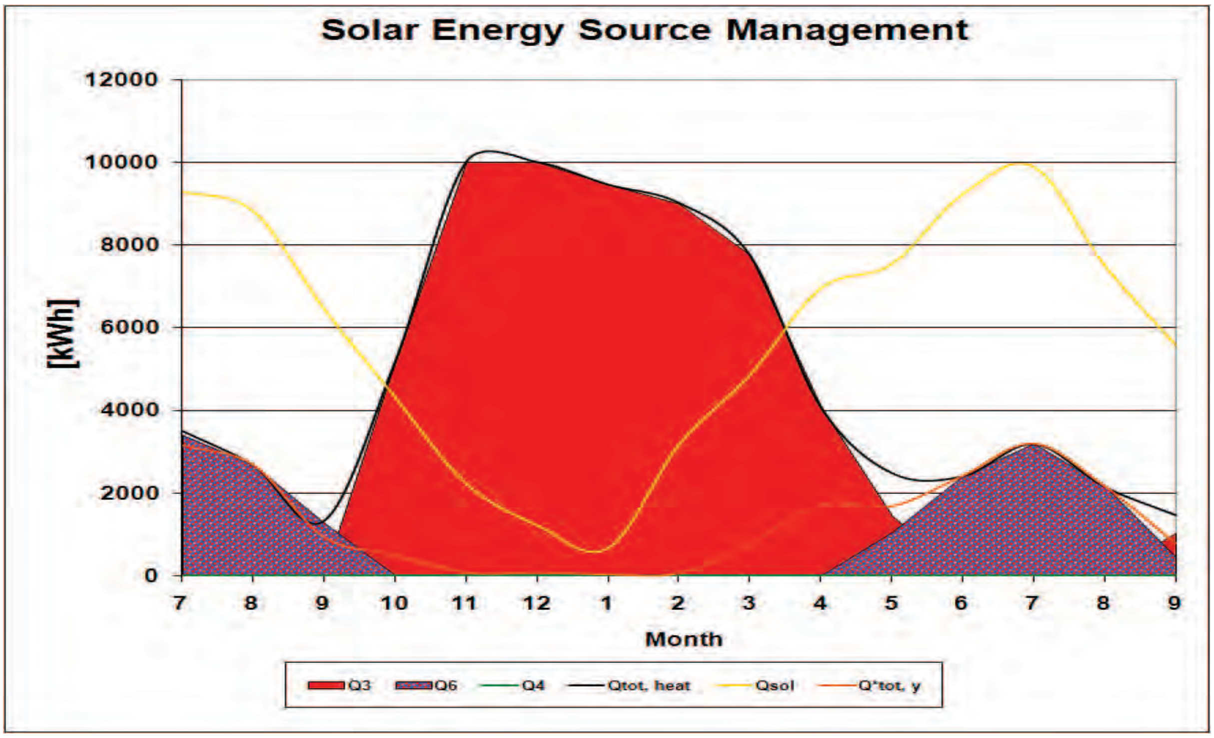


Figure 5: Solar energy management over the whole measuring period July 2009 to September 2010

In the first cooling season the primary energy ratios (PER) of the SHC plant are lower than the defined reference system (Figure 6). As a result of these PER negative savings in terms of primary energy occurred!

After improvements and changes in May 2010 the fraction of primary energy savings got slightly positive. Higher values are expected for the cooling season 2011 with further improvements. Since operation started in July 2009 savings of 14% could be achieved. If the first (negative) cooling period is excluded the fraction of primary energy savings is greater 20%. Taking into account the 100% solar cooling plant with no conventional backup this value has to be improved.

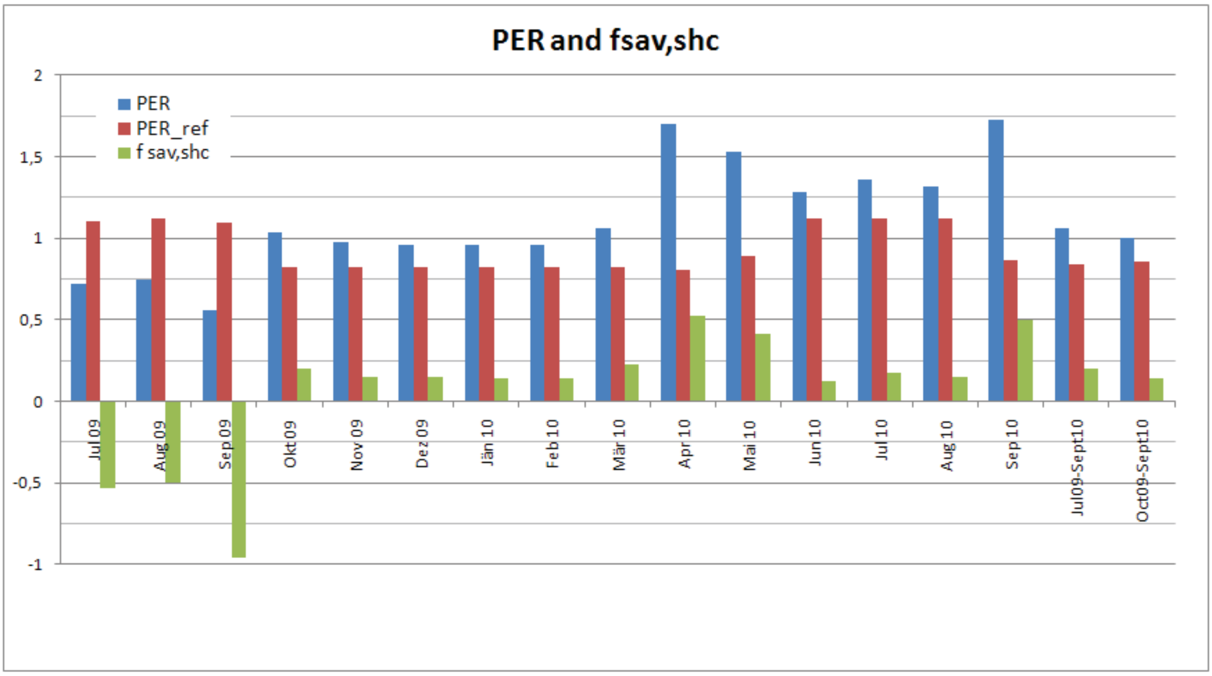


Figure 6: Primary energy ratios and fraction of primary energy saving over the whole measuring period July 2009 to September 2010

Having a look on the general energy consumption of all devices (Figure 7) the average fraction is about

36% for cooling tower including internal pump and ventilator

28% for recooling pump

18 % for hot water pump and ACM

9% for solar pumps

5% for chilled water pump

4% for distribution pump

The standby consumption of E11 and E14/15 leads to a lower average electrical COPs. If the solar system is working but without any cooling demand the electric consumption of E11/E14 is even higher than the solar pump consumption. In case of August 2010 the difference between average with standby and without is 0.1. This difference gets blatant in months with low cooling demand like September 2010. The electrical COP without standby would be 3.2 including the total electrical consumption but only a COP_{el} of 2.4 was achieved!

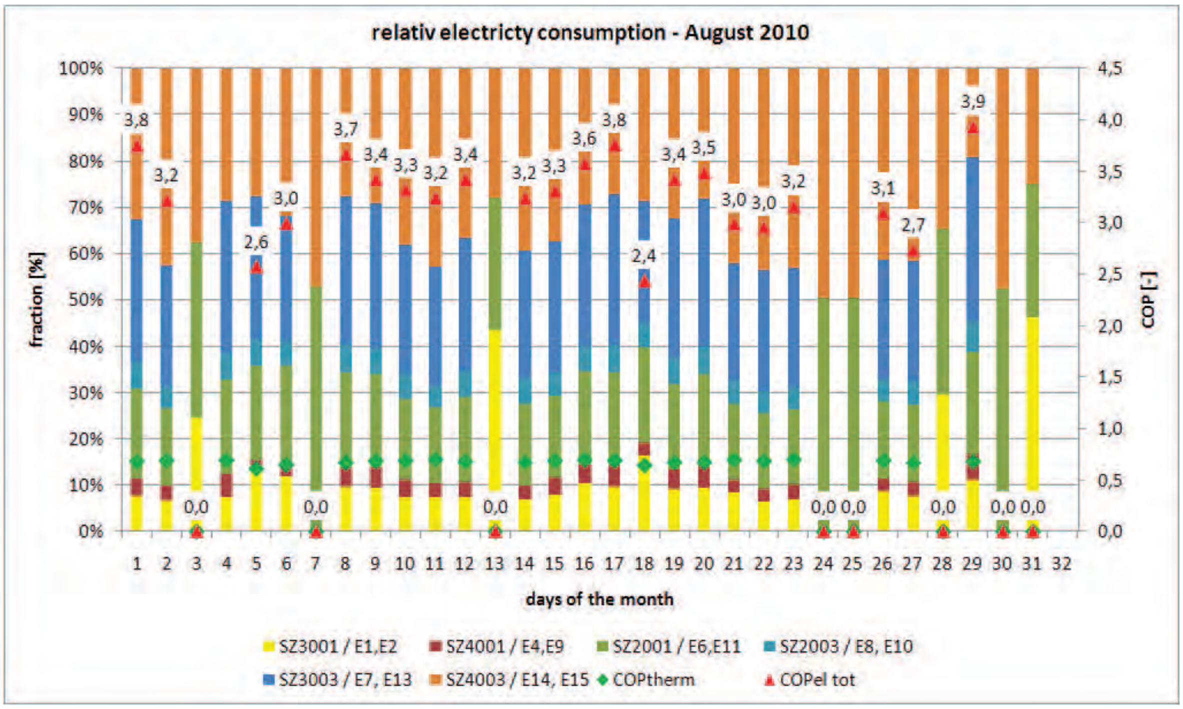


Figure 7: Relative electricity consumption of all devices in August 2010 after first improvements in May 2010

Figure 8 shows the annual load duration curve for cooling in summer 2010. Overall 500 hours of operation can be observed. The absorption chiller is running approximately 100 hours at full load conditions. The remaining period the chiller is working at part load conditions.

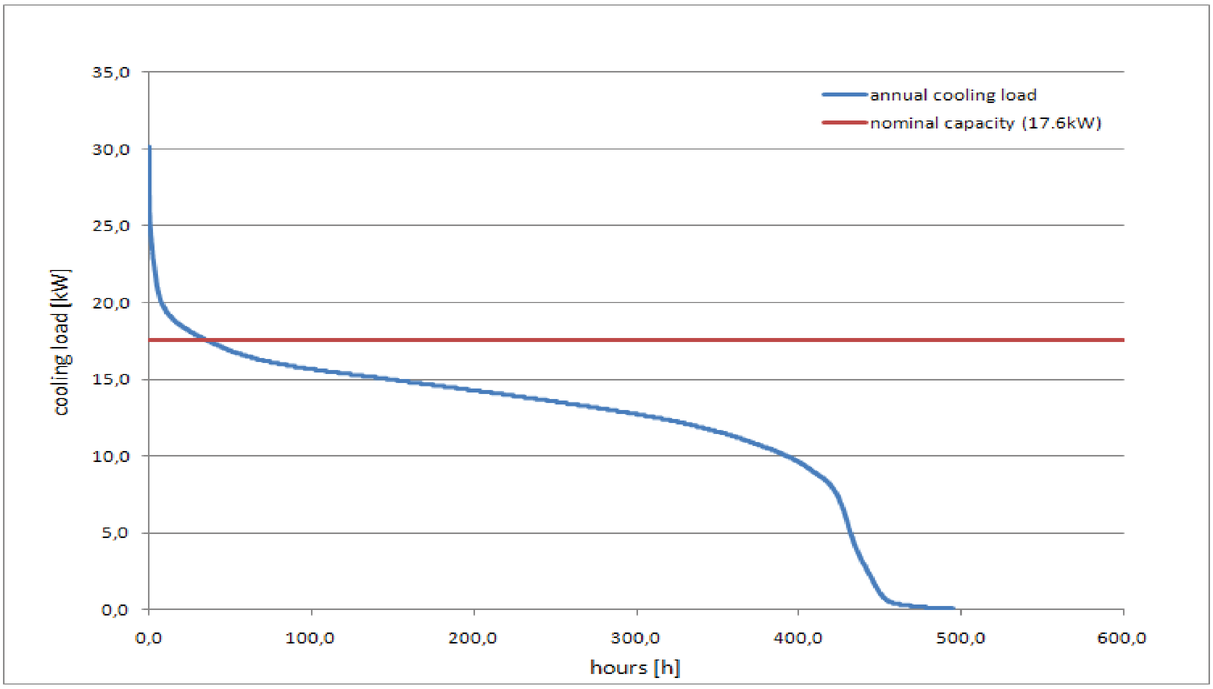


Figure 8: Cooling load curve for 2010

4.2 Analysis of Typical Days

4.2.1 Cooling period 2009

The 25th of August 2009 is an average sunny summer day in 2009. The cooling power, driving heat and solar yield are drafted together with the linked respective temperatures.

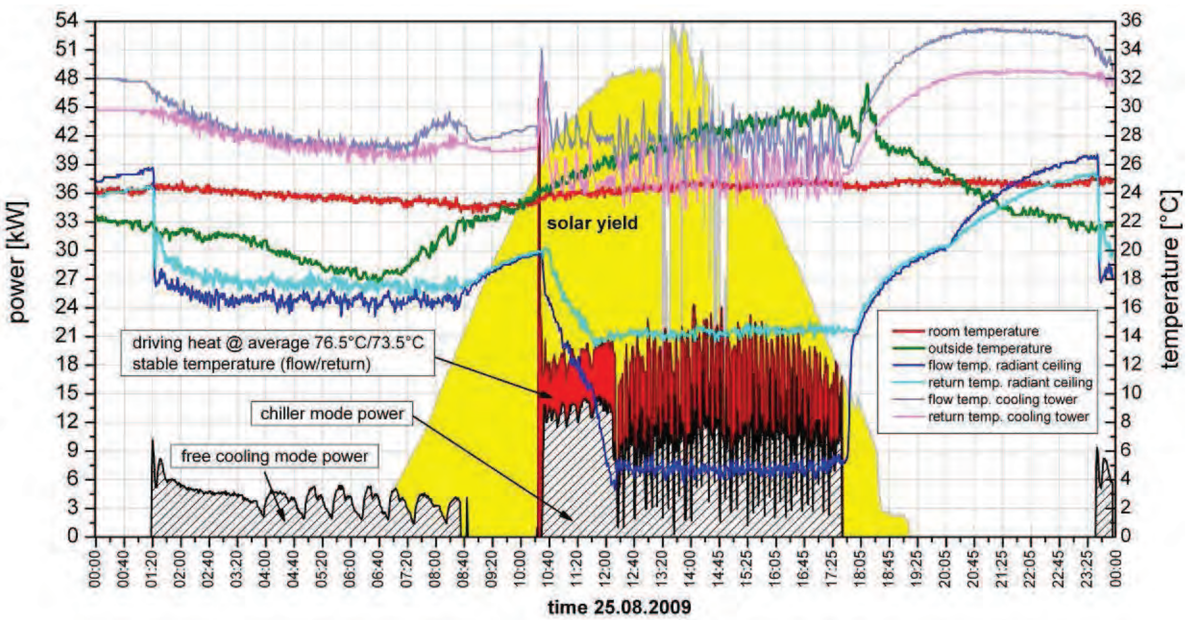


Figure 9: Power characteristics of the CoolCabin on Aug. 25, 2009

At 10:16 a.m. the WFC-SC5 chiller is switched on and is running nearly seven and a half hours without break. The average cooling power alternates between 15 and 10 kW with a matching driving heat power between 20 and 15 kW. Some strong swinging conditions can be observed in the last two-thirds of the cooling period. Referring average temperatures for the driving heat are showed in Figure 9. The heat flow temperature remains in steady conditions throughout the cooling period. It can be observed that the solar yield fits quite well with the delivered cooling load. The recooling water loop coming from the cooling tower is regulated with the valve V6 to control the output cooling power. After two hours of declining temperatures at the radiant ceiling - steady supply and return temperatures of 5℃/14℃ are reached. Bad thermal COPs and fluctuating power conditions in the machine are the consequence of the low temperature. These fluctuations can be seen in the recooling loop temperatures, the heat medium power and the cooling power. Following this daily analysis it appears that the plant works how it was designed, but some design decisions as well as some control adjustments are irreproducible.

Out of the manufacturing information of the Yasaki chiller WFC-SC5 the thermal COP reaches a value of 0.65 under standard conditions. In the beginning of this day the thermal COP reached stable values around 0.7. Afterwards the machine shows alternating COPs. In average a thermal COP of 0.616 was reached in the whole period. Considering measurement uncertainties and the lower heat medium inlet temperature mentioned before the thermal COP is feasible.

The chiller was running very reliable throughout the whole summer 2009. It also brought the expected power level and reached a feasible thermal COP. In the following section a closer look will be set to the free cooling mode in order to check the practical operation behavior and performance.

4.2.2 Free cooling

The CoolCabin is equipped with a special hydraulic design in order to run on a free cooling mode. This feature is included to ensure the cooling of the office rooms also on days without sunshine. Therefore the chiller is bypassed so that the cooling tower is able to run on wet mode and can serve the cooling load directly. The mode is activated if the outside temperature is falling below 21.5℃ and the office still has a cooling demand.

Similar to the absorption chiller above a day in August was picked out in order to evaluate the operation behavior of the free cooling mode. In Figure 10 the free cooling power, the solar yield and four temperatures are drafted in a diagram. To be able to compare the solar yield to a standard sunny summer day the solar radiation of the August 25, 2009 is added.

It can be seen that the free cooling mode started three times that day, always switching on when the outside temperature falls under 21.5℃ and stopping when the outside temperature exceeds the 22℃ level. The room temperature stays quite stable throughout the day showing a slight downtrend in a bandwidth between 25℃-23℃. Furthermore the flow and return temperature to the radiant ceiling of the office shows values between 18-16℃/19-17℃ (flow/return). The realized temperature differ ence between flow and return temperature is very small. In total 71 kWh cold were delivered to the office rooms on that day.

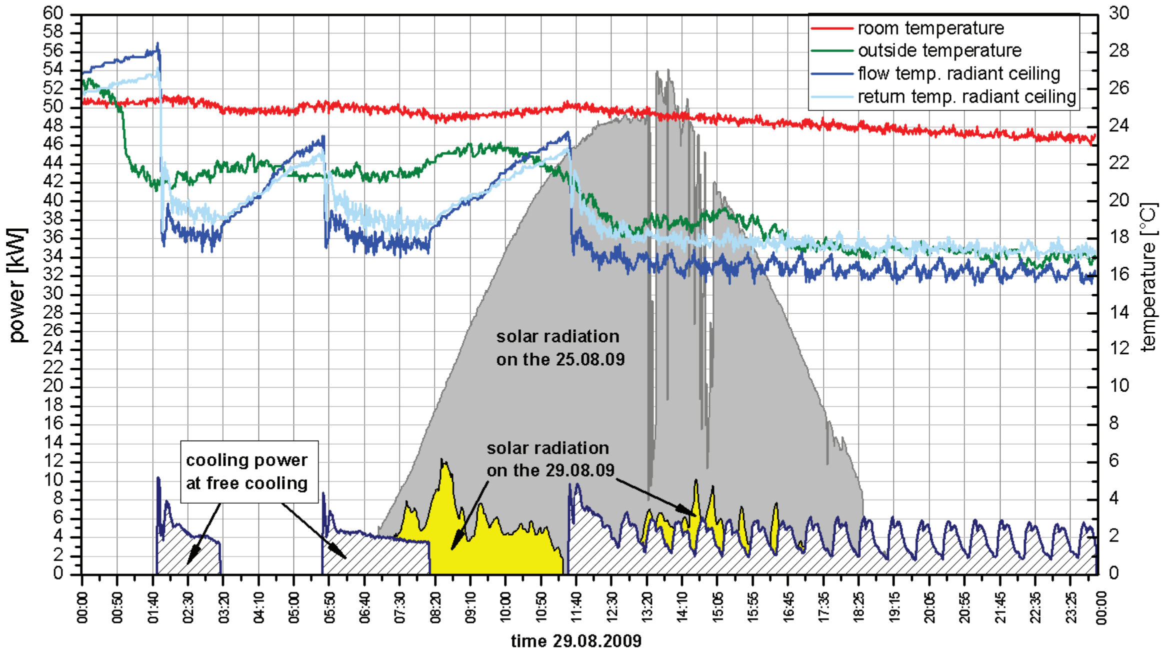


Figure 10: Free cooling operation performance of the CoolCabin on the Aug. 29, 2009

The cooling power revealed a saw tooth tread design which can also be seen in the developing of the radiant ceiling flow temperature. In Figure 11 the practical operation experience of the August 23, 2009 is marked in a diagram. The room temperature stays stable between 25℃ and 23℃. In night times the fr ee cooling mode is activated and running on a poor power level. On the day the chiller is operating at an average power level of 10 kW. The room temperature in this time declines only slightly.

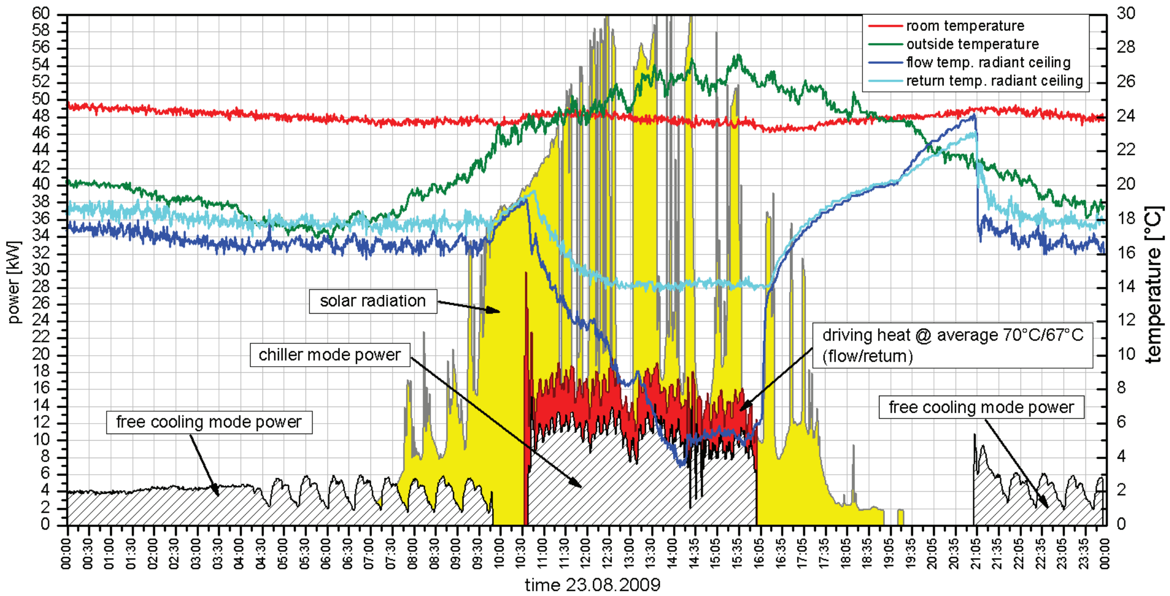
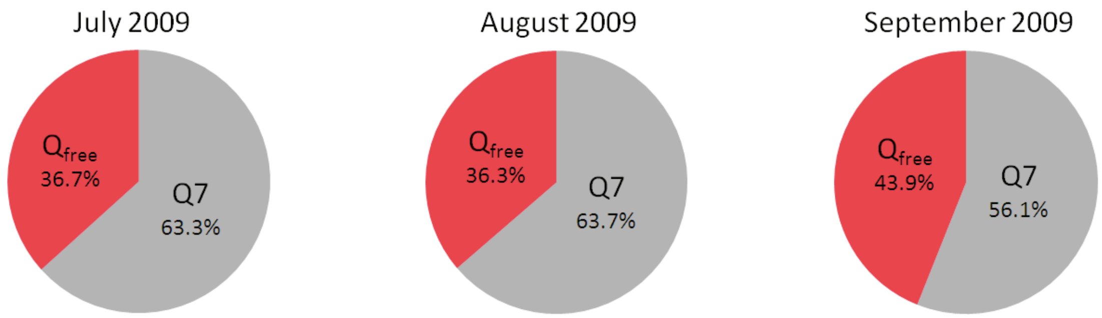


Figure 11: Operation performance of the CoolCabin on the Aug. 23, 2009

The fraction of free cooling in the months July, August and September (2009) is shown in Figure 12 including detailed numbers in the tables below.



Q10a	Q7	Q_{free}
2998.0 kWh	1898,5 kWh	1099,5 kWh
100%	63.33%	36.67%

Q10a	Q7	Q_{free}
2673.0 kWh	1702.0 kWh	971.0 kWh
100%	63.67%	36.33%

Q10a	Q7	Q_{free}
1195.0 kWh	670.0 kWh	525.0 kWh
100%	56.07%	43.93%

Figure 12: Free cooling share of the total delivered cooling load for July, August and September 2009

Approximately one third of the total cooling load was done by free cooling. The free cooling mode is especially running in night times and on rainy days.

4.2.3 Improvements / changes 2009-2010

After the monitoring period 2009 some changes and improvements have been performed.

The main changes were:

- beginning of May10: Maintenance of the cooling tower
- end of May10:
  - supply temperature of chilled ceiling from 17℃ to 15℃
  - clearance temperature (upper storage temp.) for starting the ACM from 75℃ to 88℃

- end of May10:
    - clearance temperature (upper storage temp.) for starting the ACM from 88°C to 80°C
    - Change of ventilation control strategy on/off to speed control

The effects of these changes are shown in Figure 13.

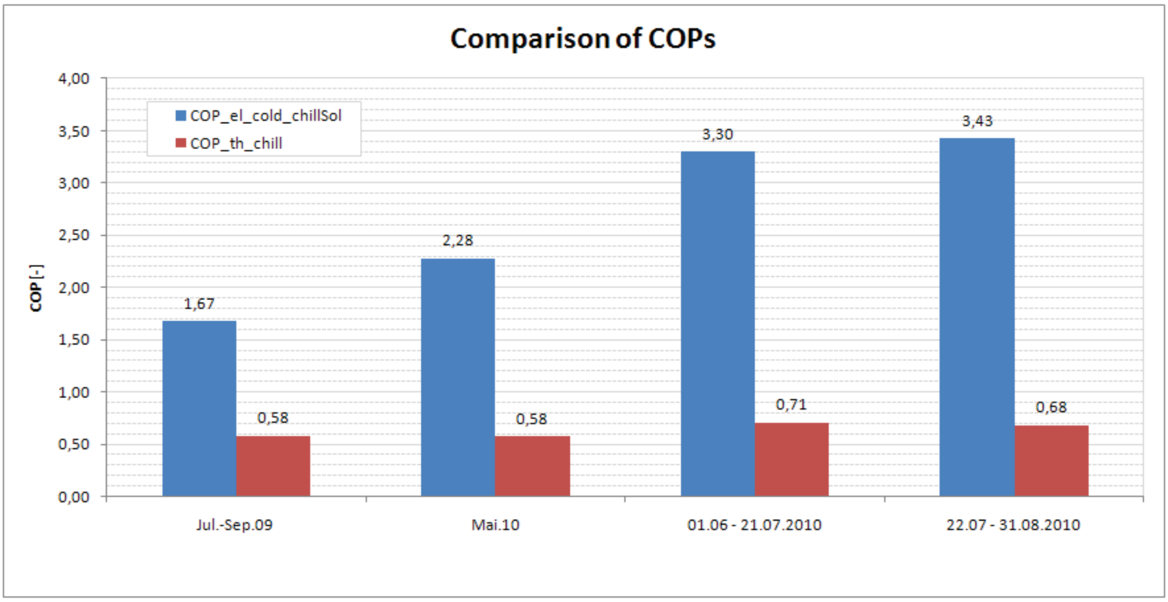


Figure 13: Comparison of thermal and electrical COPs 2009 and 2010

Compared to the cooling period 2009 (including free cooling and thermal driven chiller) the electrical COP was rising 36% up to value of 2.3 while the thermal COP was on the same level. After changes in the control parameters another increase of 44% could be achieved. Finally, including the change of the control strategy of the cooling tower fan, the electrical COP is at about 3.4. Compared to 2009 a plus of 105% in the electrical COP was obtained. The thermal COP mainly affected by the reccoling, regeneration and chilled water temperature level was enhanced by 17%.

Main reasons for the better performance are higher loads due to weather conditions and changes in the set temperatures (supply temp.) and as a result less part load condition!

4.2.4 Cooling period 2010

The daily performance of a typical sunny day is shown in Figure 14. The chiller starts its operation at 11:30 a.m. after the storage temperature reaches 80°C. The driving temperature is slightly increasing during operation and the average is at about 77°C. The chilled water temperature decreases from 20°C to 8° C while the recooling temperature is controlled to 29°C. Within these conditions a decre asing power for driving heat and cold production is visible.

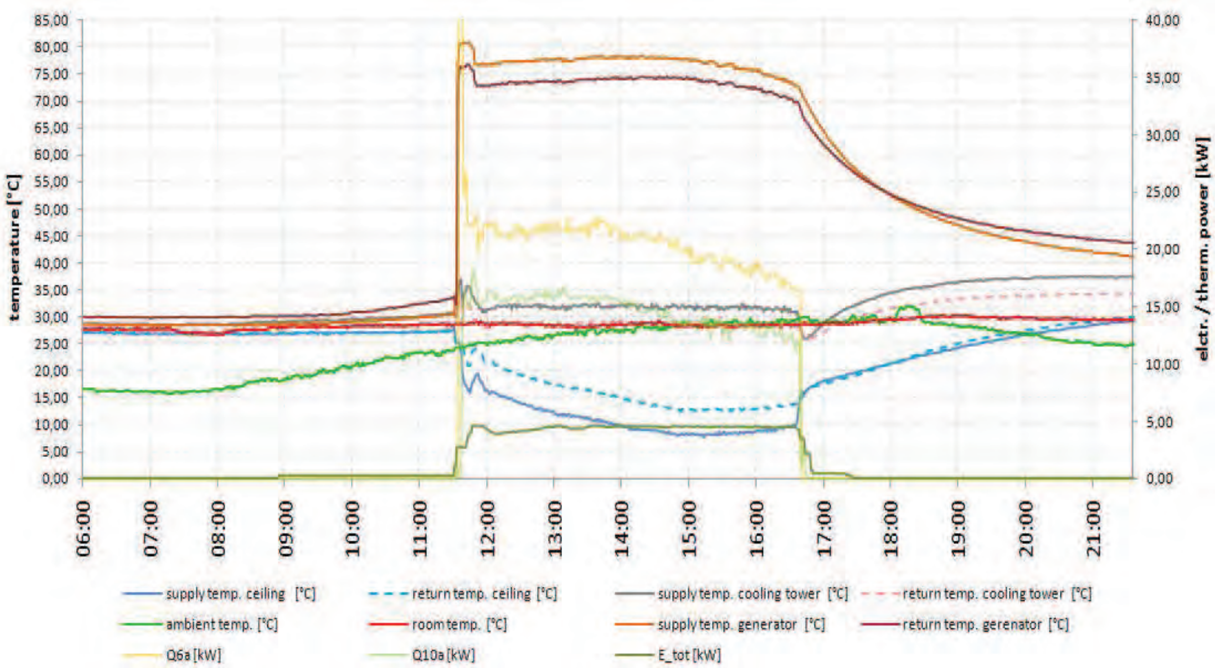


Figure 14: Typical sunny day after improvements in cooling season 2010

An average generation power of 20.7 kW and an average cooling power of 14 kW leads to an average thermal COP of 0.68. The electricity consumption is almost constant during operation at a level of 4.5 kW. Due to the part load conditions the average electrical COP amounts to 3.1. The average COP_{el} of August 2010 is 3.4; mainly due to the same reason.

The under dimensioning of the chilled ceilings can be seen by having a look on the room temperature. During operation of the cooling plant there is hardly any effect on the room temperature. To put all in a nutshell this plant shows a good performance but points out the importance of correct system integration and long term monitoring. Nevertheless the overall primary energy savings of this plant during the period Oct 09 to Sept 10 are 51% compared to the reference system.

After improvements the alternating behavior of the chiller could be stopped (Figure 9). Consequently the thermal COP was increased. The electricity consumption was slightly decreased but at the same time a increasing of the cooling load was achieved.

5 Experiences / Lessons Learned

The average electrical COP 2010 of about 3.4 is still too low and should be improved. Main focus should be the part load conditions.

- Main components focus: recooling pump, cooling tower
- Control strategy
  - Lower flow rate (constant or variable) for driving pump
  - Including the actual load to speed control of main components
- Reactivation of free cooling mode with better control strategy

6 Conclusions

Analysis of the operation stability of the system shows reliable running conditions over the whole summer with only a few days where the absorption chiller did not work due to electrical problems.

The daily operation performance indicates that the radiant ceiling is dimensioned to small for the system or vice versa. If the radiant ceiling is under-dimensioned for the cooling plant, as it is the case at the Solid plant, the whole cooling system is running in part load.

Matching the temperatures between two systems is important. The solar cooling system was designed for running on a 9℃/17℃ nominal flow/ret urn temperature, but the cold distribution (radiant ceiling) requires only 16℃/1 9℃. Consequently this leads to high mixing losses in the radiant ceiling inlet temperature regulating valve. This affects mainly the absorption chiller, because it leads to lower thermal COPs but also to higher system losses and therefore to lower electrical COPs.

The free cooling mode (2009) delivers one third of the total cooling load to the building. The electrical COPs of the free cooling mode are not satisfying. Due to the low heat storage capability of the building the cooling effect is very limited. Maybe there are other reasons that make a further running on the free cooling mode feasible. If the free cooling mode is continued a new time control has to be implemented in the control strategy in order to run the mode only in times when the office is occupied. Free cooling of an empty office with low heat storage capability just raises electricity costs.

Measurement results show that the heat rejection system including the cooling tower and the recooling pump plays a key role in reducing the electrical consumption of the plant. Replacing the existing cooling tower to a smaller and more efficient one could be one solution. With a more efficient cooling tower and more efficient pumps the electrical COP could easily be doubled. Therefore a pure wet cooling tower is preferable.

Expected improvements of these suggestions are higher thermal and electrical COPs as well as improved conditions in the office. In order to simulate other changes of the system in the course of the Solar Cool Monitoring Project simulations will be carried out.

SOLID: "…Sophisticated solar cooling systems need 1 to 3 years of surveillance and optimization after start-up. This ensures optimal operation according to the customer's needs high efficiency and long life time of the system…"

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 3

Monitoring Results of solar cooling plant at Municipality Department 34, Vienna

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 23.05.2011

By Anita Preisler and Tim Selke

Institution: AIT – Austrian Institute of Technology, Energy Department

Address: Giefinggasse 2, A-1210 Vienna, Austria

Phone: +43 (0) 50550-6634

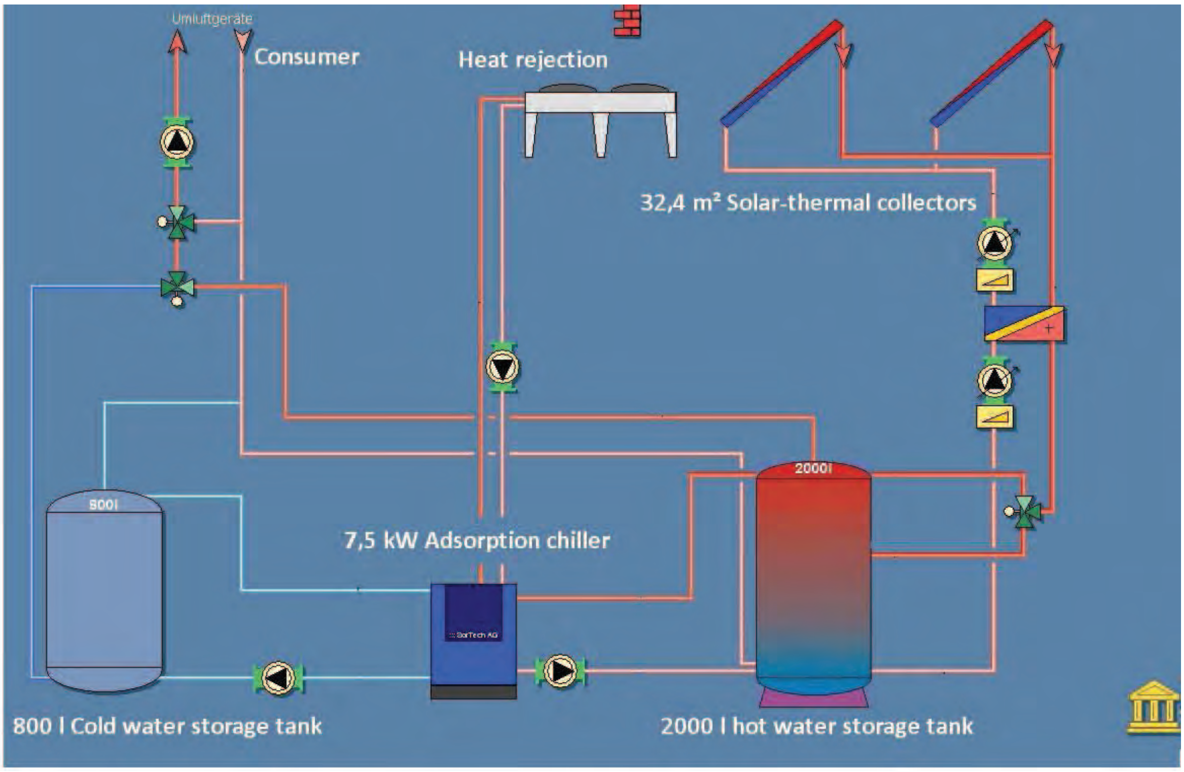
Fax: +43 (0) 50550-6613

e-mail: anita.preisler@ait.ac.at

Contents

1 Technical description of the plant

The thermally driven adsorption SOL ACS 08 - manufactured by Sortech AG in Halle a.d.S./ Germany - generates cold water with a nominal capacity of 7,5 kW (see Figure 1). The adsorption chiller is a new development and applies innovative coating, compact construction, optimized subsystem including dry heat rejection unit, which has EC-fan technology and an additional fresh water spray mode implemented. In order to provide solar energy to the adsorption chiller 12 flat-plate collectors, with an overall gross area of 32,4 m², are mounted on a garage roof, facing south with an inclination of 40 degree. The hydraulic scheme of the overall system design shown in Figure 2 contains two thermal storages, hot water storage (2000 l) as well as cold water storage (800 l). In summer fan coils unit extract heat from several office rooms in order to control the room temperature. The three pumps around the adsorption chiller (cold water pump, heat rejection pump, hot water pump) are not designed for automatic speed level control; they have three levels of switching, which can be changed only by hand. AIT had direct access to the plant management system for changing set points and analyzed their impact on the system performance.



[illegible]

2 Recording of monitoring data

The solar cooling plant was installed in spring 2009 and the system operation was monitored in summer 2009 and 2010. In summer 2009 the monitoring data transfer contained gaps for the duration of some hours to several days, caused by the used monitoring recording software. Therefore, a new monitoring server was installed at AIT for summer 2010 which led to a continuous recording of monitoring data in the second summer.

Approximately 75% of the monitoring data from the summer 2009 were recorded. This data represents all relevant weather conditions for Vienna. As a result, it was possible to make a first evaluation of the energy performance of the solar cooling plant.

Due to the new monitoring server at AIT there weren´t any losses of monitoring data in summer 2010.

3 Experience report summer 2009

The monitoring of the first cooling season showed that the adsorption chiller worked reliable and the desired air-conditioning of the office rooms was given. The room temperature in the offices was kept under 24℃ over the whole summer. There weren`t any error operations in the overall system or in the individual components. The average cooling capacity of the system lay between 1,7 kW and 2,4 kW, which is way below the nominal capacity of 7,5 kW of the adsorption chiller. This part load operation caused by the low cooling capacity on the building side has a negative impact on the thermal Coefficient of Performance (COP_{th}) as well as on the electrical Coefficient of Performance (COP_{el}) of the solar adsorption chiller system. The documentation of the control was incomplete, which made the check of the implemented control strategies in the plant through monitoring evaluation quite difficult.

In July 2009, the cold water pump showed a varying electricity consumption. The comparison between the electricity consumption of the pump and the mass flow in the cold water circle showed that there was an electricity consumption of the pump, even when the mass flow was zero. This led to the conclusion that air was in the cold water pipes. Therefore, the water pipes were de-aired which affected the desired continuous mass flow.

The energy performance of the system is evaluated by two indicators. The Thermal Coefficient of Performance (COP_{th}) gives the ratio between produced cooling energy and used thermal driving energy for a certain period of time (see Equation 1).

Thermal Coefficient of Performance:

$$COP_{th} = \frac{Gains}{Thermal\_Demand} = \frac{Q_{cooling}[kWh]}{Q_{solar\_thermal}[kWh]} \qquad \text{Equation 1}$$

The Electrical Coefficient of Performance gives the ratio between the produced cooling energy and the used electrical consumption for a certain period of time (see Equation 2).

Electrical Coefficient of Performance:

$$COP_{el} = \frac{Gains}{Electricity\_Demand} = \frac{Q_{cooling}[kWh]}{E_{driving,el}[kWh]} \qquad \text{Equation 2}$$

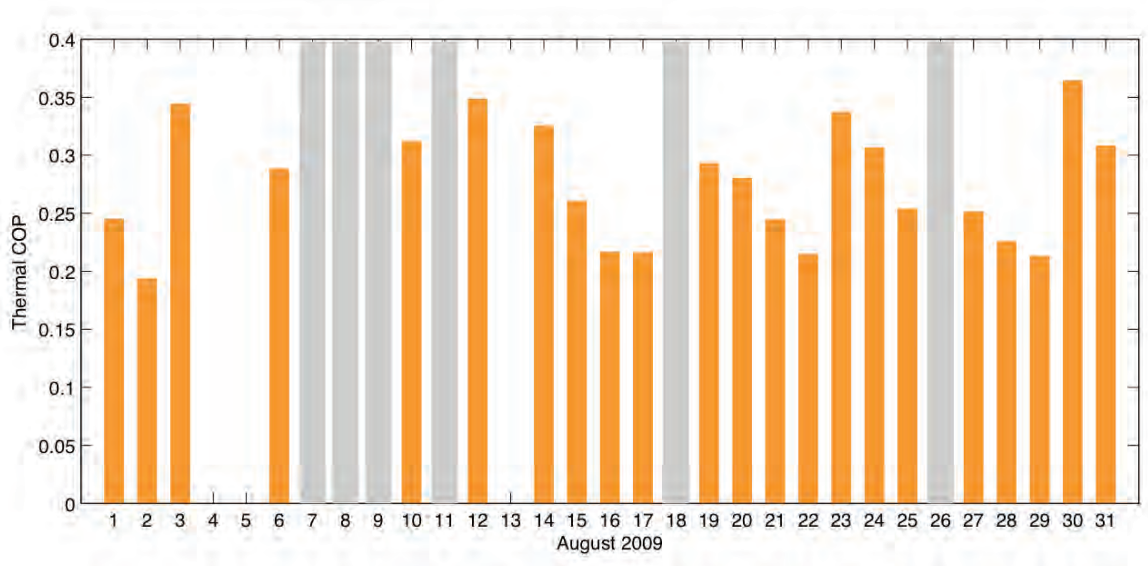


Figure 3 shows that the daily COP_{th} for August 2009 are in the range between 0,19 to 0,35. On the days with grey bares the recording of monitoring data was insufficient.

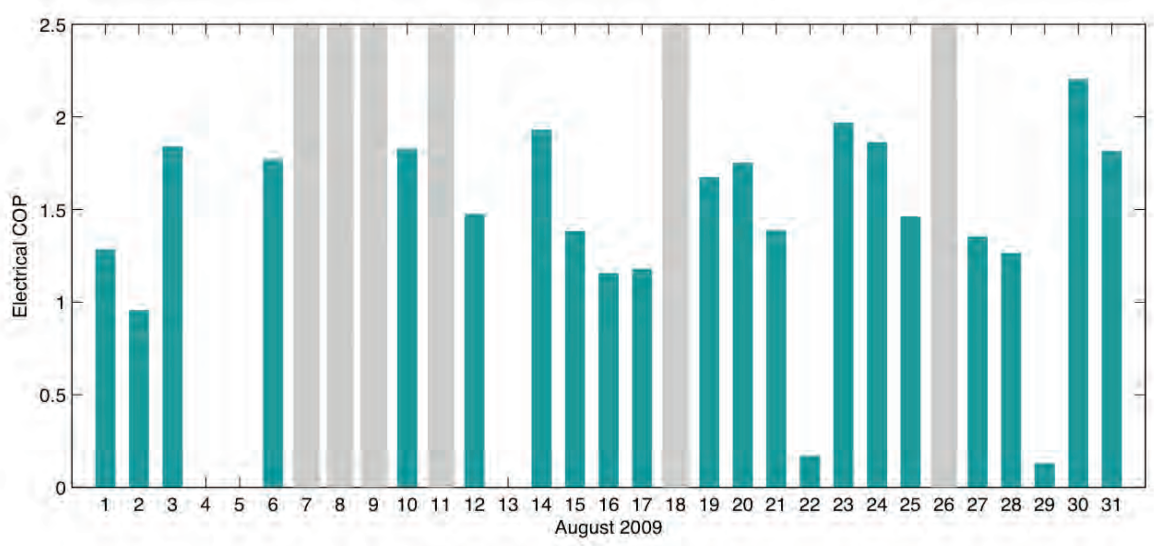
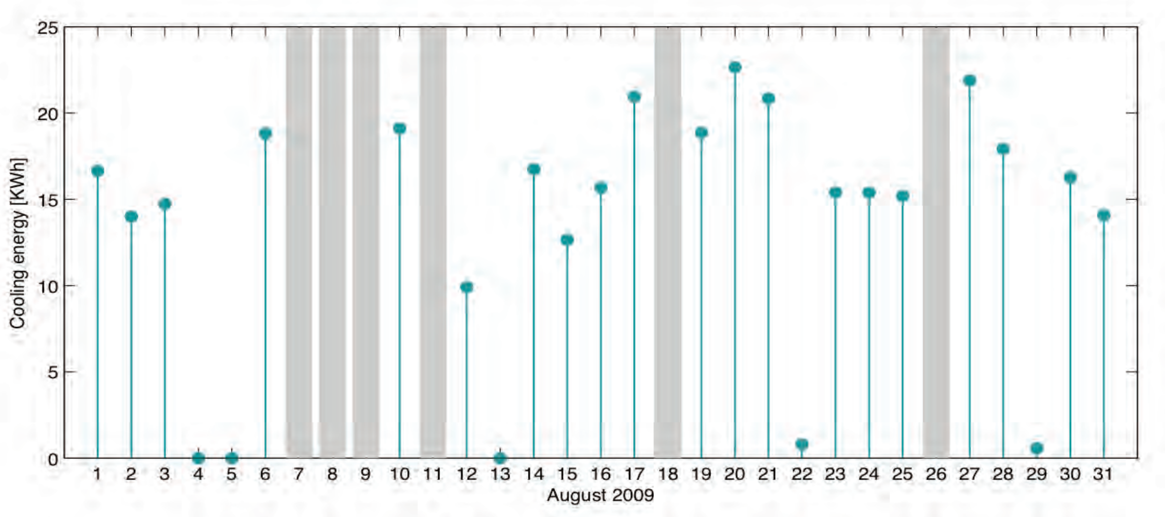


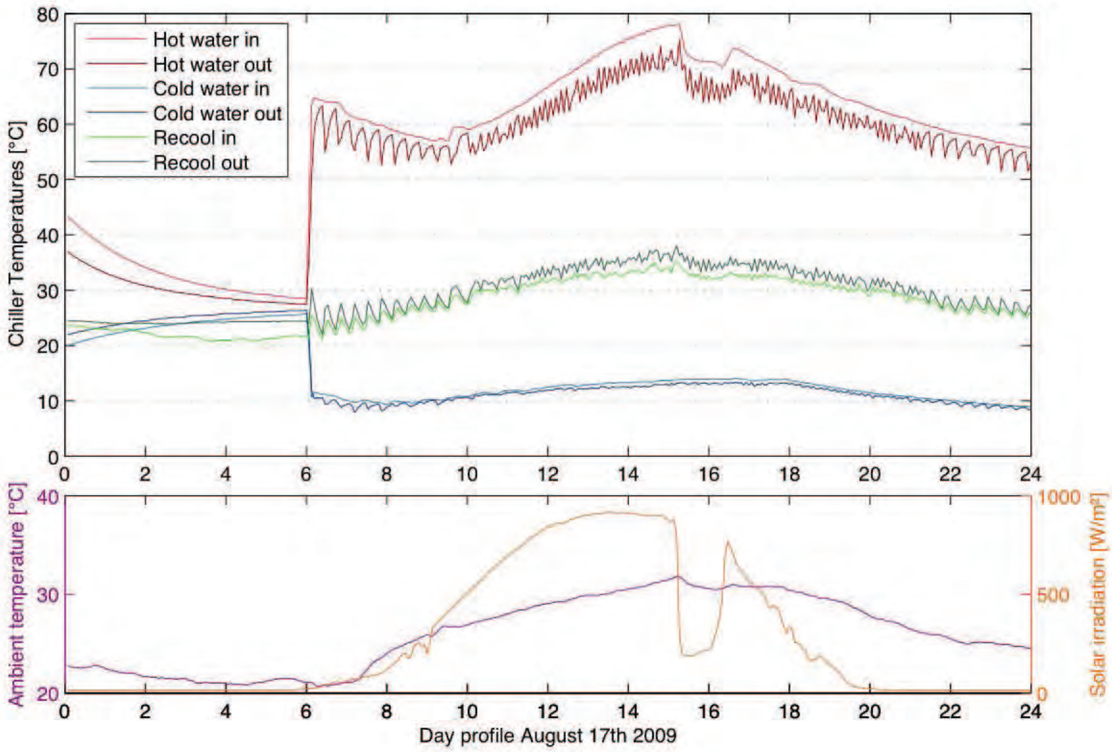
Figure 4 shows that the daily COP_{el} are in a range between 0,2 and 2,3 in August 2009. The low COP_{el} are mainly caused by the low cooling capacity needed in the offices. The adsorption chiller operated nearly all the time in a very low part load, but the three pumps around the adsorption chiller and the heat rejection were operating in full load.



Starting from the nominal cooling capacity of the SorTech chiller, the maximum daily cooling energy can be determined. A 7.5 kW chiller can yield a maximum of 180 kWh of cooling energy per day, assuming incessant operation. The number of operation hours in a solar cooling system, however, can be limited by the heat source. An six-hour long operation would ideally still yield 45 kWh of cooling energy. Figure 5 shows daily cooling energies between 0 and 23 kWh.

The values vary according to the control strategy, which indirectly reacts to the outside temperature and solar radiation. In addition, the consumer behaviour must be taken into account. Constant low values of the cooling capacity imply that the chiller is over-sized for the system.

Apart from the energy output of the system, it is also very important to look at temperature levels going in and out the adsorption chiller, as the hot side and the re-cooling temperature levels can drastically influence the performance. Figure 6 shows the inlet and outlet temperatures on all three temperature levels at the chiller for a typical August day. The monitored outside temperature and solar irradiation can be found in the lower graph. The outside temperature peaked at 32℃ and the solar ir radiation reached 917 W/m2. Overall, it was a bright sunny day with a cloudy passage in the afternoon.



The chiller turned on at 6 a.m. and was in operation through the rest of the day. Hot water, heated by the solar collectors, was driving the chiller with 57-78℃, depending on the time of the day (top curve). It is noticeable, that the loss in radiation by a cloud decreased the hot tank temperature with almost no time delay. The temperature difference between the two red curves stands for the energy intake of the adsorption chiller. Oscillations in the dark red curve reflect the timing of the chiller, which occurs when it switches between the adsorption chambers. The timing can still be seen in the re-cooling temperatures (green curves). The top green line is the temperature of the water going from the chiller to the cooling tower. After being cooled down, the water returns to the chiller (light green curve). The temperature spread generated by the cooling tower did not seem to be affected by the high outside temperature, as the ΔT was not decreasing. Chilled water, represented by the blue curves, only showed a small temperature spread of 1-2 K. However, the overall temperature level of the cold water went down towards the end of the day. This was due to a stop in the cooling demand from the offices.

This characteristic diagram of temperature levels at the MA34 adsorption chiller shows, that it was not a problem for the chiller to meet the cooling demand of the offices. Temperature levels, at which the chiller is driven, are within the recommended range of the manufacturer. Nevertheless, the ΔT for the cold water remains below expectations, which is expressed in the actual cooling capacity and also affects both the thermal and electrical COP.

ΔT_cold is dependent on both, the hot side temperature and the heat rejection temperature entering the chiller according to thermodynamic principles and the efficiency of the chiller. In order to maximize ΔT_cold, it is necessary to drive the chiller with the highest desorption temperature possible. This can be done with changing volume flows and adapting the control strategy.

It is also essential to improving ΔT_recool to create a higher ΔT_cold. The more energy the cooling tower can dissipate, the higher ΔT_cold the chiller is able to reach. Modifying volume flows and the controls could be a way to make some improvements.

Despite all attempts to increase ΔT_cold, it is also necessary to look at the actual temperature level of the cold water. The SorTech adsorption chiller can cool down to about 5℃, however, the lower the temperature level, the less efficient the chiller works. It can be seen in the above figure that the cold side temperature level drops below 10℃ at night. On other monitored days, going down of temperatures to 6℃ as not uncommon, and only then the chiller turned off. Such low temperatures are certainly not necessary for cooling offices with fan coils. Flow temperatures from 15-18℃ are sufficient and probably even more comfortable. Also, heat losses in the cold storage tank could be reduced. Therefore it is highly recommended to change the set-point for the chiller, to turn off at a higher temperature.

All previously mentioned adaptations aim at increasing ΔT_cold or the efficiency of the chiller, which results in both, higher thermal and electrical COPs.

It is, however, necessary to decide whether to optimize the solar cooling system to a high electrical COP or to a high thermal COP. Improvements in the COP_{el} can be attained by applying earlier mentioned modifications as well as by cutting down the electrical consumption.

Figure 7 shows typical COP_{th} and characteristic curves of the here analysed adsorption chiller (Type ACS08).

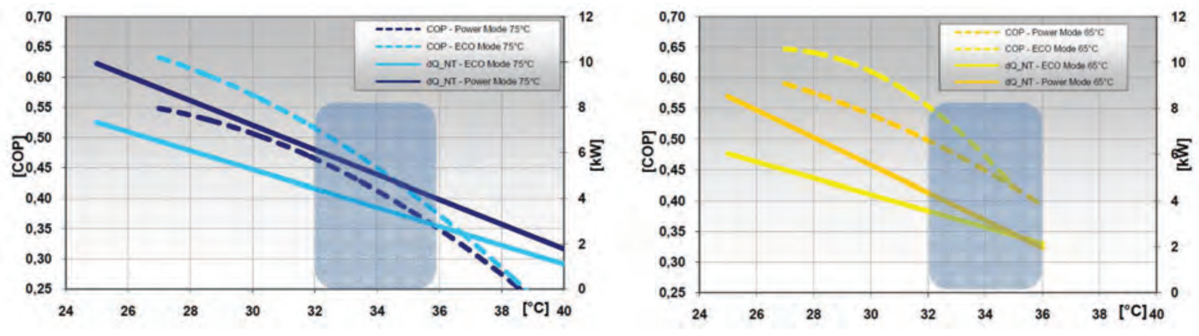
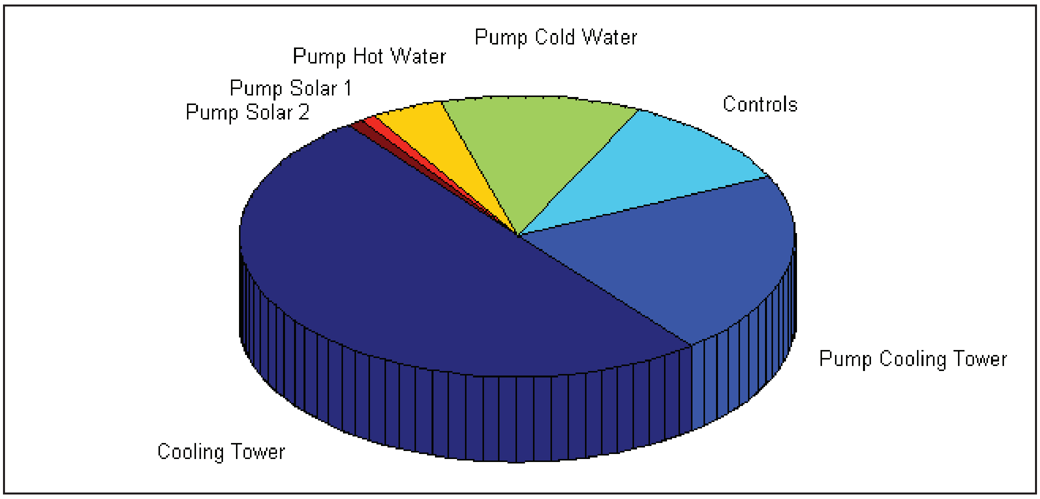


Figure 8 shows the share of all devices in electricity consumption. It can be seen that heat rejection and the associated pump make up almost two thirds of the entire electricity consumption. Hot and cold water pumps together with the control device account for almost all the rest. The pumps in the solar cycle I and II are the only energy saving pumps and they are also the only speed controlled pumps in the solar cooling system. A lot of electricity could have been saved by equipping the entire system with energy saving pumps. However, some additional improvements may be made by operating the pumps connected to the chiller at variable speeds instead of steady flow.



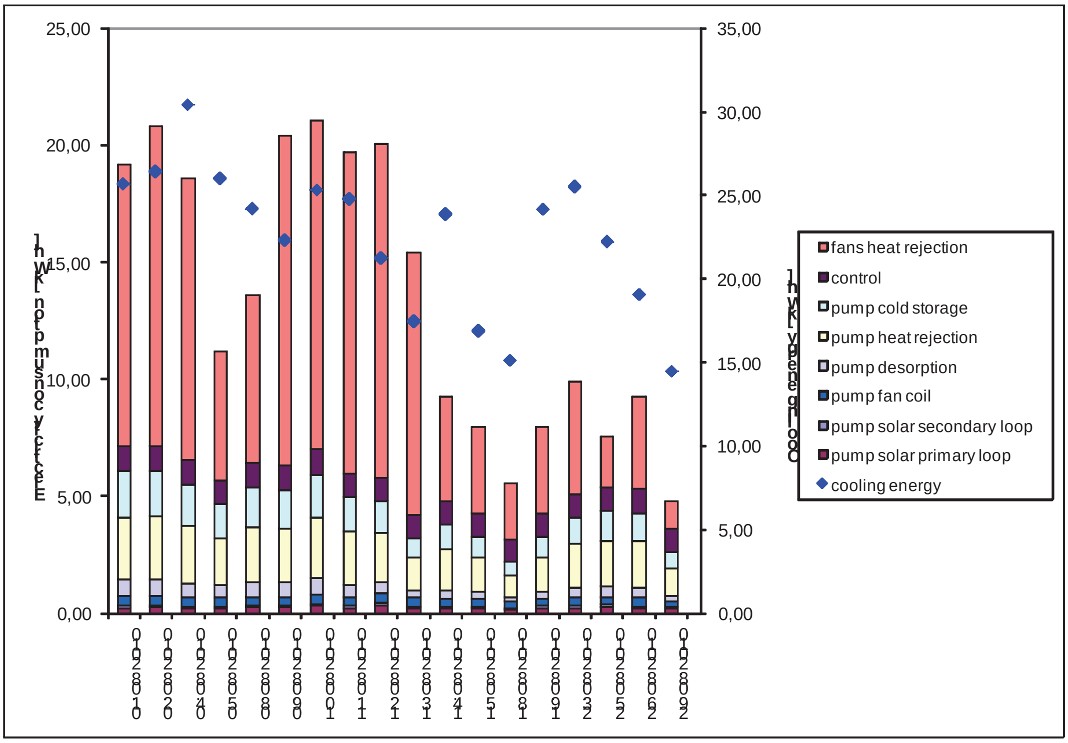
The control system also exhibited a surprisingly high energy use. Monitoring showed a constant consumption of 40 W, which is definitely higher than many other control systems available in the market. The cooling tower is by far the biggest consumer of all electrical components.

Interestingly enough, monitorings uncovered a random consumption of the cooling tower of constant 130 W on top of the regular consumption during operation times. There is no indication of a stand-by mode and speculations about an incorrectly wired anti-freeze element turned out to be wrong. So far, it has not been clarified what really causes the extra electricity consumption.

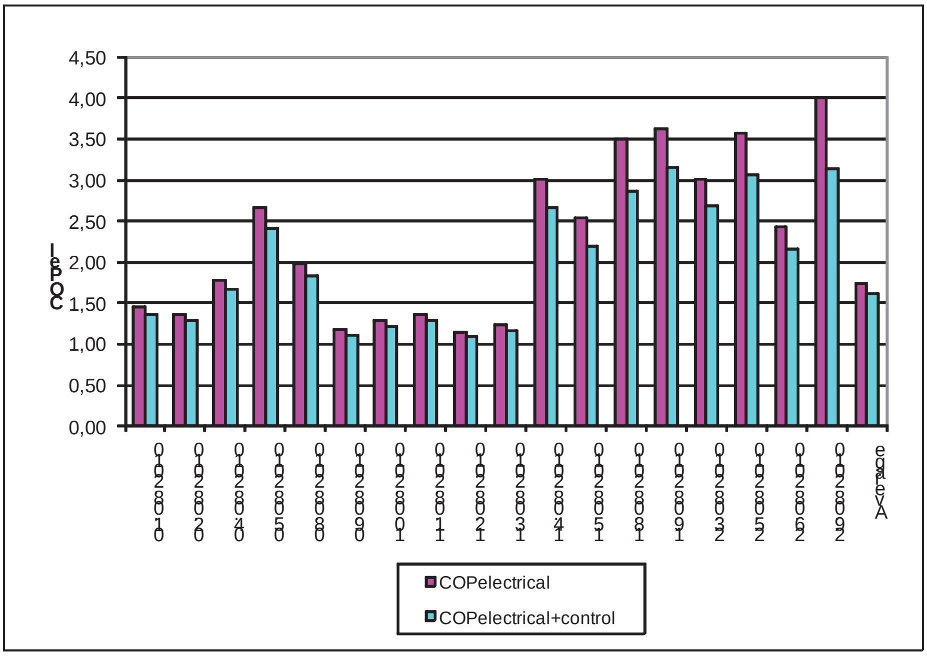
4 Experience report summer 2010

The following section contains monitoring results of the MA34 solar cooling system for August and September 2010 with a focus on the variable speed control mode of the heat rejection fans.

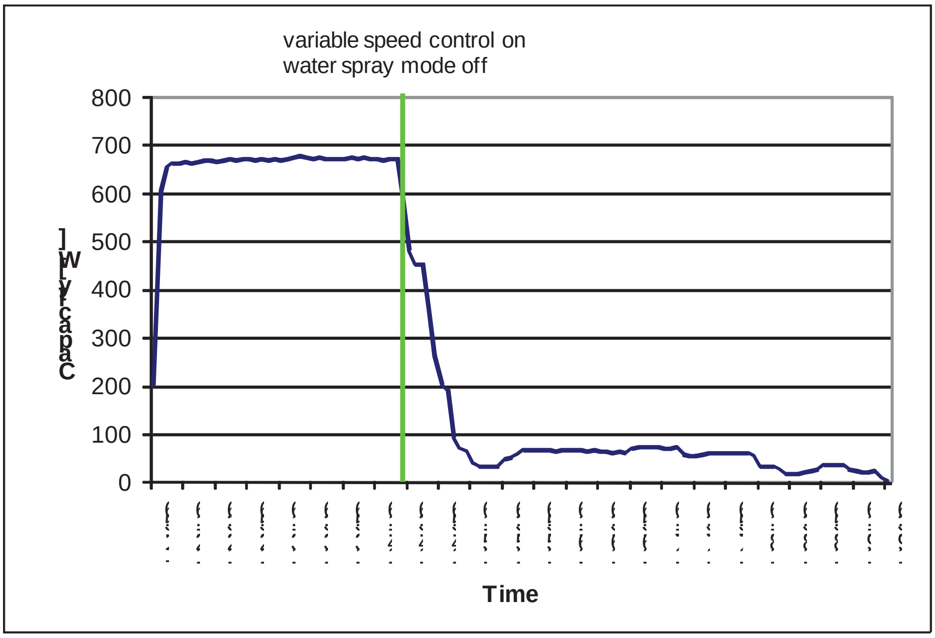
The monitoring evaluation of August showed that the heat rejection operated permanently, even when the rest of the plant wasn´t in operation (see Figure 9). That was caused by an error set at the adsorption chiller control panel, which was “manual” instead of “automatic”. This error was corrected at the 14th of August 2010; the resulting decrease of electricity demand is shown in Figure 9. Furthermore, the COP_{el} increases significantly from that date on (see Figure 10) although the heat rejection is operating with maximum capacity (ca. 650 W).

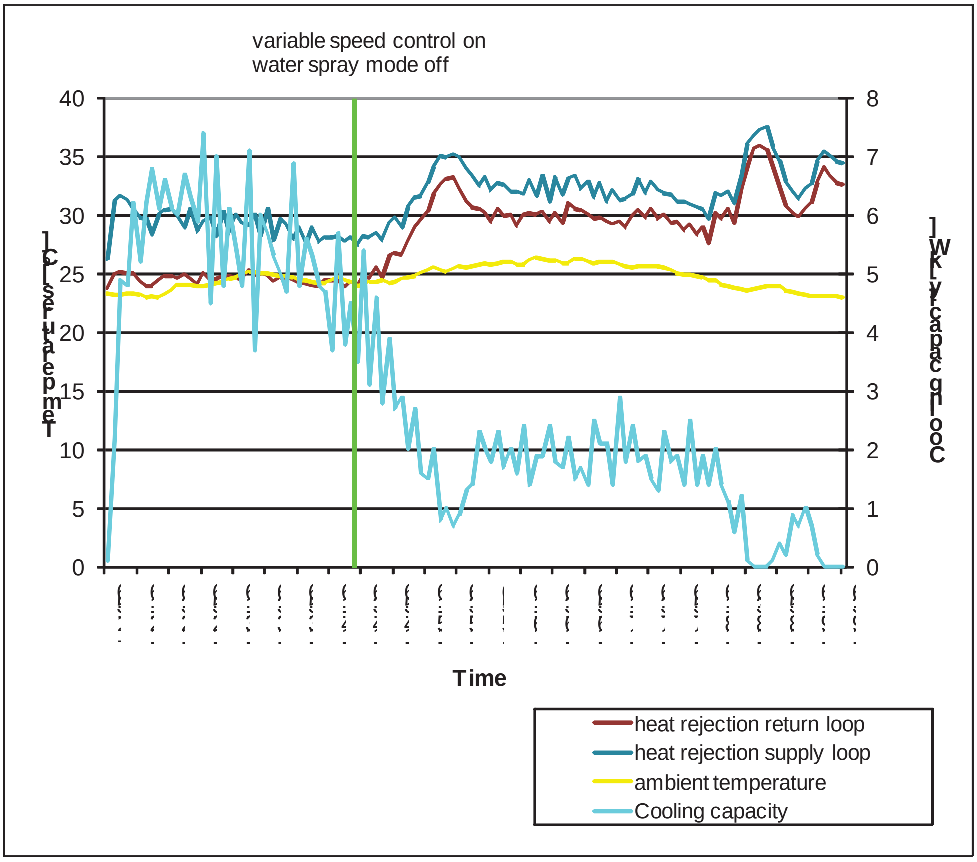


[illegible]

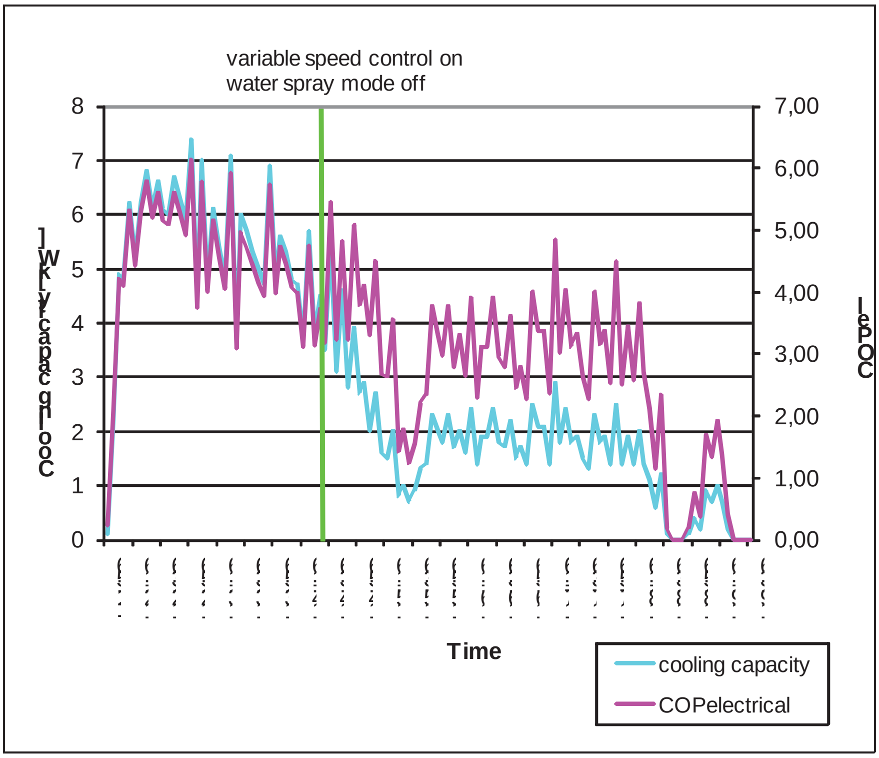


On the 25th of August 2010 the variable speed control of the heat rejection fans started to operate the first time for a longer time period. The heat rejection fans - which cause the main part of the electricity consumption – operate between 670 W and 65 W driving capacity (see Figure 11).

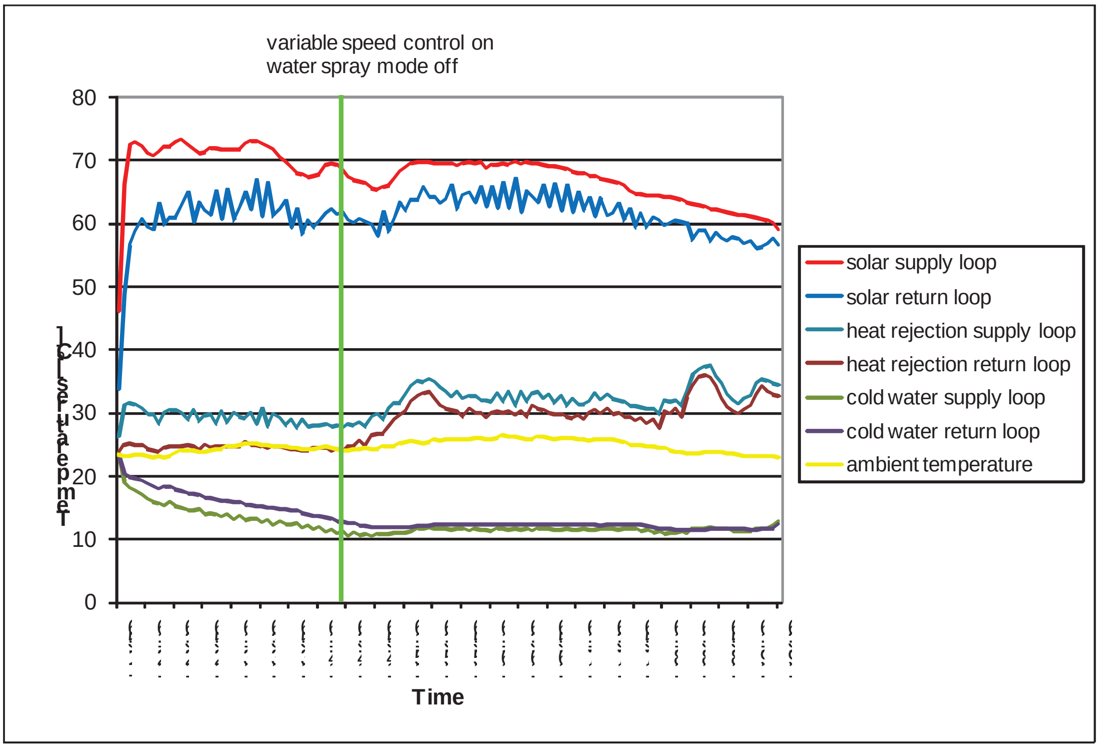




As a result, the heat rejection temperatures increased which affected the delivered cooling capacity (see Figure 12). The cooling capacity decreased, due to the higher heat rejection temperatures, significantly (see Figure 13).

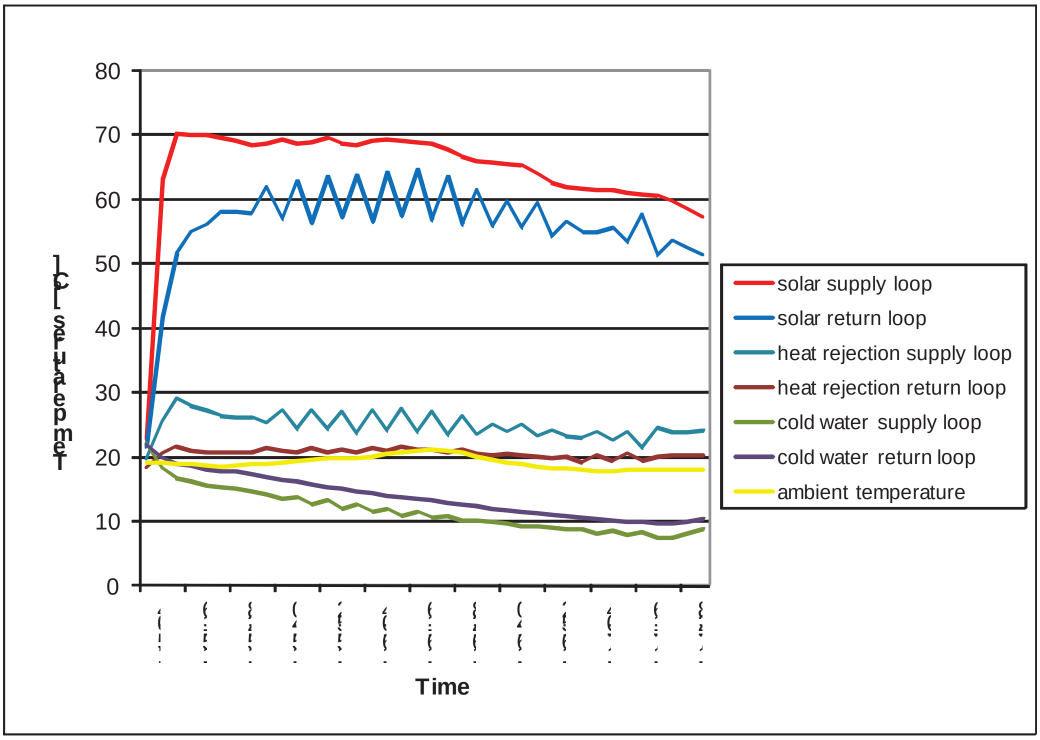


The average COP_{el} without variable speed control of the heat rejection fans is 4,55 at the 25th of August 2010 (see Figure 13). In the time period, when the variable speed control of the heat rejection fans is in operation, the average COP_{el} decreases to 2,78. The average cooling capacity of the adsorption chiller in the time period without variable speed control of the heat rejection fans amounts to 5,46 kW; with variable speed control of the heat rejection fans this value drops to 2,24 kW. Furthermore, the wet operation mode of the heat rejection turns off by starting the variable speed control mode.



In Figure 14, the temperatures around the adsorption chiller on the 25th of August 2010 are shown. It is clearly identifiable, that while the heat rejection temperatures are rising, nearly any temperature difference between supply and return cold water side happens during the variable speed control mode of the heat rejection fans. Therefore, there wasn´t nearly any cooling demand in the offices and the adsorption chiller shouldn´t have been working at all. The control of the absorption chiller is only taking a certain cooling set point temperature on the cold supply water side into account (here 12 °C). It is therefore the duty of the system controller to turn off the adsorption chiller, when an unreasonable operation like this happens, but that was not adapted by installing the variable speed control mode of the heat rejection fans in July 2010.

To avoid the operation, as it happened on the 25th of August 2010, the cooling set point temperature was changed to 6 °C until the adaptatio n of the system controller happens (see Figure 15).



5 Summary and Conclusions

Within the IEA SHC task 38 for the solar adsorption cooling plant of the Viennese Municipality Department 34 (MA34) following investigations were accomplished:

- COP_{el} and COP_{th} of the solar adsorption cooling plant
- Operation of hybrid heat rejection in dry or wet cooling mode
- Effects of variable speed control of the heat rejection fans on the energy performance

Summary of the substantial realizations:

- The heat rejection in this plant causes three quarters of the electricity demand, therefore the selection of a heat rejection device with a high efficiency class (preferably wet cooling tower or at least hybrid heat rejection) is essential to achieve a high COP_{el}.
- Selection of variable speed, energy-efficient pumps are also important to achieve high COP_{el}.
- The monitoring results of the 25th of August 2010 showed that a COP_{el} of 4,55 is already possible with this plant. Therefore, high desorption temperatures are

necessary (> 70 °C) and a wet cooling mode in the heat rejection. This value is clearly over the maximum daily value of the COP_{el} in August 2009 with 2,3.

- Due to the water temperatures in all three hydraulic circles (cold water, heat rejection and hot water circle) of the adsorption chiller daily COP_{th} were measured in summer 2010 within the range of 0,16 to 0,58. The nominal COP_{th} is about 0,56.
- The plant optimization attempt, by using variable speed control of the heat rejection fans didn´t work due to the lack of coupling between the adsorption chiller control and the system control; on the contrary, the COP_{el} decreased during the variable speed control mode. Generally, the usage of a system controller including the chiller control must be recommended to avoid such errors.

6 Publications

A. Preisler, M. Brychta, F. Dubisch, F. Stift, T. Edlinger, Solar-gestützte DEC-Anlage ENERGYbase, Wien: Evaluierung der Anlage durch Vergleich TRNSYS Simulationen mit Monitoring-Ergebnissen für den Sommer 2009, 20. Symposium Thermische Solarenergie, Bad Staffelstein, 5/2010

A. Preisler, T. Selke, Experience report on two different solar driven air-conditioning systems in Vienna/Austria based on monitoring data of summer 2008/2009, 3rd International Solar Air-Conditioning Conference, Palermo, 10/2009

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 4

Monitoring Results of Résidence du Lac, Maclas (France)

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 30th November 2010

By Romain Siré[1] and Daniel Mugnier[1]

[1]Institution	TECSOL
Address	105 av. Alfred Kastler – Tecnosud
	66004 Perpignan CEDEX
	France
Phone	+33 4 68 68 16 40
Fax	+33 4 68 68 16 41
e-mail	romain@tecsol.fr

Contents

1 Background

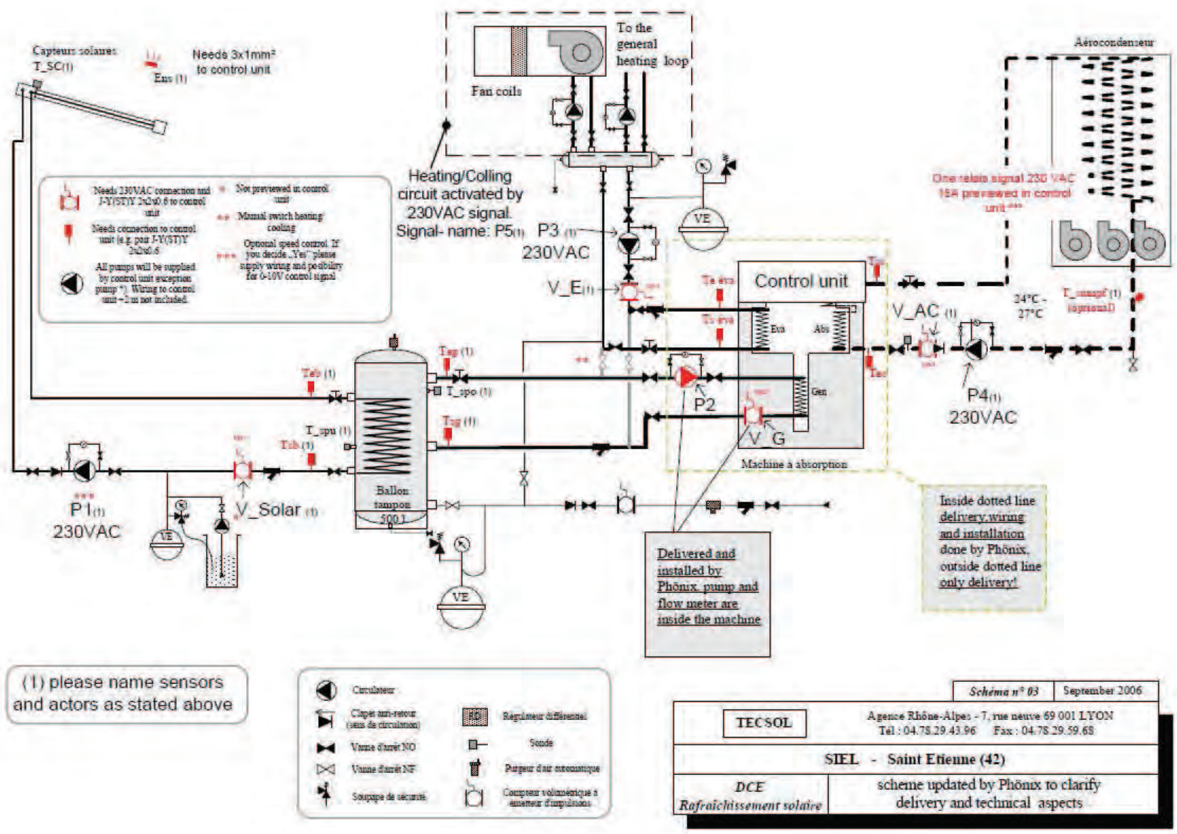
The targeted building welcoming the solar cooling application is the Résidence du Lac, a building dedicated to retired people. This building is located in the small town of Maclas in the Rhône Alpes area, close to Lyon. The town is in altitude, nearly 450 m high.

The building was created in the seventies and is of an average quality level for the energy efficiency. Only one small part of the building is cooled, the leisure space/restaurant which is compulsory since summer 2003 in retired buildings. This area is of 210 m² and includes a veranda oriented in the Southern direction. Efforts were made to increase the solar protection level in the veranda by adding dark thin protection films. Till 2007, the building owner used electric compression chillers (3 monosplits). Two of them were out of order in 2007 and the management took the decision with the help of the SIEL (Syndicat Intercommunal d'Energie de la Loire) to go for a solar cooling system. The owner of the system is the SIEL itself.

2 System Design

The system is based on an absorption chiller of 10 kW coupled with evacuated tube collectors. The system is in configuration of a quasi solar autonomous cooling system because only a small electric compression chiller (split type) is used in case of failure of the solar system. The load of a part of the entire building is based on the following scenario: cooling demand from June to mid September and heating demand from mid October to end of May. The solar system is using fan coils for the cooling and heating modes but thanks to buffer storage, it can be valorized as well in the heating mode through the central heating network of the Résidence du Lac. The heat rejection system is done by a drycooler located in the northern facade of the building.

Installation's scheme:



Hydraulic scheme of the Maclas installation

As it can be seen, the system was designed to limit as much as possible the electricity consumption of the overall installation. Indeed only 4 pumps are used to run the entire system (without distribution). Moreover, a simple system always works better than a complicated one.

Another point which must be enlightened it the choice of a dry cooler for the system's heat rejection. The "Résidence de Lac" is a building for retired peoples. As a consequence legislation is very drastic concerning the legionella bacteria, and a drycooler was the only affordable way to reject efficiently energy without any restrictive regulation.

Building:

- Type of building: Retired people residence
- Location: Maclas, France
- In operation since 2007
- System operated by: SIEL
- Air-conditioned area: 210 m²
- System is used for space cooling and heating (No DHW preparation)

System general properties:

- Technology: closed cycle
- Nominal capacity: 10 kWcold
- Type of closed system: Absorption
- Brand of chiller unit: Sonnenklima
- Chilled water application: Fan coils
- Dehumidification: no
- Heat rejection system: dry

Solar thermal:

- Collector type: Evacuated tube
- Brand of collector: Thermomax Mazdon 20
- Collector area: 24 m² absorber
- Tilt angle, orientation: 30°, 15° west
- Collector fluid: water-glycol
- Typical operation temperature: 75°C driving temperature for chiller operation

Configuration:

- Heat storage: 0.5 m3 water
- Cold storage: Buffer water (80 liters)
- The hot and cold backups are not included into the solar project, so they are not monitored. Indeed, the building is an existing building, so a conventional heater and air conditioning system were already used.

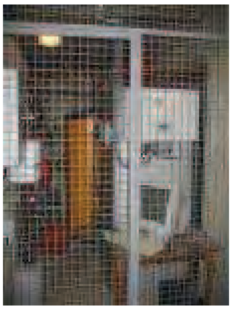
Photos:

Building

Cooled space

Solar collector field

Dry cooler

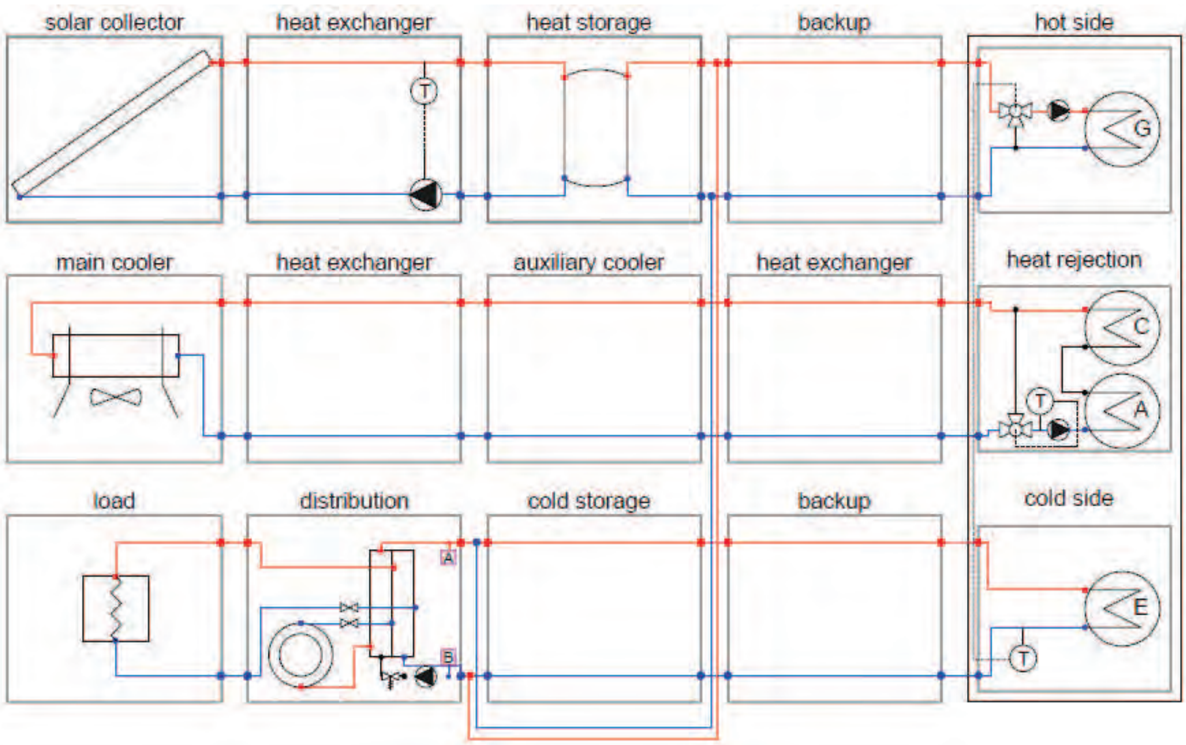


Technical premises

Distribution system

Task 38 standard Scheme for the installation:

If the standard Task 38 scheme is used to describe the installation, the scheme is as follows:



Task 38 standard hydraulic scheme for the Maclas installation

3 Control Strategy

The installation control is realized by the internal controller of the Sonnenkilma chiller. Indeed, this chiller goes with its internal regulation. This controller can be remotely programmed via an internet access and a VPN.

The control strategy is as follows:

The solar loop's pump starts when the solar radiation reaches a certain value. So the hot storage starts warming up.

Then, when the temperature at the top of the hot tank reaches a certain value (75°C), the three pumps of the chiller starts running (generator, evaporator, and condenser). And the cold production begins.

Of course, several setpoint and thresholds were established to stop the installation in case of problems or at the end of the day.

As it was explained in the previous paragraph, the system was designed to be as simple as possible. And it is the same principle for the control strategy.

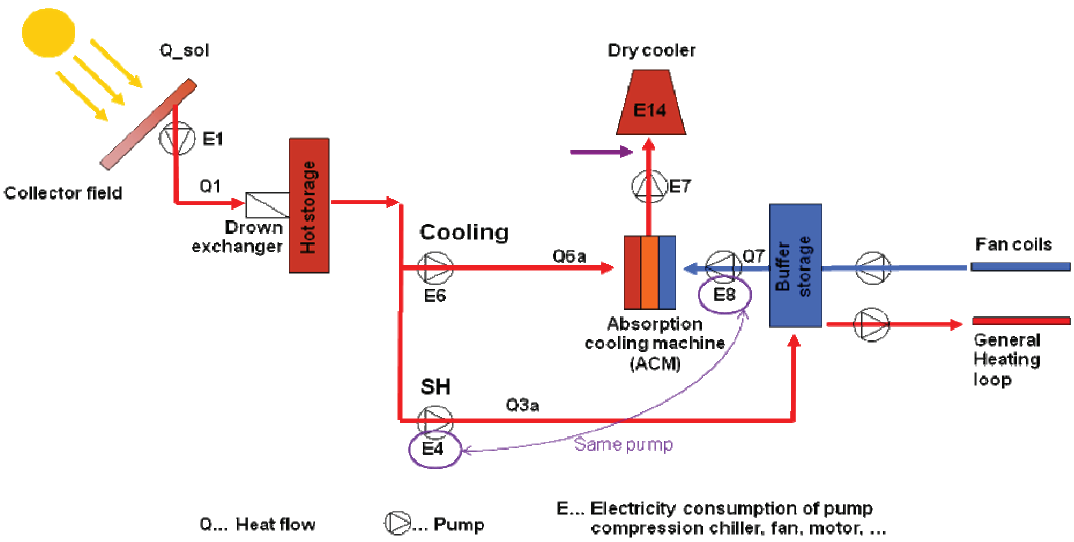
4 Monitoring Equipment

4.1 Installed Equipment

The measured data are basically the energy flow for every loop of the installation. As a consequence, at least two temperature measurement and one flowrate measurement are taken for each loop of the installation.

The solar radiation and the overall electricity consumption of the installation are also measured.

The Task 38 standard monitoring scheme below shows the different energies monitored:



Task 38 standard monitoring scheme for the Maclas installation

Going deeply into the details, the monitored data are:

VARIABLE	EXPLICIT MEANING	UNIT
Zeit	Date and time	
V_G	Flow in the generator piping	m^3/h*100
V_E	Flow in the evaporator piping	m^3/h*100
V_AC	Flow in the absorber/condenser piping	m^3/h*100
E_TS	Solar Irradiation	W/m^2
V_K	Flow in the collectors piping	m^3/h*100
t_Gh	“temperature generator hot” = temperature at the entry of the generator	°C*10
t_Gc	“temperature generator cold” = temperature at the exit of the generator	°C*10
t_Ec	“temperature evaporator cold” = temperature at the exit of the evaporator	°C*10
t_Eh	“temperature evaporator hot” = temperature at the entry of the evaporator	°C*10
t_ACc	“temperature absorber/condenser cold” = temperature at the entry of the absorber	°C*10
t_ACh	“temperature absorber/condenser hot” = temperature at the exit of the condenser	°C*10
Q_G	Energy used by the generator	kW*100
Q_E	Energy accumulated at the evaporator	kW*100
Q_AC	Energy given at the absorber/condenser	kW*100
COP	Coefficient of performance	none*100
Q_Ballon_out	Heating Power in winter operation	kW*100
t_Koll	“temperature collectors” = where is it taken exactly? You have to check at site, as far as I know directly in the collector	°C*10
t_Kh	primary circuit solar coming from field	°C*10
t_Kc	primary circuit solar going to field	°C*10
t_sp11	Upper storage temperature	°C*10
t_sp12	Lower storage temperature	°C*10
t_stout	Storage outlet temperature (to SAC=Solar absorption chiller)	°C*10
t_stin	Storage input temperature (coming from SAC)	°C* 10
Q_Solar	Power Solar Circuit	kW*100
t_room_AKA	Temperature inside control unit	°C*10
Setpoint_Ec	Cold Water Setpoint	°C*10

4.2 Period of Measurement

Every data are measured and saved in daily “historic” files with a time scale 30 seconds. These files were downloaded and analyzed frequently about every week.

The data available for the Task 38 monitoring is only the summer 2009 (from June to September). Unfortunately, after this date a few issues prevented us to get the monitoring data (problems with the building owner, and it was not possible to get the monitoring data because of the Sonnenklima insolvency).

5 Monitoring Results

5.1 Annual / Monthly Data

The table below shows the energies considered in each loop of the installation. It also shows some of the main performance factors calculated monthly and yearly for the monitoring period:

	Yearly	June	July	August	September
	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]
Total Electricity Consumption	351,9	115,0	82,2	84,3	70,4
solar irradiation on collector aperture area	8516,9	3519,0	1584,4	1727,5	1686,0
solar thermal output to hot storage	3404,6	1387,9	795,3	607,3	614,1
hot storage input to cooling machine (ACM)	1693,3	511,9	346,5	447,0	387,8
cold output ACM to cold-storage	915,5	271,0	167,3	252,4	224,8
cold storage output to cold-distribution	869,8	257,5	158,9	239,8	213,6
	[-]	[-]	[-]	[-]	[-]
Collector efficiency (-)	0,40	0,39	0,50	0,35	0,36
Thermal COP (-)	0,53	0,48	0,56	0,58	0,53
Electrical COP (-)	2,24	1,93	2,84	3,03	2,24

During the considered monitoring period about 900 kWh of cold was produced and supplied to the building. The collector efficiency were about 0,40 which is good, but expected because the collectors are evacuated tubes collectors. The performance of the chiller is correct, indeed the yearly average of the thermal COP reaches 0,53. On the other hand, the electrical COP is a little bit lower than it could be expected: the average of the electrical COP for this cooling period reaches 2,24 which is not bad because only the cooling period was considered, but it could be better for a solar cooling installation. Actually this value of electrical COP can be explained because of the use of a drycooler to reject the heat. Indeed its two fans are consuming more than a conventional cooling tower, but it was the only affordable way to avoid every legionella risk.

5.2 Analysis of a typical good day: 23th August 2009

The table below summarizes the performances of the installation on 23th August 2009:

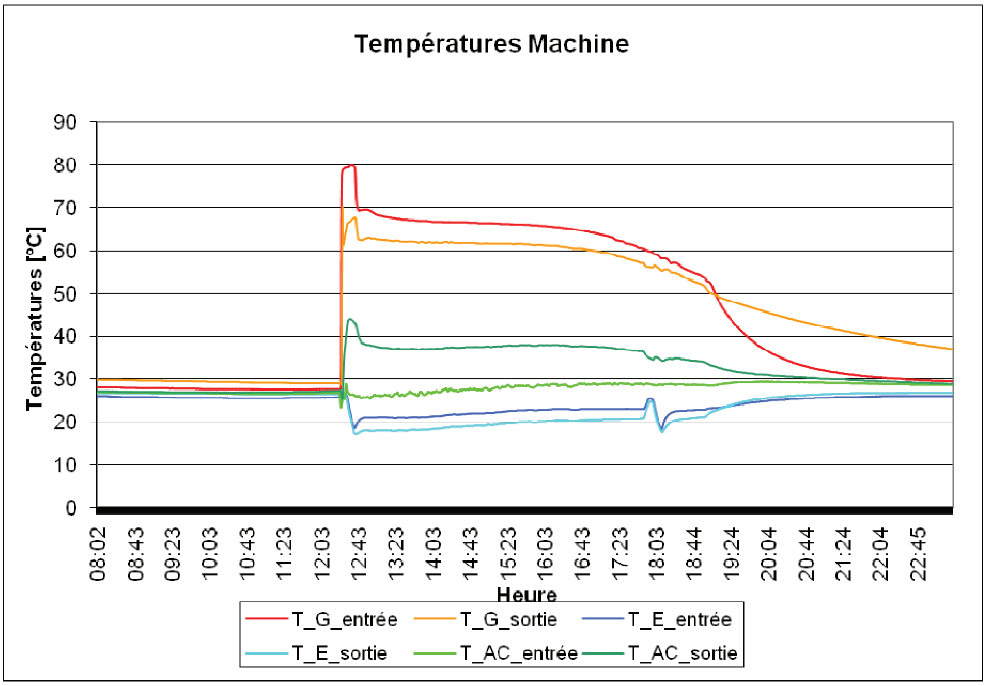
Ensoleillement	177,56	kWh
Energie solaire	60,39	kWh
Energie générateur	50,87	kWh
Energie Evaporateur	30,66	kWh
Total énergie élec utilisée	9,19	kWh
Rendement capteurs	**34,01**	**%**
COP thermique	**0,60**	
COP électrique	**3,34**	

During this day 30,66 kWh of cold was produced and supplied to the building. The collector efficiency were about 0,34 which is good, even if we could have expected a little bit better because the evacuated tubes collectors are used.

The performance of the chiller is good, indeed the daily average of the thermal COP reaches 0,60 which is a value which can be expected from this chiller.

The average of the electrical COP for this day reaches 3,34 which is good. Of course this value could have been better if another heat rejection system have been used, but this value is still good when a drycooler is used.

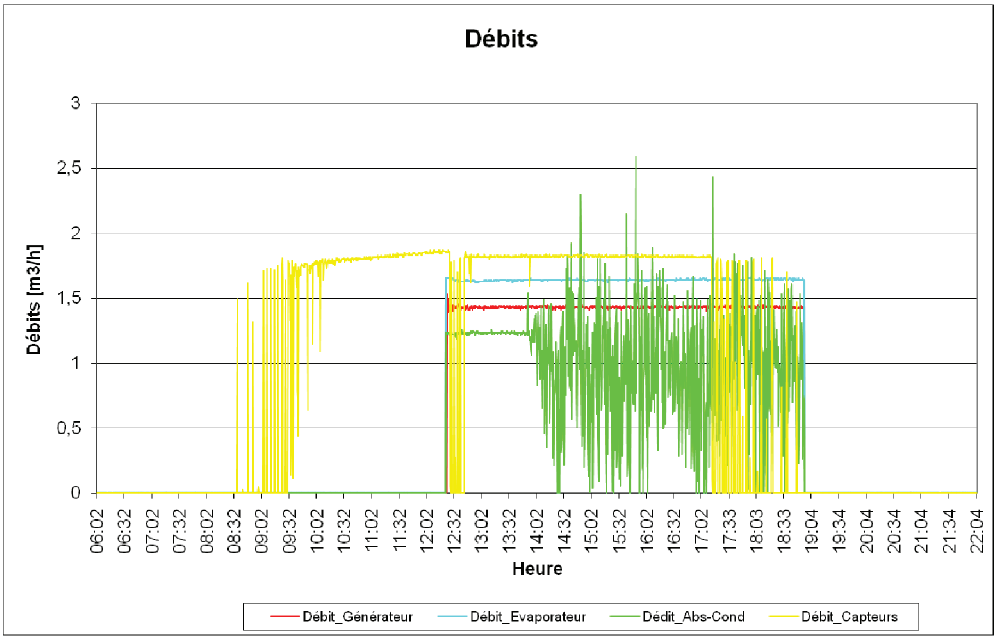
The chart below shows the temperatures at the input and output of the chiller:



Temperatures at the input and output of the chiller

It can be concluded the chiller works properly from about 12:30 to 19:00. The driving temperature at the generator is more or less always between 60℃ and 70℃, while the cold is produced at about 20℃.

The chart below shows the flowrates into the different loops of the installation:



Flowrates into the different loops of the installation

The solar pump is starting at about 9:00 and as said before the pumps for the other loops (generator, evaporator, and condenser) are starting at about 12:30.

The Generator and evaporator flowrates are pretty stable, but it's not the case for the condenser and for the solar pumps. Indeed, the solar pump works in short cycle at the beginning and at the end of the day, and when the chiller is starting running. And the condenser pump works in short cycle a few hours after the chiller started running and until the end of the day.

6 Experiences / Lessons Learned

As said before, no main problems were identified during the monitoring period. On the other hand a few problems happened after this period.

- The monitoring was not possible to do because of the insolvency of the chiller manufacturer, and because the monitoring devices were part of the chiller. But now this problem is solved, and the monitoring still can be done via a VPN.
- Because the installation stopped for a long period, the solar collectors which used a heat exchanger were cycling so now the performance of the collector field are lower than what it should be and what it was when the installation started.

7 Conclusions

The installation was able to run and have good performances for the entire monitoring period. Then the monitoring was not possible for some external reasons, and a few problems started. Now the installation is being renovated, and will start running soon.

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 5

Monitoring Results of CNRS PROMES Research Center Office, Perpignan (France)

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 30th November 2010

By Romain Siré[1] and Daniel Mugnier[1]

[1]Institution	TECSOL
Address	105 av. Alfred Kastler – Tecnosud
	66004 Perpignan CEDEX
	France
Phone	+33 4 68 68 16 40
Fax	+33 4 68 68 16 41
e-mail	romain@tecsol.fr

Contents

1 Background

The targeted building where the solar cooling and heating system is installed is the CNRS research centre PROMES in Perpignan. This building is an office building, but there is also numerous labs into this building. This building is located at the "TECNOSUD" area in Perpignan (Languedoc Roussillon, France), close to the Mediterranean Sea, with thus a Mediterranean climate.

PROMES laboratory building

The building is a large one: more than 5 000 m² on 3 floors. The collector field is located on the building roof, and the other part of the system is located on the first floor. The installation is cooling and heating only a small part of the building.

Area cooled/heated by the solar system

The solar collector field has been installed in the same axis than the building, that is to say the solar collector field is oriented to the South-East (45° East), so the system works earlier in the morning but stops faster at night. The building was built in the year 2000 and presents a good level of energy efficiency.

2 Solar cooling and heating system description

2.1 Key components of the system

The system is based on an adsorption chiller of 7.5kW capacity, and a double glazed flat plate solar collector field of 25m². The system itself is working without any backup, and is thus an autonomous solar cooling and heating system. However, comfortable conditions into the building are insured by a general cooling and heating system (multi split system) in parallel and working independently to the solar system.

Technical data about the chiller

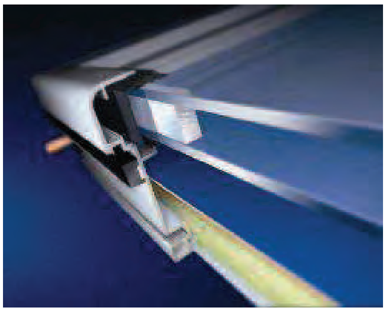
- Technology: closed cycle
- Nominal capacity: 7.5 kW cold
- Type: Adsorption
- Chiller brand: SORTECH
- Dehumidification: no

7,5 kW adsorption chiller used in Perpignan

Technical data about the collectors

- Collector type: High performances double gazed flat plate collectors
- Collectors brand: Schüco
- Collector field area: 25 m² (aperture area)
- Tilt : 30°
- Orientation: South/East (45°)
- Collector fluid: water
- Typical operation temperature: 75°C driving temperature for chiller operation

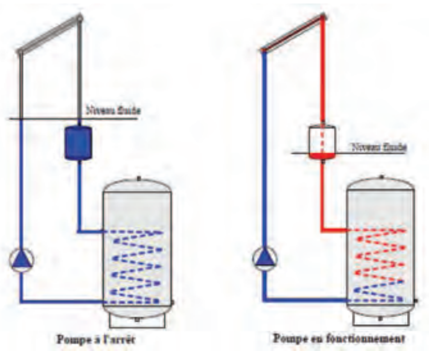
25 m² of double grazed solar collector



Double glazed technology

These solar collectors are auto-drainable. It was thus possible to design the installation with a drainback system. This system permits to have an installation safe (no risks of freezing or overheating with this system), efficient (performances of monitored drainback systems are very good), and simple (no expansion systems nor drain-cock devices).

The principle of this drainback system is as follows: when the solar loop pump is not working, the collector fluid level in slightly above than the drainback tank. The collectors are thus filled in with air at atmospheric pressure. This amount of air in the system permits to absorb the expansion of water when the system is working. When the solar loop pump is starting, the manometric pressure of the pump permits to pump up the level of fluid until a level above the solar collectors, as a consequence, the air is driven out to the drainback tank. The volume of air inside the loop permits to drain the collector field and prevent it to freeze or overheat, and it also permits to absorb the expansion of water when the system is working. In the loop neither water not air are replaced when the system is working, so there is no need to treat water, and there is no risk of corrosion.



Scheme of a drainback system

The energy distribution into the building is managed thanks to a independent loop. This loop is compatible with cold and with hot water. This loop is connected to three fan coils working at a temperature level of 14/18°C.

Fan coils for the distribution system

The heat rejection is managed by a drycooler. To insure a sufficient heat rejection for the entire cooling period, the drycooler is equipped with a water spraying system. This system is used only for the very warm days in summer, when the drycooler alone can't insure a sufficient heat rejection from the system.

Drycooler used at the perpignan installation

A small hot storage of 300 liters is installed between the solar collectors and the chiller. An internal exchanger is present in this tank. This storage is actually a buffer storage, which permits to smooth the solar radiation variations.

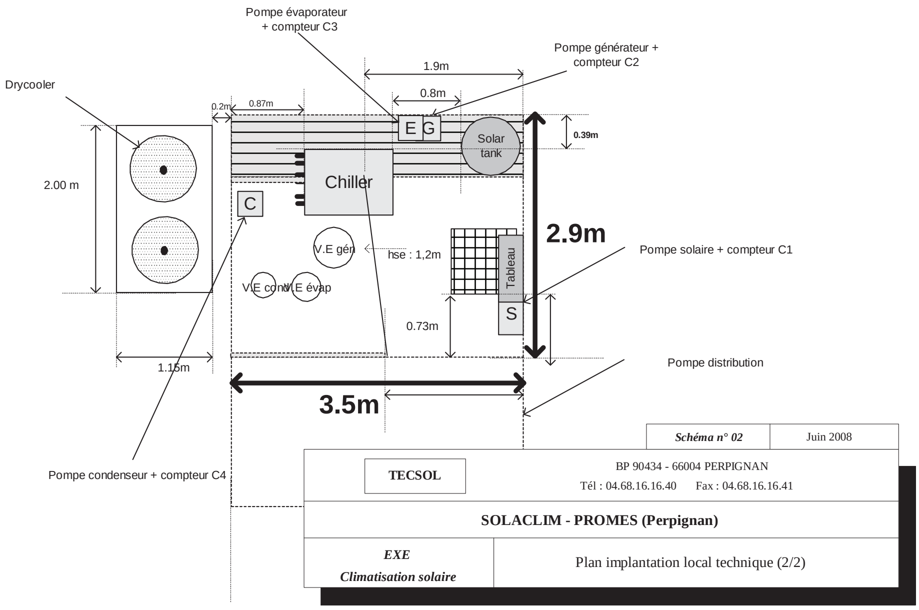
Similarly, a small cold storage is also installed between the chiller and the distribution circuit.

Hot storage tank of 300 liters

2.2 The system in its entirety

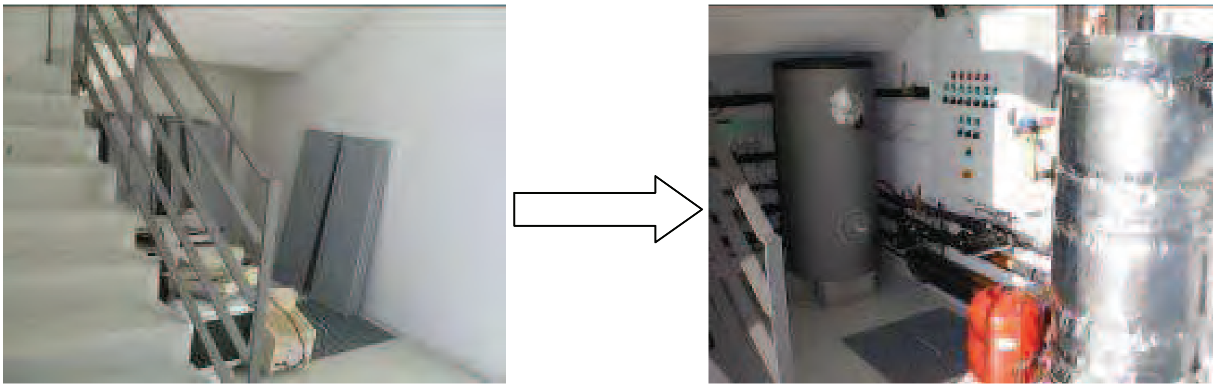
The layout constraints were very large for this installation. Indeed, the only space available for the system was located into one of the building stairwell. The system had to be designed very carefully and as cleverly as possible to avoid any trouble in the traffic of the stair.

The implantation scheme is as follows:



System implantation scheme

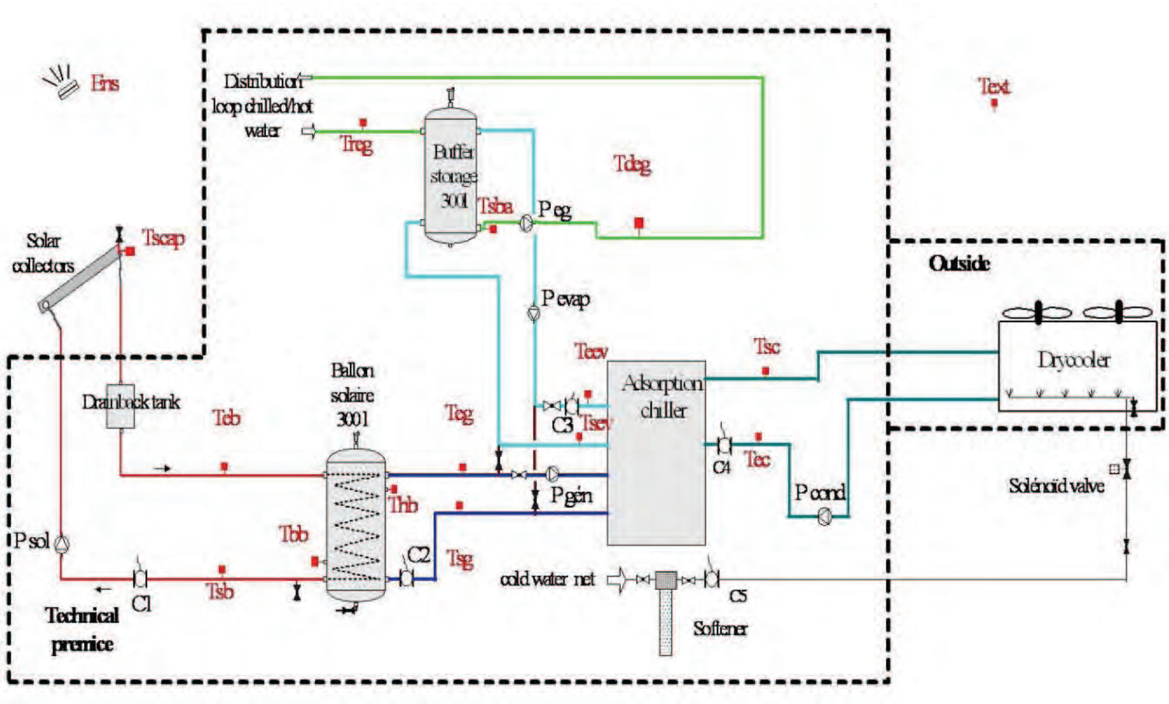
So it was possible to implant the entire system into this small spot of about 10 m², and a large part of it is located under a stair.



Technical premises before and after the works

The installation was designed to be used following to working mode: the first one is the cooling mode which is used in summer from May to October, and the second one is the heating mode which will be used during winter from November to April.

The installation's hydraulic scheme is showed below:



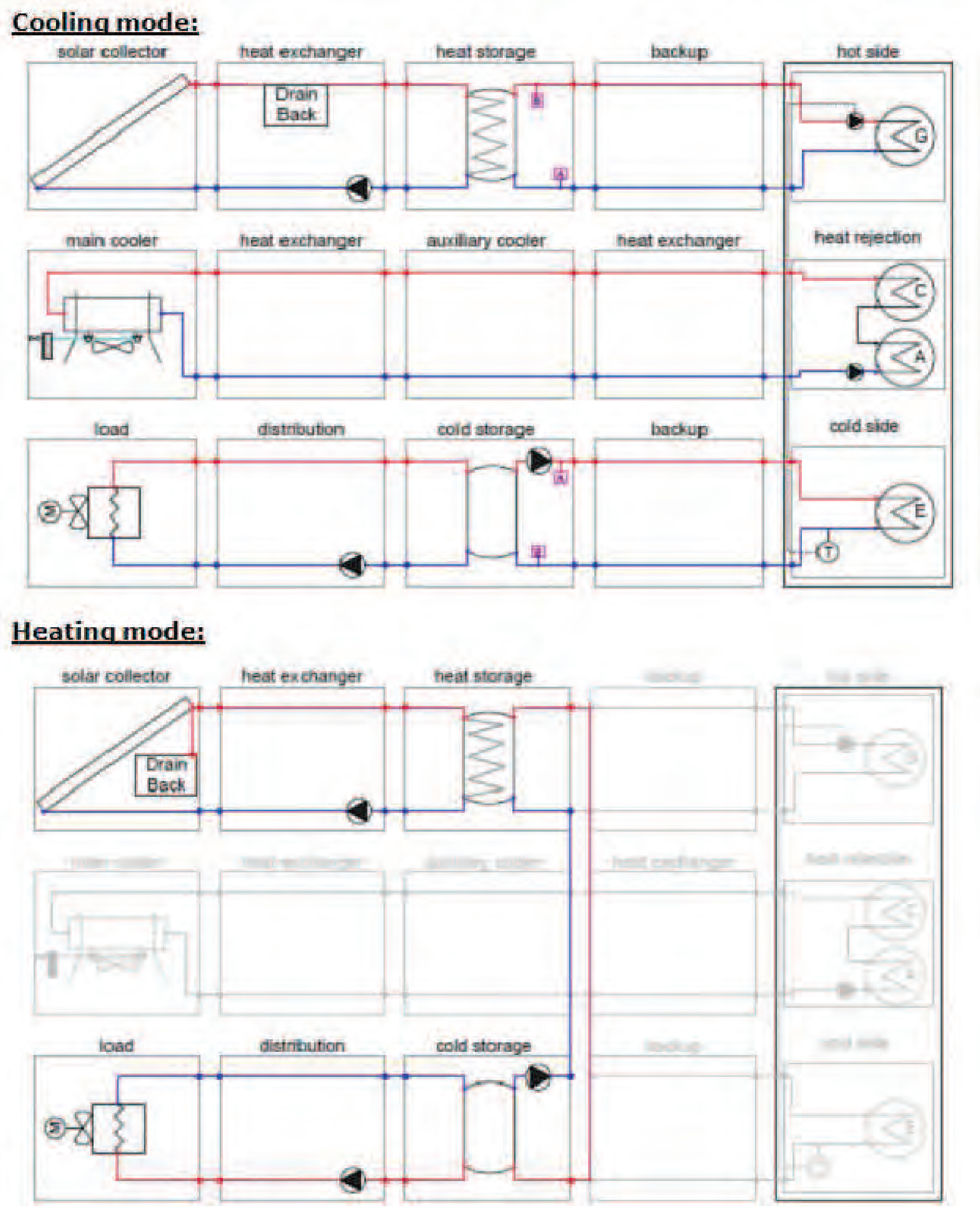
Installation's hydraulic scheme

When the installation is used in cooling mode, the circuit between the generator circuit to the evaporator circuit (in brown on the scheme above) is not used.

When the installation is used in heating mode, this circuit is used, and the chiller is by-passed, and the drycooler is not used neither.

For the monitoring of the installation, the thermal energies of each circuit are calculated by measuring the temperatures and the flow rates. The solar radiation is also measured as well as the total electricity consumption to characterize the energy required to drive the system..

The standard task 38 hydraulic scheme for this installation is shown below:



3 System monitoring and performance calculation

The first start of the system was done in July 2008.

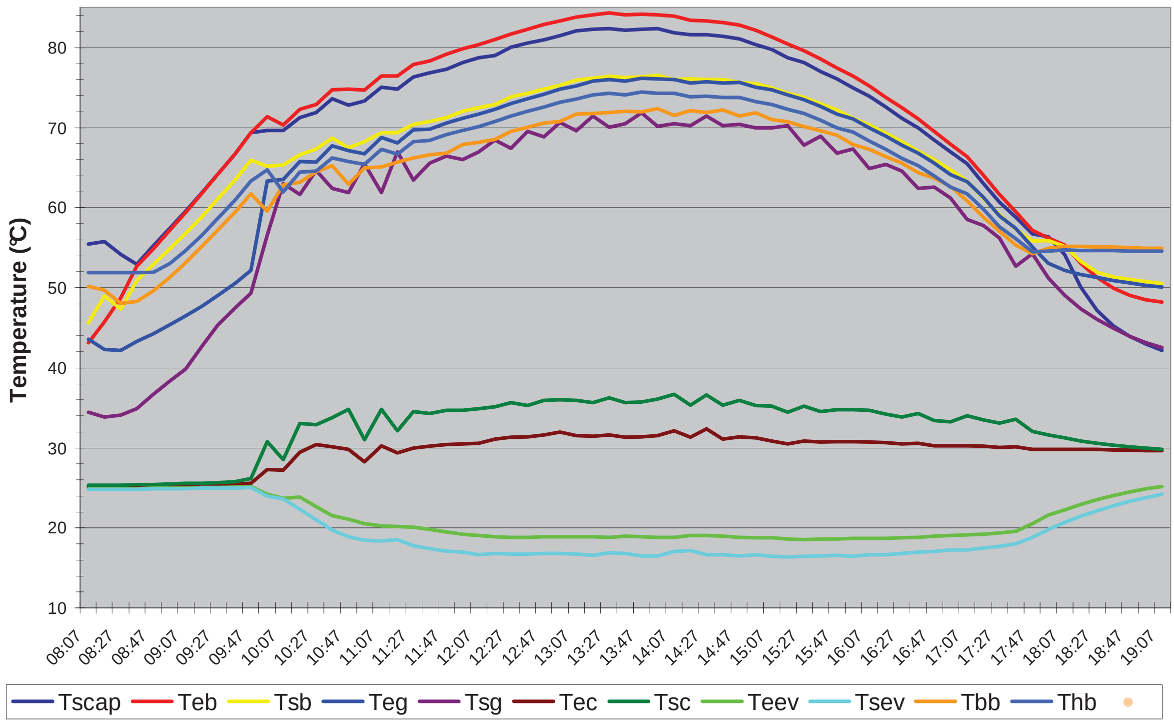
Measurement with a time scale of 1 and 10 minutes were automatically sent daily. It was then possible to calculate balances on a day long, a month long or a year long.

3.1 Daily analysis

3.1.1 <u>Sunny day for the cooling mode: 15 July 2008</u>

The 15th July 2008 was considered to evaluate the performances of the installation in cooling mode.

<u>Temperatures:</u>

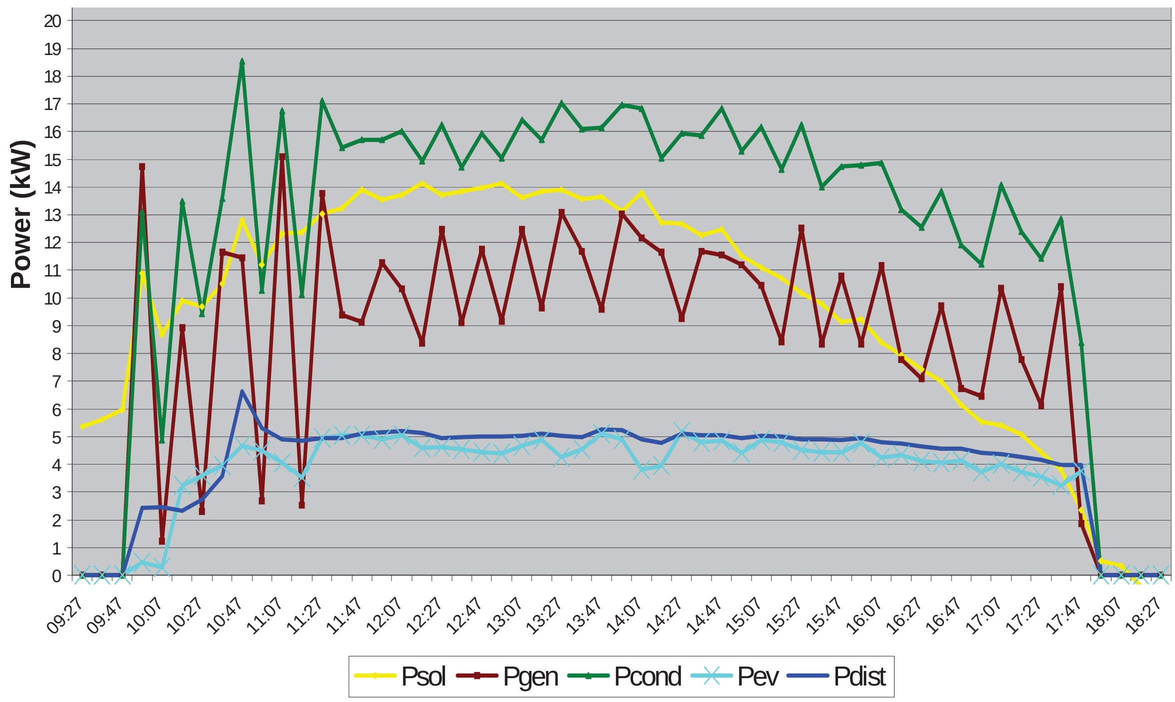


Temperatures for the day 15/07/2008

The behavior of the installation is very normal and correct for an adsorption technology. The temperatures into the solar collectors are about 80/75 °C at the maximum capacity. This level of temperature allows the chiller to produce chilled water at about 15 to 20 °C. And the heat is rejected via the drycooler at about 35 to 30°C.

Power:

The power at each circuit could be calculated ad are shown below:



Powers for the day 15/07/2008

The power in the solar loop depends of the solar radiation coming to the collector field, and reaches about 14 kW. When the installation is working, the power going to the chiller at the generator loop reaches about 11kW. Thanks to this power at the generator, the chiller is able to produce about 5 kW cold, so a value not far from its nominal capacity. The power rejected by the drycooler is thus about 16 kW.

Energy balance:

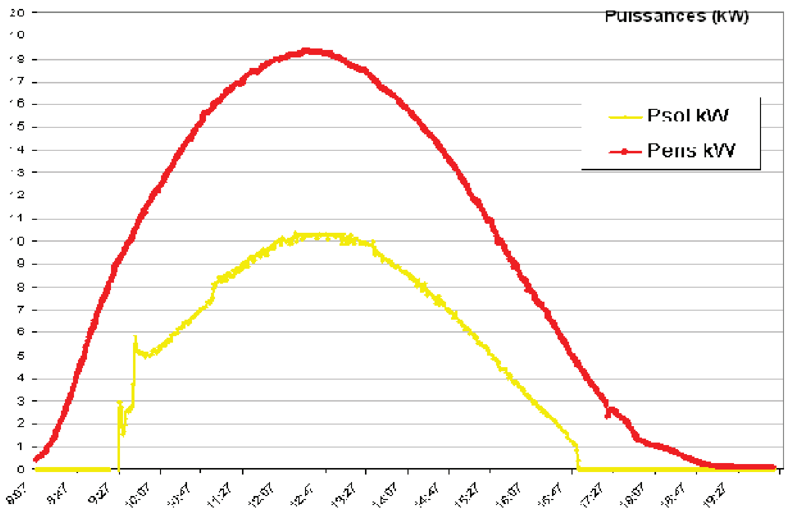
- The energies for the day 15/07/2008 are:
- Solar radiation: 260 kWh
- Energy collected by the solar collectors: 90 kWh
- Energy coming to the generator: 76 kWh
- Energy produced by the chiller at the evaporator: 33 kWh
- Energy rejected by the drycooler: 116 kWh
- Electricity consumption: 5,8 kWh
- Thermal COP = Eevap/Egen: 0.44
- Electrical COP = Eevap/electricity consumption: 5.8

Very good performances were thus monitored when the Perpignan installation was in cooling mode.

3.1.2 Sunny day for the heating mode: 15 July 2008

The switch to the heating mode was performed in October. The day studied here is the 15/10/2008.

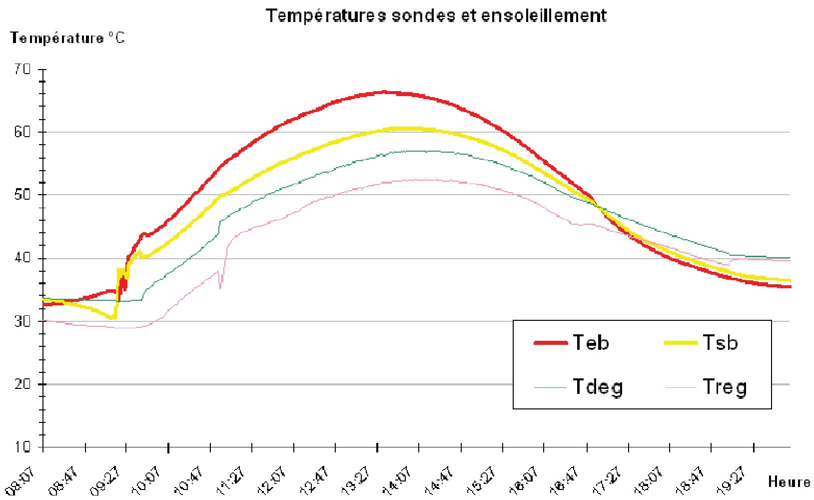
It was possible to evaluate the collector performances in heating mode. The solar radiation power and the power collected by the solar loop are showed on next chart:



Solar radiation and solar loop power

A daily average was calculated and leads to a collector field efficiency of 52%.

It was also possible to calculate the energies collected by the collectors, but also the energy produced and distributed to the building.

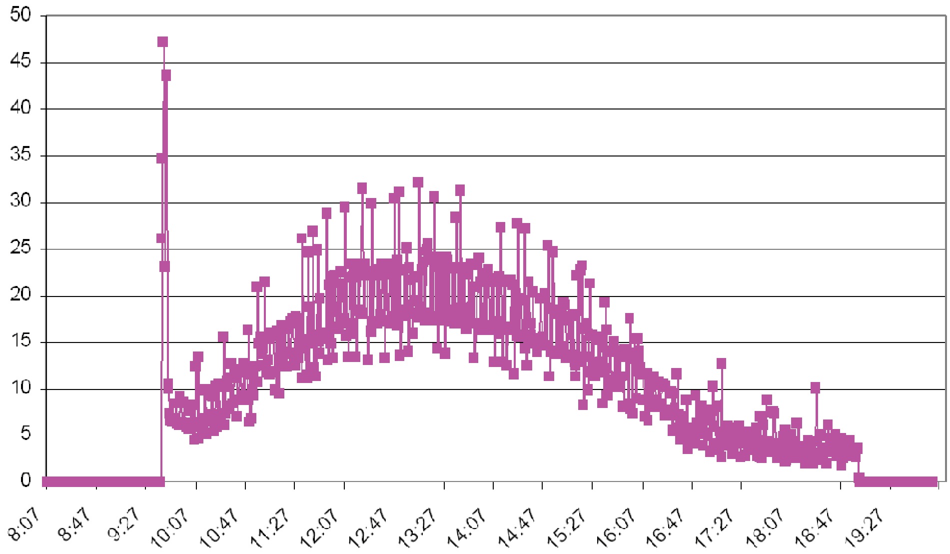


Inlet and outlet temperature of the collector and of the distribution

After calculation of the energy balance, the heating production is about 27 kWh.

The electrical COP was also calculated following the equation :

Electrical COP = heating energy produced / electrical energy consumed.



The average electrical COP for the day is 12.

3.2 Monthly and yearly analysis

The complete analysis of an entire working year (2008-2009) showed excellent results. It is noticeable that the overall electric efficiency of the system for one entire year reaches about 10. That means this solar cooling and heating system produces 10 times more energy (hot and cold) than it consumes electricity. As a consequence the system is far more efficient that the best heat pumps currently on the market.

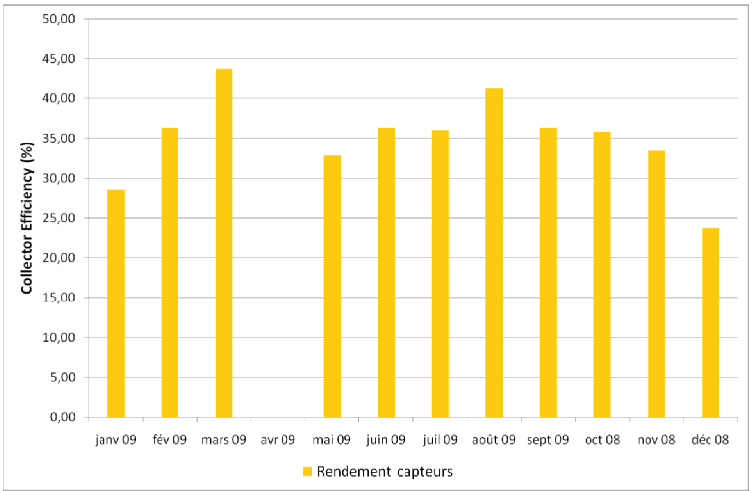
Bilan thermique :

	TOTAL	janv.-09	févr.-09	mars-09	avr.-09	mai-09	juin-09	juil.-09	août-09	sept.-09	oct.-08	nov.-08	déc.-08
Ensoleillement	28357,04	1765,67	2527,81	3253,03		2328,92	4414,36	4657,99	2679,11	1460,24	1902,27	1807,95	1559,67
Energie Solaire collectée	10186,57	504,57	918,08	1422,82		765,12	1603,17	1678,43	1107,34	530,28	682,08	605,06	369,64
Energie distribution	5573,79	436,21	838,38	1305,40		226,61	457,35	454,43	290,21	157,68	579,77	536,12	291,63
Rendement capteurs	35,92	28,58	36,32	43,74		32,85	36,32	36,03	41,33	36,31	35,86	33,47	23,70
COP thermique	0,33					0,35	0,33	0,31	0,29	0,35			

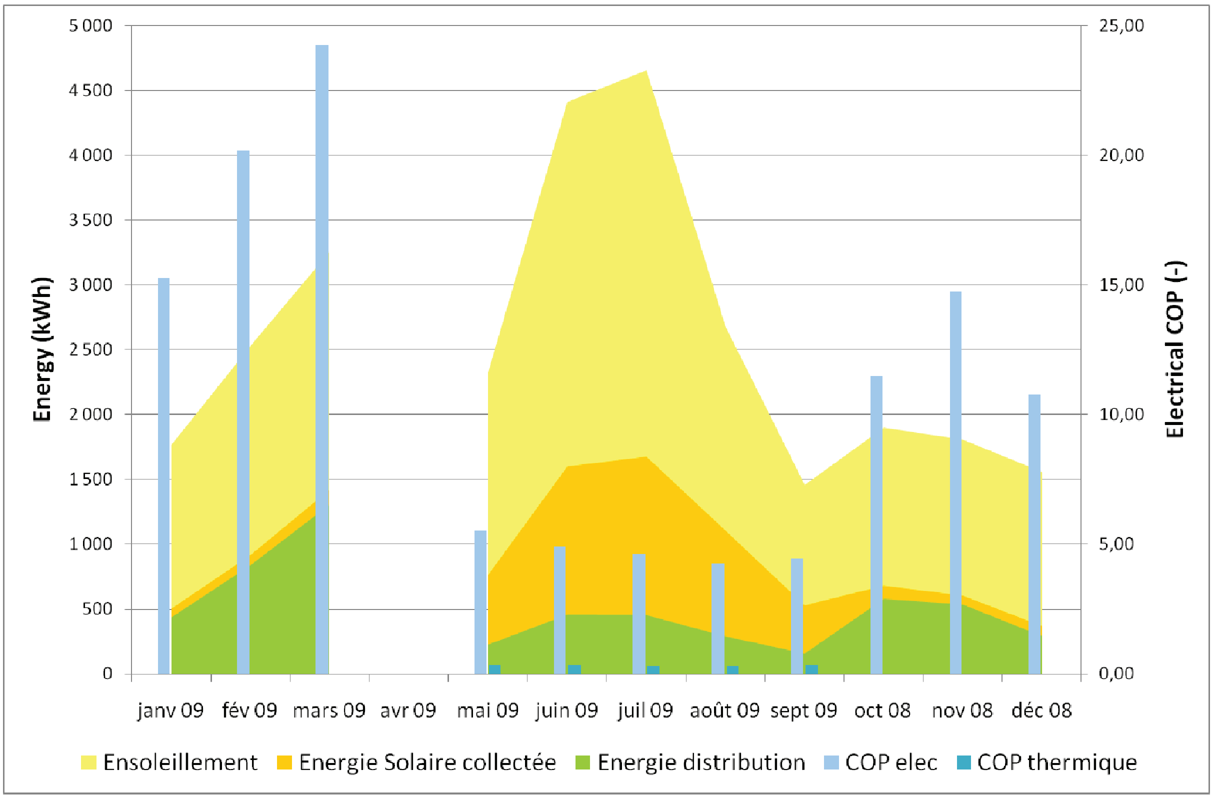
Bilan électrique :

	TOTAL	janv.-09	févr.-09	mars-09	avr.-09	mai-09	juin-09	juil.-09	août-09	sept.-09	oct.-08	nov.-08	déc.-08
COP elec	9,72	15,28	20,17	24,28		5,52	4,89	4,64	4,27	4,46	11,49	14,74	18,78
conso elec	615,97	31,07	45,18	58,48		43,51	98,93	104,66	72,70	37,70	54,83	39,53	29,37
conso sans distrib	573,47	28,59	41,57	53,80		41,02	93,48	97,90	67,93	35,35	50,44	36,37	27,02

Performances of the installation for the entire year October 2008 to September 2009



Efficiency of the solar collector field during the year 2008-2009



Graphic representation of the installation performances for the year 2008-2009

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 6

Monitoring Results of INES Research Center Offices, Chambéry (France)

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 30th November 2010

By Romain Siré[1] and Daniel Mugnier[1]

[1]Institution	TECSOL
Address	105 av. Alfred Kastler – Tecnosud
	66004 Perpignan CEDEX
	France
Phone	+33 4 68 68 16 40
Fax	+33 4 68 68 16 41
e-mail	romain@tecsol.fr

Contents

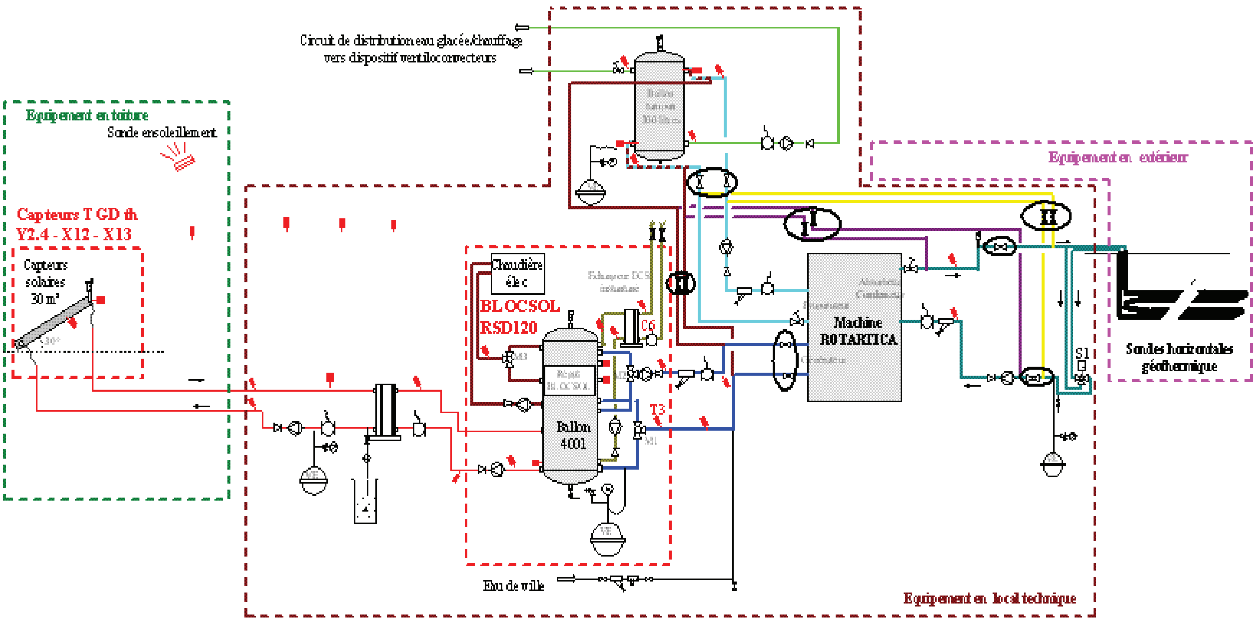
1 Introduction

The targeted building welcoming the solar cooling application is the PUMA3's INES research office. The INES (National Institute of Solar Energy) was created in 2006 by the public institutions to promote and develop solar technologies in France. To reach these objectives, the INES is divided into two centers: research, development and innovation on the one hand, and training and education on the other hand. The INES is located in the "Savoie Technolac" area which is very close to Chambéry in Rhônes-Alpes area (close to the French Alps and Lyon). The PUMA3 building is large, but only 3 mezzanine offices are cooled down by the solar system. The building was created recently so it has a good level of energy efficiency.

2 System design

The system is based on an absorption chiller of 4.5 kW coupled with 30 m² flat plate collectors. The installation is cooling the building in summer, heating in winter, and it also producing a small amount of domestic hot water all year long. The 400 liters heat storage tank is included in a packaged device supplied by Clipsol (SSC BlocSol RSD 120). Included in this device there is also an electric backup and all the security devices related to the heat storage. Another tank is installed in the system which is ensuring the cold storage in summer and a second heat storage in winter. The heat rejection system is done by a horizontal geothermal field.

<u>Installation's scheme:</u>



<u>Building:</u>

- Type of building: Office building
- Location: Chambery, France
- In operation since: April 2009
- System operated by: INES
- Air-conditioned area: 21m²
- System used for space cooling and heating (DHW production possible but not used)

System general properties:

- Technology: Closed cycle
- Nominal capacity: 4.5 kWcold
- Type of closed system: Absorption
- Brand of chiller unit: ROTARTICA
- Chilled water application: Fan coil
- Dehumidification: No
- Heat rejection system: Geothermal probes, exchange area about 138m²

Solar thermal:

- Collector type: Flat plate collectors
- Brand of collector: CLIPSOL
- Collector area: 30m²
- Tilt angle, orientation: 30°, 10°
- Collector fluid: Water glycol
- Typical operation temperature: 80°C

Configuration:

- Heat storage: 0.4 m3 water (part of:SSC BlocSol RSD 120 CLIPSOL)
- Cold storage: 0.3 m3 water
- Use of auxiliary heating system: Hot backup (heating the heat storage tank)

Photos:

Technical premises

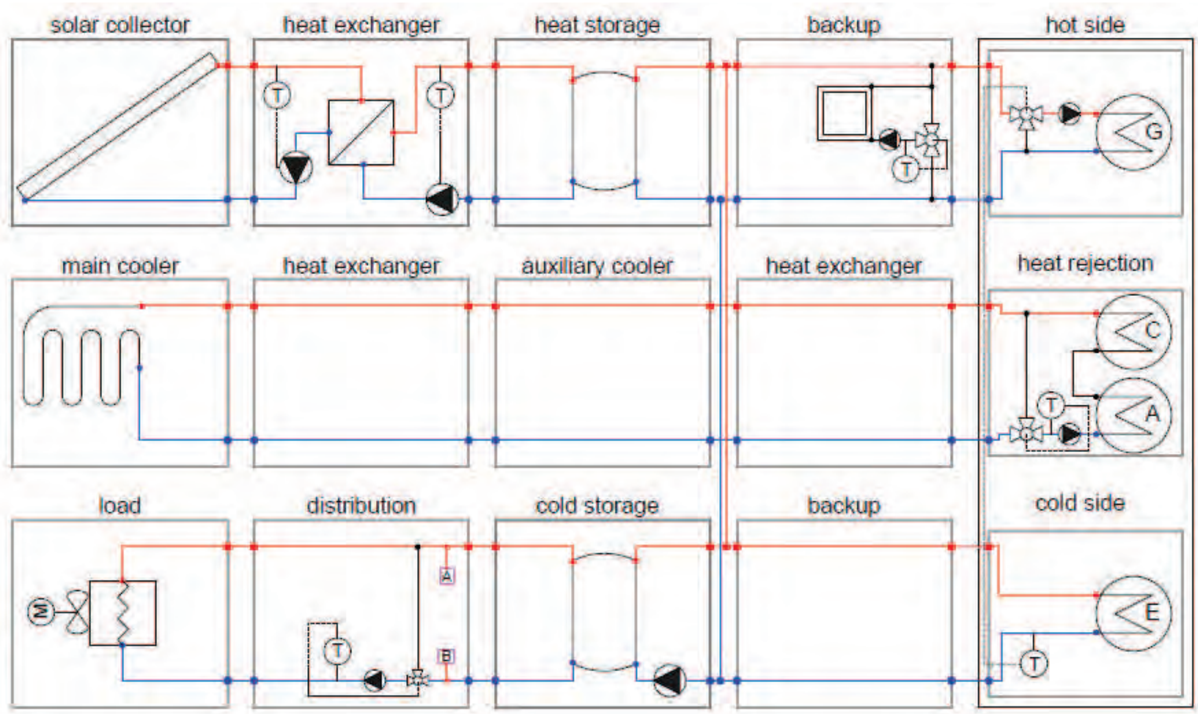
Collector field

Geothermal field

Distribution system

<u>Task 38 standard Scheme for the installation:</u>

If the standard Task 38 scheme is used to describe the installation, the scheme is as follows:



3 Performance factor used for the Chambery installation

3.1 Background

Within IEA Task 38 a universal monitoring procedure for solar heating and cooling systems was developed. This monitoring procedure can be conducted in three different levels of detail:

1. First level: Basic Information on Primary Energy Ratio and Costs
2. Second level: Simple analysis of the solar energy source management
3. Third level: Advanced monitoring procedure

The IEA Task 38 also developed a standard basic scheme adjustable for the large majority of systems (See **Fehler! Verweisquelle konnte nicht gefunden werden.**). In this scheme, the energy flows of a solar heating and cooling system are nominated and shown.

Abbreviations:

SH Space Heating

DHW Domestic Hot Water

SC Space Cooling

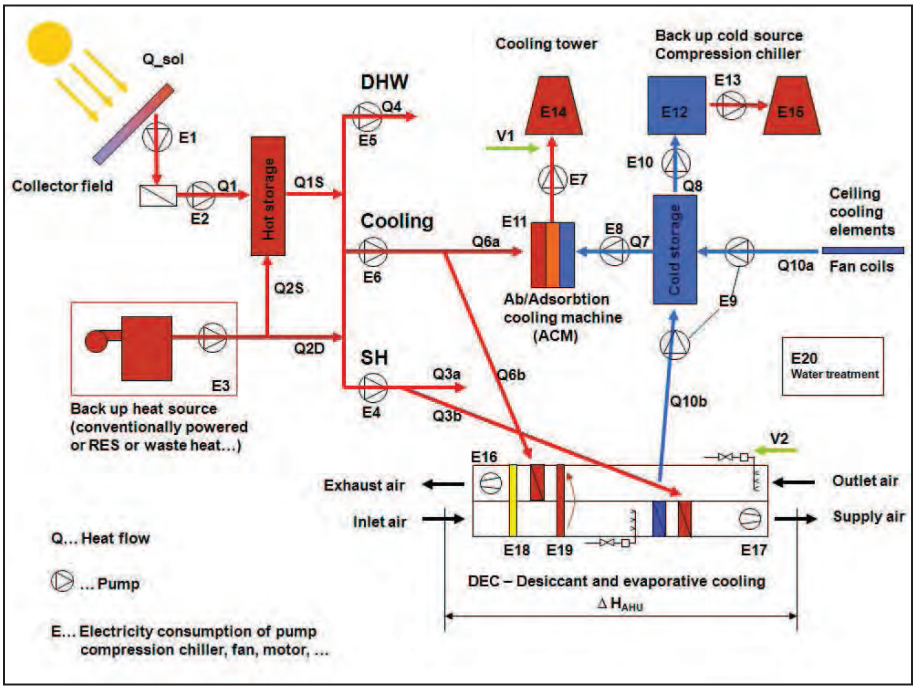
RES Heat recovery from cogeneration unit or biomass...

AHU Air Handling Unit

εelec Primary energy factor for electricity production (kWhel/kWhprim-fossil)

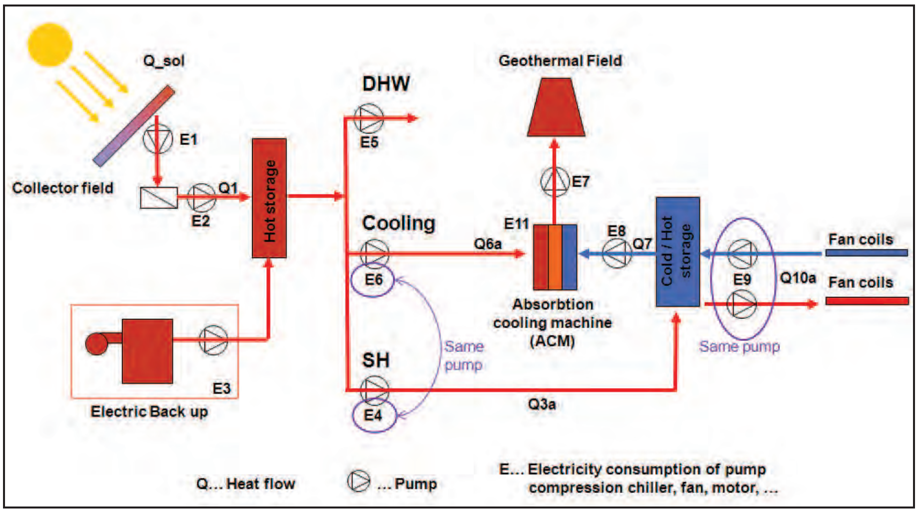
εfossil Primary energy factor for fossil fuel (kWhfinal/kWhprim-fossil)

εRES Primary energy factor for renewable sources (kWhfinal/kWhprim-fossil)



Task 38 base SAC maximum system scheme

This standard scheme was modified to match the characteristics of the Chambery installation (See **Fehler! Verweisquelle konnte nicht gefunden werden.**)



Task 38 scheme adapted the the Chambery installation

Several performance factor were selected to characterize the installation performances:

3.2 Collector efficiency

Basically the collector efficiency characterise the performances of the solar collector field.

It is calculated as follows:

$$\eta_{coll} = \frac{Q_{sol}}{G}$$

With:

η_{coll} Collector efficiency (-)

Q_{sol} Solar energy collected in the solar loop (kWh)

G Solar radiation (kWh)

This performance factor is calculated for a certain period. Usually it is calculated for one day, or for one month.

Indirectly, the collector efficiency permits also to characterize the overall performances of the installation. Indeed, if the installation is turned off, because of a technical issue, or because there is no charge, the energy collected is equals to zero while the solar radiation is still there. So the collector efficiency value is low.

3.3 Thermal COP

The thermal coefficient of performance (thermal COP) characterizes the internal performances of the chiller. This performance factor is calculated only in summer (cooling mode of the installation) as follows:

$$thermal\ COP = \frac{Q_{evap}}{Q_{gen}}$$

With:

Thermal COP Thermal coefficient of performance (-)

Q_{evap} Energy taken from the evaporator loop (kWh)

Q_{gen} Energy coming from the generator loop (kWh)

Usually adsorption chillers have a thermal COP around 0.4 to 0.5 and absorption chillers have a thermal COP around 0.6 to 0.7.

3.4 Electrical COP

The electrical coefficient of performance (electrical COP) characterizes the electric performances of the entire solar installation. This performance factor is calculated all year long as follows:

$$electrial\ COP = \frac{Q_{distrib}}{E_{tot}}$$

With:

Electrical COP Electrical coefficient of performance (-)

$Q_{distrib}$ Energy (hot or cold) given to the distribution loop (kWh)

E_{tot} Electrical energy consumed by the solar system (except backups) (kWh)

This ratio is used to compare the performances of the installation to the performances of a conventional system such as heat pumps.

3.5 Primary Energy Ratio (PER)

This performance factor characterizes the overall efficiency of the installation in terms of primary energy. It is the ratio between all the energies given to the user (heating, cooling, and DHW) and all the primary energies consumed by the system (including backups).

The more this performance factor is high, the more the solar fraction is high and/or the overall efficiency of the installation is high.

$$\boldsymbol{PER} = \frac{\boldsymbol{Q}10 + \boldsymbol{Q}3 + \boldsymbol{Q}4}{\boldsymbol{E}_{\boldsymbol{syst\grave{e}me}} \cdot \quad_{\boldsymbol{\acute{e}lec}.} + \dfrac{\boldsymbol{Q}2}{\boldsymbol{Rg}} \cdot \quad_{\boldsymbol{x}} + \dfrac{\boldsymbol{Q}8}{\boldsymbol{EER}} \cdot \quad_{\boldsymbol{x}}}$$

See the SOLERA deliverable No. D 6.1, or the Task38 deliverables for more explanation about the meanings of the different terms of the equation.

3.6 Fractional solar heating and cooling savings

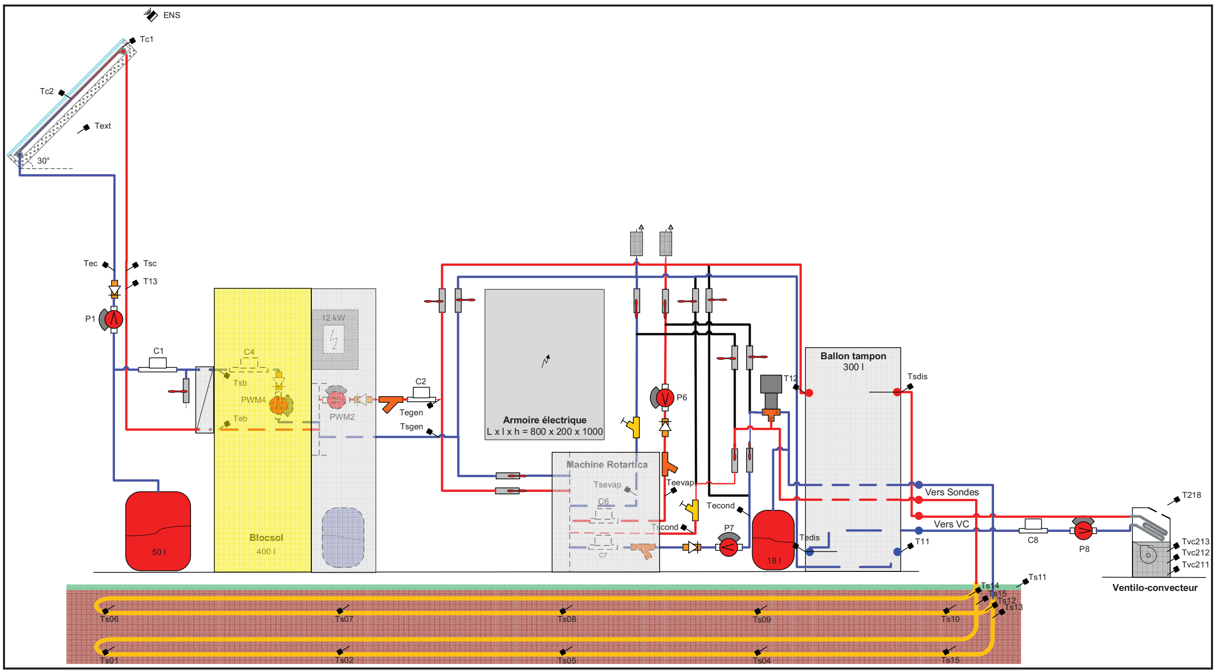
The global evaluation of the whole system was performed by using the fractional solar heating and cooling savings fsav,SHC as actually defined in the IEA-SHC Task 38. It describes the fraction of energy savings of the solar system compared to a conventional system that provides the same service for heating, cooling and domestic hot water demand. It is calculated as follows:

$$f_{sav.shc} = 1 - \frac{\dfrac{Q_{boiler}}{\varepsilon_{fossil} \cdot \eta_{boiler}} + \dfrac{Q_{RES}}{\varepsilon_{RES} \cdot \eta_{RES}} + \dfrac{E_{el}}{\varepsilon_{elec}} + \dfrac{Q_{cooling.missed}}{SPF \cdot \varepsilon_{elec}}}{\dfrac{Q_{boiler.ref}}{\varepsilon_{fossil} \cdot \eta_{boiler.ref}} + \dfrac{E_{el.ref}}{\varepsilon_{elec}} + \dfrac{Q_{cooling.ref}}{SPF \cdot \varepsilon_{elec}}}$$

See the SOLERA deliverable No. D 6.1 for more explanation about the meanings of the different terms of the equation.

4 Monitoring scheme and list of measurements

In order to control the system, to obtain good monitoring data, and to be able to calculate the performance factors described above, the following sensors are used:



Position of the monitoring equipment

The sensors used are listed below:

Table 1: list of the Chambery installation's sensors

Type	***Locations***	***Names***
Temperature sensor	on the collector field	Tc1 & Tc2
Temperature sensor	in the primary solar loop	Tec, Tsc & T13 (Blocsol)
Temperature sensor	in the secondary solar loop	Tsb, Teb & 1 inside the Blocsol
Temperature sensor	at the top and bottom of the hot water storage tank	inside the Blocsol
Temperature sensor	in the generator loop	Teg, Tsg & 2 inside the Blocsol
Temperature sensor	for the electric back up	inside the Blocsol
Temperature sensor	in each branch of the condenser loop	Tecond & Tscond
Temperature sensor	in each branch of the evaporator loop	Teevap & Tsevap
Temperature sensor	At the top and bottom of the cold water storage tank	T11 & T10 (Blocsol);

Temperature sensor	in each circuit of the distribution loop	Tsdis & Tedis
Temperature sensor	in the sanitary hot water loop	inside the Blocsol
Temperature sensor	ambient temperature sensor	Text
Temperature sensor	in each office heated and cooled	Tvc211 → Tvc213
Temperature sensor	in the office located in the middle	T212
Temperature sensor	in the ground next to the geothermal probes	Ts1 → Ts11
Temperature sensor	on each loop of the two geothermal probes	Ts12 → Ts15
Flow meter	in the primary solar loop, on the inlet pipe of the pump	C1
Flow meter	in the secondary solar loop, on the inlet pipe of the exchanger	C4
Flow meter	in the generator loop on the outlet pipe of the pump	C2
Flow meter	in the condenser loop, on the inlet pipe of the absorption machine	C7
Flow meter	in the evaporator loop, on the inlet pipe of the machine	C6
Flow meter	in the distribution loop, on the inlet pipe of the pump	C8
Irradiation sensor	On the roof tilt & orientation like the collectors	ENS

See the SOLERA deliverables for more information about the monitoring itself.

5 Analyse of the monitoring data

5.1 Collector efficiency

The collector efficiency can be calculated and followed every two minutes thanks to an efficient monitoring. It is then possible to draw the collector efficiency during one day.

The Figure 1 shows the collector efficiency of the Chambery installation on the 6th June 2010.

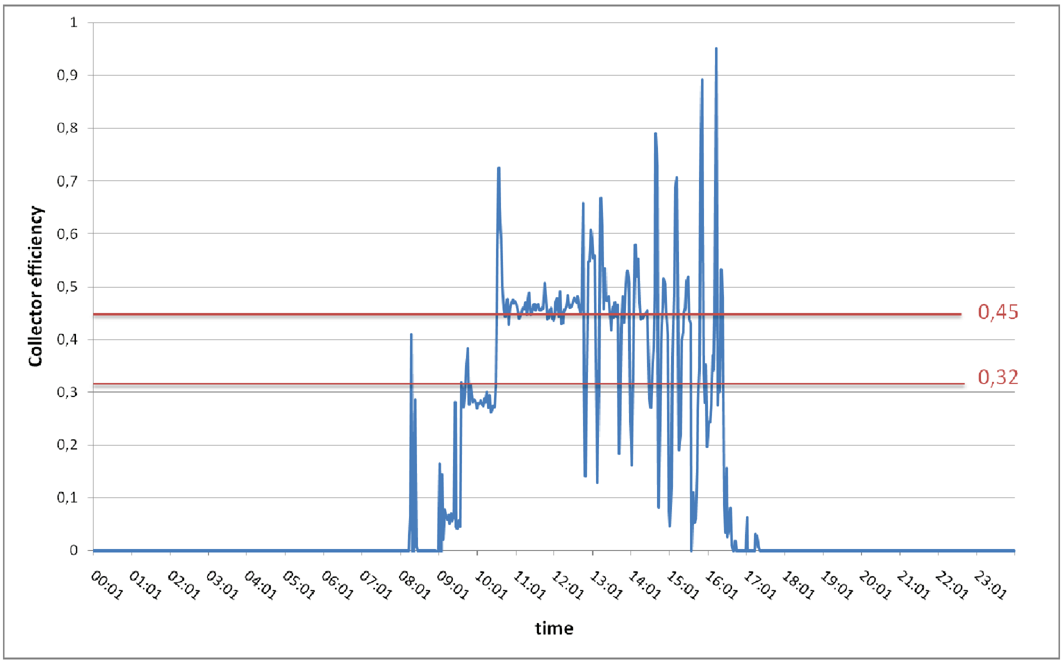


Figure 1: Collector efficiency of the Chamberry installation on 06/06/2010

During this day the solar collector field is working properly as the installation is running in stationary state between 10:30am and 4:30pm, the collector efficiency is around 0,45 which is a correct value with regard to the expected performances.

However, when the calculation is performed during the whole day, solar irradiation is 184,13kWh while solar energy received is 59,77kWh: the whole day collector efficiency is then 0,32.

This difference between the collector efficiency function of the time of observation can be explained because in the morning and in the evening, when the solar radiation is low, solar collectors can't work at they full potential. Anyway, a collector efficiency value of 0,32 for the entire day is still a correct value.

The calculation for one month follows the same principles.

For the period between 01/05/2009 and 31/10/2010, the monthly collector efficiency is showed on Table 2.

Table 2: Collector efficiency of the Chambery installation during the observation period

May 2009	Jun 2009	Jul 2009	Aug 2009	Sep 2009	Oct 2009	Nov 2009	Dec 2009	Jan 2010	Feb 2010	Mar 2010	Apr 2010	May 2010	Jun 2010	Jul 2010
23,2 %	**27,7 %**	**30,8 %**	**31,2 %**	**20,3 %**	**21,2 %**	**11,9 %**	**9,7%**	**10,0 %**	**17,9 %**	**22,9 %**	**24,5 %**	**26,2 %**	**23,7 %**	**26,1 %**

The collector efficiency is good for flat plate collectors in the summer and for the sunny months. However in winter the efficiency goes low. It can be explained by the large amount of days in winter that are not sunny enough to start the solar system, or not sunny enough to keep the system working. However, for those days a solar radiation is still there even if it's not very high. As a consequence, the average collector efficiency is not very high.

Indeed, even in December, a few days appears with good collector efficiency, like on 08/12/2009 (see Figure 2) where the collector efficiency reaches 0,4. These days are just not so numerous than in summer.

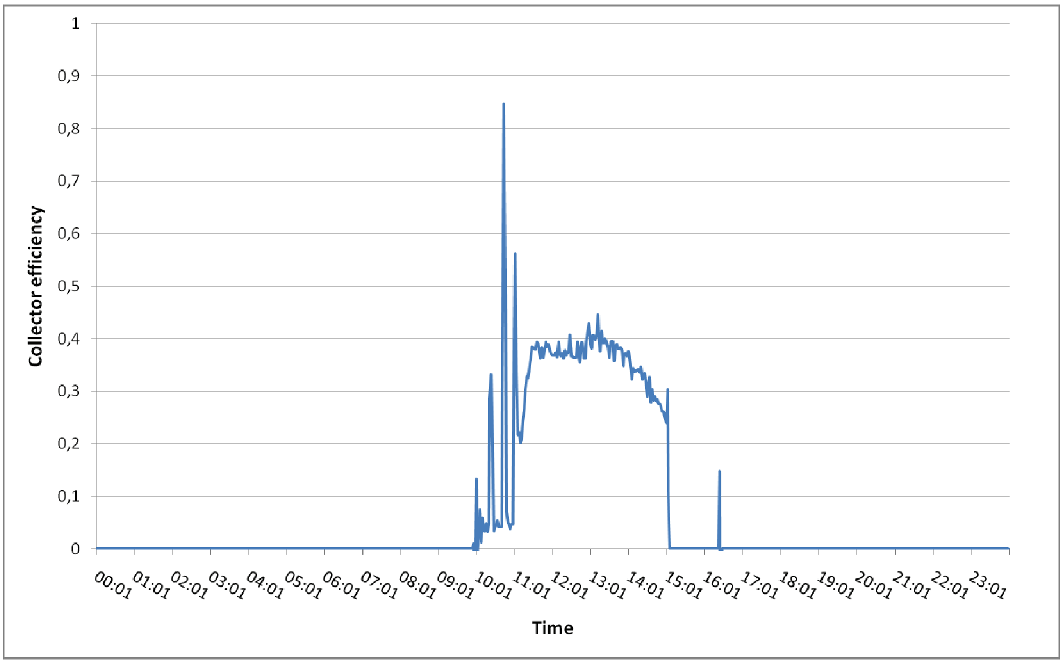


Figure 2: Collector efficiency of the Chamberry installation on 08/12/2009

5.2 Thermal COP

The Thermal COP can be calculated and followed every two minutes thanks to an efficient monitoring. It is then possible to draw the thermal COP during one day.

The Figure 3 shows the thermal COP of the Chambery installation on the 14th July 2009.

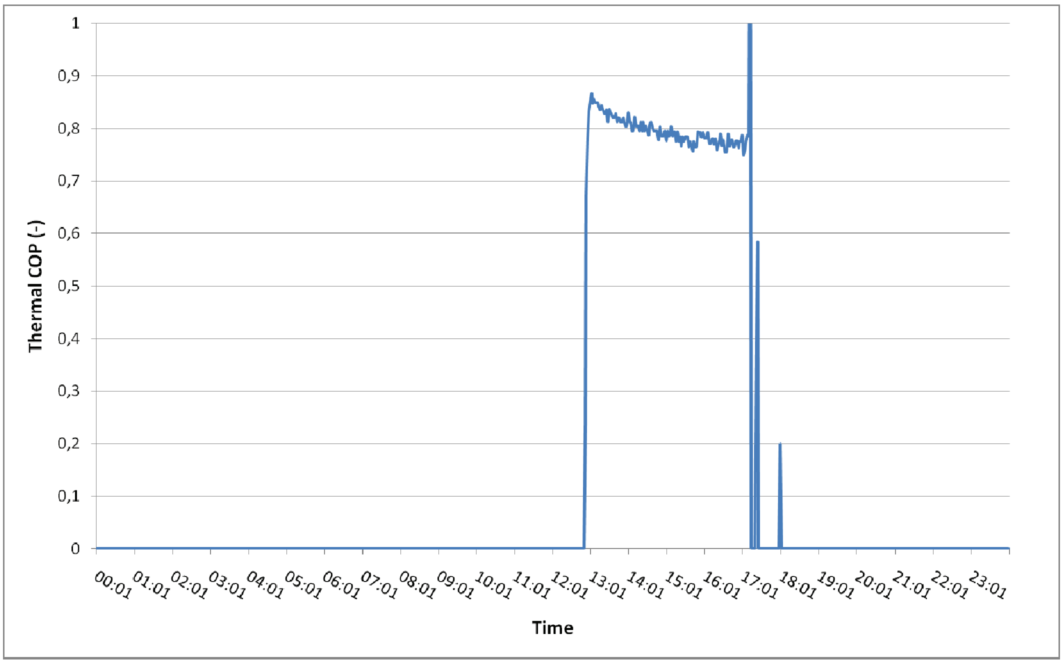


Figure 3: Thermal COP of the Chamberry installation on 14/07/2009

Looking at this day the Rotartica chiller exhibited an excellent (high and stable) coefficient of performance.

The chiller started on 14/07/2009 at about 12:50 and stopped in the evening around 5:10pm. During this time as the installation was producing cold, the internal thermal COP of the chiller was about 0,8 which is an excellent value even for an absorption chiller.

If the overall working period is studied, the thermal COP can be calculated month by month, and are showed on Table 3and on Figure 4:

Table 3: Thermal COP of the Chambery installation during the working period

May	Jun	Jul	Aug	Sep	Oct		May	Jun	Jul
2009	**2009**	**2009**	**2009**	**2009**	**2009**		**2010**	**2010**	**2010**
0,68	**0,68**	**0,73**	**0,70**	**0,73**	**0,75**		**0,75**	**0,72**	**0,72**

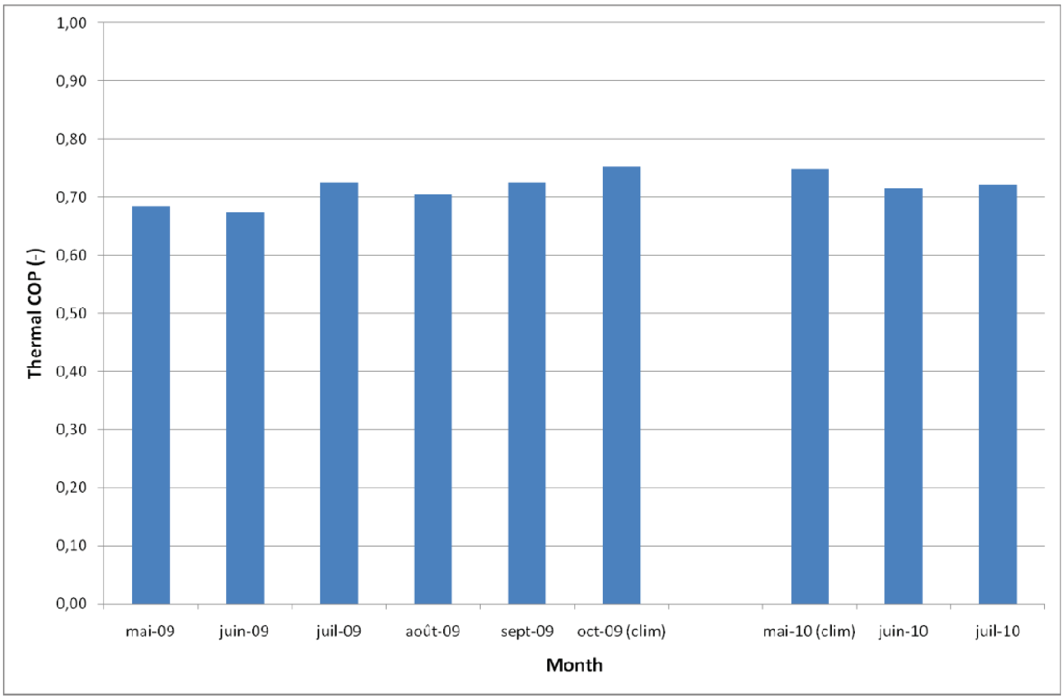


Figure 4: Thermal COP of the Chambery installation during the working period

So, during the cooling period, the thermal COP of the Rotartica chiller is quite excellent. Indeed, thermal COP reaches at least a value of 0,7 almost every month. Besides, this value is quite stable for the whole cooling period.

5.3 Electrical COP

The Electrical COP can be calculated month by month for the working period, and is showed in Table 4 and in Figure 5.

Table 4: Electrical COP of the Chambery installation during the working period

May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Jun	Jul
2009	**2009**	**2009**	**2009**	**2009**	**2009**	**2009**	**2009**	**2010**	**2010**	**2010**	**2010**	**2010**	**2010**	**2010**
1,94	**2,60**	**3,20**	**3,53**	**3,60**	**0,65**	**3,72**	**2,03**	**0**	**4,55**	**9,31**	**10,6**	**3,55**	**3,71**	**3,62**

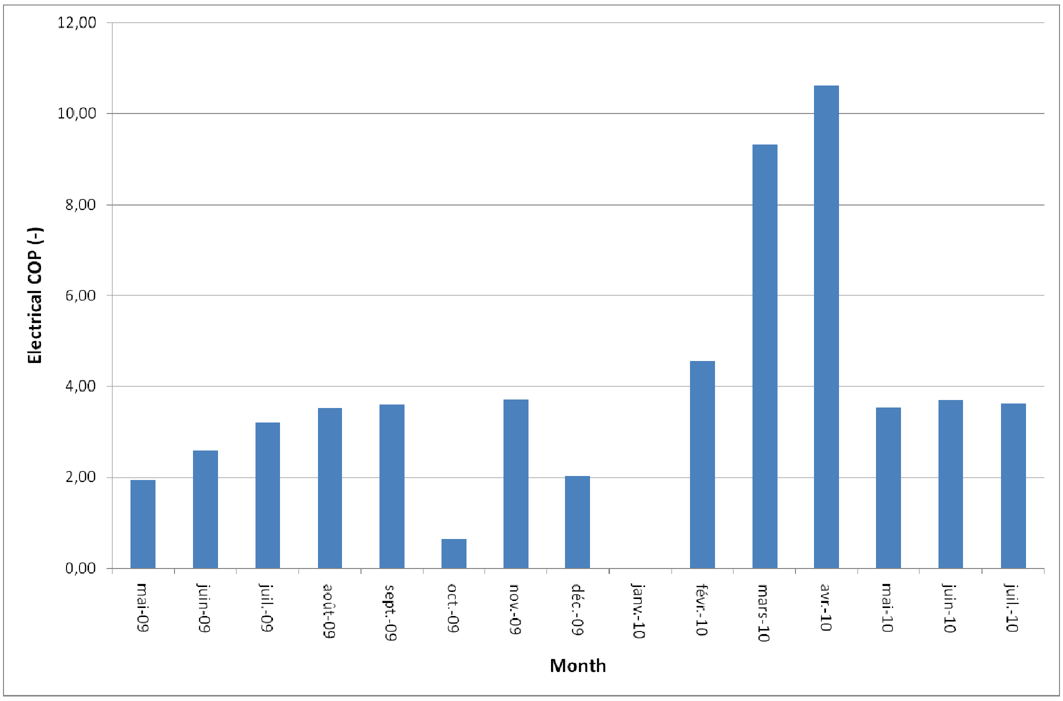


Figure 5: Electrical COP of the Chambery installation during the working period

Even with some very good month (March and April 2010), it appears that the overall average electrical COP for the working period is 3,14, which is not an extremely good result (even if it could have been worse). Indeed the objective of a solar cooling and heating installation is usually to reach an overall electrical COP of 5. In fact, an electrical COP around 3 could be almost reached by a conventional heat pump and the objective of a solar installation is to be far better than a conventional solution.

However, this low electrical COP can be at least partly explained by the design of the installation. Indeed, the installation, because of its "experimental state" has numerous devices consuming electricity. The number of pumps is high, the Clipsol's Blocsol device consumes electricity too, as well as the Rotartica which consumes more electricity than an "average" absorption chiller, and finally a not negligible part of the electricity consumption is due to the metrology devices (sensors more numerous than in a "basic" solar cooling installation).

5.4 Primary Energy Ratio (PER)

The overall efficiency of the installation in terms of primary energy was calculated using the PER (Performance Energy Ratio). It is the ratio between all the energies given to the user (heating, cooling, and DHW) and all the primary energies consumed by the system (including backups). This value is calculated monthly for the working period, and the results are shown in Table 5.

Table 5: Monthly Primary Energy Ratio for the Chambery installation

May **2009**	Jun **2009**	Jul **2009**	Aug **2009**	Sep **2009**	Oct **2009**	Nov **2009**	Dec **2009**
0,68	**0,94**	**1,14**	**1,29**	**1,18**	**0,38**	**-**	**0,73**

Jan 2010	Feb 2010	Mar 2010	Apr 2010	May 2010	Jun 2010	Jul 2010
0,77	**0,90**	**2,05**	**3,68**	**1,22**	**1,45**	**1,43**

Again the results are strongly related to the studied month. The Primary energy ratio can reach 3,68 in April 2010. But they also can be lower than 1.

An average based on the year from July 2009 to June 2010 was calculated, and it leads to a PER equals to 1,10. Once again, even if this value is higher than 1, it is not very high.

In addition to the large amount of electricity consumed by the different electrical devices (as explained in the previous paragraph 5.3), the backup is an electrical one. And the amount of primary energy to create one kWh of electrical energy is very high, as a consequence, the PER is lower than in the case where another type of backup is used.

5.5 Fractional solar heating and cooling savings

The global evaluation of the whole system was performed by using the fractional solar heating and cooling savings fsav,SHC performance criteria. It calculates the ratio of the fraction of energy savings of the solar system to energy consumption of a conventional system that fulfils the same demand for heating, cooling and domestic hot water. The calculation was achieved by considering the conventional system as using fossil electricity. This value is calculated monthly for the working period, and the results are shown in Table 6.

Table 6: Mothly fractional solar heating and cooling savings for the Chambery installation

May 2009	Jun 2009	Jul 2009	Aug 2009	Sep 2009	Oct 2009	Nov 2009	Dec 2009
-63,74%	**-19,18%**	**2,14%**	**13,15%**	**4,73%**	**-195,8%**		**-12,99%**

Jan 2010	Feb 2010	Mar 2010	Apr 2010	May 2010	Jun 2010	Jul 2010
-5,88%	**9,12%**	**60,08%**	**77,70%**	**10,11%**	**22,83%**	**21,51%**

The results are closely related to the studied month. The fractional solar heating and cooling savings can be high up to 77,7% (for April 2010), but they can also be negative.

An average value based on the year from July 2009 to June 2010 was calculated, and it leads to a fractional solar heating and cooling savings equals to 14,80%. Even if this value is positive, it is not very high, and the explanations are the same than in the previous paragraph (5.4): because of the "experimental state" of the system, a lot of devices consuming electricity are part of the installation design, and in addition an electrical boiler is used in the hot tank which is disadvantageous when the calculations are based on primary energy.

6 Conclusion

Then, thanks to a precise monitoring it was possible to obtain the necessary values to calculate these performance factors with the real experimental data.

After the calculation and analysis of this set of performance factors, the performance of the overall installation has been evaluated as well as the performance of some specific element of the system.

The main conclusions obtained regarding the performances analysis carried on are:

- ✓ Excellent performances of the Rotartica chiller.
- ✓ Good behaviour of the geothermal horizontal probes.
- ✓ Good performances of the collector field considering they are flat plate collectors, but a lot of days are not sunny enough to start the solar system.
- ✓ Difficulties to reach a high electrical COP due to the “experimental state” of the installation: explained by the presence of numerous pumps, of devices consuming more electricity than usually (hot tank, chiller), and of a large set of sensors (more numerous than in a “basic” solar cooling installation). The same conclusions are also valid for the Primary Energy Ratio and for the fractional solar heating and cooling savings.

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 7

Monitoring Results of Spain: Gymnasium of the University of Zaragoza, Zaragoza

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date:

By Fernando Palacín[1] and Carlos Monné[2]

[1]Institution	**National Renewable Energy Centre (CENER)**
Address	**Ciudad de la Innovación 7, Sarriguren, Navarra (Spain)**
Phone	**+34 948 252 800**
Fax	**+34 948 270 774**
e-mail	**fpalacin@cener.com**

[2]Institution Group of Thermal Engineering and Energy Systems (GITSE), University of Zaragoza (Spain)

Address	**C/María de Luna s/n, 50018, Zaragoza, Spain.**
Phone	**+34 976 762 042**
Fax	**+34 976 762 616**
e-mail	**cmmb@unizar.es**

Contents

1 Background

The installation is located in Zaragoza (Spain) at the indoor sports centre of the University of Zaragoza and it is used to cool a gymnasium. This installation was designed as a consequence of the overheating in the existing solar collectors. In summer, the solar field was oversized because solar power was higher than needed.

2 System Design

- System scheme

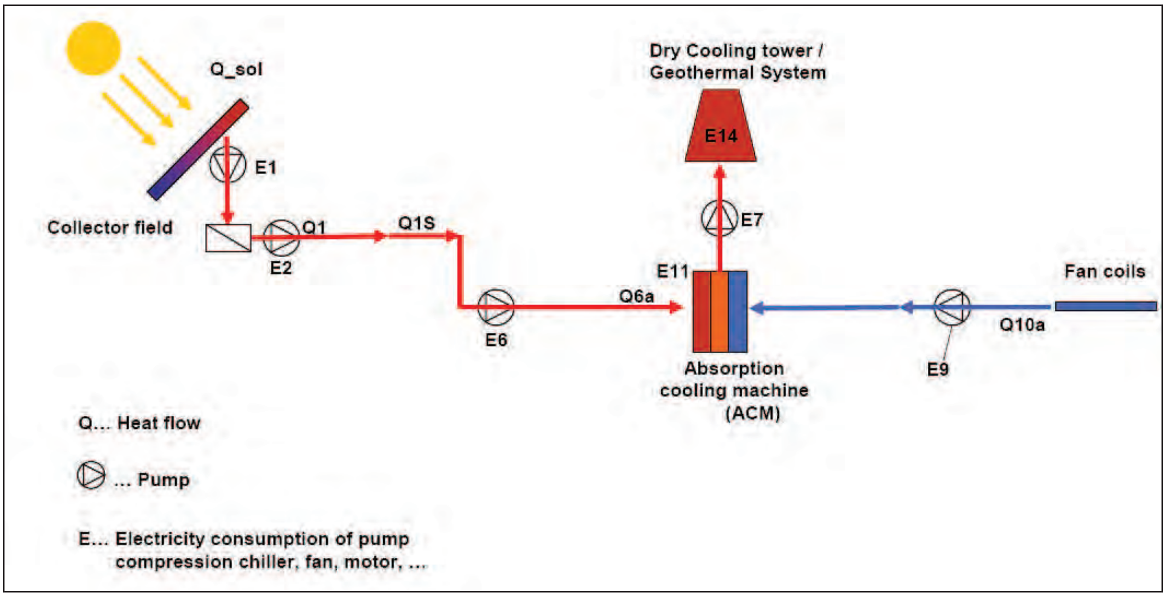


Fig. 1. – Solar Cooling System scheme

- Reasons for designing the system the way it has been designed

This solar cooling installation was designed as a consequence of the overheating problems of the existing solar filed used to contribute to the domestic hot water supply of the building. In the summer, to solve this problem and to use this solar waste energy, the chosen solution was the installation of an absorption chiller. Therefore, the solar collector field of the solar air-conditioning system has 37.5 m^2 of useful area. Problems encountered

- Photo documentation

		
Solar field	Machinery room	Gymnasium

		
Fan coil	Initial heat rejection system	Geothermal heat rejection system (2009)

3 Control Strategy

The solar pump of the system starts to work when the temperature of the solar field is around 86ºC, in this moment the water circulates in the primary loop. The secondary pump, which supplies thermal energy to the absorption chiller, is activated when the solar heat exchange temperature on this secondary side is over 80ºC.

The monitoring system is configured so that the absorption machine is turned on when the inlet temperature of the generator is 80ºC.

It has to be noticed that the installation contains a hot water storage tank and an auxiliary boiler as the back-up system. The aim of the hot water tank is to store the solar energy in those periods without cooling demand. And the mission of the gas boilers is to supply thermal energy to the absorption chiller when the solar energy isn't enough to produce chilled water. However, both devices have never been used. On the one hand the gymnasium always demands cooling demand when there is enough solar energy to activate the chiller, so it isn't necessary a buffer to store the heat water. On the other hand the installation is focused on the chiller performance when the chiller is only solar powered.

4 Monitoring Equipment

4.1 Installed Equipment

The installation is completely monitored. The monitoring system was designed perform the energy balances of the different components of the installation. There are a PLC unit and a web controller, which form the controlling and recording system. In this way, the outdoor and indoor conditions of the installation are well defined.

The monitoring system consists mainly of two temperature probes (near to the absorber and the condenser of the absorption machine) which measure the inlet and outlet temperature of the flow between the absorption chiller and the dry cooling tower (finned tube heat exchanger). Two temperature probes measure the inlet and outlet temperature of the flow between the absorption chiller and the fan coils (the measure is taken near the evaporator in the absorption machine).

Two temperature probes measure the inlet and outlet temperature of the flow between the absorption chiller and the heat exchanger (the measure is taken near to the generator in the absorption machine). While the temperature values for the exterior temperature are measured with NTC sensors, the ones located around the absorption chiller are registered with PTC sensors.

Furthermore, there is a flow meter in each one of the circuits of the chiller; the water flow that goes to the generator, the water flow that goes to the fan coils and the water flow that goes to the finned tube heat exchanger.

In figure 2 it can be seen the location of probe of the monitoring system.

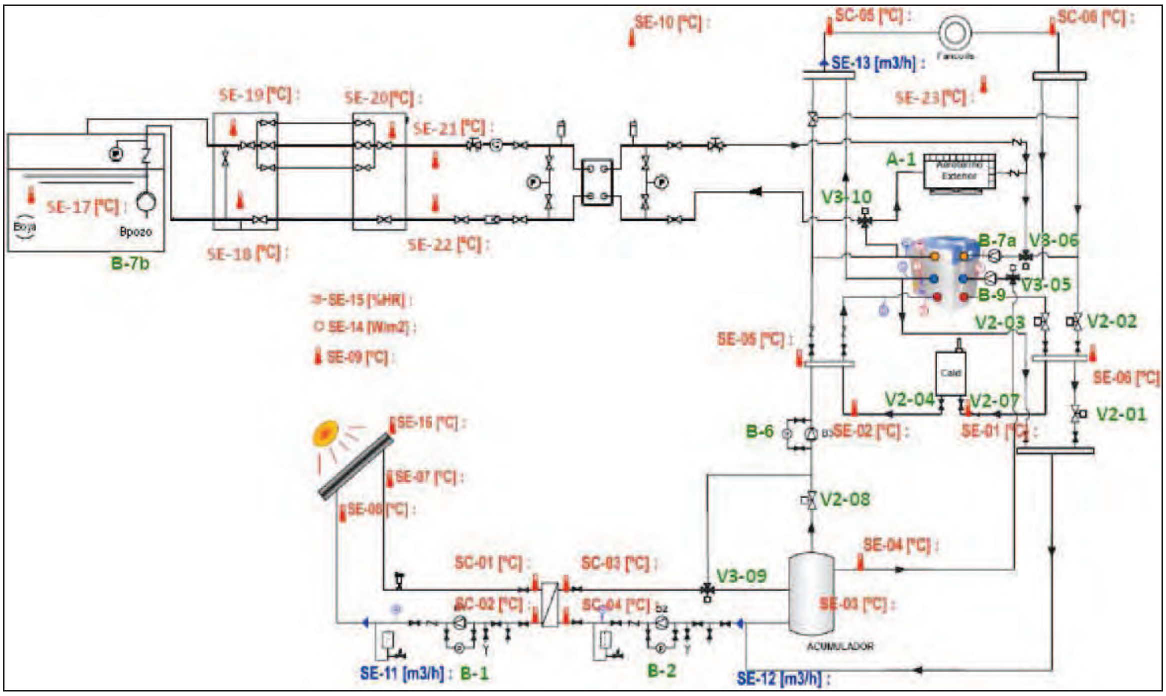


Fig. 2. – Monitoring Equipment

In Table 1 the features of the main sensors of the monitoring system are shown.

DEVICE	MEASUREMENT	UNITS	COMMERCIAL MODEL	RANGE	ACCURANCY
SC-01	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-02	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-03	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-04	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-05	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-06	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+1 80ºC	Máx. ±1,5%
SE-01	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-02	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-03	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-04	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-05	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-06	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-07	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-08	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-09	Temperature	℃	SIEMENS QFA 3160	0ºC;+50ºC	±0,8ºC (from 15ºC to 30ºC); ±1ºC (0ºC a 15ºC; 30ºC a 50ºC)
SE-10	Temperature	℃	SIEMENS QAC22	-35ºC;+50ºC	±0,05ºC in 0ºC (DIN 43760)
SE-11	Flowmeter	m^3/h	SIEMENS Ultraheat 2WR5	15l/h-3m^3/h	Máx. ±4%
SE-12	Flowmeter	m^3/h	SIEMENS Ultraheat 2WR5	15l/h-3m^3/h	Máx. 4%.
SE-13	Flowmeter	m^3/h	SIEMENS Ultraheat 2WR5	15l/h-3m^3/h	Máx. ± 4%
SE-14	Solar Radiation	W/m^2	QUIMISUR IQ-5.0	0-2000W/m^2	±2%
SE-15	Relative Humidity	%HR	SIEMENS QFA 3160	0-100% h.r.	±2%h.r.
SE-16	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-17	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-18	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-19	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-20	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-21	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-22	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-23	Temperature	℃	SIEMENS QAC22	-35ºC;+50ºC	±0,05ºC en 0ºC (DIN 43760)

Table 1.- Features of the installed equipment

4.2 Period of Measurement

The solar cooling system has been measured from June to September since 2007.

5 Monitoring Results

5.1 Annual / Monthly Data

Table 2 shows the average experimental results of the chiller analysis in the year 2007 and 2008, operating in the steady state period.

Year	W_{ch} (kW)	W_c (kW)	W_g (kW)	***COP***	T_{dbo} (ºC)
2007	5.78	9.7	15.4	0.57	27.7
2008	4.4	8,0	12.5	0.51	31.2

Table 2.- Experimental mean values of the installation in the years 2007 and 2008

5.2 Analysis of Typical Days

In Fig. 3 it is shown the chiller operation temperatures for one day (11/07/2008). The operation in this day can be considered as representative for the chiller performance. The daily operation process is described as follows.

At 11:30 due to the solar field temperature, the pump of the secondary circuit starts to pump water heated in the solar heat exchanger to the generator of the chiller. As soon as the temperature of this water flow overcomes 80 ºC at the generator inlet, the chiller begins to produce chilled water.

The temperature difference in the heat driven of the absorption cycle is up to 7 ºC between the inlet and the outlet of the generator. The average generator power for this day is 7.9 kW. This generator power makes the evaporator outlet temperature ($T_{ev,o}$) decrease, reaching 12 ºC on this day. When the climatic conditions are the optimal ones, these values can decrease to 9 ºC. The temperature difference in the evaporator is around 2 - 3ºC, achieving a chilling capacity of 5.8 kW in the last part of the 2007 and 4.4 kW in 2008. According to the heat rejection system, the inlet temperature ($T_{he,i}$) increases when the absorption chiller operates, being its maximum value 42 ºC, coinciding with the maximum values of $T_{g,i}$.

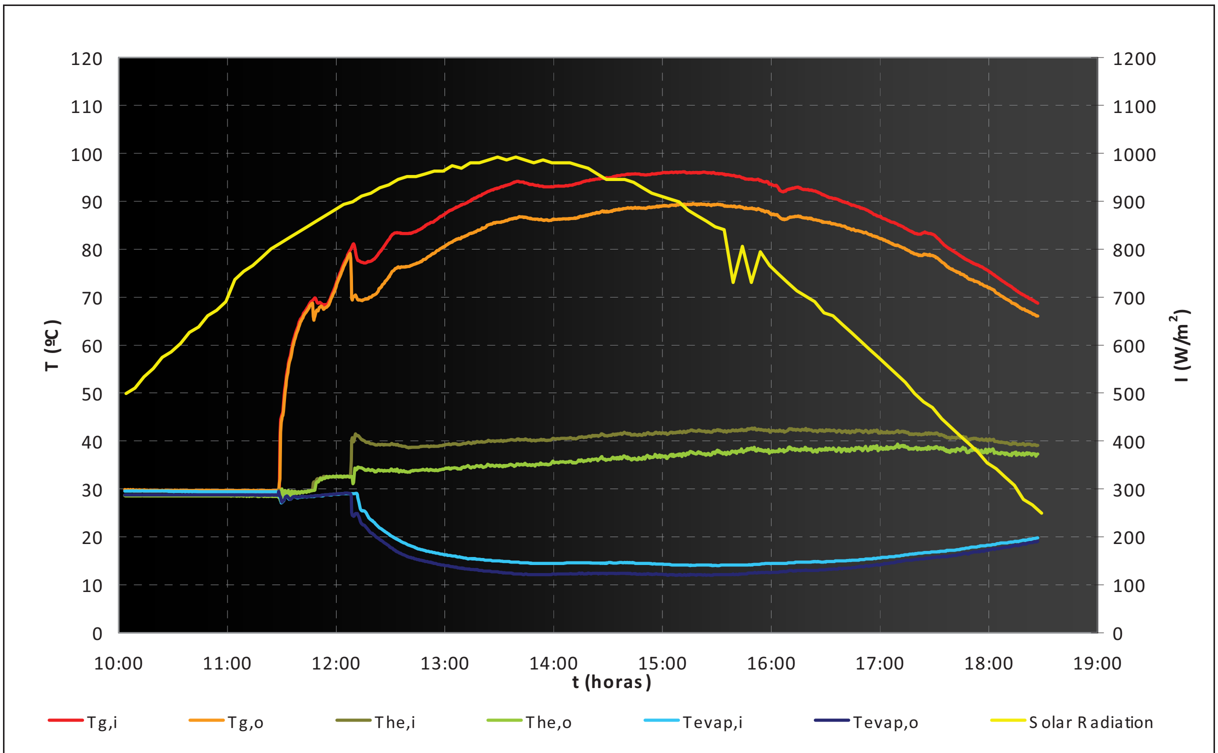


Fig. 3. - Operation temperatures of the chiller and solar radiation in 11/07/2008.

Because of the fact that a dry cooling tower is used to reject the waste heat of the absorption chiller, in figure 3 it can be seen that the heat rejection temperatures depend on the outdoor temperature. The influence of the ambient temperature can be seen in the Table 3 too. The average values of the most important parameters of the chiller are presented for three representative days in this table. It can be seen that the average generator power is similar during the three days, but the deviations between the chilling capacities differ up to 19%. This happens because at high values of the outdoor temperature, the capacity of the dry cooler to reject the heat of the absorption cycle to the ambient decreases. Therefore, the higher the outdoor temperature is, the lower the chilling capacity and the COP produced by the absorption chiller are.

Date	T_{dbo} (ºC)	$T_{g,i}$ (ºC)	$T_{ev,o}$ (ºC)	W_{ch} (kW)	W_c (kW)	W_g (kW)	***COP***
11/07/2008	35	94	12.4	4.02	12	7.9	0.48
29/07/2008	30.4	92.2	11.6	4.5	12.8	8.2	0.53
08/08/2008	29.4	91.9	9.5	4.8	13.3	8.7	0.53

Table 3.- Average values of three typical days.

5.3 Detailed Analysis

The steady state performance of the chiller was analyzed. In this way figure 4 shows the influence of the outdoor temperature on the COP during 2007 and 2008 when the chiller operated in the steady state period. The COP decreases when the sink temperature increases. Both trends have the same slope, although the mean values of the year 2008 are lower because the mean ambient temperature during 2008 was higher.

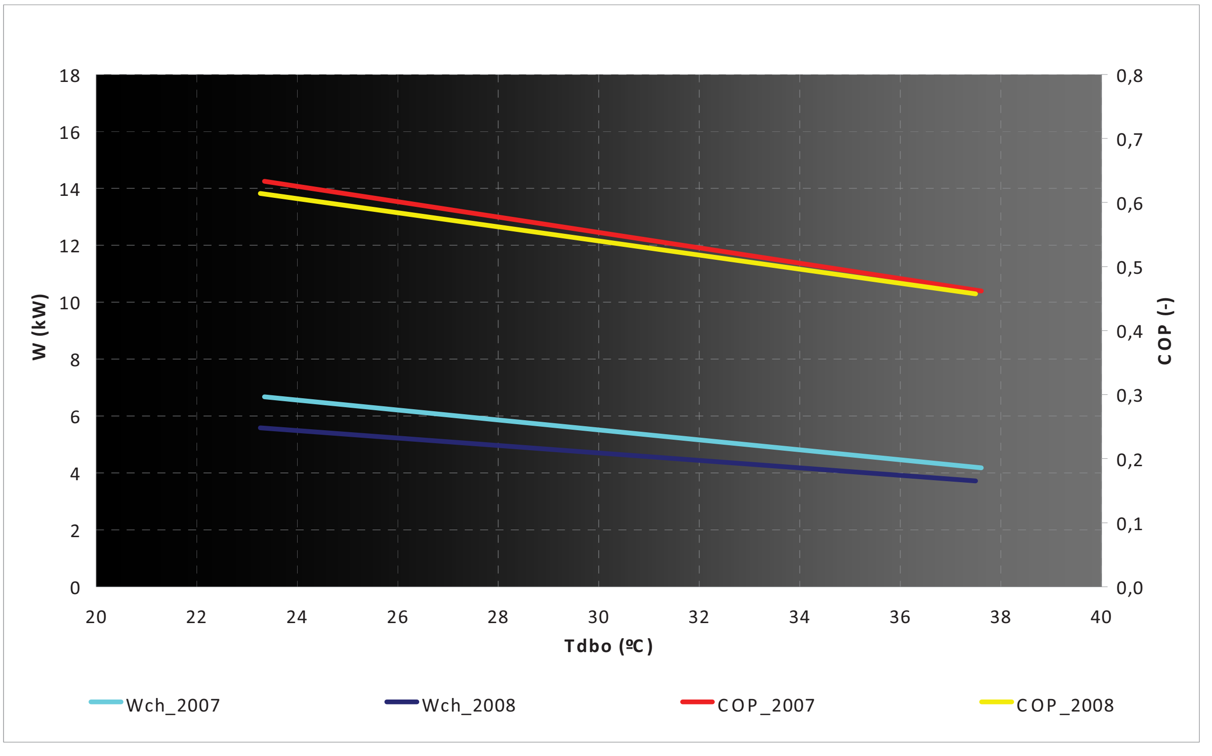


Fig. 4. – Influence of the outdoor temperature on the COP and in the chilling capacity

6 Experiences / Lessons Learned

As it was mentioned, during this first two years, the installation worked with a dry cooler tower. The studies showed the great influence of the temperature of the heat rejection sink on the machine performance. Therefore, the substitution of the initial heat rejection system was proposed in order to improve the performance of the absorption chiller. Finally, it was decided to use a geothermal system using a water well placed in the surroundings of the solar cooling installation. With this modification the chiller operated with a constant temperature in the heat rejection sink. This temperature, according to others water wells placed near by was around 17 ºC. This water well supply water to a 25 m^3 water tank, in which the heat produced in the absorption cycle, is rejected. The tank is usually used to irrigate the sport grounds placed in the surroundings of the solar cooling installation in summer. This use means the contained water of the tank is renewed every day, so, daily, the water temperature kept daily a constant value, resolving the possible problems of thermal saturation in the tank.

Another feature of the geothermal system is the following. The overall length of the new circuit is 190.5 m, of which 90.5 m was divided into three pipes with a diameter smaller than the rest of the circuit. This has been done in order to increase the heat exchange surface between the pipes and the ground. Hence the rejection of the heat generated by the absorption machine will take place in two places: the water tank and the geothermal horizontal exchanger. Besides this, the initial heat rejection sink, the dry cooling tower hasn't been removed from the installation, so that the solar cooling has a hybrid heat rejection system.

In order to estimate the performance of the chiller working with the geothermal sink a TRNSYS model was developed. Previously the model of the initial installation was created and validated with the experimental results taken from the monitoring system.

In 2009 the new heat rejection system started to work. Unfortunately the improvement of the chiller performance was partial. Although the influence of the ambient temperature was removed, due to the operational temperature of the water tank was 25 ºC instead of 17 ºC, the mean values of the chiller capacities were not the expected ones (Figure 5).

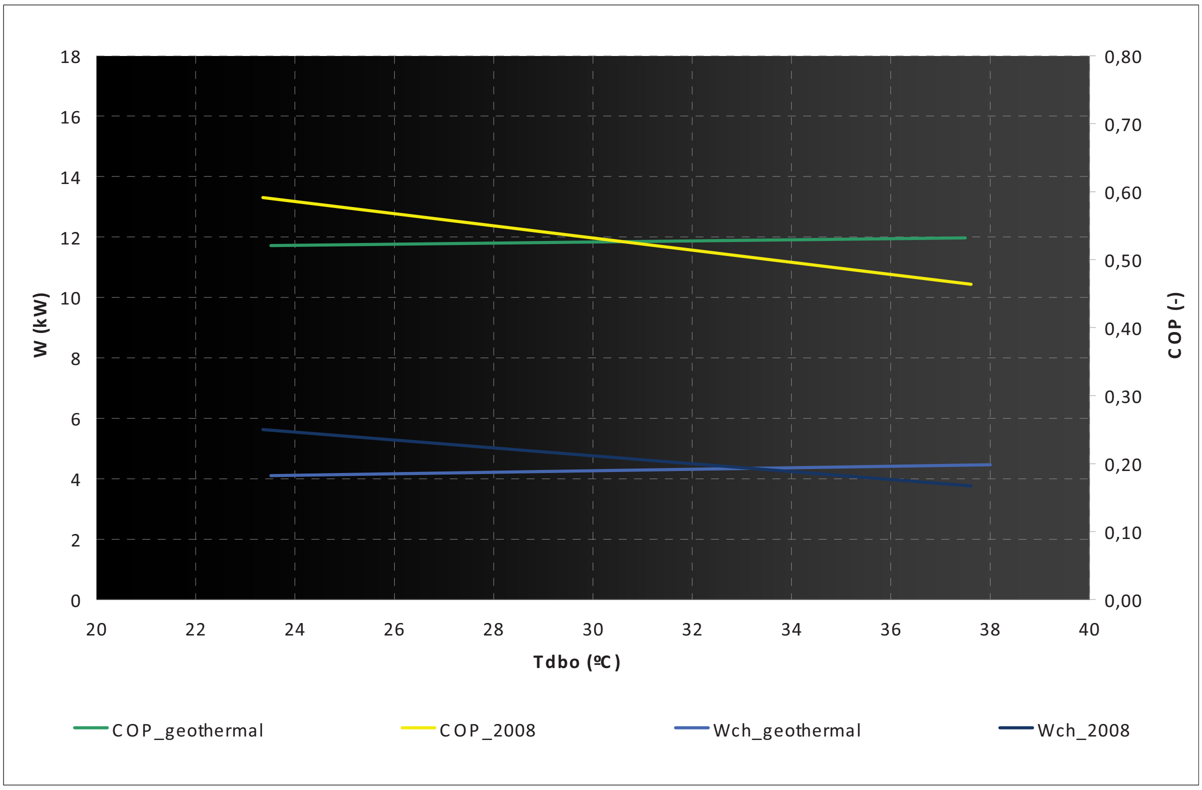


Fig. 5. – Comparison of the COP and the chilling capacity

7 Conclusions

The analyses of the solar cooling installation placed in the University of Zaragoza have allowed enlarging the knowledge of this kind of solar thermal systems.

The absorption chiller was installed in order to use the solar waste energy in summer time. Before its installation, the solar thermal system suffered overheating problems in this period.

Besides that the users of the gymnasium are satisfactory because the ambient conditions in the gym have been improved.

From the initial configuration of the solar installation, the system has been modified in order to increase its energy efficiency.

8 Bibliography

- Monné, C.; Alonso, S.; Palacín, F.; Serra, L., (2011). ***Monitoring and simulation of an existing solar powered absorption cooling system in Zaragoza (Spain).*** Applied Thermal Engineering, 31, 28-35. doi:10.1016/j.applthermaleng.2010.08.002.
- Monné, C.; Alonso, S.; Palacín, F., (2011). ***Experiencia en una instalación de refrigeración por absorción con energía solar***. Información tecnológica, 22 (3).
- Monné, C.; Palacín, F.; Alonso, S., (2011). ***Analyses of a new dissipation system for a solar cooling Installation in Zaragoza (Spain***). Journal of thermodynamics.
- Monné, C.; Alonso, S.; Guallar, J.; Palacín, F., (2011). ***Stationary analysis of a solar LiBr-H_2O absorption refrigeration system.*** International Journal of Refrigeration. Article in Press. doi: 10.1016/j.ijrefrig.2010.11.009.
- Palacín, F.; Monné, C.; Alonso, S., (2010***). Evaluation, diagnosis and improvement of a solar cooling plant by means of experimental analysis and dynamic simulation***. 8th International Congress on Solar Heating, Cooling and Buildings. Eurosun 2010. 28th September to 1st October 2010. Graz (Austria).
- Palacín, F.; Monné, C.; Alonso, S.; Izquierdo, J. M.; Heredero, J., (2010). ***Design and operational experiences of an alternative heat rejection sink for an existing absorption solar cooling system.*** 8th International Congress on Solar Heating, Cooling and Buildings. Eurosun 2010. 28th September to 1st October 2010. Graz (Austria).
- Palacín, F.; Monné, C.; Alonso, S., (2010). ***Experiences on heat rejection sinks of a solar absorption cooling system.*** 3rd International Conference on Passive and Low Energy Cooling for the Built Environment (PALENC 2010), 5th European Conference on Energy Performance & Indoor Climate in Buildings (EPIC 2010) and 1st Cool Roofs Conference. 30th September to 2nd October 2009. Rhodas Island (Greece).
- Palacín, F.; Monné, C.; Alonso, S., (2010). ***Comparing different sinks of heat rejection of an existing solar powered absorption cooling system.*** International conference on renewable energies and power quality (ICREPQ'10). 25th to 26th March 2010. Granada (Spain).
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Instalación experimental de refrigeración solar - primeros resultados.*** XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th to 21th June 2008. Vigo (Spain). SPE.
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Influencia de la velocidad de giro en las máquinas rotativas de absorción (parte I)***. XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th al 21th June 2008. Vigo (Spain). SPE.
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Influencia de la velocidad de giro en las máquinas rotativas de absorción (parte II)***. XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th al 21th June 2008. Vigo (Spain). SPE.

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 8
Monitoring Results of Technical College Butzbach

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 16.12.2010

By Edo Wiemken[1] and Anna R. Petry Elias[2]

[1]Fraunhofer-Institut für Solare Energiesysteme
Heidenhofstraße 2
79110 Freiburg
Phone 0761 4588 5412
Fax 0761 4588 9000
edo.wiemken@ise.fraunhofer.de

[2]Fraunhofer-Institut für Solare Energiesysteme
Heidenhofstraße 2
79110 Freiburg
Phone 0761 4588 5436
Fax 0761 4588 9000
anna.raquel.petry.elias@ise.fraunhofer.de

Contents

1 Background

The low-energy building of the Technical College in Butzbach has a demand for summer air-conditioning due to high occupation rates and the frequent use of computational equipment. Besides the regular school days the building is occupied throughout the summer season intensively as well. This fact was determining for the promotion of this project within the frame of the German Solarthermie 2000plus funding programme.

Two ventilation systems with heat recovery and a 1,250 m³/h air volume flow rate each were already installed in the building. These systems were not sufficient in order to remove the sensible and latent cooling loads in summer. Therefore, a solar autonomous chilling plant was installed which consists of two absorption chillers of the type Suninverse from Sonnenklima, Berlin. Each of the chillers has a nominal chilling capacity of 10 kW. They are driven exclusively by solar heat from a collector field of 60 m² aperture area. It consists of evacuated tube collectors of the type CPC Star azzurro, Paradigma. In addition, the ventilation units were extended by cooling coils. Furthermore, chilled ceilings and a cooling shaft were installed. The area which is air-conditioned comprises 335 m².

2 System Design

The Fraunhofer ISE supported the preliminary planning by carrying out simplified system simulations. With the help of the simulation results estimations could be given for the dimensioning of the system. The variation of the building simulation, which considers an external shading device, leads to estimations on the peak cooling loads (sensible and latent) of more than 20 kW. The sensible cooling loads often showed daily peaks of about 15 to 20 kW. As the type of chiller was almost already decided (Suninverse, 10 kW nominal chilling capacity) the results supported the installation of two absorption chillers.

A simplified scheme of the complete air-conditioning plant of the Technical College in Butzbach can be seen in figure 1.

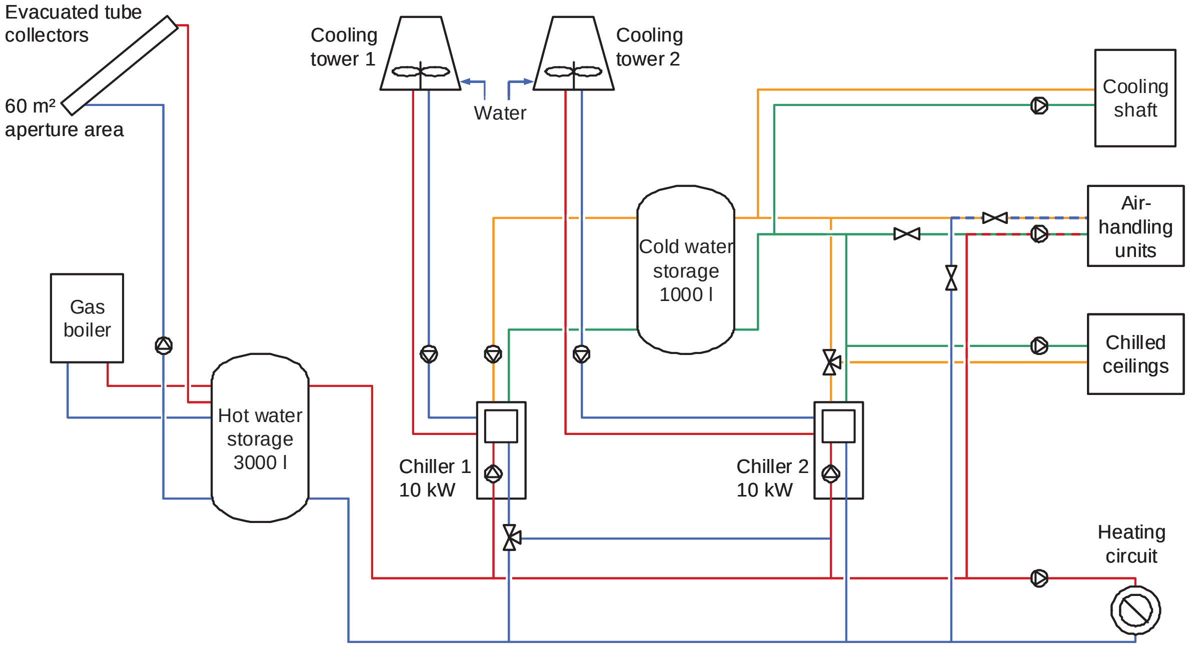


Figure 1: Simplified scheme of the heating and cooling system in the low energy building of the Technical College Butzbach. Source: Fraunhofer ISE

The hydraulic connection permits a single or a simultaneous operation of the chillers. Both chillers can deliver chilled water to the cold distribution systems. In the system concept it is planned that for high cooling demands in summer one of the chillers provides chilled water on a low temperature level for the dehumidification of the supply air. The second chiller supplies chilled ceilings on a higher cold water temperature level. This chiller cannot charge the cold water buffer storage. The last mentioned chiller could not be operated in the cooling season of 2009 due to a defect. Therefore, only operating results from one chiller can be provided for 2009. Since the end of June 2010 the second chiller could also start its operation.

The whole heat production and distribution system is one continuous water circuit. There is no heat exchanger integrated into the loop. This means also that the collector is a pure water system. In times where anti-freeze protection is necessary heat from the hot water storage is returned to the collector in short pump intervals.

Both, the solar collectors as well as the gas boiler can charge the hot water buffer storage. However, the boiler is switched off in the cooling period. By doing so, it can be assured that only solar thermal energy is used for the cooling of the building.

Figure 2: Cold distribution systems in the seminar rooms. Top: chilled ceiling elements; right: cooling shaft (without cover) for silent cooling and air dehumidification. Bottom: one of two air handling units with heat recovery and supply air cooling and dehumidification. The air handling unit in the seminar room was insulated additionally against acoustic noise. Source: Fraunhofer ISE

Figure 3: Evacuated tube collector field at the main building of the Technical College Butzbach. Also an effective shading of the rooms can be achieved by the assembly of the collectors. Source: Fraunhofer ISE

3 Control Strategy

The control strategy of the system was realized by one of the project partners (Hindenburg Consulting). Information on the strategy is not at hand.

The collector sub-control system for freezing protection runs independently from the overall system control and was prepared by the collector manufacturer Paradigma.

4 Monitoring Equipment

4.1 Installed Equipment

For the system monitoring different measurement devices have been installed. Beside several temperature and volume flow meters also the specific radiant power is measured by the help of two irradiation pyranometers (CMP11); one in horizontal position and the other one in the collector plane. The pressure in the collector circuit is measured as well. Ten electricity meters record the electrical power input of pumps, absorption chillers, cooling towers, air handling units and the controller (three-phase electricity counters with 1000 impulses per kWh). The operating hours from most pumps and some valve engines are registered as well.

The devices to measure the volume flow rates are multiple-jet impeller water meters with impulse transmitter. For measuring the temperatures in the system Pt100 temperature sensors (accuracy 1/3 DIN class B) were installed.

The following figure 4 shows all measurement points in the system.

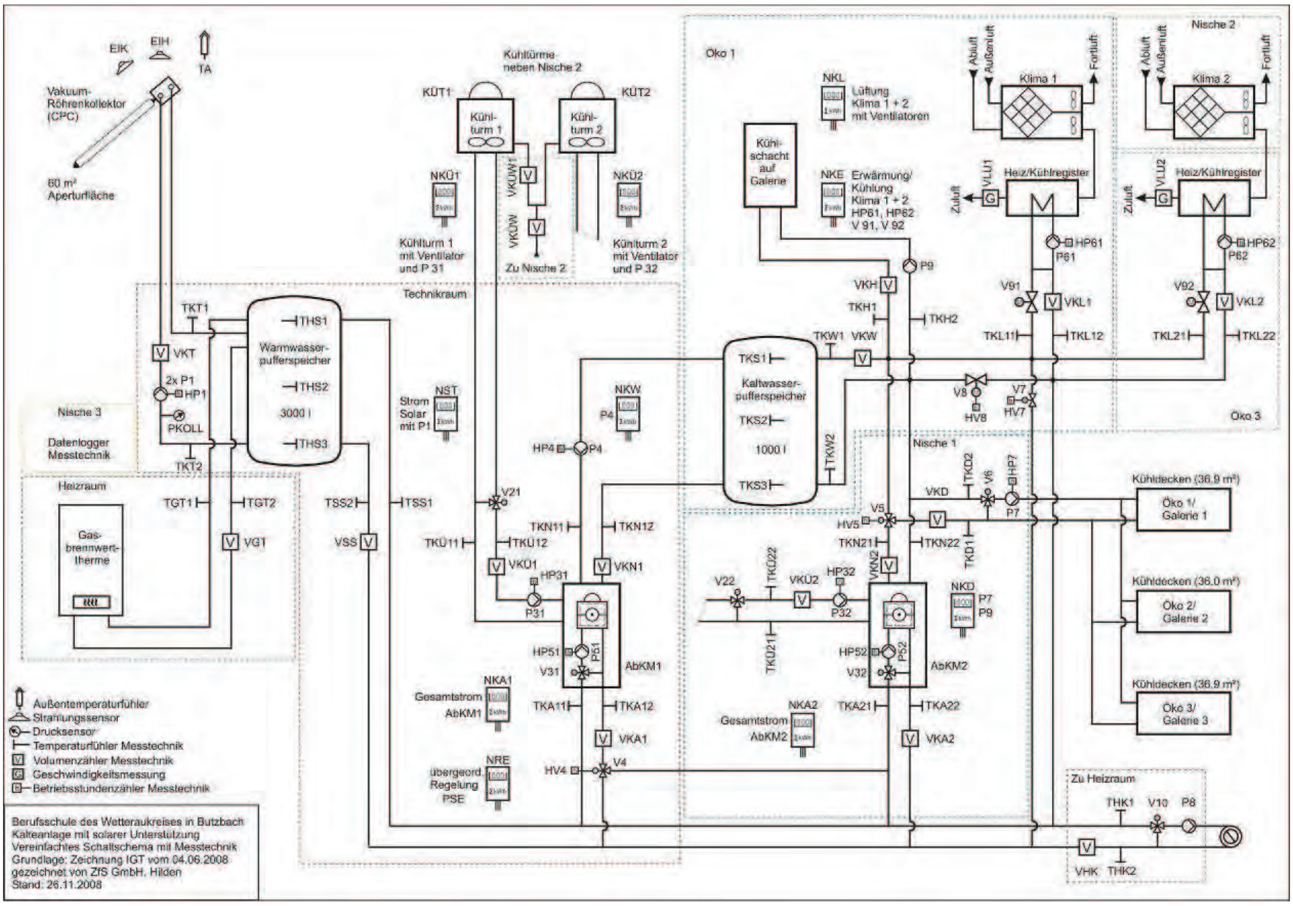


Figure 4: Scheme of the solar cooling plant with the measurement points. Source: ZfS GmbH

4.2 Period of Measurement

The monitoring system started its operation in December 18th 2008. Since then, the Fraunhofer ISE receives the data (one minute values) each day from the project partner ZfS. Since the beginning of the data monitoring there did not occur any complete or severe data loss. Just some measurement points needed to be adjusted at the beginning.

The Fraunhofer ISE will receive the data presumably until the end of 2011.

5 Monitoring Results

5.1 Annual / Monthly Data

As figure 5 shows the specific collector yields as well as the collector efficiency tend to be slightly higher in 2010 than the monthly values in 2009. A reason for this tendency can be a reduced shading of the collector in 2010 compared to 2009. In 2009 a scaffold shaded the collector.

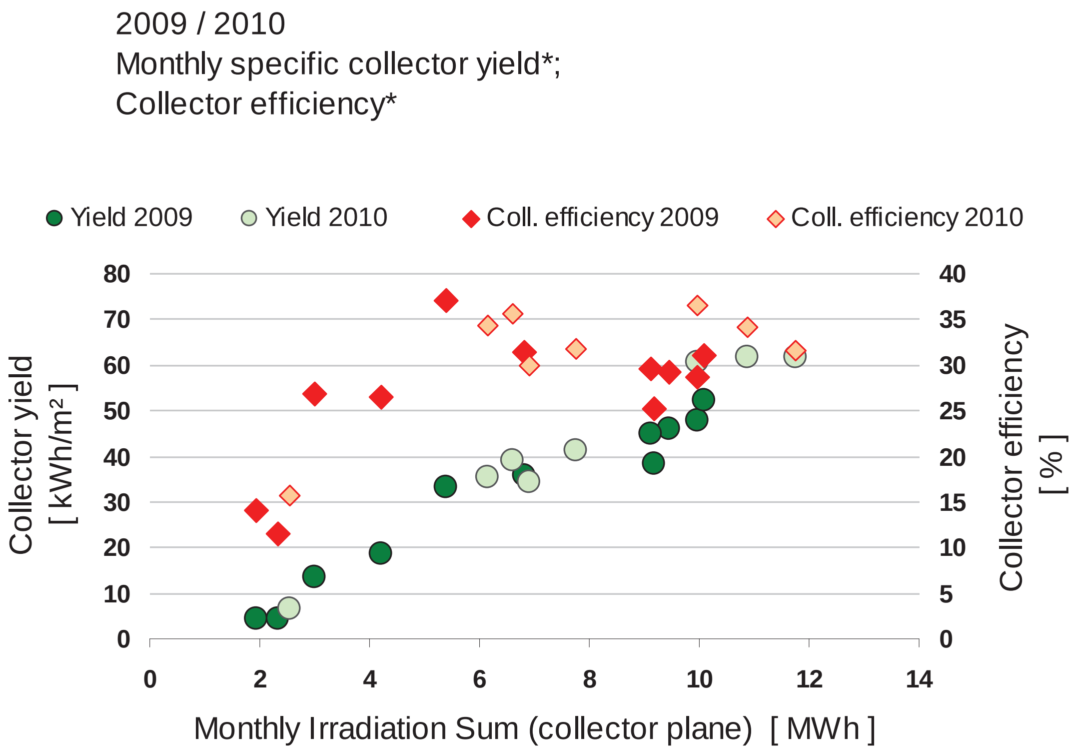


Figure 5: Monthly values of the specific collector yield and the collector efficiency in 2009 and 2010 (up to and including October 2010) plotted against the monthly irradiation sum on the collector plane. The values refer to the heat which the collector supplied to the storage; * includes returned heat to collector; no storage losses considered. Source: Fraunhofer ISE

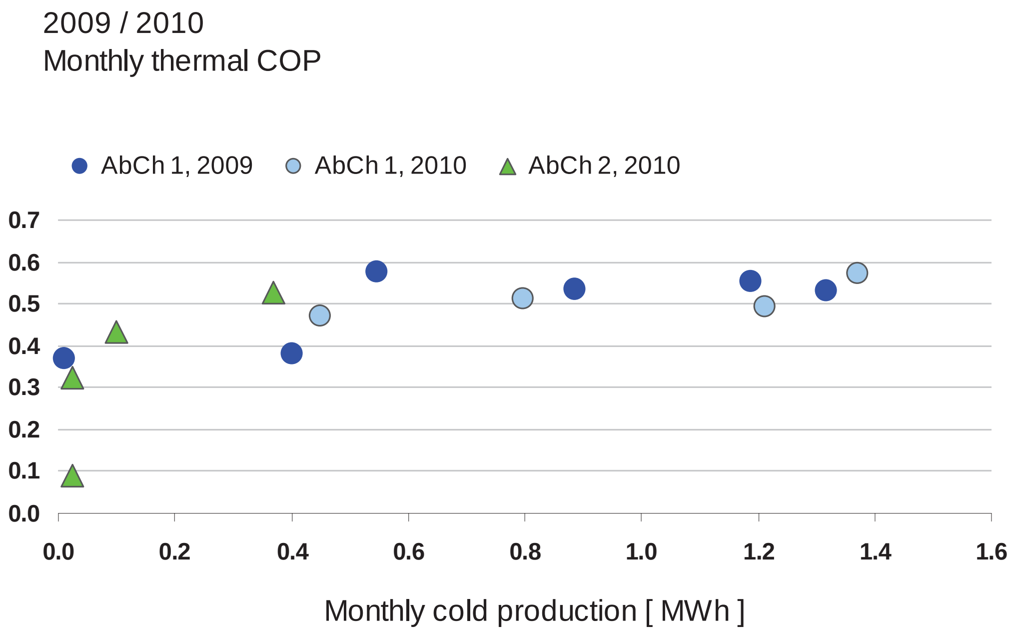


Figure 6: Monthly values of the thermal COP of the absorption chillers plotted against the monthly produced cold in 2009 and until September 2010. The cold which was produced by the absorption chiller 2 in June and September 2010 was very low. In June the chiller 2 started its proper operation at the end of the month and in September there was only little cooling load. Source: Fraunhofer ISE

2009	Collector efficiency* [%]	Share of solar heat returned to collector [%]	Share of solar heat on total heat input*** [%]	COP thermal absorption chiller 1 [-]	COP electric absorption chiller 1 [-]
January	11.5	57.4	4.1	---	---
February	26.9	20.6	16.5	---	---
March	37.0	6.7	48.7	---	---
April	29.2	9.4	100	---	---
May	29.7	5.0	92.8	0.39	2.9
June	25.2	7.1	100	0.55	4.6
July	28.7	4.3	100	0.56	5.4
August	31.1	3.5	100	0.53	5.0
September	31.5	4.7	100	0.58	5.6
October	26.5	12.6	29.8	0.37	2.4
November	14.0	39.5	5.1	---	---
December	< 0 **	165.2	< 0 **	---	---
Annual	27.7	10.8	40.6	0.53	4.7

* includes returned heat to collector; no storage losses considered

** collector yields slightly below heat for freezing protection

*** corrections in winter due to measurement error in heat input from gas boiler

Table 1: Monthly and annual balances for some system operation data of 2009. In the heating season when the gas boiler is in operation the share of solar heat is based on an estimate because the heat input of the boiler into the storage was measured incorrectly due to pollution in the hydraulic circuit. The values were adjusted by fixing the monthly hot water storage losses at 10% in winter. Source: Fraunhofer ISE

5.2 Analysis of Typical Days

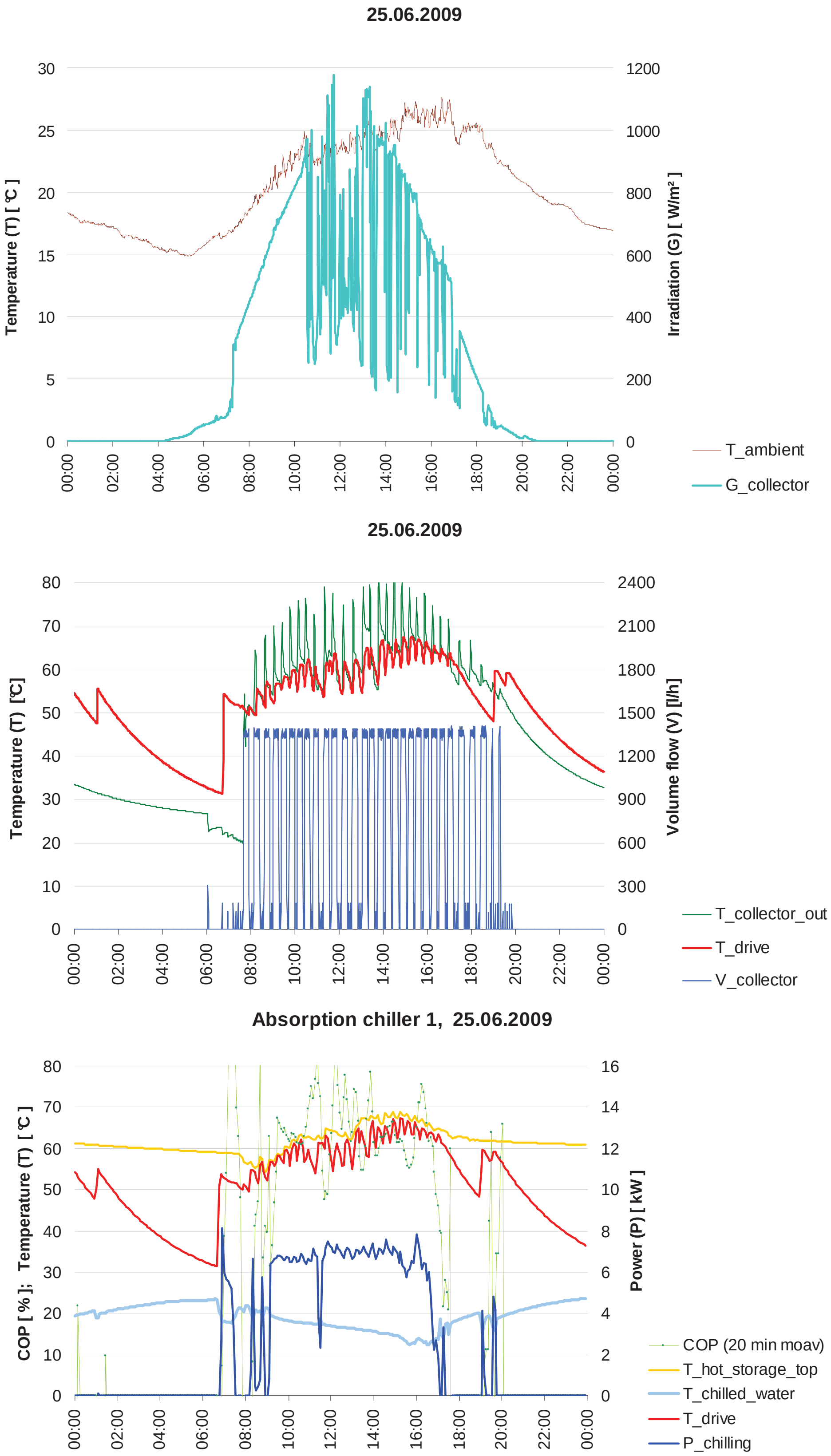


Figure 7: From the irradiation on the collector plane up to the cold production: Cooling with the absorption chiller 1 on June 25th 2009. (Values based on one minute average; just the thermal COP is based on the moving average of 20 minutes for a better visibility.) Source: Fraunhofer ISE

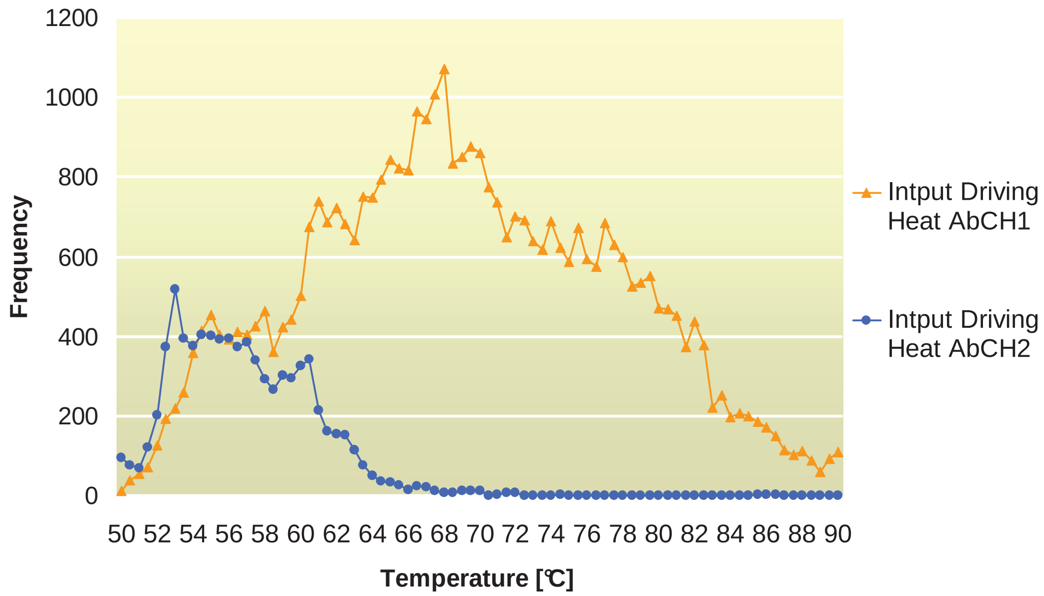
Often it can be observed that the chillers run continuously for several hours thereby taking advantage of the big range in the driving temperature. Figure 7 shows an exemplary day with cooling operation of absorption chiller 1. In contrast to the relatively steady temperature in the upper part of the storage the driving temperature to the chiller oscillates periodically in a stronger manner. This oscillations correlate exactly with the collector outlet temperature and the periodical pump operation in the collector circuit (“bucket principle”). It is still not clear why the driving temperature is evened by the storage just on a small scale. The measuring point of the upper storage temperature shows clearly a temperature course which is more straightened.

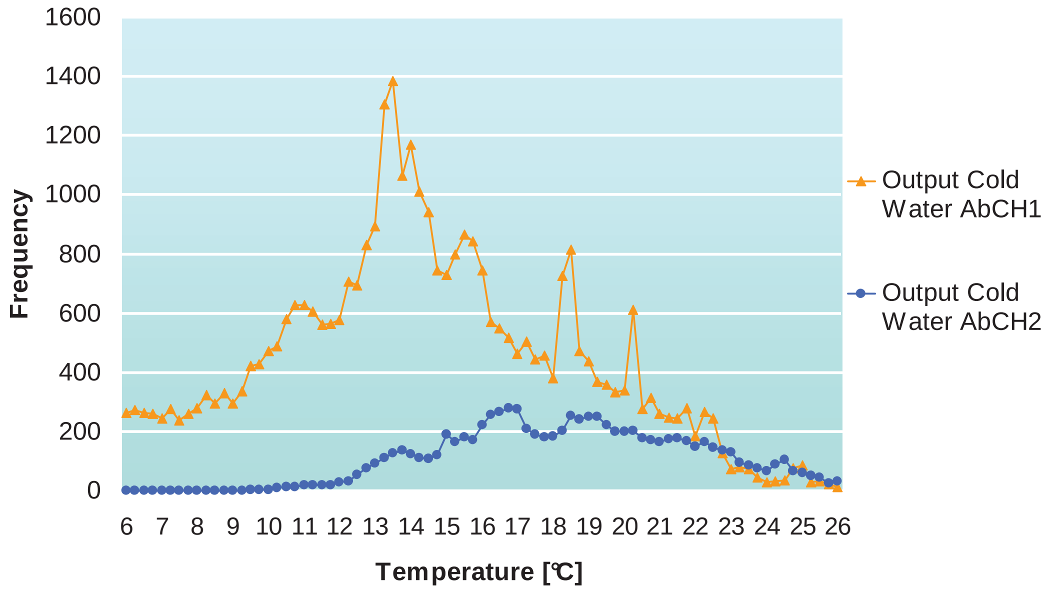
In contrast to the pump in the collector circuit the pump in the driving circuit of the chiller runs continuously from 9:00 until 17:00 with a short interruption before 12:00. Altogether with the low driving temperature chilling capacities between 6 and 8 kW are achieved. According to the temperature and capacity oscillations in the hot water circuit of the chiller the thermal COP varies strongly as well.

5.3 Detailed Analysis

Figure 8 shows the frequency distribution of driving, cooling water and chilled water temperatures of both chillers in serial connection in 2010. The frequency maximum for the driving temperature of chiller 1 lies between 65℃ and 70℃. Temperatures greater than 83℃ appear rarely. Chiller 1 produces most frequen tly chilled water temperatures in the range of 13℃ to 15℃. The chilled water range of c hiller 2 is widely distributed and on a higher temperature level as well. The driving temperature level of chiller 2 is shifted about 10 K downwards. The maxima of the cooling water temperatures are relatively clear. However, the values differ with 4 K from each other.

The frequency distributions show just values when the pumps in the three circuits are in operation. Though, the first operation minutes after the chiller starts can for example show higher chilled water temperatures until a steady operation is reached. However, these values are not filtered out from the shown graphics.





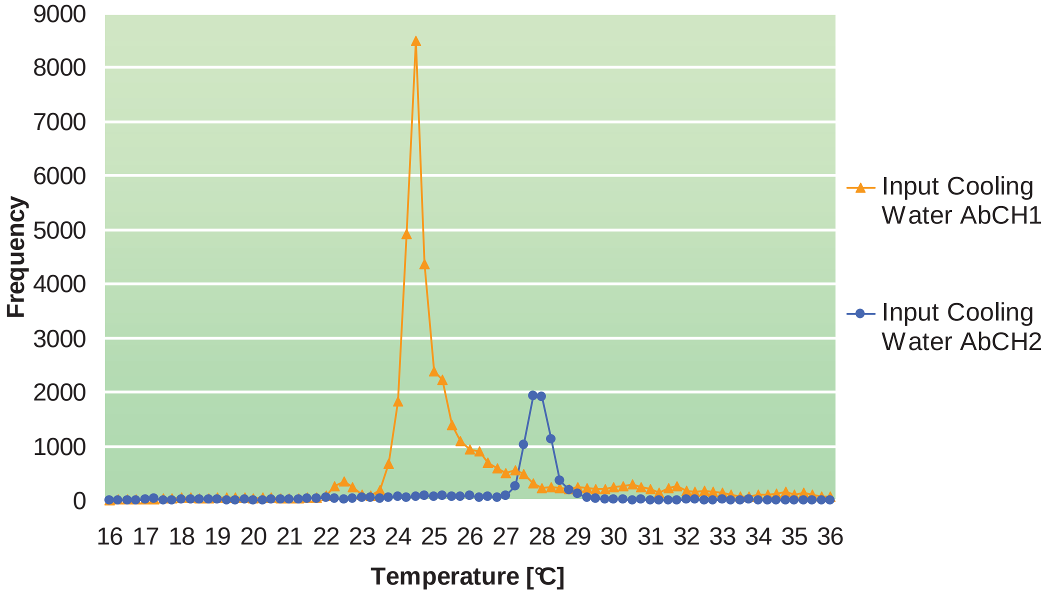


Figure 8: Frequency of the operation temperature ranges of both chillers in 2010 at serial connection in the driving circuit. (Values based on one minute average.) Source: Fraunhofer ISE

6 Experiences / Lessons Learned

As the collector is a pure water system it is necessary to return heat from the storage to the collector for the anti-freeze protection. Therefore, the pump runs in short intervals (bucket principle). The data analysis on an annual base showed that in one year about 10% of the collector heat gain is returned to the collector. In these 10% it is not only included the heat for freezing protection but also the losses which appear during the normal collector pump starts (cold water in the pipes). It should be kept in mind that in normal collector monitoring these start losses are not recorded. Rough estimated: approximately 1/2 to 2/3 of the 10% heat losses may be assigned to freezing protection.

As the plant is a pure water system no heat exchanger is installed. Consequently, the collector water is identical with the water in the heating system and driving circuit of the chillers. Thus, small particles from fabrication, e.g. swarfs, circulate through the whole hot water circuits and may cause pollution of filters which can lead to measurement errors. Actually, the flow meter in the gas boiler circuit was not working accurately due to the problem mentioned above. For avoiding this kind of problem a careful flushing of the total circuit after the installation is recommended.

The entry of small particles through the open wet cooling towers caused the pollution of the cooling water circuit. This led to a serious problem as the volume flow rates were much lower than normally and consequently too high cooling water temperatures entered the chiller. The solution of the problem was reached by the installation of an automated filter backwashing system, which starts every 16 minutes, and by a daily elutriation of the cooling towers. This circumstance should be considered by the installation of open cooling towers in areas with a high density of plants.

At the beginning, one of the two absorption chillers was not working due to a vacuum leakage. As an insolvency procedure of the chiller manufacturer was in progress in 2009 it took one year to solve the problem in the respective chiller. However, a general recommendation in order to avoid such problems cannot be given.

A special characteristic of the suninverse absorption chiller is the great flexibility in terms of driving temperature range. After the start phase of the chiller the chilled water production can be kept up until the driving temperatures drops slightly below 60°C. Since the second chiller was repaired and thus both chillers could be operated, the driving circuits were connected in series in order to test the above mentioned characteristic. The advantage of the serial connection is an increase in the temperature spread for the collector system. The experience that could be made with this connection variation was consistently positive. The first absorption chiller produces chilled water for supply air cooling and dehumidification, the second chiller supplies preferentially chilled ceilings and a cooling shaft on a higher chilled water temperature level. The operation of the chillers is stable; there are no start and stop phases and the operation of the chillers can be kept up continuously for several hours. Anyway, the common parallel connection of the chillers' generators with the hot water storage is also possible by switching a valve.

The acceptance of the solar autonomous air-conditioning during summer by the occupants (teachers, students) is big and it is stated as fully sufficient.

Furthermore, there is no extra input of fossil fuel for the chillers' operation necessary in order to overcome short periods with room air states outside the comfort range.

7 Conclusions

The plant for solar autonomous air-conditioning in summer in the low energy building of the Technical College Butzbach is built up relatively complex. In this case the reason lies in the testing of different connection variations in the driving and chilled water circuit, also for educational purposes.

The intended system operation was not possible due to the malfunction of one chiller in 2009 and the big delay of the reparation due to the insolvency process of the chiller manufacturing company. Further disturbances of the system's operation were caused by increased shading of the collector as a result of construction measures and pollution problems in the cooling water circuit. It is evident that the heat rejection is still a weak link in the component chain and it deserves closer attention also from the manufacturer's part.

Beyond that, the operation of the collector, of the absorption chiller 1 and the other hydraulic components has been very reliable. No damage results from collector stagnations. In spite of the above mentioned disturbances the occupants affirmed their satisfaction with the system at the final project meeting in September 2010. They also pointed out that the room air states in the seminar rooms improved significantly. The chosen concept, especially with respect to the autonomous solar thermally driven cooling, can be applied very well to the conditions of the cooling loads in the building.

The system serves at the same time as support in the education of students from the Technical College. This results in a big multiplier effect. In case the system should be transferred to a similar application it can be simplified considerably, as for example by installing just one chiller (for appropriate cooling loads) and thereby reducing respectively the hydraulic complexity. On the hot water part, the system is already simplified by being a pure water system and therefore not needing a heat exchanger in the collector circuit.

In the following points optimization potential can be given or a further revision is reasonable:

- possibly testing of the chiller operation with higher driving temperatures;
- a further examination of the second chiller's capacity in the coming cooling season;
- a further examination of the backwashing intervals in the cooling water circuit: at present a high fresh water demand is required. There might be found an optimum between water consumption, backwashing intervals and pollution degree in the cooling water circuit.
- checking if a careful pruning of the trees shading the collector is possible.

Estimations show that in comparison to conventional heating and cooling systems primary energy savings and CO_2 emission avoidance was achieved.

8 Bibliography

E. Wiemken, J. W. Wewiór, A. R. Petry Elias: Betriebserfahrungen aus solar unterstützter Kaltwassererzeugung zu Gebäudeklimatisierung. Tagungsbeitrag zum 20. Symposium Thermische Solarenergie, 5. – 7. Mai 2010, Bad Staffelstein

E. Wiemken, J. W. Wewiór, A. R. Petry Elias, B. Nienborg, L. Koch: Performance and Perspectives of Solar Cooling. Tagungsbeitrag zur EuroSun2010, 28. September – 1. Oktober 2010, Graz, Österreich

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 9

Monitoring Results of Germany: ZAE Bayern Office Building, Garching

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 20.12.2010

By M. Helm, M. Himpel, C. Schweigler

Institution	**The Bavarian Center for Applied Energy Research (ZAE Bayern)** Division 1: Technology for Energy Systems and Renewable Energy Walther-Meißner-Str. 6 D-85748 Garching, Germany
Phone	+49(0)89/329442-0
Fax	+49(0)89/329442-12
e-mail	helm@muc.zae-bayern.de

Contents

1 Background

In solar thermal installations, both solar cooling and solar heating can be provided synergistically, yielding a complete annual utilization. During the cold season, solar heat serves for space heating. During the warm season, solar heat can be converted into useful cold by means of sorption cooling. A favorable situation is given when low temperature heating and cooling facilities, e.g. floor or wall heating systems or activated ceilings, are applied for heating and cooling. During heating operation, a low temperature latent heat storage can be used to balance the heat generation by the solar thermal system and the heat supply to the heating system. Thus, a low operating temperature of the solar thermal system is accomplished yielding efficient operation with optimum solar gain.

In cooling mode, the same storage is used in combination with a dry cooling tower to absorb/reject the waste heat in order to replace a wet cooling tower. By that means heat rejection of the chiller is shifted partly to periods with lower ambient temperatures, i.e. night time, or to off-peak hours.

2 System Design

In conventional absorption cooling installations, wet cooling towers designed for a coolant supply/return temperature 27/35 °C are applied. To use a dry air-cooler, cooling water temperatures have to be increased to 40/45 °C. As a consequence of the increase of the cooling water temperature, the temperature level of the driving heat supplied to the regenerator of the absorption chiller has to be increased accordingly.

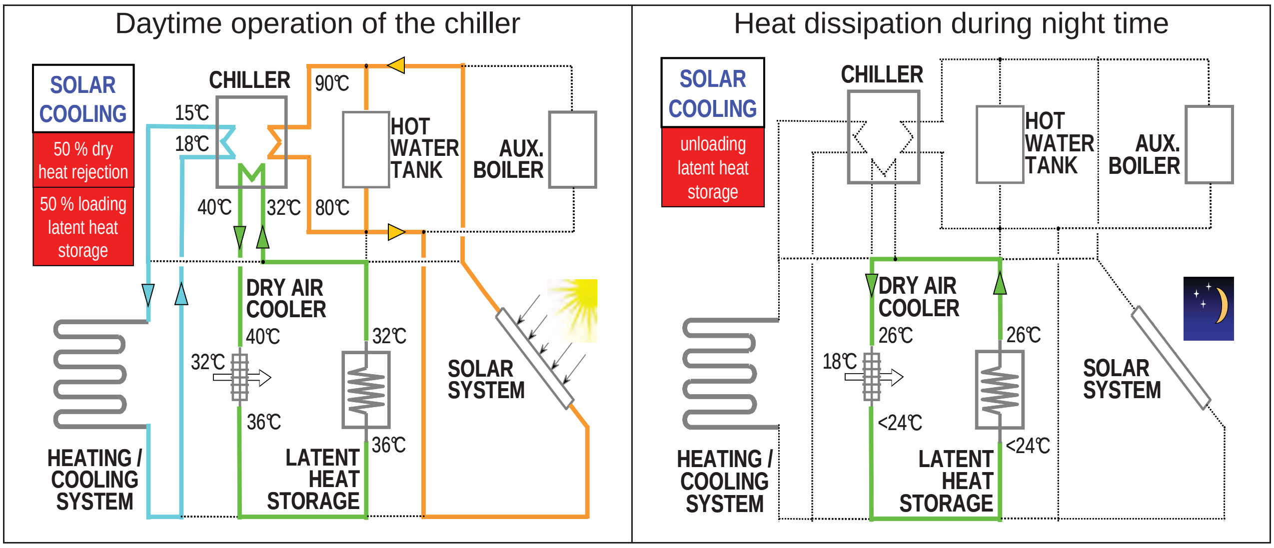


Fig. 1. System scheme of the solar cooling installation supported by latent heat storage.

By integrating a heat storage into the heat rejection system of the absorption chiller, a part of the reject heat can be buffered during peak load operation of the chiller, allowing the application of a dry air cooler with coolant temperatures of 32/40 °C. During off-peak operation or at night time when lower ambient temperatures are available, the stored reject heat can be discharged.

As a consequence of the reduced coolant temperature arising from the integration of the latent heat storage, lower temperatures of the driving solar heat are feasible to operate the absorption chiller. Thus a higher solar gain is obtained for a given size of the solar collector system (see Figure 1).

During the heating season, the latent heat storage buffers the solar surplus heat and balances the heat supply to the consumer by boosting the return temperature of the heating system (see Figure 2). Thus, a low operating temperature of the solar thermal system is accomplished yielding efficient operation with optimum solar gain.

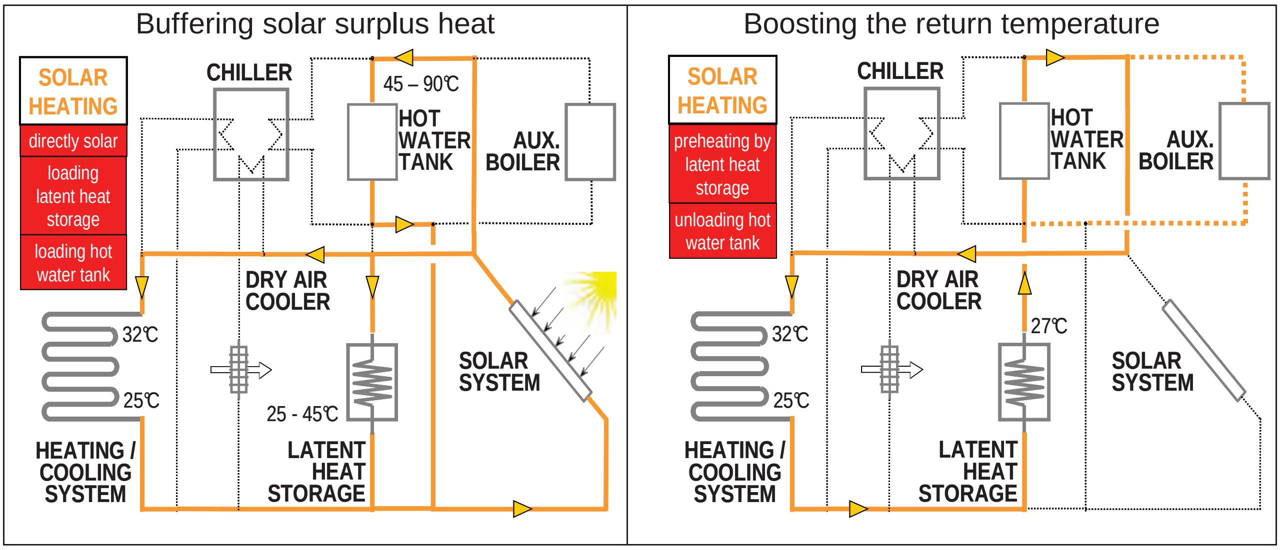


Fig. 2. System scheme for solar heating with latent heat storage

Simplified system configurations for cooling and heating operation are given in Figure1 and Figure 2. The realised piping and instrumentation is given in Figure 3. This kind of installation facilitates a very flexible and efficient use of solar energy in existing buildings with heating and cooling systems operating at moderate heating and chilled water temperatures.

- Conventional design of the solar thermal system with primary loop in water/glycol and a secondary loop connected to the heating system, the hot water tank, and the generator of the absorption chiller via the high temperature (HT) heat distributor.

- The dry air-cooler and the latent heat storage are integrated into a secondary loop, linked to the heating system by the plate heat exchangers WT 4-2. A set of valves is applied for switching between the different operating modes in summer and winter, enabling for boosting the temperature of the return flow of the activated ceilings, the loading and unloading of the latent heat storage, and the emergency cooling of the solar thermal system during summer operation. The latent heat storage had to be integrated in the secondary reject heat loop due to safety reasons and simplified unloading by the dry air cooler in cooling mode. In further installations also the chiller’s absorber and condenser should be integrated into this water/glycol loop to avoid a temperature increase of the cooling water due to the heat transfer in the plate heat exchanger. Furthermore an increase of the electrical COP could be achieved.

- During the heating season low (NT) and high temperature (HT) distributor are connected to each other.

- A ground water well linked to the NT distributor and a pellet boiler linked to the HT distributor serve for backup in cooling and heating mode, respectively.

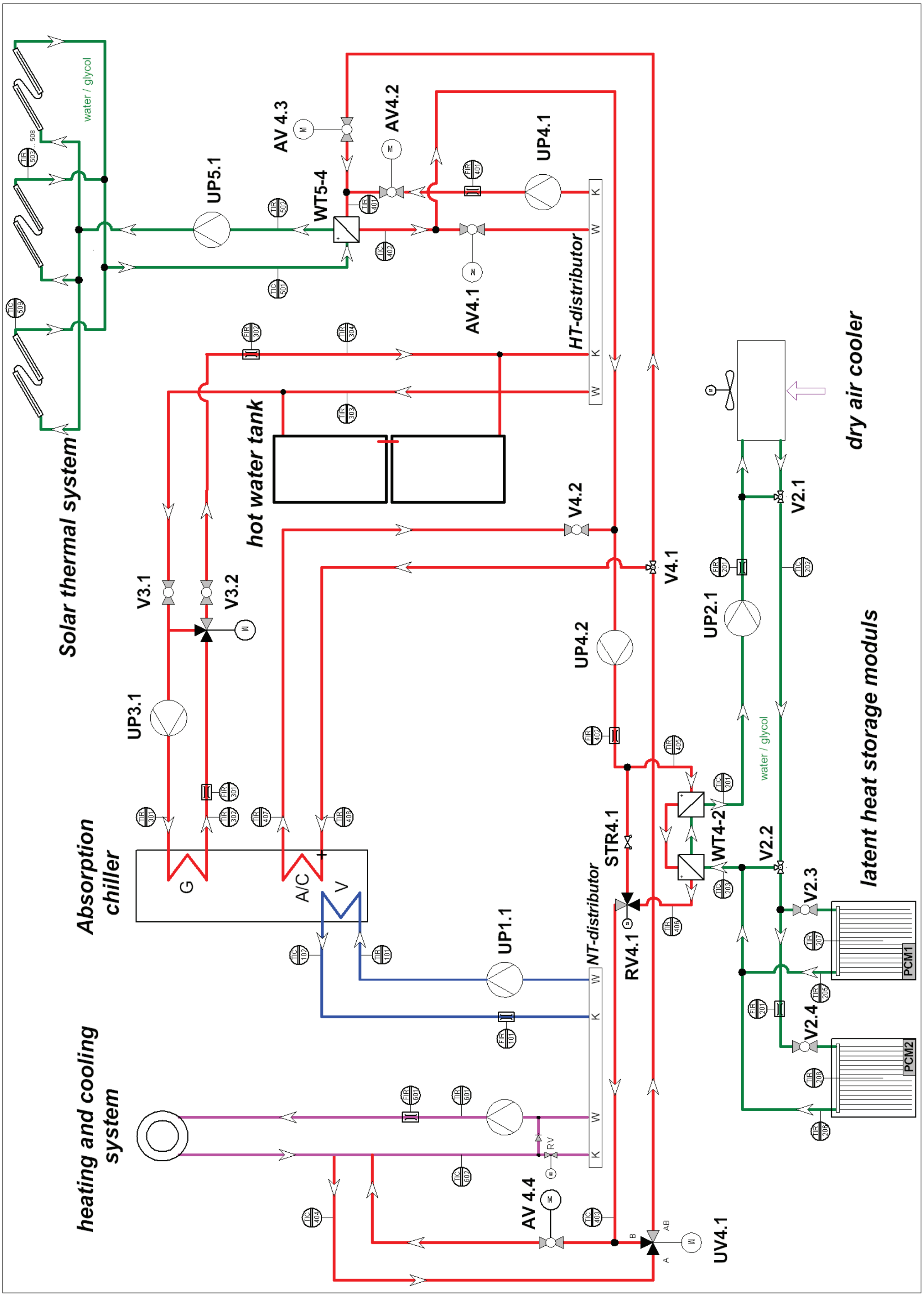


Fig. 3. Hydraulic scheme for the solar heating and cooling system with latent heat storage

3 Control Strategy

The control strategy for this solar cooling and heating system with all parameters, interactions and safety switches is very complex and can not be discussed in detail. The following list only gives a short overview about the control units and their major tasks.

(BMS) Building management system

- Selects heating or cooling mode (dis/connects NT and HT distributor).
- Controls supply temperature of the heating and cooling system as a function of the ambient temperature or the dew point temperature, respectively.
- Activates backup sources and valves of the building.

(Beckhoff SPS) PLC control of the absorption chiller

- Controls the internal pumps, hot and chilled water pump.
- Controls the chilled capacity: Controlled parameter is the outlet temperature of the chilled water; manipulated parameter is the driving heat input with control of the chilled water supply temperature.
- Requests external pumps and dry air cooler.
- Observes internal process parameters and activates the automatic purge procedure with 30 minutes purging every five days.

(UVR1611) Universal programmable logic controller

- Enables the solar thermal system when the temperature of the collector exceeds the:
  → Winter: return temperature of the heating system (activated ceilings) + 5 K
  → Winter: phase change temperature of the latent heat storage + 5 K
  → Winter: temperature in the top of the hot water tank + 5 K
  → Summer: minimum driving temperature of the absorption chiller 60 °C + 5 K
- Adjusts the speed of the different pumps (solar secondary loop to the HT distributor, heating loop to activated ceilings, flow rate through latent heat storage), according to the requested temperature.

  → direct solar heating < 35 °C
  → direct solar heating and/or loading the latent heat storage <50 °C
  → loading the hot water tank 50 °C up to 90 °C
  → emergency cooling of the solar thermal system 90°C
- Switches and controls all valves in the solar heating and cooling system.
- Enables the absorption chiller when cooling mode is selected and driving heat in the solar thermal system or buffer tank is sufficiently high (> 65 °C).
- Enables the reject heat pumps and the dry air cooler and adjusts fan speed for constant cooling water outlet temperature 30 °C.
- Calculates optimized unloading time and duration for the latent heat storage depending on the ambient air temperature or the return temperature of the activated ceilings.

4 Monitoring Equipment

4.1 Installed Equipment

Each heat sink or source is metered with two temperature sensors (inlet and outlet) and one electromagnetic flow meter. Apart from the TIC sensors for control purposes all sensors are connected to an Agilent 34980A Multifunction Switch/Measure Unit with shielded and grounded cables.

Installed sensors:

32	TIR	Pt-100 temperature sensors	1/10 DIN B	4-wire	resistance
	- Ambient air - Surface temperature of one solar collector - Outlet of each solar collector array - Solar heat exchanger, primary and secondary inlet and outlet - hot water tank - absorption chiller inlet and outlet of generator, evaporator and absorber/condenser - Reject heat exchanger, primary and secondary inlet and outlet - Dry air cooler outlet - Latent heat storage modules outlet and PCM temperature				
18	TIC	Pt-1000 temperature sensors	1/10 DIN B	2-wire	resistance
	- only for control purposes				
8	FICR	Electromagnetic Flow meters	± 0.2 %	2-wire	0-10 V
	- solar secondary loop - primary and secondary reject heat loop - latent heat storage module1 loop - absorption chiller, chilled water loop - absorption chiller hot water and hot water tank loop - heating and cooling system loop of the building				
4	LIR	Wattmeter	± 0.1 %	2-wire	0-10 V
	- Solar thermal system components and UVR1611 control unit - Reject heat loop and latent heat storage components - fan of the dry air cooler - absorption chiller and control unit				
1	RIR	Pyrometer	< 2 %	2-wire	mV
	- direct and indirect solar insolation on the aperture area of the solar collector				

4.2 Period of Measurement

The period of measurement started in January 2008 and is ongoing. Due to latent heat storage performance tests and capacity tests the data of the following days have not been taken into account for the evaluation of the annual performance data:

30.01. – 31.01.2008
20.04. – 23.04.2008
17.06. – 18.06.2008
29.10. – 30.10.2008

21.04. – 22.04.2009
09.07. – 10.07.2009
17.09. – 18.09.2009

23.03. – 28.03.2010

4.3 Annual / Monthly Data

Monthly data of the solar cooling and heating system for the years 2008, 2009, and 2010 are given in the tables 1 to 3 below. By replacing some pumps with new high efficiency models in June 2009 the electrical COP in cooling mode has been increased significantly.

Table 1: Monthly energy values for the measuring period 2008

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
2008	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]
Solar thermal system												
solar irradiation on collector	3182	6029	6404	6811	10009	8439	9660	8736	5980	4689	3094	1567
solar thermal gain	1167	2300	2110	2006	2804	2220	2831	2743	1968	1793	1060	749
Absorption chiller												
consumed heat	0	0	0	77	1419	1428	1874	1938	701	26	0	0
provided cold	0	0	0	44	972	949	1279	1378	490	11	0	0
Building												
cold demand	0	0	0	56	1897	4132	4035	3585	1134	95	0	0
heat demand	4763	3116	2456	958	0	0	0	0	1120	2277	3809	8ς
Secondary energy												
Total Electricity Consumption	125	262	208	204	211	267	230	255	203	165	130	104

Table 2: Monthly energy values for the measuring period 2009

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
2009	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]
Solar thermal system												
solar irradiation on collector	2797	3034	4162	6776	9674	8683	9168	10074	7553	4475	3376	2097
solar thermal gain	938	919	1364	3143	2825	2195	2717	3421	1827	1367	1215	504
Absorption chiller												
consumed heat	0	0	0	61	731	981	1676	2225	540	53	0	0
provided cold	0	0	0	31	503	671	1221	1639	392	32	0	0
Building												
cold demand	0	0	0	133	1368	2198	3596	3781	965	176	0	0
heat demand	6046	4975	3209	426	116	30	0	0	8	2127	2742	♦*ϑ
Secondary energy												
Total Electricity Consumption	88	102	155	125	155	129	175	227	83	111	130	62

Table 3: Monthly energy values for the measuring period 2010

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
2010	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]
Solar thermal system												
solar irradiation on collector	1671	3239	5385	8782	6266	8374	9683	7417	6980	4989	2808	
solar thermal gain	255	1022	1731	2369	1253	2388	3227	2128	2126	1770	880	
Absorption chiller												
consumed heat	0	0	0	140	0	1292	2061	1134		0	0	
provided cold	0	0	0	92	0	956	1488	828		0	0	
Building												
cold demand	6234	4135	2407	779	639	284	0	55	575	2320	3278	
heat demand	0	0	0	215	751	2974	4661	2171	137	0	0	
Secondary energy												
Total Electricity Consumption	42	91	112	141	165	194	216	139	82	139	101	

4.4 Analysis of the absorption chiller and the reject heat loop

Figure 4 shows the capacity of the chiller and the heat rejection loop during the course of a typical summer day. The absorption chiller provides about 11 kW chilled water by means of 14.5 kW driving heat from 10 a.m. to 5 p.m. Until 9 p.m. chiller operation is continued with surplus driving heat from the buffer tank, gained during daytime. The chiller's reject heat is primarily dissipated by the dry cooler. Although auxiliary energy saving has not been the major focus of this first pilot installation, the electrical COP (chilled water capacity per driving electric input) is about 7 to 10. The daily electrical COP comprises also the pumping energy during regeneration of the latent heat storage during night time and reaches values between 4.5 and 8 depending on the stored heat in the latent heat storage and the ambient temperature.

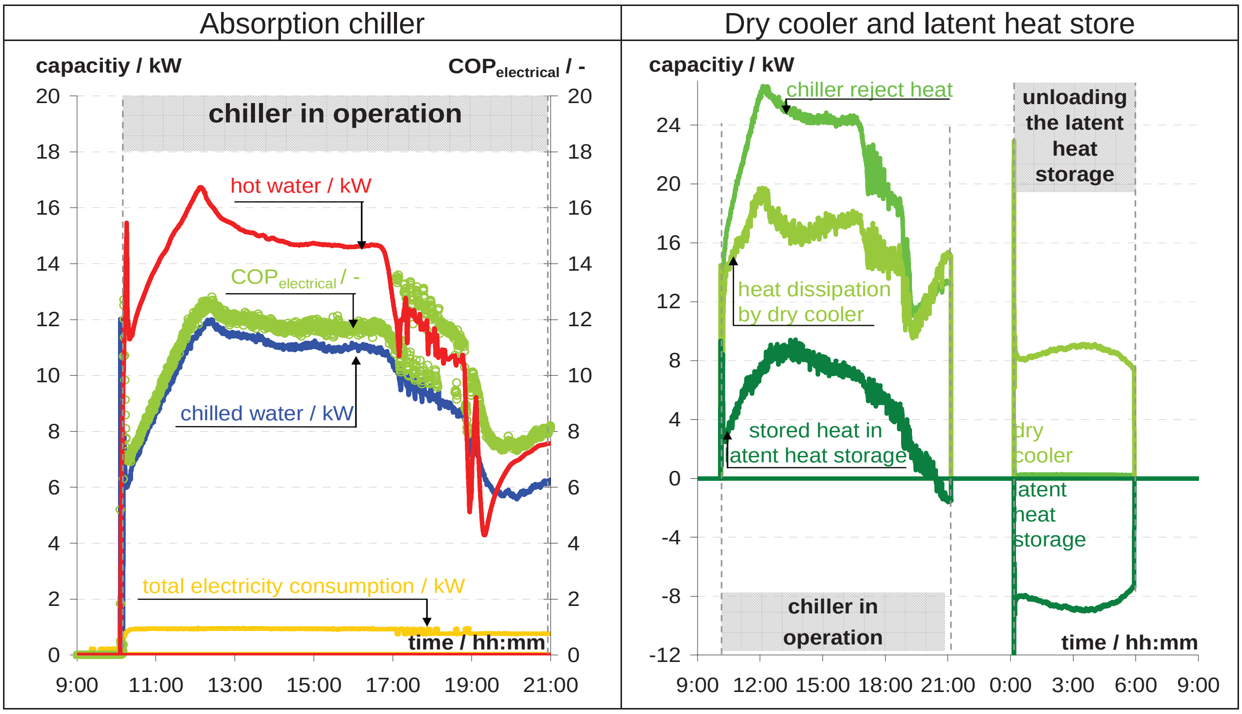


Fig. 4. Operational data of the solar cooling system during the course of a typical day

During hot ambient conditions (10 a.m. to 8 p.m.) the latent heat storage supports the dry cooler with maximum cooling water temperature about 33℃. The discharging of the latent heat storage is controlled with regard to the actual heat content (here from 0 to 6 a.m.).

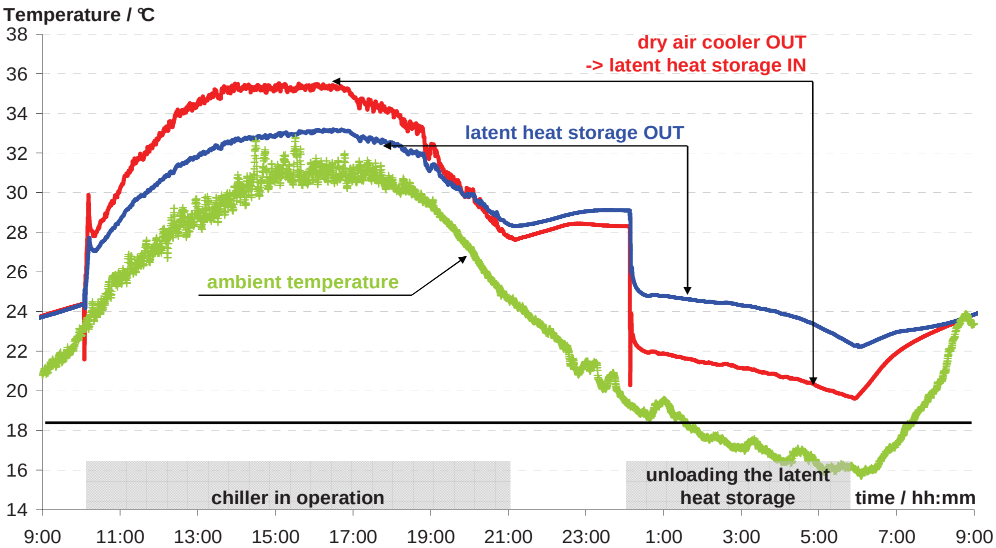


Fig. 5. Temperature profile of the reject heat loop

4.5 Detailed Analysis of the long term stability of the latent heat storage

Between 2007 and 2010 recurring measurements of loading and unloading the storage were carried out to determine long term effects on the PCM. These measurements showed no degradation of thermal power or capacity (see Figure 6), confirming the assumption that the separation of the PCM could be prevented successfully.

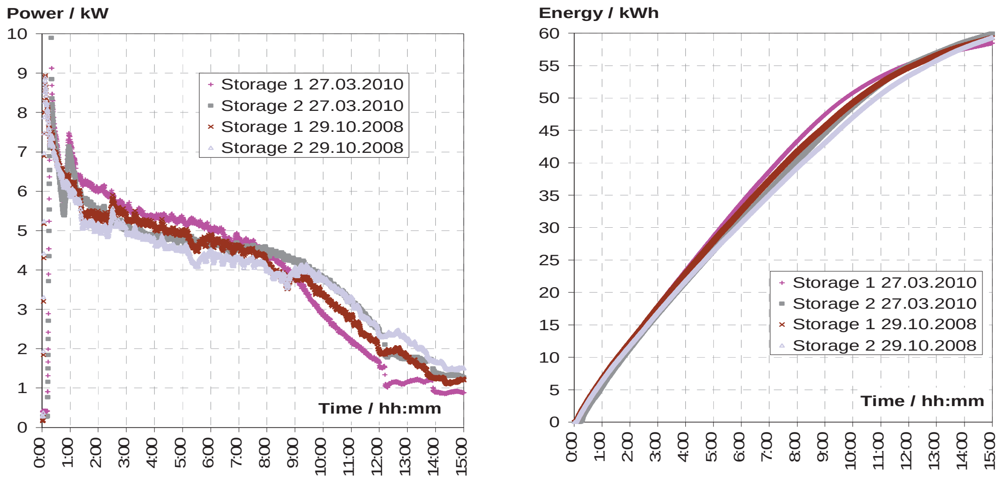
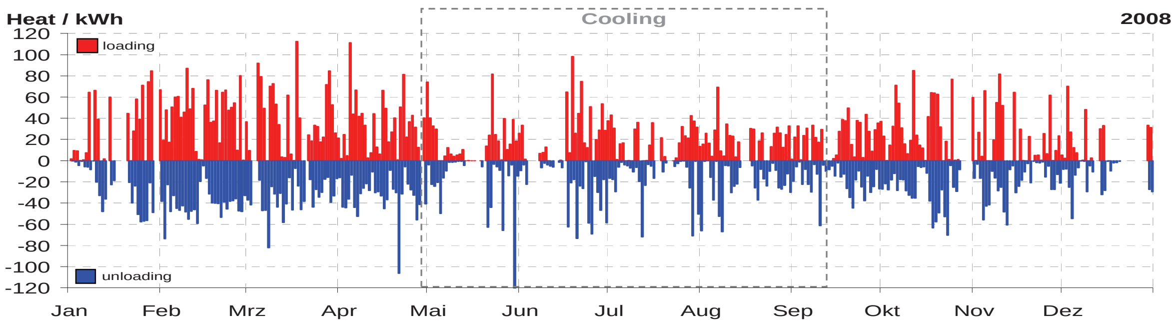


Fig. 6. Power and the stored energy of the two PCM storage modules during loading with a heat carrier supply temperature of 36 ℃ and a flow rate of 1.5 m^3/h

Figure 7 shows the charging and discharging cycles of the latent heat storage during summer and winter operation in 2008 and 2009 (data for 2009 in the following text in brackets), respectively. In total 293 (223) charging and discharging cycles have been performed. During the heating season a total of 6478 (3922) kWh solar heat has been stored, whereof 5741 (3323) kWh heat could be discharged and transferred to the heating system. In the cooling period 2105 (2053) kWh reject heat of the absorption chiller have been stored. In this case with close temporal coherence of loading and unloading a storage efficiency of 96,6% (85,4%) has been accomplished. Figure 7 furthermore illustrates the large impact of the weather situation on the utilization of the latent heat storage: Due to mild and sunny weather in spring 2008 rather large amounts of heat have been processed whereas a substantially lower utilization of the storage has been accomplished in winter 2008/09.



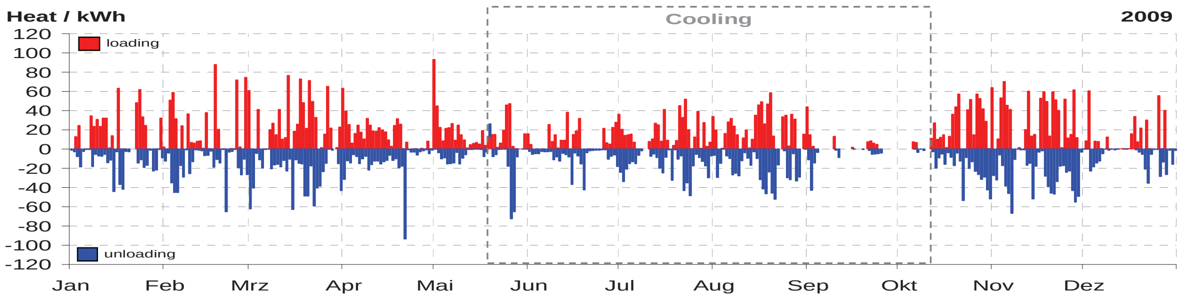
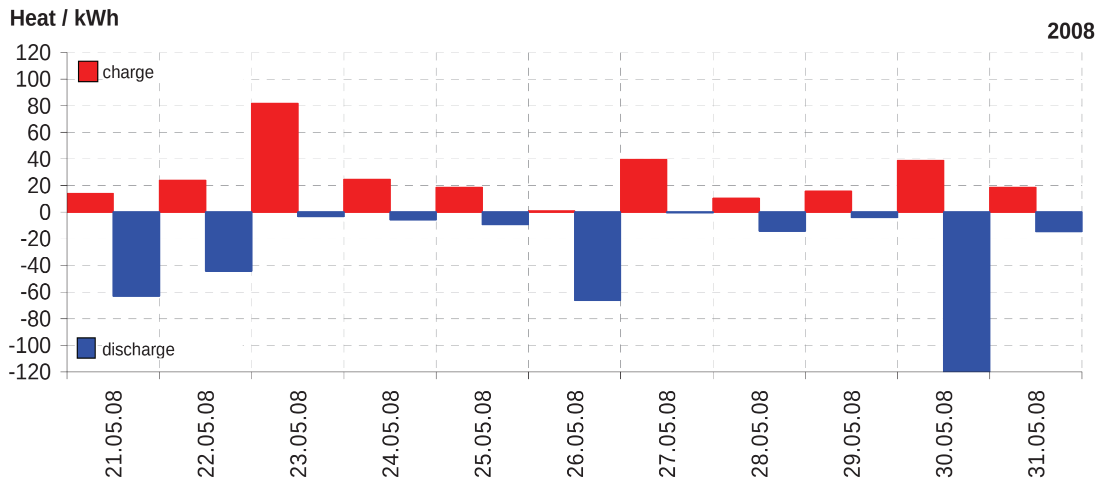


Fig. 7. Loading and unloading cycles of the PCM storage

In order to operate a latent heat storage as efficient part of a heating or cooling system, precise information about the actual energy content of the storage is essential. Due to the fact that a metering method for the thermal content of a latent heat storage is still a matter of development, up to now the potential for efficient charging and discharging of the storage could not be fully exploited. As a first attempt, different approaches using a heat meter to indicate the actual state of charge had only little success due to the small temperature difference of the heat carrier between storage inlet and outlet and the resulting inaccuracy of the control. To integrate the storage into the solar heating and cooling system despite the missing charge control, for the moment different temperature criteria are used to achieve a rough estimate of the actual state of charge. The development of a more precise metering procedure is ongoing.

Figure 8 shows the transferred heat for a series of loading and unloading cycles of the mentioned latent heat storage during solar cooling operation in summer 2008 an 2009, respectively. In cooling mode complete discharge of the accumulated heat during the following night is strived for. Yet , due to the imperfect charge control a substantial mismatch between loading and unloading of the storage occurred in 2008. Finally, by improving the control strategy in 2009 the daily disbalance of the heat storage has been minimized.

As a second conclusion from Figure 8, it can be stated that the storage allows for efficient mid-term storage, e.g. storage periods of some days, without substantial heat loss: A proof is given by the period May 23 to 30, 2008, when the solar gain accumulated during 8 days is almost completely extracted on two days only.



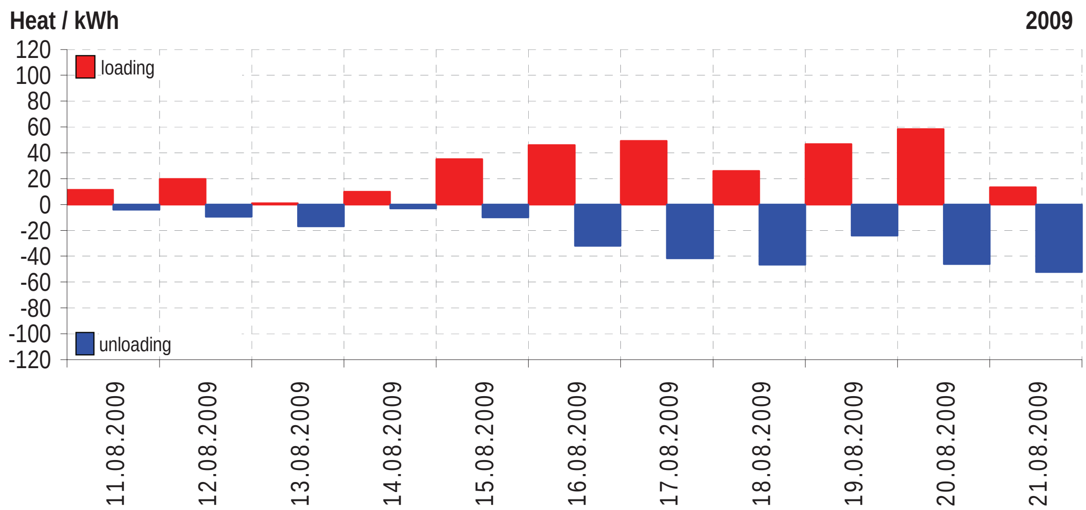


Fig. 8. Loading / unloading cycles of the PCM Storage 2008 / 2009

In 2008 a substantially higher level of insolation compared to 2009 allowed for a higher solar heat accumulation accompanied by a lower building heat demand. Consequently, a monthly solar fraction of 24 % to 85 % was reached in the period Jan – March 2008, whereas under severe winter condition in January and February 2009 the solar coverage was limited to 15 to 18 % only. In the transitional period March/April 2008 and April/May 2009 almost fully solar coverage has been reached. For the whole winter periods 2008 and 2009 solar fractions of 48 % and 33 % have been achieved, as shown in Fig. 9.

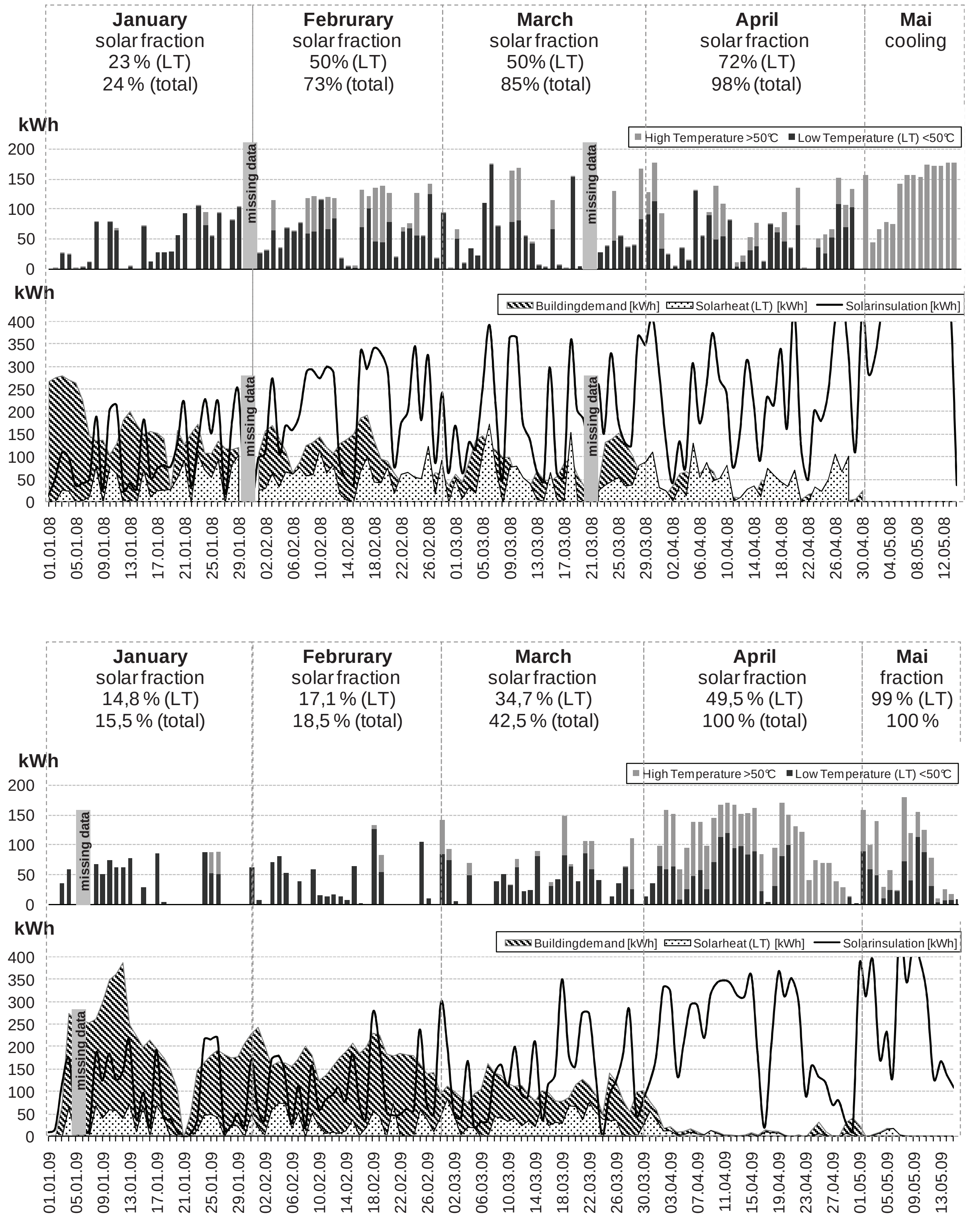


Fig. 9. Measurement data from solar heating period 2008 (top) and 2009 (bottom)

Simulations confirm the significant increase of the solar gain e.g. during the heating period from October to March of about 20 %, compared to a hot water tank system with the same specific cubic content per square meter collector surface area (in this case 40 Liters/m²).

All data of the solar fraction given above refer to the total amount of solar heat supplied to the building. These results by far exceed the performance of conventional large-scale solar installations. Pre-requisite for the obtained efficiency of the solar thermal system is the operation at moderate collector temperatures stabilized by means of the low-temperature latent heat storage. As a consequence, the major part of the solar heat supplied to the building passes through the low-temperature (LT) heat storage and only minor parts of the heat are generated at higher temperature levels, as given by the monthly data for the total heat supply to the building (total) and the contribution of the latent heat storage (LT).

5 Experiences / Lessons Learned

Within the monitoring period of 3 years various efforts in increasing the energy efficiency have been made. As a result reliable and efficient components, e.g. EC technology for pumps and fans, have been applied. Apart from this, the system effectiveness also depends on:

- the thermal COP has a crucial impact on the electrical COP. A poor COP results in an increase of the secondary energy used for dissipating the reject heat to the ambient.
- the standby electricity consumption should be reduced to a minimum. At best, unutilized components are shut down completely.
- the knowledge of the real heat content in the latent heat storage is essential for an optimized and efficient storage management aiming at high solar fraction and minimized electricity consumption.
- the pressure drop in all components should be minimized, obeying economic and thermodynamic limitations.

Further improvements will be realized in the near future:

- Replacement of the 3-phase AC fan of the dry air cooler by a 1-phase EC model. This reduces the electric energy consumption of the cooler by about 50 % and the parasitic energy demand in standby by about 15 %.
- Integration of an optimized latent heat storage with increased performance and reduced setup area (-50%).
- Implementation of a 3-way mixing valve in parallel to the latent heat storage allowing for an easy adjustment of the cooling water temperature and preventing unintended loading of the latent heat storage.
- Test of a heat content sensor for the latent heat storage.
- Control of the flow rate of the primary solar loop depending on the current insolation on the solar collectors.

6 Conclusions

A solar heating and cooling system with absorption chiller and latent heat storage has been operated for more than three years with high energetic efficiency and only minimal maintenance effort. In recurring measurements from 2007 until 2010 no aging effects of the phase change material (PCM) used in the latent heat storage could be observed.

Due to the implemented latent heat storage solar cooling with a dry cooling tower has been accomplished during the whole cooling period independently from ambient temperatures. In the heating periods an average solar fraction of 48% in 2008 and 33% in 2009 has been achieved owing to the reduced collector temperature stabilized by the latent heat storage. Aiming at series-production and wider distribution of the heat storage, in Oct 2009 a follow-up project for further development of the solar heating and cooling system has been started. By integrating a new optimized latent heat storage, accompanied by further system improvements, an electrical COP of 12 seems to be feasible for this solar cooling system.

7 Acknowledgment

The project is supported by funds of the German Federal Ministry of Environment (BMU) under contract number 0329605O and 0329605D.

8 Bibliography

M. Himpel, S. Hiebler, M. Helm, C. Schweigler, “Long term test results from a Latent Heat Storage developed for a Solar Heating and Cooling System”, Proceedings of EuroSun 2010, Graz, 2010

M. Helm, C. Keil, S. Hiebler, H. Mehling, C. Schweigler, Solar heating and cooling system with absorption chiller and low temperature latent heat storage: Energetic performance and operational experience, International Journal of Refrigeration, 32, 2009, 596–606.

S. Hiebler, H. Mehling, M. Helm, C. Schweigler, Latent heat storage developed for solar heating and cooling with melting temperature 29 °C, Proceedings of Effstock 2009 – 11th International Conference on Thermal Energy Storage, Stockholm, 2009

M. Helm, S. Hiebler, H. Mehling, C. Schweigler, “Solar Heating and Cooling with Absorption Chiller and Latent Heat Storage”, 3rd International Conference Solar Air-Conditioning, Palermo, Italy, 30.09.–02.10.2009

C.Schweigler, S.Hiebler, C.Keil, H.Köbel, C.Kren, H.Mehling, “Low temperature Heat Storage for Solar Heating and Cooling Applications”, ASHRAE Transactions, ASHRAE Winter-Meeting, Dallas, USA, Jan. 2007

F.Storkenmaier, M.Harm, C.Schweigler, F.Ziegler, P.Kohlenbach, T.Sengewald, “Small-capacity LiBr Absorption Chiller for Solar Cooling or Waste-Heat driven Cooling”, Proc. of the 30th International Congress of Refrigeration, Washington, USA, Sept. 2003

Task 38
Solar Air-Conditioning
and Refrigeration

D-A5:

Installation, Operation and Maintenance Guidelines for Pre-Engineered Systems

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: June 2011

Edited by Anita Preisler[1]

Institution	AIT – Austrian Institute of Technology, Energy Department
Address	Giefinggasse 2, A-1210 Vienna, Austria
Phone	+43 (0) 50550-6634
Fax	+43 (0) 50550-6613
e-mail	anita.preisler@ait.ac.at

Contribution from Dagmar Jähnig[2], Amandine LeDenn[3], Uli Jakob[4], Nicole Olsacher[5], Hilbert Focke[6], Daniel Neyer[7], Luis Angel Bujedo Nieto[8], Assunta Napolitano[9]

[2]Institution	**AEE INTEC**
Address	**Feldgasse 19, A-8200 Gleisdorf, Austria**
Phone	**+43-3112-5886-28**
Fax	**+43-3112-5886-18**
e-mail	**d.jaehnig@aee.at**

[3]Institution	**TECSOL**
Address	**Perpignan, France**
Phone	**+33 (0)4 68 68 16 39**
Fax	**+33 (0) 4 68 68 16 41**
e-mail	**amandine.ledenn@tecsol.fr**

[4]Company	**Solem Consulting**
Address	**Postfach 2127, D-71370 Weinstadt, Germany**
Phone	**+49 174 4130921**
Fax	**+49 7151 60486-25**
e-mail	**uli.jakob@solem-consulting.com**

[5]Company	**S.O.L.I.D.**
Address	**Puchstrasse 85, 8020 Graz, Austria**
Phone	**+43 316 292840-48**
Fax	**+43 316 292840-28**
e-mail	**n.olsacher@solid.at**

[6]Institution	**ASIC – Austria Solar Innovation Center**
Address	**Roseggerstraße 12, A-4600 Wels, Austria**
Phone	**+43 (0) 7242 / 9396 5563**
Fax	**+43 (0)7242 / 9396 49 5560**
e-mail	**focke.hilbert@asic.at**

[7]Institution	**University Innsbruck - Energieeffizientes Bauen, Institut für Konstruktion und Materialwissenschaften**
Address	**Techniker Str. 13, A-6020 Innsbruck, Austria**
Phone	**+43 512 507-6618**
Fax	**+43 512 507-36092**
e-mail	**daniel.neyer@uibk.ac.at**

[8]Institution	**CARTIF**
Address	**Parque Tecnológico , 47151 Boecillo, Spain**
Phone	**+34 983 546 504**
Fax	**+34 983 546 521**
e-mail	**luibuj@cartif.es**

[9]Institution	**EURAC**
Address	**Viale Druso 1, 39100 Bolzano, Italy**
Phone	**+39 0471 055651**
Fax	**+39 0471 055699**
e-mail	**assunta.napolitano@eurac.edu**

Contents

1 Introduction

The idea of this working group was to set up guidelines for installation, operation and maintenance based on experience of already existing pre-engineered small scale solar heating and cooling plants. Therefore, an end-user survey was invented by setting up a questionnaire including the relevant information for this purpose (see chapter 2.1). From the results of the interviews (see 2.2) it became clear that most of the analyzed solar heating and cooling plants can´t be categorized in “pre-engineered” systems. As a result it was decided in the working group that the information gained from the end-user survey should be used for a list of recommendations (see chapter 2.3) to bring the idea of pre-engineered systems forward.

As a second step interviews with companies, which already offer package solutions for solar heating and cooling plants were carried out (see chapter 3) to investigate the pre-engineered status of the offered packages available. Furthermore, by comparing the different package solutions and also taking the experiences from monitoring of such systems into account, suggestions for high quality package solutions were set up.

2 End-User Survey

The purpose of the end-user survey was to give an impression on the pre-engineered status of the already existing solar heating and cooling plants by collecting the experiences (and expectations) of the plant owners regarding installation, operation and maintenance of their systems. The end-user survey is based on interviews which were held personally by authors of this report as well as filled in questionnaires by the plant owners themselves. To get in contact with the relevant plant owners a list of existing small scale solar cooling plants was put together. Therefore, in addition to the already known plants, due to monitoring evaluation by IEA SHC Task 38 institutions, manufacturer of small scale ab-/adsorption chillers were asked for existing installations. In total, data about 122 solar cooling plants was collected (see Appendix 1).

2.1 Interview guideline

To get the desired information from the plant owners of small scale solar heating and cooling plants a questionnaire was set up as an interview guideline. The content of the interview guidelines is divided into the following sections (see Appendix 2):

1. General data
2. General questions
3. System configuration

4. Planning and installation
5. System operation
6. Maintenance
7. Costs

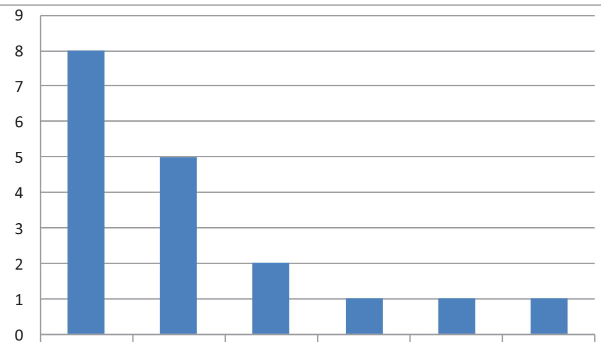
As the list of existing small scale solar cooling plants shows, most of the more than 120 elaborated plants are located in Spain or in German speaking countries (see Appendix 1), the questionnaires were translated into Spanish and German (see Appendix 2) to increase the number of filled in questionnaires by plant owners.

2.2 Results

The collected results are presented in anonymous form, which was part of the interview guidelines general conditions.

2.2.1 General data

In total, 18 interviews were conducted in 6 countries (see Figure 1) on 8 application types (see Figure 2). No statistical conclusions can be taken from this survey because of the small sample size but it reflects the average ongoing trend.



	Spain	Austria	Germany	Switzer-land	France	Italy
Interviews per Country	8	5	2	1	1	1

Figure 1: Evaluated interviews per country

Although most of the application types are office buildings (see Figure 2), a mix of applications for small scale solar heating and cooling plants can be verified, which also corresponds with the application types in all listed plants shown in Appendix 1.

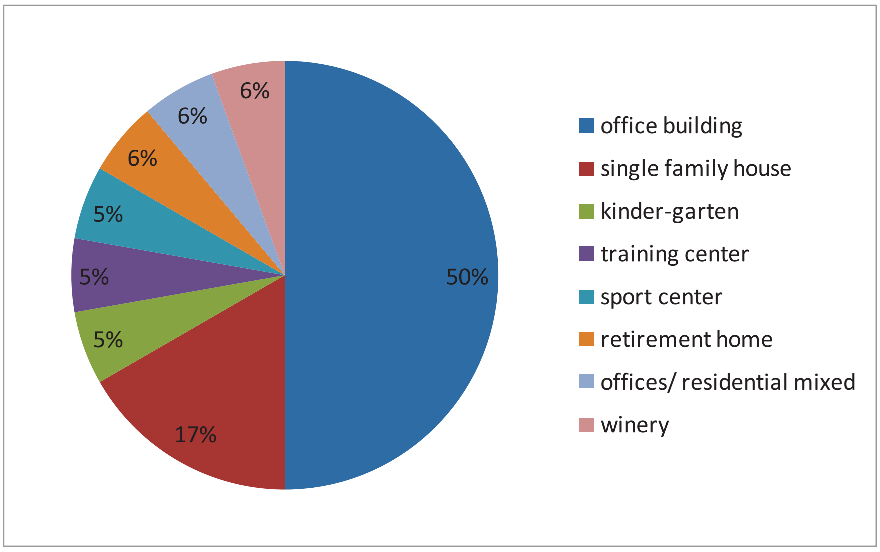


Figure 2: Application types of solar heating and cooling plants

2.2.2 General questions

To the question, if the plant did fulfil owners expectations, two third answered with yes and one third with no, which is rather a high percentage of dissatisfied plant owners (see Figure 3). For “no” the following reasons were mentioned:

- System is still not ready for operation
- Plant was only fully operating after the third season. The energy savings were much smaller than predicted by consultant
- Lots of adjustments for the operation
- Commissioning didn’t happen (2 plants)
- Start up phase is too late in the morning (first time of cold production at noon)
- Lack of continuity on the management of the project after commissioning (chiller provider is unable to maintain correctly the chiller and its control/monitoring system)

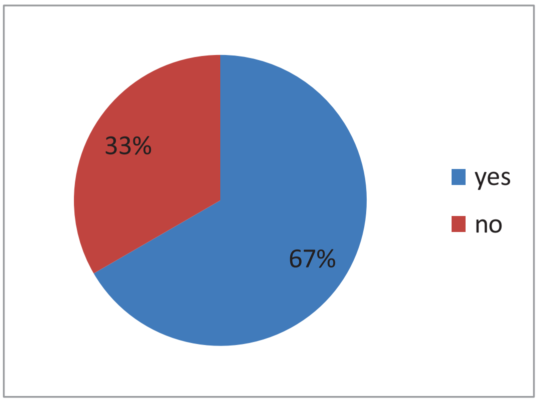


Figure 3: Fulfilment of expectations

The plant quality was judged quite well by the plant owners (see Figure 4); for “moderate” and “bad” following additional information or impressions were given:

- The plant was not ready for daily use and needed a lot of manual adjustments, therefore, the plant has been dismounted
- The majority of solar thermal installation companies do not understand solar thermal systems

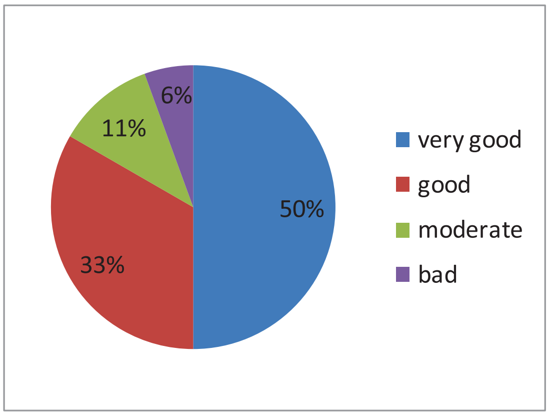


Figure 4: Judgment of plant quality by plant owner

2.2.3 System configuration

From the technology side 14 of the evaluated plants are equipped with absorption chillers only four are adsorption chiller systems. The preferably used technology on the solar thermal side are flat plate collectors (14 plants), only three plants operate with evacuated tube collectors and one uses both types of solar collectors.

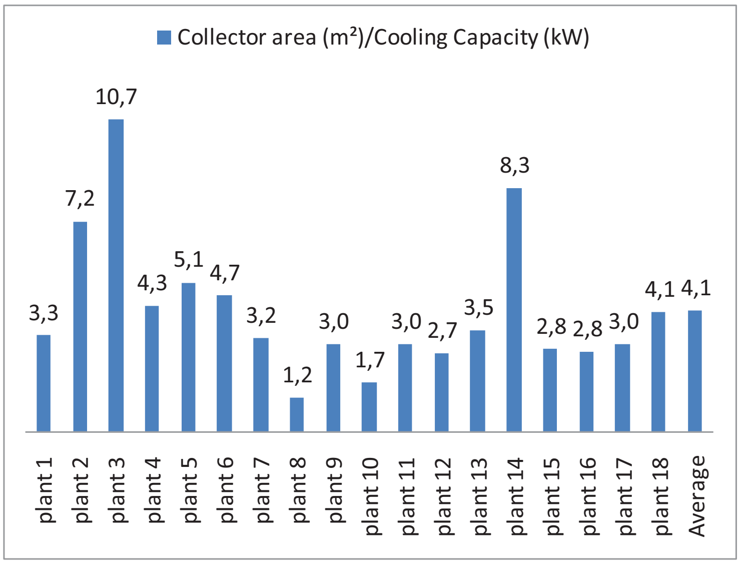


Figure 5: Configuration of solar collector field

Concerning dimensioning of the solar collector area in correlation to the installed cooling capacity, which is a typical planning indicator for solar thermal cooling plants, the values vary from 1,2 m²/kW to 10,7 m²/kW (see Figure 5). Also the configuration of the hot storage tank in correlation to the solar collector area has quite a large range from 9,4 l/m² to 166,7 l/m² (see Figure 6).

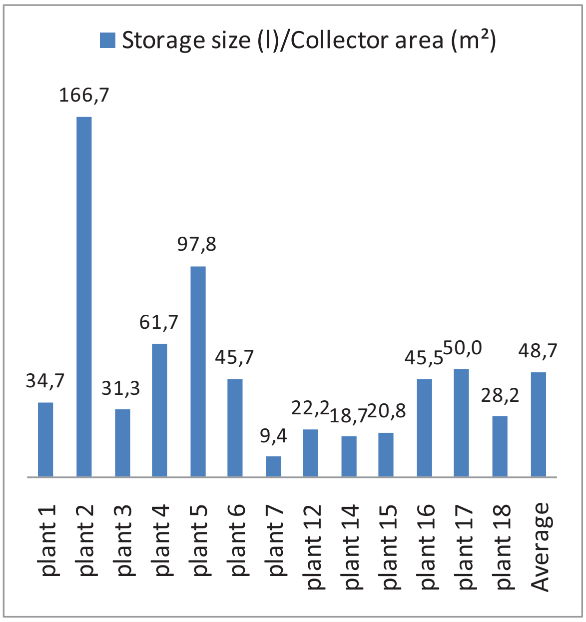


Figure 6: Configuration of solar storage size

The usage of solar thermal heat is limited to air-conditioning in three of the evaluated plants, all of the other plants use the solar thermal heat either for space heating support and/or domestic hot water (DHW) preparation (see Figure 7).

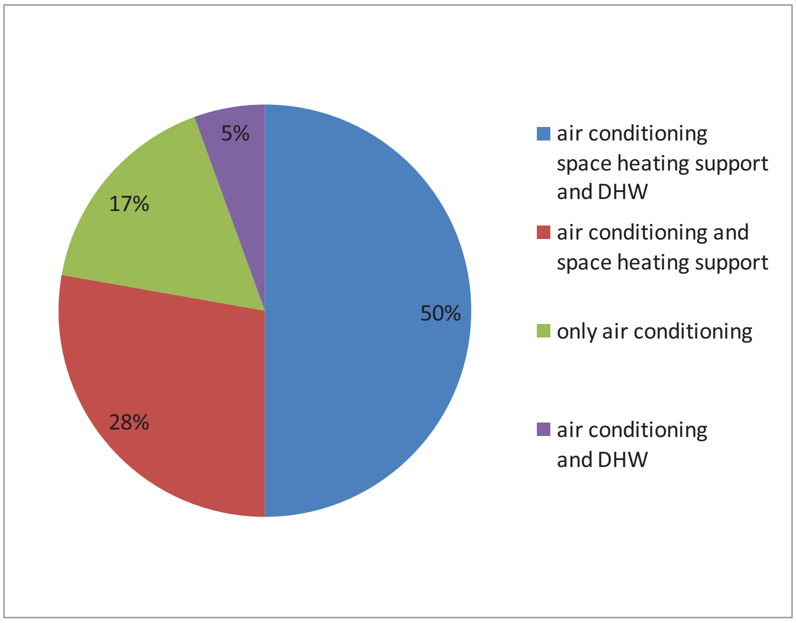


Figure 7: Use of solar heating and cooling plants

The back up strategies also varies significantly (see Figure 8). Most of evaluated plants use hot side back up. Based on the experiences of monitoring it is the most critical configuration in terms of achieving high primary energy savings.

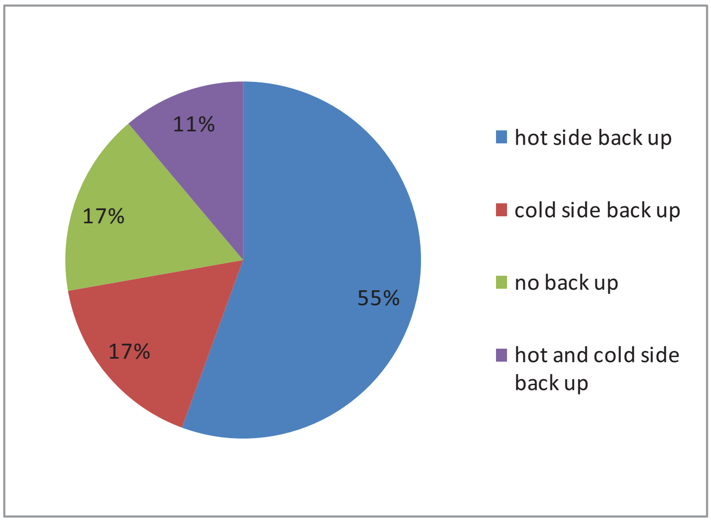


Figure 8: Back up strategy of solar heating and cooling plants

2.2.4 Planning and installation

The installation period of the evaluated plants varies as well. It was finished in some cases within three days and reaches in some cases up to three month (see Figure 9). It is quite interesting that it is already possible to install a solar heating and cooling plant within three days and a short installation time should be one goal for pre-engineered systems.

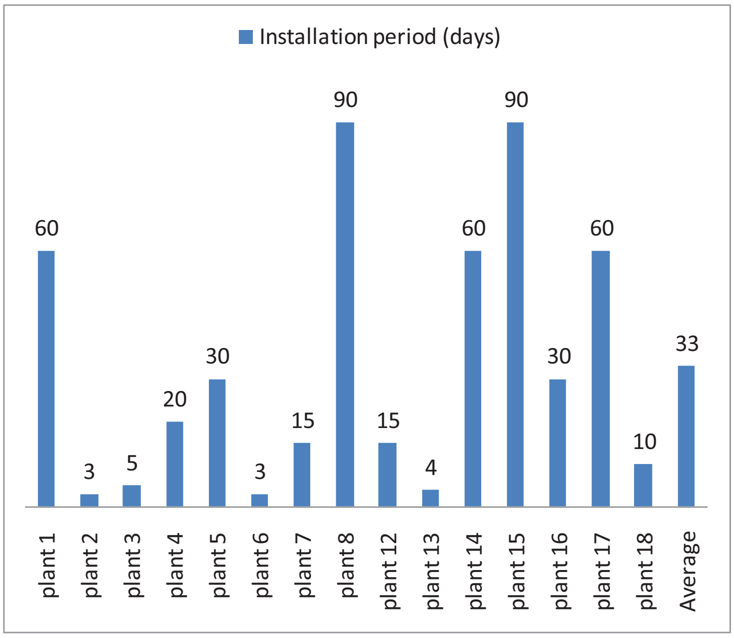


Figure 9: Installation period of plants

The number of people involved for installation for the evaluated plants shows that for most of the plants less than five people were necessary (see Figure 10). Also this can be used as an indicator for pre-engineered systems to avoid misunderstandings and time delay during installation due to the amount of involved people and companies.

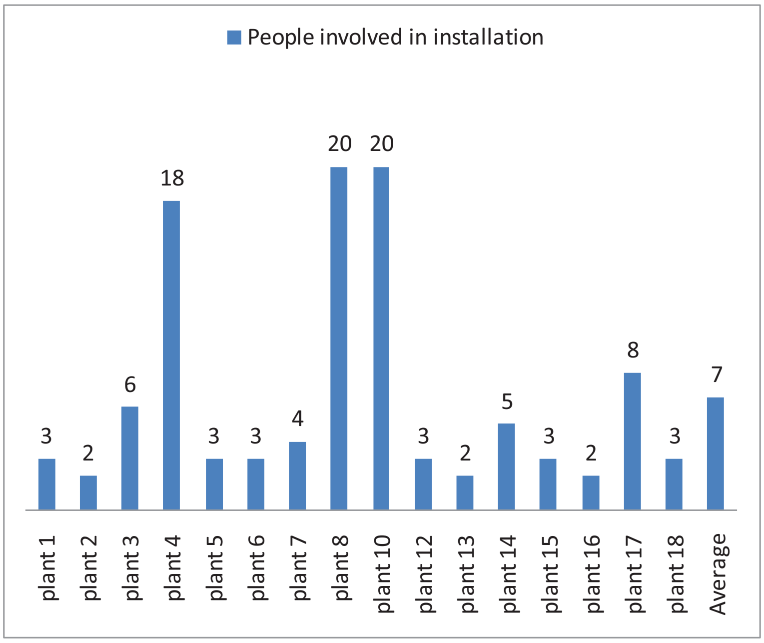


Figure 10: People involved for installation

In seven of the evaluated plants time delays during installation (see Figure 11) were recorded.

The reasons for such time delays mentioned are as follows:

- caused by plant owner: negotiations and ordering is regulated by public law and took quite long (delay 3 month)
- pump didn't work
- caused by discrepancies with installer
- caused by installer and subcontractor of installer (two weeks)
- different time period of material delivery
- difficulties in finding the desired pumps
- water damage caused by poorly compressed pipes

The list of reasons mentioned above show that most of the problems occurred were not particularly related to solar heating and cooling technology but to HVAC technology in general.

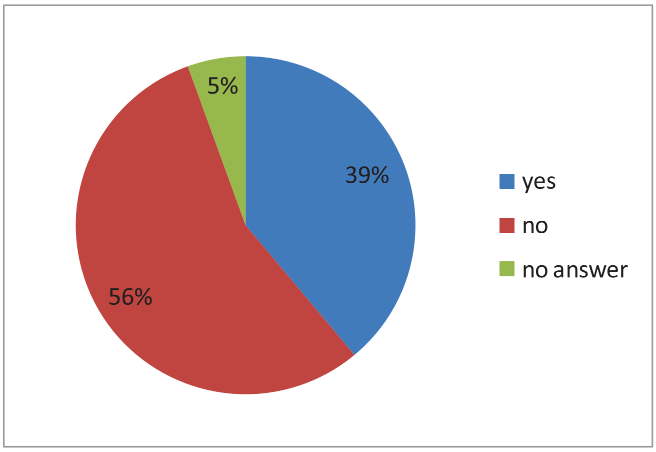


Figure 11: Delays during installation

In Table 1 the main problems mentioned during installation are listed as well as the solutions and the time for solving the problem. Pre-designed control strategies as well as quality standards for piping would have avoided most of the problems mentioned.

Problem	Solution	Time for solving the problem
programming was difficult	new pump and new program from supplier	2 years
discrepancies with low qualified installer	with calmness and a continuous discussions	for each case various days
air in the system	repeated blowing off	1 day
chiller without manual in the desired language	patience and motivation of the different actors	6 month
issue on the wireless connection of the monitoring system to the Internet	as above	as above
connection and settings of the control system	as above	as above
manufacturing defect of the chiller	on site by the installer and the planner	1 day

Table 1: Issues recorded during installation

2.2.5 System operation

In response to the question, if the plant owners are comfortable in the solar cooled area, most of the evaluated plants were perceived as comfortable (see Figure 12).

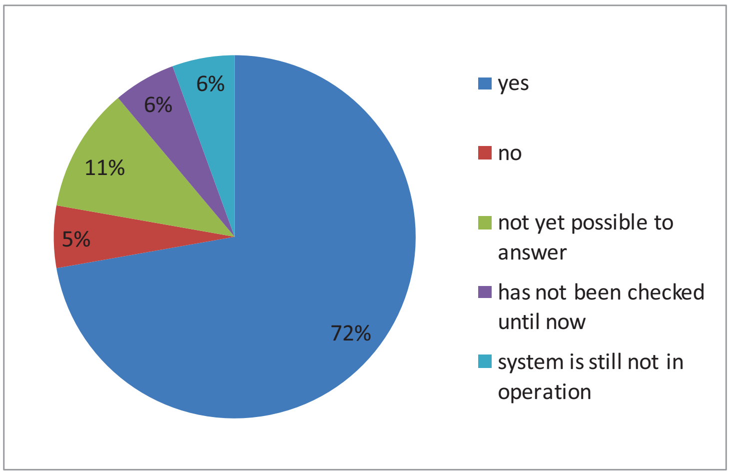


Figure 12: Indoor comfort

Positive experiences:

- first day of operation it was too cold; following the adjustment of room air temperature to a minimum of 24°C, the expectations of comfort were met;
- the operation is fully automatic, except the switching from cooling to heating season

Negative experiences:

- insufficient dimensioning of the cooling ceiling
- floor gets too cold for children to play, pre-cooling of the building is necessary
- the control system is not optimized
- to many manual adjustments necessary

2.2.6 Maintenance

Table 2 shows a summary of the maintenance items mentioned by the plant owners. No similarity between the mentioned maintenance actions was visible by comparing the answers. Only some of the plant owners had enough information about necessary maintenance actions. In half of the evaluated cases, hardly any maintenance items were listed. This situation gives the impression that in most cases the plant owners lack information about necessary maintenance actions to be undertaken. These recordings expose possible reasons for a non satisfactory energy performance of the plants.

Regular basis	Maintenance item
Once a week	check of main variables
Once a month	adjustment of software cleaning of the solar collectors
Twice a year	Chiller: evacuation Heat rejection: • emptying/filling • cleaning cooling/heating switch
Once a year	Whole system: • pressure • temperatures • pumps • probes checking (water, glycol) • elimination of air from the pipes

Table 2: List of maintenance actions mentioned

2.2.7 Costs

The specific investment costs, given by the plant owners, are in a very big range (see Figure 13). Compared to presently achievable average investment costs from 1.500 - 2.800 €/kW$_{cooling\ capacity}$ for medium to large size solar cooling plants between

70 - 400 $kW_{cooling\ capacity}$[1] most of the small-scale plants evaluated, are obviously much more expensive. Interestingly, there is one plant within the average values of investment costs of medium and large size solar cooling plants (plant 6) and one plant is much below these values (plant 13).

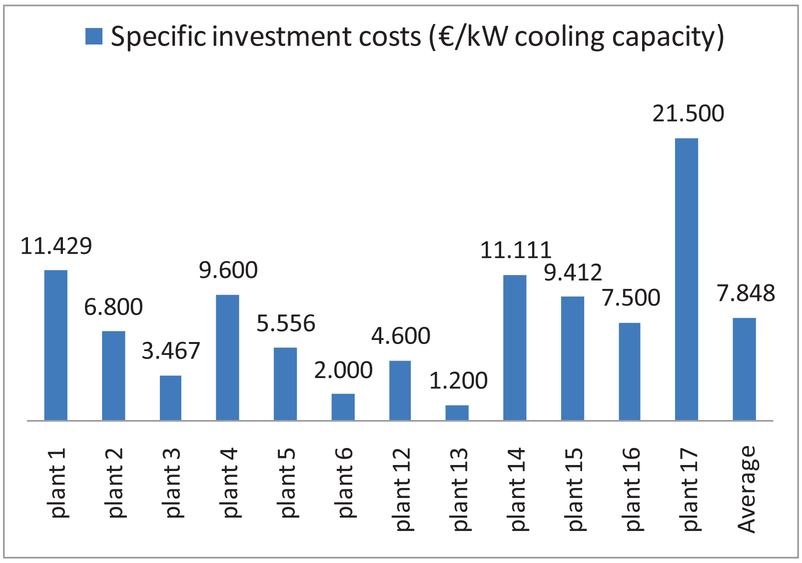


Figure 13: Specific investment costs of plants

2.3 Recommendations for installation, operation and maintenance

As already mentioned in chapter 1, the information gained from the end-user survey is transferred into the following list of recommendations to bring the idea of pre-engineered systems forward.

Installation:

- One person (company) with responsibility for the full installation and co-ordination of communication between the involved companies and the end-user
- Set up of installation time schedule which allows parallel work of different professionals and exact milestones, to shorten the time of installation
- Planning of quality checks during installation; A list of components, typically used for solar heating and cooling plants, should be set up and applied

[1]A. Preisler, T. Selke, L. Sisó, A. LeDenn, R. Ungerböck, ROCOCO – Reduction of Costs of Solar Cooling Systems, European Project in 6th Framework Program, Specific Support Action, Wien, 06/2008

Operation:

- Setting up commissioning plans according to the possible technology combinations for solar heating and cooling plants from which the accurate plan can be selected
- Thorough commissioning, checking of all relevant parts in the system (pumps, heat rejection, water treatment, valves, control, etc.) over a longer period of time (at least some days)
- Handbook for end-user, explaining the main operation modes, values to be monitored, maintenance plan, supplier contact information
- System controller should be pre-designed for the following purposes:
    - Smooth operation of various technology combinations for solar heating and cooling plants also considering the distribution side
    - Recording of monitoring data
    - Simple to modify for improving the plants
- Continuous recording of monitoring data of the plant (stored at system controller and/or sent to online data base at system supplier) and analyzed by system supplier should ensure high energy performance of the plant

Maintenance:

- Maintenance should be carried out according to a maintenance plan set up by the system supplier beforehand
- Typical maintenance items are shown in Table 2 but must be adjusted according to the actual application
- Company or companies responsible for maintenance should be defined at the latest for the beginning of plant operation

3 Interview Package Solution Provider

As already mentioned in chapter 1, interviews with companies which already offer package solutions for solar heating and cooling plants were carried out to investigate the pre-engineered status of the offered packages available. Furthermore, by comparing the different package solutions and also taking the experiences from monitoring of such systems into account, suggestions for high quality package solutions were set up.

3.1 Interview guideline

The interview guideline for package solution providers (see Appendix 3) was set up for the purpose of information collection about package definition (included components, control items, maintenance and monitoring) as well as design indicators. Furthermore, the components which have to be designed separately were investigated.

3.2 Results

Following results correspond to the feedback of five package solution providers of small-scale solar heating and cooling systems. The collected results are also presented in anonymous form, always referring to package solution provider one to five as “PSP 1” to “PSP 5”.

3.2.1 Design indicators

PSP 1 has developed a design tool for selection of the cooling package size, based on weather data, calculation of heat rejection parameters and/or temperature levels of hot water, cooling water or chilled water. PSP 2 estimated the package size from available pre-designed packages; Separated planning is necessary for the dimensioning of pumps and cold water distribution system. The packages of PSP 3, PSP 4 and PSP 5 were mainly custom-made by the package solution provider themselves or external planners.

3.2.2 Components included

Table 3 compiles the components included in different package solutions provided by the interviewed package solution providers. Differences occur in the variety of components offered, for example PSP 1 and PSP 3 can provide dry, hybrid and wet heat rejection units while PSP 4 only offers wet cooling towers. Also the range of different chiller manufacturers is larger from PSP 1 and PSP 4 than from others. Back up is only mentioned as part of the package by PSP 1 and PSP 3. PSP 5 does not include the three pumps around the chiller in the package, only the solar thermal pumps are included. Water treatment is only mentioned as part of the package by PSP 1 and PSP 4. Cold storage is given as an option from PSP 3 and PSP 4. All of the interviewed package solution providers, except PSP 4, offer one system controller for the whole system up to the cold storage mainly developed by themselves.

	PSP 1	PSP 2	PSP 3	PSP 4	PSP 5
ab/-adsorption chiller	✓	✓	✓	✓	✓
solar collectors	✓	✓	✓	✓	✓
solar pumps/heat exchanger	✓	✓	✓	✓	✓
hot water storage	✓	✓	✓	✓	✓
cold water storage	✓	✓	✓	✓	✓
pumps around chiller	✓	✓	✓	✓	×
piping and hydraulics	✓	✓	✓	✓	✓
heat rejection	✓	✓	✓	✓	✓
water treatment	✓	×	×	✓	×
back up	✓	×	✓	×	×
system controller	✓	✓	✓	×	✓

Table 3: Components included in package solutions

3.2.3 Control items included

PSP 1 offers a very sophisticated system controller, taking into account the sorption chiller, heat rejection, solar thermal system, several other heat sources (e.g. CHP unit, biomass, etc.), distribution system (four mixed heating/cooling circuits) and room conditions; pre-defined control strategies, referring to 43 million hydraulic options, available in the system controller.

Also PSP 2 system controller is part of the package, which takes the solar cooling system up to the cold water storage into account (not the water distribution system); pre-defined control strategies are stored at the system controller.

The system controller of PSP 3 is custom-made and used for the whole installation including back-up and distribution, if required. The control system runs the chiller, the heat rejection system, pumps of all the loops and the back-up.

PSP 4 offers two different controllers, which can communicate with each other; pre-defined control strategies are foreseen for applications with fan coils, chilled ceilings and sub zero cold production.

The system controller of PSP 5 is custom-made and controls the system up to hot storage and cold storage; except some parameters, such as minimum temperature of chiller, no control strategies are pre-defined.

Only the system controllers of PSP 1 and PSP 3 can take into account both, the distribution system to the cooled area (fan-coils, cooled ceiling, etc.) and the room conditions (temperature, humidity).

3.2.4 Maintenance included

All interviewed package solution providers, except PSP 5 offer maintenance as part of the full package implemented by them or a recommended specialized company; commissioning is part of all packages, implemented by the package solution providers. PSP 1 and PSP 2 have pre-defined maintenance plans according to the chosen package solution. PSP 4 has included a visual inspection once a year and check of pump running times by telemaintenance. PSP 5 just recommends to the plant owner to empty the heat rejection unit in winter and to refill it in summer.

3.2.5 Monitoring included

Table 4 shows the present status of monitoring services included in different package solutions of interviewed package solution providers. PSP 1 has put most effort into this item. In this case monitoring was included in the system controller and made accessible for the plant owner, either directly at the system controller or put into graphs via internet access to the system supplier.

	PSP 1	PSP 2	PSP 3	PSP 4	PSP 5
part of package	✓	×	✓	✓	×
included in system controller	✓	×	×	×	×
online data available	✓	×	×	×	×

Table 4: Monitoring included in package solution

PSP 3 and PSP 4 also offer monitoring as part of the package; PSP 4 integrates sensors into the machine for all three circuits; PSP 3 didn´t give any detailed information about the included monitoring. Monitoring is offered by PSP 1, PSP 2 and PSP 3 also to improve the systems.

3.3 High quality package solutions

Following services should be included by a package solution provider to achieve high quality solar heating and cooling plants:

- Design of the entire solar heating and cooling plant
- Thorough commissioning
- Handbook for end-users (including maintenance plan)
- Maintenance according to maintenance plan
- Monitoring data from main parts of the plant (solar thermal system, three circuits around chiller, heat rejection, storages, distribution to building); available on system controller or/and online data
- Improvement of the system (at least yearly reporting)

When selecting components, following focus should be given to achieve high energy efficient plants:

- Energy efficiency class of fans in heat rejection
- Sufficient water treatment system included in the package
- Energy efficiency class of pumps; variable speed pumps

As already mentioned, the degree of pre-fabrication varies from supplier to supplier. It is recommended to choose a package, which is as pre-engineered as possible. This way, it can be made sure that all components work well together. An important aspect is that the controller is included in the package and the control strategies, necessary to ensure proper operation of all system components, have been pre-defined by the manufacturer. Most manufacturers offer pre-defined control strategies, depending on the exact configuration of the system e.g. the temperature level in the chilled water circuit.

Not all packages on the market include a control system which manages, besides the solar thermal plant and the chiller, the heat and cold distribution systems. In these cases, a separate controller is needed. However, the overall control concept of the building often has a big influence on the performance of the entire system. As an example, the cold distribution system should take into account whether there is cold from the thermally driven chiller available or not. That shows that the design matching of the controller for cold production and the controller of the heat and cold distribution is an important issue. Therefore packages that include the control algorithms also for the distribution system are recommended.

4 Summary and conclusions

The end-user survey showed that many installed small-scale solar heating and cooling plants are not truly pre-engineered. This explains many of the problems with the installed plants mentioned in the survey.

For example, the evaluation of the installation period showed that some installations have much too long periods of time for installation and many people involved. The communication between the involved parties seemed to be difficult in some cases (chiller provider, solar thermal collector provider, installer, planner and owner). In some cases the commissioning didn’t happen at all. During operation, the range of work for the plant owners reaches from hardly anything to do (everything runs automatically) to daily adjustments. The answers given by the plant owners concerning maintenance show that only some plant owners were well informed about the main maintenance items of a solar cooling system; the rest mentioned only parts of the necessary maintenance items or nothing at all. In most of the evaluated plants, the average investment costs are too high for what is already possible to achieve.

Small-scale solar heating and cooling systems can be bought from specialized companies as complete packages, including the entire solar thermal plant with hydraulics, heat storage tank, absorption or adsorption chiller, heat rejection system, possibly cold storage tank and a control system of the whole installation. The degree of pre-fabrication varies from supplier to supplier. In some cases, all pumps are pre-defined depending on the flow rate and an assumed pressure drop, taking distances between components into account. In other cases, these components need to be sized specifically for each installed system. These pre-engineered packages are then connected to heat, cold and domestic hot water distribution systems.

Packages, available on the market, offer different cooling capacities. The number of small-scale thermally driven chillers available on the market is not vast, often the chosen chiller represents the starting point for design. The rest of the system is then pre-engineered to fit the chiller size. For the purpose of choosing the correct size of the package for a given application, an assessment of the cooling load of the building has to be carried out. System suppliers help the customer in the planning process or recommend charging a planning office, to identify the cooling load of the building.

Suppliers of such systems typically have affiliated installers that carry out the installation work. The commissioning is then done by the supplying companies themselves, to ensure proper installation and operation of the plant. Many companies offer monitoring and optimization processes of the system. Since this is a relatively new technology, this should systematically be used in order to avoid possible malfunctioning, minimize energy consumption and therefore operation costs of the system.

Maintenance is normally carried out by the system suppliers themselves or by an affiliated company typically at least once a year. However, some packages include a wet cooling tower for heat rejection, which requires following specific tasks: emptying of the water before the winter and refilling in the spring, water treatment management process (compulsory in some countries). With a pre-engineered system, it should not be necessary for the end-users to do any maintenance themselves.

.

5 List of figures

6 List of tables

7 Annexes

Appendix 1 – List of existing small scale solar heating and cooling plants

Appendix 2 – Questionnaires for owners of small-scale solar cooling plants

Appendix 3 – Questionnaire for package solution provider

Task 38
Solar Air-Conditioning and Refrigeration

D-A5: Appendix 1

List of existing small scale solar heating and cooling plants

Date: September 2009

Edited by Anita Preisler

Institution	**AIT – Austrian Institute of Technology, Energy Department**
Address	**Giefinggasse 2, A-1210 Vienna, Austria**
Phone	**+43 (0) 50550-6634**
Fax	**+43 (0) 50550-6613**
e-mail	**anita.preisler@ait.ac.at**

Country	Name/ Company	City	Application type	Technology cold production Absorption kW	Adsorption kW	Flat plate m²	Solar thermal technology Vacuum Tube m²	CPC m²	Parabolic Through m²	Hot water l	Storage Ice storage l	PCM kg	Year of completion	Divers
1 Australia	CSIRO Energy Centre	Newcastle	office building	17,5					60	-			2008	Yasaki (Water/LiBr)
2 Australia	Milk Factory	Wyong	industry		10		34,8			1.500			2009	Invensor (Water/Zeolith)
3 Austria	Weinbaubetrieb Peitler	Leutschach	wine cellar	10			100			4.000			2003	Pink (Ammonia/Water)
4 Austria	Fa. SOLID	Graz	office building	2		7				-			2003	Kunze (Ammonia/Water)
5 Austria	Fa. SOLution	Sattledt	office building	15		40				2.000			2005	EAW (Water/LiBr)
6 Austria	Privathaus Jungreithmayr	Sattledt	residential building		5,5	38				3x2.000			2007	Sortech (Water/Silicagel)
7 Austria	Fa. Bachler	Gröbming	training center	9		46				3x1.500			2007	Pink (Ammonia/Water)
8 Austria	Fa. Manschein	Gaweinstal	office and industrial building		7,5	32				2.000			2008	Sortech (Water/Silicagel)
9 Austria	Fa. Kreuzroither	Schörfling	office and industrial building		15	160				1.500 cold water			2008	Sortech (Water/Silicagel)
10 Austria	Fa. SOLID	Graz	office building	17		57,5				2000			2008	Yasaki (Water/LiBr)
11 Austria	Municipality of Vienna Section 34	Vienna	office building		7	32,4				2000			2009	Sortech (Water/Silicagel)
12 China	Public building	Hong Kong	Public building	4,5		?				?			2007	Rotartica (Water/LiBr)
13 China	residential building	Beijing	residential building		7,5		20,5			1.000			2008	Sortech (Water/Silicagel)
14 Egypt	Hostel	Sharm el Sheikh	Hostel	13,5		?				?			2008	Rotartica (Water/LiBr)
15 France	Laboratory	Lannion	laboratory set up	4,5		?				?			2006	Rotartica (Water/LiBr)
16 France	Laboratory	Moret sur Loing	laboratory set up	4,5		?				?			2007	Rotartica (Water/LiBr)
17 France	Résidence du Lac / SIEL	Maclas	residential building	10			24			500			2007	Sonnenklima (Water/LiBr)
18 France	SOLACLIM	Perpignan	laboratory set up		7,5	25				?			2008	Sortech (Water/Silicagel)
19 Germany	FH Ingolstadt	Ingolstadt	test plant	15		30				2.000			2004	EAW (Water/LiBr)
20 Germany	WEGRA-Anlagenbau GmbH	Westenfeld	Office building	15		50				2.000			2004	EAW (Water/LiBr)
21 Germany	Schüco International KG	Bielefeld	office building	15		45				1.500			2005	EAW (Water/LiBr)
22 Germany	Private company	Braunschweig	offices	4,5		?				?			2006	Rotartica (Water/LiBr)
23 Germany	Conergy AG	Rangsdorf	office building	15		50				800			2006	EAW (Water/LiBr)
24 Germany	Fa. Solar Next	Rimsting	office building	15		37	34			2.000			2007	EAW (Water/LiBr)
25 Germany	Sonnenalternativtechnik	Strukum	officebuilding / workshop	15		45				1.000			2007	EAW (Water/LiBr)
26 Germany	QS-Grimm GmbH	Gutach	Office / production rooms	15		45				1.500			2007	EAW (Water/LiBr)
27 Germany	ZAE Bayern	Garching	laboratory set up	10		20				2.000		2.500 Calcium Hexahydrat	2007	Sonnenklima (Water/LiBr)
28 Germany	Fraunhofer ISE	Freiburg	canteen		5,5	20				?			2007	Sortech (Water/Silicagel)
29 Germany	Citrin Solar	Moosburg	office building		5,5	?				?			2007	Sortech (Water/Silicagel)
30 Germany	TKIZ GmbH	Barby	office building	15		45				1.000			2008	EAW (Water/LiBr)
31 Germany	radiological practice	Berlin	radiological practice	10			40			1000			2008	Sonnenklima (Water/LiBr)
32 Germany	residential building	Wiesloch	residential building		7,5	36				2.000			2008	Sortech (Water/Silicagel)
33 Germany	Office and apartment building	Bad Waldsee	Office and apartment building		7,5	?				?			2008	Sortech (Water/Silicagel)
34 Germany	residential building	Alzenau	residential building		7,5	24				2.000			2008	Sortech (Water/Silicagel)
35 Germany	Fa. Schneider	Megesheim	office building		5,5	?				?			2008	Sortech (Water/Silicagel)
36 Germany	Fa. Zink	Unterensingen	office building		5,5	?				?			2008	Sortech (Water/Silicagel)
37 Germany	Solarzentrum	Wietow	office building / showroom	15		45				1.000			2009	EAW (Water/LiBr)
38 Germany	Siemens Erlangen	Erlangen	test plant	15		50				1.000			2009	EAW (Water/LiBr)

39	Germany	University Stuttgart	Stuttgart	laboratory set up	10		32			-	2x500	2009	ITW-Stuttgart (Ammonia/Water)
40	Germany	Raiffeisenbank Miesbach	Miesbach	bank	2x10		100			7.500		2009	Pink (Ammonia/Water)
41	Germany	Technologiezentrum Köthen	Köthen	office building	15			79		?		?	?
42	Italy	Private company	Arezzo	Private company	4,5		?			?		2007	Rotartica (Water/LiBr)
43	Italy	Private company	Perugia	Private company			?			?		2007	Rotartica (Water/LiBr)
44	Italy	Paradigma	Darzo	office building	15		45			1.500		2006	EAW (Water/LiBr)
45	Italy	ISSA	Milano	office building	4,5			19,5		1.800		2007	Rotartica (Water/LiBr)
46	Italy	CRA-VIV	Pescia	green house		7,5		19		1.500		2008	Sortech (Water/Silicagel)
47	Italy	Universitiy Palermo	Palermo	test plant	12			40,5		2.000		2008	Pink (Ammonia/Water)
48	Italy	Schüco International	Sarmeola	office building	15		45			1.000		2008	EAW (Water/LiBr)
49	Italy	Kindergarden Polimi	Milano	kindergarden		8,5	25,7			1.000		2009	Sortech (Water/Silicagel)
50	Italy	Ebner Solartechnik	Eppan	office building	15		150			2 x 8.000		2009	EAW (Water/LiBr)
51	Lichtenstein	Baugemeinschaft Rietli	Schaan	Wohnungen	15			50		1.500		2008	EAW (Water/LiBr)
52	Malta	Retirement home Kalkhara	Kalkhara	retirement home	10		38			1.000		2008	Pink (Ammonia/Water)
53	Malta	Headquater of Eco Group	Kordin	office building	10		30,5	7		400		2009	Pink (Ammonia/Water)
54	Portugal	Private company	Lisbon	Private company	4,5		?			?		2007	Rotartica (Water/LiBr)
55	Portugal	Private company	Chegancas	Private company	4,5		?			?		2008	Rotartica (Water/LiBr)
56	Portugal	AoSol	Lisbon	office building	8				36	1.000		2009	AO SOL (Ammonia/Water)
57	Spain	Private House	Barcelona	residential building	4,5		?			?		2006	Rotartica (Water/LiBr)
58	Spain	Sports Centre	Zaragoza	Sports Centre	4,5		?			?		2006	Rotartica (Water/LiBr)
59	Spain	Private Company	Sarriguren-Navarra	Offices	4,5		?			?		2006	Rotartica (Water/LiBr)
60	Spain	Public Building	Puerto Lumbreras-Murcia	Sport centre	4,5		?			?		2006	Rotartica (Water/LiBr)
61	Spain	Private Company	Sevilla	Offices	4,5		?			?		2006	Rotartica (Water/LiBr)
62	Spain	Private House	Madrid	residential building	4,5		?			?		2006	Rotartica (Water/LiBr)
63	Spain	Private Company (Food)	Noreña-Asturias	Private Company (Food)	4,5		?			?		2006	Rotartica (Water/LiBr)
64	Spain	Private House	Bilbao	residential building	4,5		?			?		2006	Rotartica (Water/LiBr)
65	Spain	School	Segovia	School	4,5		?			?		2006	Rotartica (Water/LiBr)
66	Spain	School	Santurtzi	School	4,5		?			?		2006	Rotartica (Water/LiBr)
67	Spain	Private hospital	Tenerife	hospital	4,5		?			?		2006	Rotartica (Water/LiBr)
68	Spain	Retirement home	Las Palmas-Canarias	Retirement home	4,5		?			?		2006	Rotartica (Water/LiBr)
69	Spain	Private Company	Sevilla	Offices	9		?			?		2006	Rotartica (Water/LiBr)
70	Spain	Private Company	Basauri-Bizkaia	Offices	9		?			?		2006	Rotartica (Water/LiBr)
71	Spain	Public Building	Errenteria-Gipuzkoa	Public Building	4,5		?			?		2006	Rotartica (Water/LiBr)
72	Spain	Private Company	Basauri-Bizkaia	Offices	13,5		?			?		2006	Rotartica (Water/LiBr)
73	Spain	Trainning Centre	Guadalajara	Trainning Centre	4,5		?			?		2006	Rotartica (Water/LiBr)
74	Spain	Trade Exhibition Center	Gijón	Trade Exhibition Center	4,5		?			?		2006	Rotartica (Water/LiBr)
75	Spain	Experimental Installation	Arganda del Rey-Madrid	Experimental Installation	4,5		?			?		2006	Rotartica (Water/LiBr)
76	Spain	Private House	Villajoyosa-Alicante	residential building	4,5		?			?		2007	Rotartica (Water/LiBr)
77	Spain	School	Molina de Segura	School	4,5		?			?		2007	Rotartica (Water/LiBr)
78	Spain	Private House	Madrid	residential building	4,5		?			?		2007	Rotartica (Water/LiBr)

79	Spain	Private House	Olivares-Sevilla	residential building	4,5	?		?	2007	Rotartica (Water/LiBr)
80	Spain	Private Company (Food)	Estepa-Sevilla	Private Company (Food)	4,5	?		?	2007	Rotartica (Water/LiBr)
81	Spain	Private House	Brunete-Madrid	residential building	4,5	?		?	2007	Rotartica (Water/LiBr)
82	Spain	Private House	Zuia-Araba	residential building	4,5	?		?	2007	Rotartica (Water/LiBr)
83	Spain	University	Yecla-Murcia	Common Zones	4,5	?		?	2007	Rotartica (Water/LiBr)
84	Spain	Camping	Aznalcazar-Sevilla	Camping	4,5	?		?	2007	Rotartica (Water/LiBr)
85	Spain	Private Company	Gazteiz-Araba	Show Room	4,5	?		?	2007	Rotartica (Water/LiBr)
86	Spain	Technological Research Centre	Miñano-Araba	Technological Research Centre	4,5	?		?	2007	Rotartica (Water/LiBr)
87	Spain	University	Santander	University	4,5	?		?	2007	Rotartica (Water/LiBr)
88	Spain	Office	Sanlucar Lamayor-Sevilla	Office	4,5	?		?	2007	Rotartica (Water/LiBr)
89	Spain	Museum	Granada	Museum	4,5	?		?	2007	Rotartica (Water/LiBr)
90	Spain	Public Building	La Moncloa-Madrid	Public Building	18	?		?	2007	Rotartica (Water/LiBr)
91	Spain	Private House	Puerto Santamaría-Cádiz	residential building	4,5	?		?	2007	Rotartica (Water/LiBr)
92	Spain	University Rooms	Univ. Cartagena	University Rooms	4,5	?		?	2007	Rotartica (Water/LiBr)
93	Spain	Tourism Office	Matalascañas-Huelva	Tourism Office	4,5	?		?	2007	Rotartica (Water/LiBr)
94	Spain	Public Building	Molina de Segura-Murcia	Public Building	4,5	?		?	2007	Rotartica (Water/LiBr)
95	Spain	Sports Centre	La Carlota-Córdoba	Sports Centre	4,5	?		?	2007	Rotartica (Water/LiBr)
96	Spain	Private company	Fuente Alamo	Private company	9	?		?	2007	Rotartica (Water/LiBr)
97	Spain	Public Building (Room)	Tivissa-Tarragona	Public Building (Room)	9	?		?	2007	Rotartica (Water/LiBr)
98	Spain	Public Building	León	Public Building	4,5	?		?	2007	Rotartica (Water/LiBr)
99	Spain	University Laboratory	Tarragona	University Laboratory	4,5	?		?	2007	Rotartica (Water/LiBr)
100	Spain	Domolab	Miñao	showroom	4,5	22,7		-	2008	Rotartica (Water/LiBr)
101	Spain	Private Company (Animals)	La Muela-Zaragoza	Private Company (Animals)	4,5	?		?	2008	Rotartica (Water/LiBr)
102	Spain	Private Company	Espinardo-Murcia	Private Company	4,5	?		?	2008	Rotartica (Water/LiBr)
103	Spain	Biodiesel Installation	Kaparroso-Navarra	Biodiesel Installation	9	?		?	2008	Rotartica (Water/LiBr)
104	Spain	Golf (shop and bar)	La Manga-Murcia	Golf (shop and bar)	4,5	?		?	2008	Rotartica (Water/LiBr)
105	Spain	Retirement home	Terrassa-Barcelona	Retirement home	9	?		?	2008	Rotartica (Water/LiBr)
106	Spain	Private House	Girona	residential building	18	?		?	2008	Rotartica (Water/LiBr)
107	Spain	Multi Family House	Laguna de Duero-Valladolid	Multi Family House	4,5	?		?	2008	Rotartica (Water/LiBr)
108	Spain	Hotel	Lanzarote	Hotel	4,5	?		?	2008	Rotartica (Water/LiBr)
109	Spain	Public Building	Bullas-Murcia	Public Building	4,5	?		?	2008	Rotartica (Water/LiBr)
110	Spain	Public Building	San José de la Vega-Murcia	Public Building	4,5	?		?	2008	Rotartica (Water/LiBr)
111	Spain	Public Building	Alcantarilla-Murcia	Public Building	4,5	?		?	2008	Rotartica (Water/LiBr)
112	Spain	School	Portugalete	School	4,5	?		?	2008	Rotartica (Water/LiBr)
113	Spain	Private Company (Food)	Noreña-Asturias	Private Company (Food)	4,5	?		?	2008	Rotartica (Water/LiBr)
114	Spain	Private House	Brunete-Madrid	residential building	4,5	?		?	2008	Rotartica (Water/LiBr)
115	Spain	Experimental Installation	Arganda del Rey-Madrid	Experimental Installation	4,5	?		?	2008	Rotartica (Water/LiBr)
116	Spain	University	Jaén	University	4,5	?		?	2008	Rotartica (Water/LiBr)
117	Spain	National Institute of Airospacial Techniques INTA	Huelva	laboratory set up	10	25	18	?	?	?
118	Switzerland	Weingut Schloss Salenegg	Mainfeld	wine cellar	15	70		3.200	2005	EAW (Water/LiBr)

119	Syria	Private House	Damascus	residential building	9	?	?	2008	Rotartica (Water/LiBr)
120	UK	University	Cardif	University	4,5	?	?	2007	Rotartica (Water/LiBr)
121	United Arab Emirates	Private house	Ras Al Khaimah	residential building	4,5	?	?	2008	Rotartica (Water/LiBr)
122	USA	Private house	Cincinnaty	residential building	4,5	?	?	2007	Rotartica (Water/LiBr)

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning
and Refrigeration

D-A5: Appendix 2

Questionnaires for Owners of Small-Scale Solar Cooling Plants

Languages: English, German, Spanish

Date: October 2009

Edited by Anita Preisler

Institution	**AIT – Austrian Institute of Technology, Energy Department**
Address	**Giefinggasse 2, A-1210 Vienna, Austria**
Phone	**+43 (0) 50550-6634**
Fax	**+43 (0) 50550-6613**
e-mail	**anita.preisler@ait.ac.at**

Telephone Interview ☐
Personal Interview ☐
Filled in by plant owner ☒

To
Organisation: AIT - Austrian Institute of Technology
Person: Anita Preisler
e-mail: anita.preisler@ait.ac.at

Date:

Dear owner of a small scale solar cooling system,

We are a group of scientists and engineers who are working in the frame of the International Energy Agency in the field of solar air conditioning. For scientific and statistic purposes we set up this questionnaire which helps to collect basic data of small scale solar cooling systems. The data are evaluated scientifically and treated confidentially. It is not necessary to answer all the questions; leave the check box open if you don't know the answer. The questionnaire is divided in four main parts: general questions – technical questions about the system - questions about the economical (cost) situation and at the end questions about design plans and measuring data. The estimated time necessary to fill in the questionnaire is 20 minutes - thank you very much for your help!
If you want to know more about this project, please visit
http://www.iea-shc.org/task38/index.html

General data: (location, building type, building orientation, user profile ....)

Name:

Object: ☐ single family house ☐ multi family house ☐ office building ☐ other

Cooled useful area: ca. m²

Cooled room type(s):

Year of construction (building):

County: Town: Postal Code:

Street name and number:

Telephone no.: Mobile phone no.:

E-mail:

General questions:

1. **What/who induced you to the installation of a solar cooling plant? (Multiple denominations possible)**

   ☐ recommendation of other plant owners ☐ fair visit ☐ advice of consultant

   ☐ installer ☐ energy conservation ☐ other

2. **Did the plant fulfil your expectations?** ☐ yes ☐ no

   If no: why?

3. **When was the plant installed?** date (year)

4. **How do you judge the plant quality?**

   ☐ very good ☐ good ☐ moderate ☐ bad

   Comments:

System configuration:

5. **Technology used?**

Cold side:

☐ Absorption ☐ Adsorption ☐ DEC (Desiccant evaporative cooling)

cooling capacity: kW

Hot side:

☐ Flat plate collectors ☐ evacuated tube collectors ☐ concentrated collectors

size: m^2

6. **What is the plant used for?**

☐ air conditioning ☐ domestic hot water preparation ☐ space heating support

☐ other

7. **What kind of cold distribution system is used?**

☐ ventilation ☐ fan coils ☐ radiant floor ☐ ceiling

☐ TABS (thermally activated building structure), which kind

☐ other

8. **Do you have a hot storage?**

☐ yes ☐ no Litres?

9. **Do you have a cold storage?**

☐ yes ☐ no Litres?

10. **Where is the chiller located?**

☐ technical room ☐ under the roof ☐ on the roof ☐cellar

☐ other

11. Where is the hot storage located?

☐ technical room ☐ under the roof ☐ on the roof ☐cellar

☐ other

12. Where are the solar panels installed?

☐ on the roof ☐ garden ☐ other

orientation of collectors:

inclination of collectors:

13. **How far is the hot storage from the solar panels?** meters

14. Do you have a backup system? ☐ yes ☐ no

If yes: What kind of backup do you have?

☐ compression chiller ☐ ground water usage ☐ earth collector

☐ hot side back up ☐ other

15. Why did you choose this system configuration?

☐ offered by installer ☐ recommendation of other plant owners

☐ other

Planning and Installation:

16. **Who did the planning for the system?** Name of company:

☐ local installer ☐ company specialised on renewable technologies

☐ other

17. **How long did it take to install the plant?** days

18. **Who was in charge of the installation?** Name of company:

☐ local installer ☐ company specialised on renewable technologies

☐ other

19. **How many people were involved in the installation?** people

20. **Were there any delays during installation?** ☐ yes ☐ no
    If yes: what were these delays caused by?

21. **What kinds of problems were encountered during installation?**

22. **How were these problems solved?**

23. **How long did it take to solve the problem?**

24. **Did this lead to extra costs?** ☐ yes ☐ no

If yes: how much? €

System operation:

25. **Are you comfortable in the solar cooled area?** ☐ yes ☐ no

Comments:

26. **How convenient is the operation of the system? What do you have to do as the user?** Please describe any problems you may have.

27. **Do you know when the solar thermal system is working and when the backup system is working?**

☐ yes ☐ no

Comments:

If yes: How do you know which system is working?

☐ shown at the local controller ☐ via internet access to my system supplier

☐ other:

28. **Is there any kind of monitoring of the performance of your system?**

☐ yes, please describe

☐ no

Maintenance:

What maintenance actions are necessary on a regular basis and how often?

Maintenance item	Every day	Once a week	Once a month	Once a year	Less than once a year	Other
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐

Comments:

29. Who does this maintenance?

☐ owner himself ☐ installer ☐ specialised company on maintenance

☐ other

Comments:

30. Have there been any problems with the solar cooling system since the installation was completed?

☐ yes ☐ no

If yes: what kind of problems have occurred?

31. Have these problems been solved?

☐ yes ☐ no

Comments:

32. Was it easy to solve these problems?

☐ yes ☐ no

Comments:

33. How could these problems have been avoided?

Costs:

34. **How much did the solar cooling plant cost?** €

35. Have you saved money since you installed your solar cooling system?

☐ yes ☐ no

Comments:

36. **How much have you saved?** €

Comments:

37. Is it what you expected? Are you satisfied?

☐ yes ☐ no

Comments:

38. Would you recommend a solar cooling plant to a friend?

☐ yes ☐ not at the moment

Comments:

Do you have following information?

Cooling load dimensioning of each room	☐ yes	☐ installer	☐ no
Installation plan of pipes	☐ yes	☐ installer	☐ no
Operation instructions and/or technical manual	☐ yes	☐ installer	☐ no
Photos of the plant installation	☐ yes	☐ installer	☐ no
Would you hand out/send us this information?	☐ yes	☐ no	

Are you interested in further investigations of your solar cooling plant?

☐ I am very interested in a plant investigation

☐ I do not have special interest – however my plant is available for further investigation

☐ Unfortunately my plant is not available for further investigation

Telefoninterview ☐
Persönliches Interview ☐
Von Anlagebesitzer selbst ausgefüllt ☒

An
Organisation: AIT - Austrian Institute of Technology
Person: Anita Preisler
e-mail: anita.preisler@ait.ac.at

Datum:

Lieber Besitzer einer solaren Kühlanlage,

Wir sind eine Gruppe von ForscherInnen und IngenieurInnen die im Rahmen der Internationalen Energieagentur (IEA) im Bereich solare Kühlung arbeiten. Dieser Fragebogen wurde mit dem Hintergrund erstellt um für wissenschaftliche und statistische Zwecke weltweite Basisinformationen zu installierten solaren Kühlanlagen kleiner Leistung zu sammeln. Die gesammelten Daten werden mit wissenschaftlichen Methoden ausgewertet und vertraulich behandelt. Es ist nicht unbedingt notwendig alle Fragen zu beantworte; wenn Sie die Antwort nicht wissen lassen Sie die Frage aus. Der Fragebogen ist in vier Teile unterteilt: (1) allgemeine Fragen, (2) technische Fragen zur Anlagenkonfiguration, (3) Fragen zu Planung, Installation und Betrieb und (4) Kosten. Das Ausfüllen des Fragebogens sollte nicht mehr als 20 Minuten dauern. Danke für Ihre Unterstützung!
Mehr Informationen zum gegenständlichen Projekt finden Sie unter:
http://www.iea-shc.org/task38/index.html

Allgemeine Daten: (Standort, Gebäudetyp, Nutzung, …)

Name:

Objekt: ☐ Einfamilienhaus ☐ Mehrfamilienhaus ☐ Bürogebäude

☐ Sonstiges

Gekühlte Nutzfläche: ca. m²

Raumarten gekühlt:

Erbauungsjahr Gebäude:

Land: Stadt/Gemeinde: Postleitzahl:

Straße, Nr:

Telefonnr.: e-mail:

Allgemeine Fragen:

1. Was oder wer hat Sie zur Installation einer Solaren Kühlanlage veranlasst? (Mehrere Nennungen möglich)

☐ Empfehlung eines anderen Anlagenbesitzer ☐ Messebesuch

☐ Vorschlag eines technischen Beraters (Energieberater, Haustechnikplaner)

☐ Installateur ☐ Energieeinsparung ☐ Sonstiges

2. **Hat die Anlage Ihre Erwartungen erfüllt?** ☐ Ja ☐ Nein

Wenn nicht, warum?

3. **Wann wurde die solare Kühlanlage installiert?** Datum (Jahr)

4. Wie würden Sie die Qualität der solaren Kühlanlage beurteilen?

☐ Sehr gut ☐ Gut ☐ Mittel ☐ Schlecht

Kommentare:

Anlagenkonfiguration:

5. Welche Technologie wurde eingesetzt?

Kalte Seite:

☐ Absorption ☐ Adsorption ☐ DEC (Desiccant evaporative cooling)

Installierte Kälteleistung: kW

Heiße Seite:

☐ Flachkollektoren ☐ Vakuumröhrenkollektoren ☐ Konzentrierende Kollektoren

Kollektorfläche: m²

6. Wofür wird die Anlage genutzt?

☐ Klimatisierung ☐ Warmwasserbereitung ☐ Heizungsunterstützung

☐ Sonstiges

7. Was für ein Energieverteilsystem wird eingesetzt?

☐ Lüftung ☐ Fan-Coils ☐ Fußbodenheizung ☐ Decken

☐ Bauteilaktivierung, welche Art?

☐ Sonstiges

8. Gibt es einen Warmwasserspeicher?

☐ Ja ☐ Nein Liter?

9. Gibt es einen Kaltwasserspeicher?

☐ Ja ☐ Nein Liter?

10. Wo steht die Kältemaschine?

☐ Technikraum ☐ unterm Dach ☐ am Dach ☐ Keller

☐ Sonstiges

11. Wo steht der Heißwasserspeicher?

☐ Technikraum ☐ unterm Dach ☐ am Dach ☐ Keller

☐ Sonstiges

12. Wo sind die Solarkollektoren montiert?

☐ am Dach ☐ Garten ☐ Sonstiges

Orientierung der Kollektoren:

Neigung der Kollektoren:

13. **Wie weit ist der Heißwasserspeicher von den Solarkollektoren entfernt?** Meter

14. Gibt es ein Back-up System? ☐ Ja ☐ Nein

Wenn ja: Welche Art von Back-up System?

☐ Kompressionskältemaschine ☐ Grundwassernutzung ☐ Erdregister

☐ Back-up auf Heißwasserseite ☐ Sonstiges

15. Warum haben Sie diese Anlagenkonfiguration gewählt?

☐ Angebot des Installateurs ☐ Empfehlung anderer Anlagenbesitzer

☐ Sonstiges

Planung und Installation:

16. **Wer führte die Planung der Anlage durch?** Name des Unternehmens:

☐ lokaler Installateur ☐ auf erneuerbare Energien spezialisiertes Unternehmen

☐ Sonstiges

17. **Wie lange hat die Installation der Anlage gedauert?** Tage

18. **Wer war für die Installation verantwortlich?** Name des Unternehmens:

☐ lokaler Installateur ☐ auf erneuerbare Energien spezialisiertes Unternehmen

☐ Sonstiges

19. **Wie viele Personen waren bei der Installation der Anlage involviert?**

Personen

20. **Gab es Verzögerungen währen der Installationsphase?** ☐ Ja ☐ Nein

Wenn ja: was hat diese Verzögerungen verursacht?

21. Welche Probleme gab es während der Installation?

22. Wie wurden diese Probleme gelöst?

23. Wie lange hat es gedauert diese Probleme zu lösen?

24. **Hat das zu zusätzlichen Kosten geführt?** ☐ Ja ☐ Nein

Wenn ja: wie viel? €

Anlagenbetrieb:

25. **Fühlen Sie sich wohl in den gekühlten bereichen?** ☐ Ja ☐ Nein

Kommentare:

26. Wie leicht ist die Handhabung der Anlage im Betrieb? Was müssen Sie als Nutzer tun?
Bitte beschreiben Sie eventuelle auftretende Schwierigkeiten die Sie mit der Anlage haben.

27. Wissen Sie wann die Solaranlage arbeitet und wann das das Back-up System?

☐ Ja ☐ Nein

Kommentare:

Wenn ja: Wie wissen Sie, dass welches System gerade im Betrieb ist?

☐ zeigt der lokale Regler ☐ über Internetzugang zum Anlagenanbieter

☐ Sonstiges:

28. Gibt es ein Monitoring der Energieeffizienz der Anlage?

☐ Ja, bitte beschreiben Sie:

☐ Nein

Wartung:

Welche Wartungsarbeiten sind regelmäßig notwendig und in welchem Intervall?

Wartungsarbeiten	**Jeden Tag**	**Einmal in der Woche**	**Einmal im Monat**	**Einmal im Jahr**	**Weniger als einmal im Jahr**	**Sonstiger Intervall:**
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐

Kommentare:

29. Wer macht diese Wartungsarbeiten?

☐ Besitzer selber ☐ Installateur

☐ auf erneuerbare Energien spezialisiertes Unternehmen ☐ Sonstiges

Kommentar:

30. Gab es irgendwelche Probleme mit der solaren Kühlanlage seit Fertigstellung der Installation?

☐ Ja ☐ Nein

Wenn ja: welche Art von Probleme sind aufgetreten?

31. Wurden diese Probleme gelöst?

☐ Ja ☐ Nein

Kommentar:

32. War es leicht diese Probleme zu lösen?

☐ Ja ☐ Nein

Kommentar:

33. Wie könnten diese Probleme vermieden werden?

Kosten:

34. **Wie viel hat die solare Kühlanlage gekostet?** €

35. **Haben Sie seit die Anlage im Betrieb ist Geld eingespart?**

☐ Ja ☐ Nein

Kommentar:

36. **Wie viel haben Sie eingespart?** €

Kommentar:

37. Entspricht es dem was Sie erwartet haben? Sind Sie zufrieden?

☐ Ja ☐ Nein

Kommentar:

38. Wüden Sie eine solare Kühlanlage an Freunde weiterempfehlen?

☐ Ja ☐ Nicht im Moment

Kommentar:

Haben Sie folgende Unterlagen/Informationen?

Kühllastberechnung von jedem gekühlten Raum ☐ Ja ☐ Installateur ☐ Nein

Dokumentation der Installationen ☐ Ja ☐ Installateur ☐ Nein

Betriebsanleitung und/oder technisches Manual ☐ Ja ☐ Installateur ☐ Nein

Photos der durchgeführten Installationen ☐ Ja ☐ Installateur ☐ Nein

Würden Sie uns diese Unterlagen zur Verfügung stellen? ☐ Ja ☐ Nein

Sind Sie an weiteren Untersuchungen an Ihrer solaren Kühlanlage interessiert?

☐ Ich bin sehr interessiert an weiteren Untersuchungen

☐ Ich habe kein spezielles Interesse – ich stelle meine Anlage trotzdem für weitere Untersuchungen zur Verfügung

☐ Leider kann ich meine Anlage für weitere Untersuchungen nicht zur Verfügung stellen

Entrevista Telefónica ☐
Entrevista Personal ☐
Cumplimentada por el dueño de la instalación ☐

Enviada a:
Organización: CARTIF
Personal de Contacto: Luis Ángel Bujedo Nieto
Correo electrónico: luibuj@cartif.es

Fecha:

Estimado propietario de una instalación de pequeño tamaño de refrigeración solar:

Somos un grupo de científicos e ingenieros que trabajamos dentro del marco de la Agencia Internacional de la Energía (AIE) en el campo de la producción de aire acondicionado mediante energía solar. Con fines estrictamente estadísticos y científicos, hemos elaborado el presente cuestionario con el que poder recoger información básica sobre las instalaciones de refrigeración solar de pequeña potencia (inferior a 20 kW). Los datos recogidos, serán evaluados científicamente y tratados de forma confidencial. No es necesario responder a todas las cuestiones, deje la casilla sin marcar si no conoce la respuesta.

El cuestionario está dividido en cuatro bloques: Cuestiones de carácter general – Cuestiones técnicas sobre el sistema – Cuestiones relativas a los resultados económicos y finalmente aspectos relativos al diseño y la medida de datos. El tiempo aproximado para la realización del cuestionario es de 20 minutos. Muchas gracias por su colaboración

Si desea más información acerca de la presente iniciativa, visite la página http://www.iea-shc.org/task38/index.html.

Datos de Carácter General (localización, tipo de edificio, orientación del edificio, perfil de usuario, etc.)

Nombre:

Tipo: ☐ Vivienda unifamiliar ☐ Viviendas en bloque ☐ Edificio de oficinas
☐ Otra

Superficie Acondicionada: m²

Tipo de espacio acondicionado (habitado, pasillos, etc.):

Año de construcción del edificio:

País: Ciudad: Código Postal:

Calle y número:

Teléfono.: Móvil.:

Correo electrónico:

Cuestiones de carácter General sobre la instalación:

1. **¿Por qué se animó a realizar la instalación de refrigeración solar (Son posibles varias respuestas)?**

☐ Recomendado por otros usuarios de instalaciones semejantes ☐ visita a una feria
☐ Recomendado por consultores ☐ Instalador ☐ Ahorro energético
☐ Otros

2. **¿La instalación ha cumplido sus expectativas?** ☐ Sí ☐ No

¿En caso de no, por qué?

3. **¿Cuando se realizó la instalación?** Fecha (año)

4. **¿Cómo valora la calidad de la instalación?**

☐ Muy buena ☐ Buena ☐ Suficiente ☐ Baja

Comentarios:

Configuración del sistema:

5. **¿Tecnología empleada?**

Producción de frío:

☐ Absorción ☐ Adsorción ☐ DEC (Refrigeración desecante evaporativa)

Potencia nominal de refrigeración: kW

Producción de calor (Instalación solar):

☐ Captadores planos ☐ Captadores de vacío ☐ Captadores de concentración

Superficie: m²

6. **¿Usos de la instalación?**

☐ Aire Acondicionado ☐ Agua Caliente Sanitaria (ACS) ☐ Calefacción

☐ Otras

7. **¿Qué sistema de distribución se emplea?**

☐ Aire ☐ fan coils ☐ Suelo radiante ☐ Techo radiante

☐ Cerramientos activos (Estructuras activadas térmicamente), ¿Qué tipo?

☐ Otros

8. ¿Dispone de almacenamiento de agua caliente (solar)?

☐ Sí ☐ No ¿Litros?

9. ¿Dispone de almacenamiento de frío?

☐ Sí ☐ No ¿Litros?

10. ¿Dónde está colocada la enfriadora?

☐ Sala de máquinas ☐ Bajo techo ☐ sobre la cubierta ☐Sótano

☐ Otro

11. ¿Dónde está el almacenamiento de calor?

☐ Sala de máquinas ☐ Bajo techo ☐ Sobre la cubierta ☐Sótano

☐ Otro

12. ¿Donde están los captadores solares?

☐ Sobre la cubierta ☐ Jardín ☐ Otro

Orientación de los captadores:

Inclinación de los captadores:

13. **¿A qué distancia está la acumulación solar de los captadores?** metros

14. ¿Dispone de sistema auxiliar? ☐ Sí ☐ No

¿Qué sistema? (solo en caso de Sí)

☐ Sistemas de compresión ☐ Aguas subterráneas ☐ Geotermia cerrada

☐ Sistema de calentamiento auxiliar ☐ Otros

15. ¿Por qué eligió la configuración de su sistema?

☐ Sugerida por el instalador ☐ ¿Recomendada por dueños de otras instalaciones?

☐ Otro

Diseño e instalación:

16. **¿Quién diseñó la instalación?** Compañía:

☐ Instalador local ☐ Compañía especializada en energías renovables

☐ Otro

17. **¿Cuanto tardó en diseñar la instalación?** días

18. **¿Quién realizó la instalación?** Compañía:

☐ Instalador local ☐ Compañía especializada en energías renovables

☐ Otro

19. **¿Cuántas personas montaron la instalación?** personas

20. **¿Hubo retrasos durante la ejecución?** ☐ Sí ☐ No
    (En caso de sí, cuanto tiempo y porqué)

21. ¿Qué tipo de problemas surgieron durante el montaje de la instalación?

22. ¿Cómo se resolvieron?

23. ¿Cuanto tiempo se tardó en encontrar la solución?

24. **¿Tuvo un sobrecoste?** ☐ Sí ☐ No

(En caso de sí, cuanto) €

Operación del sistema:

25. **¿Se encuentra confortable en el área acondicionado por la instalación?**
☐ Sí ☐ No

Comentarios:

26. ¿Cómo opera el sistema? ¿Qué labores rutinarias realiza?
Por favor, describa cualquier problema que se haya presentado.

27. ¿El sistema le permite conocer fácilmente, cuando el aporte de calor es solar y cuando es con el sistema auxiliar?

☐ Sí ☐ No

Comentarios:

(Cuando sí, ¿Cómo conoce la fuente de aporte?

☐ A través del control local ☐ A través de una conexión a Internet de mi proveedor

☐ Otro:

28. ¿Hay algún sistema de medida y registro del comportamiento de la instalación?

☐ Sí, Por favor, describala

☐ No

Mantenimiento:

¿Qué acciones regulares de mantenimiento son necesarias, y con qué frecuencia?

Acción de mantenimiento	Diariamente	Semanalmente	Mensualmente	Anualmente	Menos de una vez al año	Otra
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐
	☐	☐	☐	☐	☐	☐

Comentarios:

29. ¿Quién realiza este mantenimiento?

☐ El propietario ☐ instalación ☐ Empresa de mantenimiento

☐ Otro

Comentarios:

30. ¿Ha habido problemas en la instalación de refrigeración solar desde que fue puesta en marcha?

☐ Si ☐ No

(En caso de sí) ¿Qué tipo de problemas?

31. ¿Han sido resueltos?

☐ Sí ☐ No

Comentarios:

32. ¿Fue sencillo resolver dichos problemas?

☐ Sí ☐ No

Comentarios:

33. ¿Como han sido resueltos?

Costes:

34. **¿Cuanto costó la realización de la instalación?** €

35. ¿Ha tenido ahorros desde que la instalación se puso en marcha?

☐ Sí ☐ No

Comentarios:

36. **¿Cuanto se ha ahorrado?** €

Comentarios:

37. ¿Ha cubierto sus expectativas?, ¿está satisfecho?

☐ Sí ☐ No

Comentarios:

38. ¿Recomendaría una planta de refrigeración sola a un amigo?

☐ Sí ☐ No de momento

Comentarios:

¿Dispone de la siguiente información?

Dimensionado de la carga para cada habitación	☐ Sí	☐ Instalador	☐ No
Plano de instalación de tuberías	☐ Sí	☐ Instalador	☐ No
Instrucciones de operación y/o manual técnico del instalador	☐ Sí	☐ Instalador	☐ No
Fotos de la planta	☐ Sí	☐ Instalador	☐ No
Estaría dispuesto a entregarnos/enviarnos esta información?	☐ Sí	☐ No	

Está interesado en nuevas estudios sobre su instalación

☐ Tengo mucho interés en que alguien estudie mi instalación.

☐ No tengo especial interés, sin embargo mi instalación está disponible para estudiarla.

☐ Desafortunadamente, mi planta no está disponible para ningún tipo de estudio

Task 38
Solar Air-Conditioning and Refrigeration

D-A5: Appendix 3
Questionnaire for Package Solution Provider

Date: June 2010

Edited by Anita Preisler

Institution	AIT – Austrian Institute of Technology, Energy Department
Address	Giefinggasse 2, A-1210 Vienna, Austria
Phone	+43 (0) 50550-6634
Fax	+43 (0) 50550-6613
e-mail	anita.preisler@ait.ac.at

Questions to Package Solution Providers

1. Design indicators
   - How to choose the size of the package? (planer is recommended, rule of thumb, …)
   - What do you recommend your customers?

2. Which components are part of the package?

3. Is the controller part of the package?

4. Does the controller only take the chiller into account or also following parts:
   - Solar thermal system
   - Distribution system to cooled area (fan-coils, cooled ceiling, etc.)
   - Room conditions (temperature, humidity)

5. Do you have pre-defined control strategies for different framework conditions and applications?

6. Which service is supplied by your company and what has to be done by any partner (installer/planer/..)?

7. What has to be sized separately?
    - E.g. pump for heat rejection loop depends on distance between cooling tower and chiller

8. Specific requirements (space in technical room and for heat rejection, water treatment)

9. Maintenance
    - What type of maintenance does the end user have to do? How often?
    - What type of maintenance needs to be done by a specialized company? How often?

10. Do you offer Monitoring as part of the package?

Task 38
Solar Air-Conditioning and Refrigeration

State of the art on existing solar heating and cooling systems

A technical report of subtask B

Date: 2009 November 12

By W. Sparber, A. Napolitano, G. Eckert[1] and A. Preisler[2]

[1]Institution	Institute for Renewable Energy, EURAC Research of Bolzano, Italy
Address	Viale Druso 1, 39100 Bolzano, Italy
Phone	+39 0471 055 651
Fax	+39 0471 055 699
e-mail	assunta.napolitano@eurac.edu

[2]Institution	AIT Austrian Institute of Technology
Address	Giefinggasse 2, 1210 Vienna, Austria
Phone	+43 50550 6634
Fax	+43 50550 6613
e-mail	anita.preisler@ait.ac.at

Table of Contents

1. Introduction

Within the subtask B of the "Task 38 - Solar Air Conditioning and Refrigeration", an overview has been carried out on thermally driven chillers and solar thermal technologies used in realized systems. The aim of this overview is to collect information on existing solar heating and cooling systems in order to derive a first identification of proven design solutions in terms of selection of technologies and dimensioning in relationship with location, final use and size of the building. The analysis of existing design solutions can help planners in the first steps of the decision making by addressing them towards the identification of optimal design solutions for solar heating and cooling applications.

Such activity has been carried out in the work package B1 in the framework of subtask B about solar cooling systems with overall thermal driven cooling power largely higher than 20 kW.

Collection of data on existing solar cooling systems has also been carried out in the work package A 5 of the subtask A concerning small solar cooling applications (overall thermal driven cooling power below 20 kW).

The present report summarizes the results derived from both the activities and working teams but goes in detail with special features of large scale systems as they are custom made systems, hence they are not as standard as package solutions for small applications and much more information could be found.

2. Research approach

Two lists of existing solar heating and cooling systems have been created, one for small scale systems and one for large scale systems.

For each system the following general data have been collected:

- name of the building project,
- location,
- final use,
- status (working, monitored,…).

Concerning technical aspects the following data have been collected:

- type and cooling capacity of the thermally driven chillers;
- type of solar collectors and their gross area;
- the capacity of energy storage on cold and hot side;
- the type and the power of heat and cold back up systems.

The data about existing solar cooling plants have been collected through a survey intended for:

- task 38 participants responsible for single systems,
- owner institutions and
- design and installation companies.

Table 1 summarizes the content of the survey.

Tab. 1 General and technical aspects selected to describe single solar cooling installations.

<table>
<tr><th colspan="8">Survey On Existing Solar Heating And Cooling Installations</th></tr>
<tr><td rowspan="4">General data</td><td colspan="7">Name of the building/project</td></tr>
<tr><td colspan="7">Location</td></tr>
<tr><td colspan="7">Final use</td></tr>
<tr><td colspan="7">Status of the SHC system</td></tr>
<tr><td colspan="8"></td></tr>
<tr><td></td><td>Components</td><td colspan="5">Technologies</td><td>Unit</td></tr>
<tr><td rowspan="5">Technical data</td><td>Thermal Driven chiller (TDC)</td><td>Absorption</td><td colspan="2">Adsorption</td><td colspan="2">DEC solid</td><td>DEC liquid</td><td>kW or m³/hr</td></tr>
<tr><td>Solar Thermal Collectors (STC)</td><td>Flat Plate (FP)</td><td>Evacuate Tube (ETC)</td><td>Compound Parabolic (CPC)</td><td>Air (Air)</td><td>Parabolic Through (PTC)</td><td>m² (gross area)</td></tr>
<tr><td>Storage</td><td colspan="5">Water, PCM, ICE…</td><td>l</td></tr>
<tr><td>Heat back up</td><td colspan="5">Gas boilers, heat pumps, district heating…</td><td>kW</td></tr>
<tr><td>Cold back up</td><td colspan="5">Compression chiller, heat pump…</td><td>kW</td></tr>
</table>

A first amount of data has been derived from other international projects like IEA-Task 25 [1], RoCoCo [2] and SACE [3].

In spite of the effort spent in collecting information, detailed data are still missing, especially for the small installations which are often private residential installations where acquiring data is difficult. In fact, for small scale systems most data have been collected thanks to manufacturers which counts the chillers sold per year. For this reason, quite complete information is available on the type and size of installed thermal driven chillers but most left required data are missing.

3. General overview on installed systems

To the knowledge of the authors, 113 large scale solar cooling systems and 163 small scale systems have been installed worldwide, eventually including systems which are currently not in operation. 254 installations are located in Europe, 13 in Asia, mostly in China and Japan, 4 in America (3 in USA and 1 in Mexico), 3 in Australia and 2 in Africa (Egypt and South Africa). Figure 3.1 shows the distribution of the worldwide number of installations on Countries, classified in small or large scale.

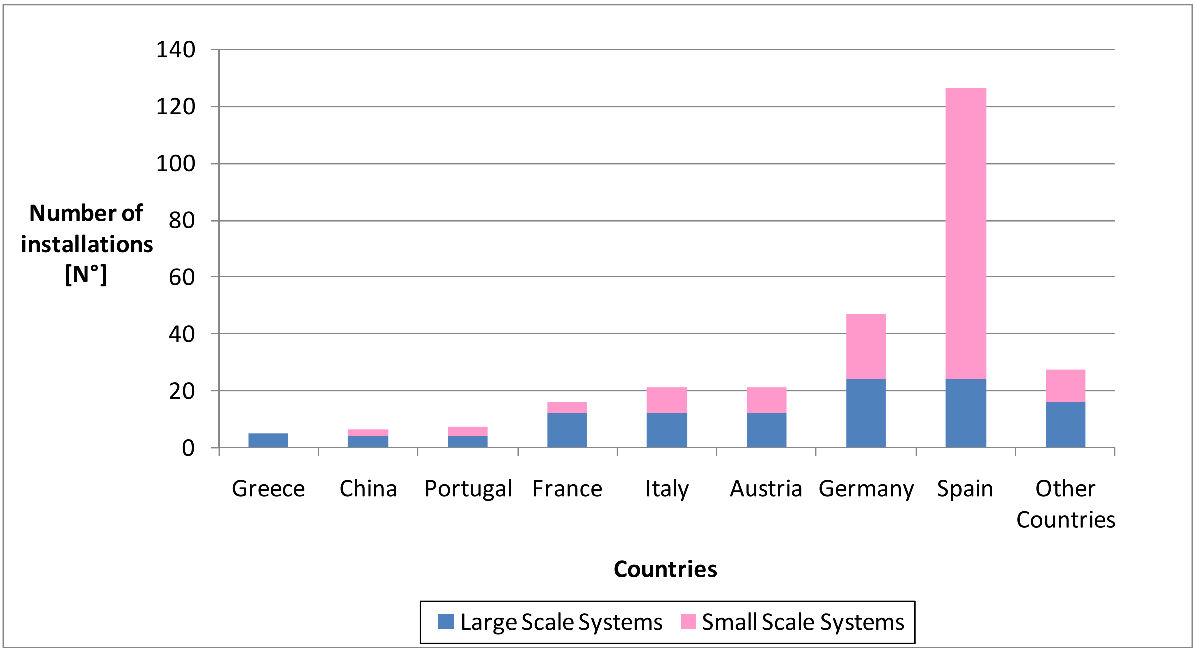


Figure 3.1 Total number of solar heating and cooling systems installed in different countries and classified in small or large scale systems. "Other Countries" include: Armenia, Australia, Belgium, Denmark, Egypt, Japan, Kosovo, Lichtenstein, Malta, Mexico, Netherlands, Singapore, South Africa, Switzerland, Syria, Turkey, UK, United Arab Emirates and USA.

Most installations are dedicated to office buildings as shown in Figure 3.2 and Figure 3.3, but 28% of the overall small scale installations serve residential buildings.

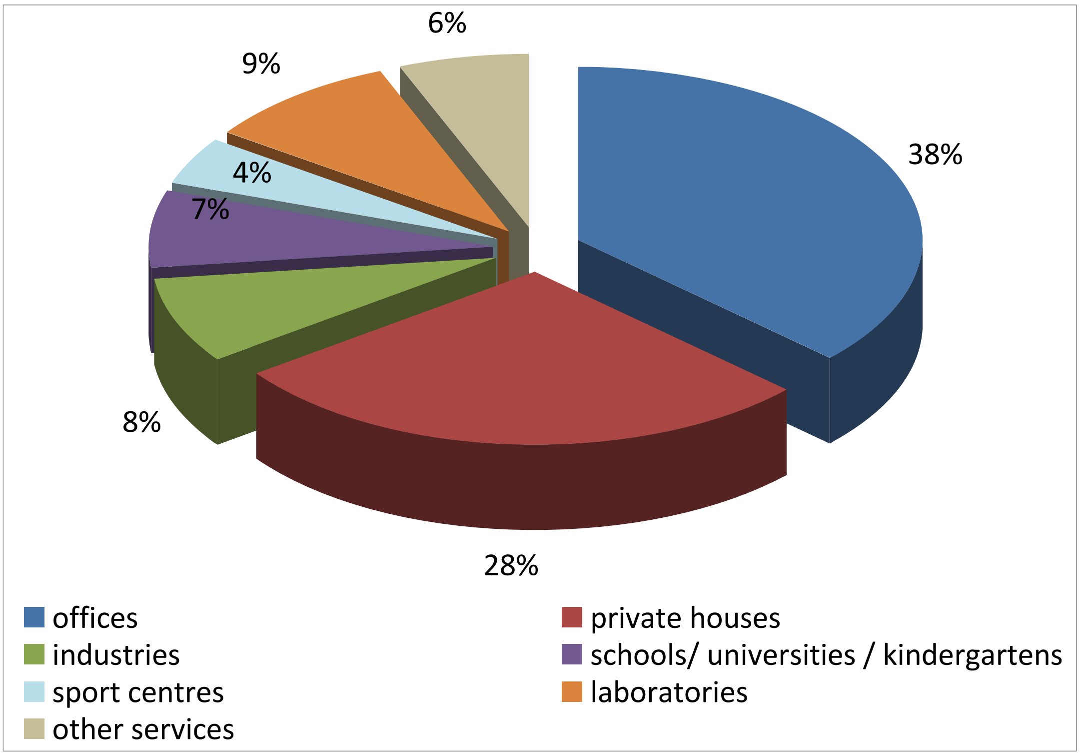


Figure 3.2 Use of small scale solar heating and cooling installations. ("Other services" include: schools, show rooms, hotels, etc)

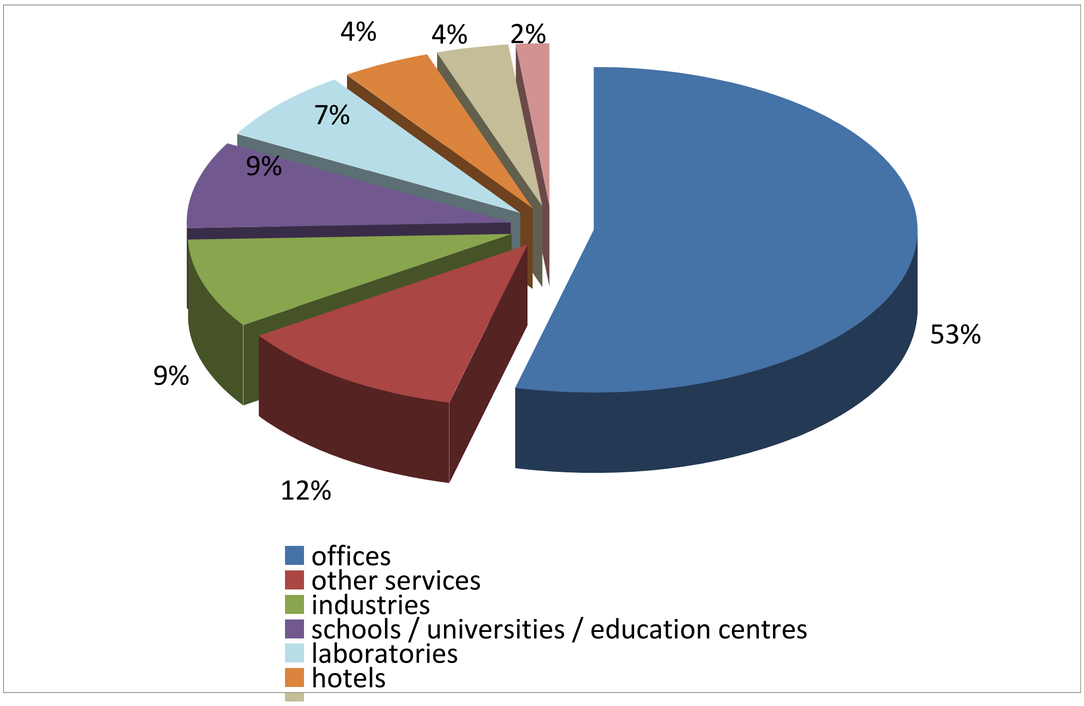


Figure 3.3 Use of large scale solar heating and cooling installations. ("Other services" include: hospitals, canteen, sport center, etc)

The overall cooling capacity of thermally driven chillers assisted by solar energy calculated on 268 systems amounts to 15.7 MW: 24.4% of it is installed in Spain, 19,5% in Germany and 17.4% in Italy 7 (Figure 3.4). 14.1 MW of such cooling capacity goes to large scale systems and 1.6 MW to small scale systems.

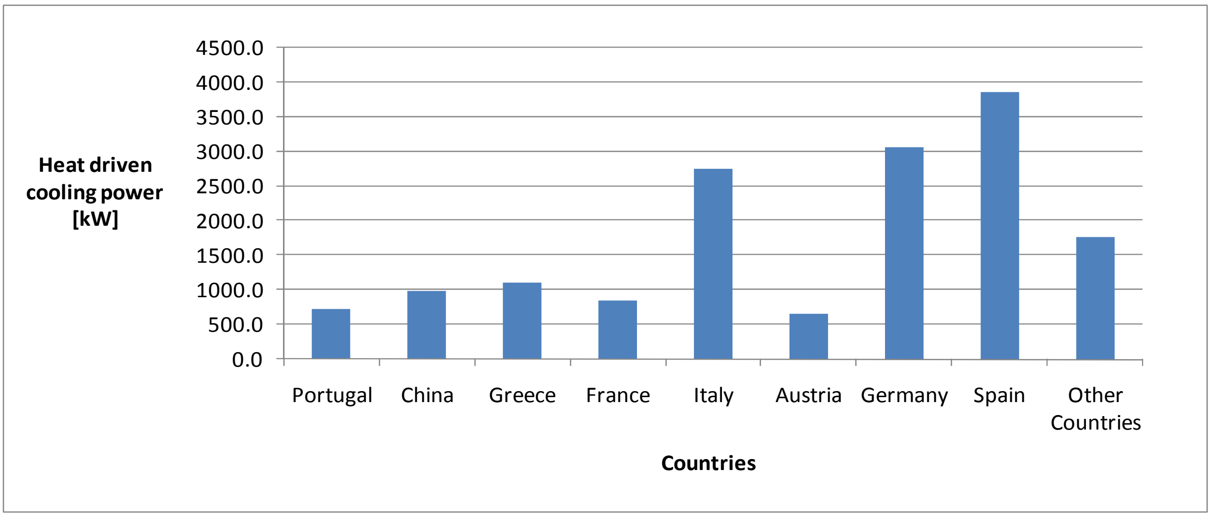


Figure 3.4 Cooling power of thermally driven chillers assisted by solar energy installed in different world Countries. "Other Countries" include the cooling power installed in Australia, Belgium, Denmark, Egypt, Kosovo, Lichtenstein, Malta, Mexico, Netherlands, Singapore, Switzerland, Syria, Turkey, UK, United Arab Emirates and in one US plant. As the cooling capacity of DEC systems is often expressed in m³/hr, information on kW installed in 5 DEC based systems (2 Austrian, 2 Italian and one Armenian) is missing.

Within 269 installations, the technology of thermally driven chillers most used is based on absorption principle (Figure 3.5 and Figure 3.6) whereas DEC systems are only used in large applications.

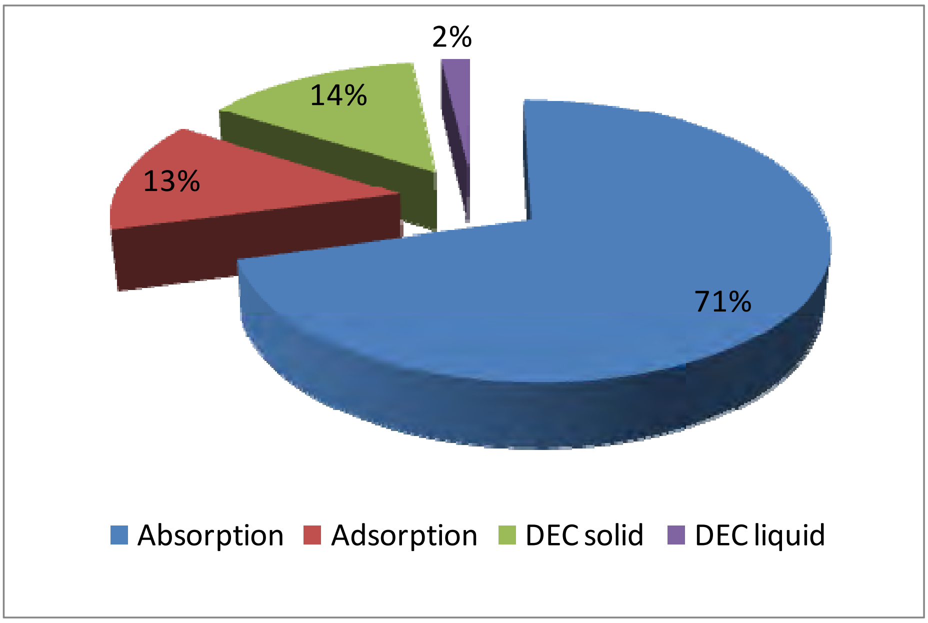


Figure 3.5 Percentage of use of different technologies for thermally driven chillers within 113 large scale systems.

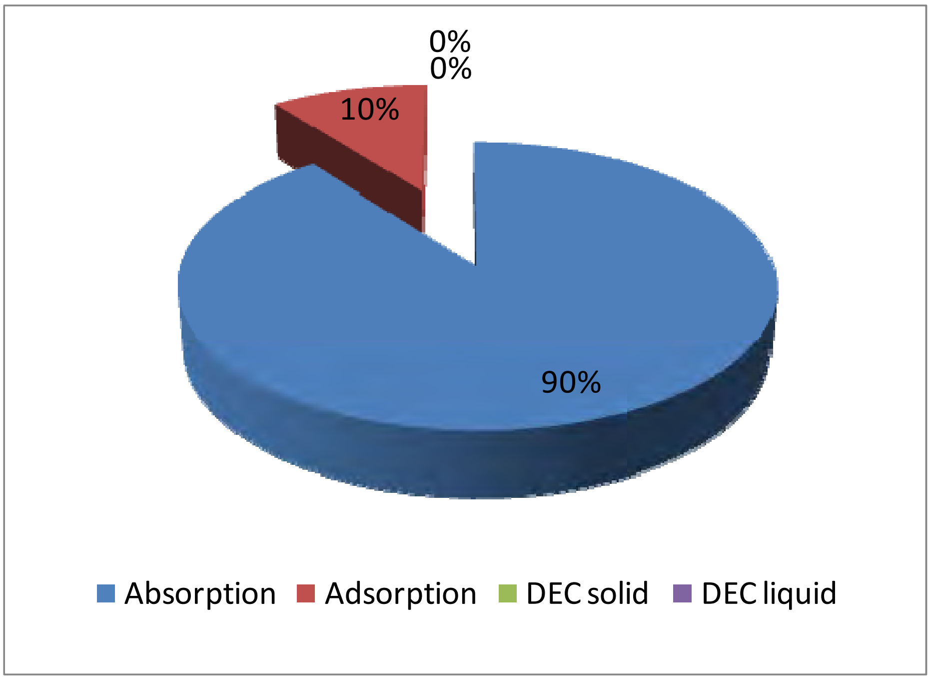


Figure 3.6 Percentage of use of different technologies for thermally driven chillers within 156 small scale systems.

Within around 105 small scale solar cooling systems based on absorption, the chillers manufactured by Rotartica with 4.5 kW are present in 50% of the installations, whereas Sorthech is the only manufacturer providing adsorption chillers in 16 systems.

For large scale systems no data of manufacturers are available and the nominal cooling capacities installed are so different within the existing installations that it is not possible recognizing special manufacturers, except one: 18 large scale installations are probably equipped with Yazaki chillers with a power of 35 kW.

4. Technical data and statistics on large scale systems

Solar thermal collectors and thermally driven cooling systems

A focus on large scale solar cooling systems shows that the technology selected for thermally driven cooling systems can vary country by country. For instance, adsorption chillers are largely used in Germany whereas Italy and Spain feature a large adoption of absorption chillers as shown in Figure 4.1.

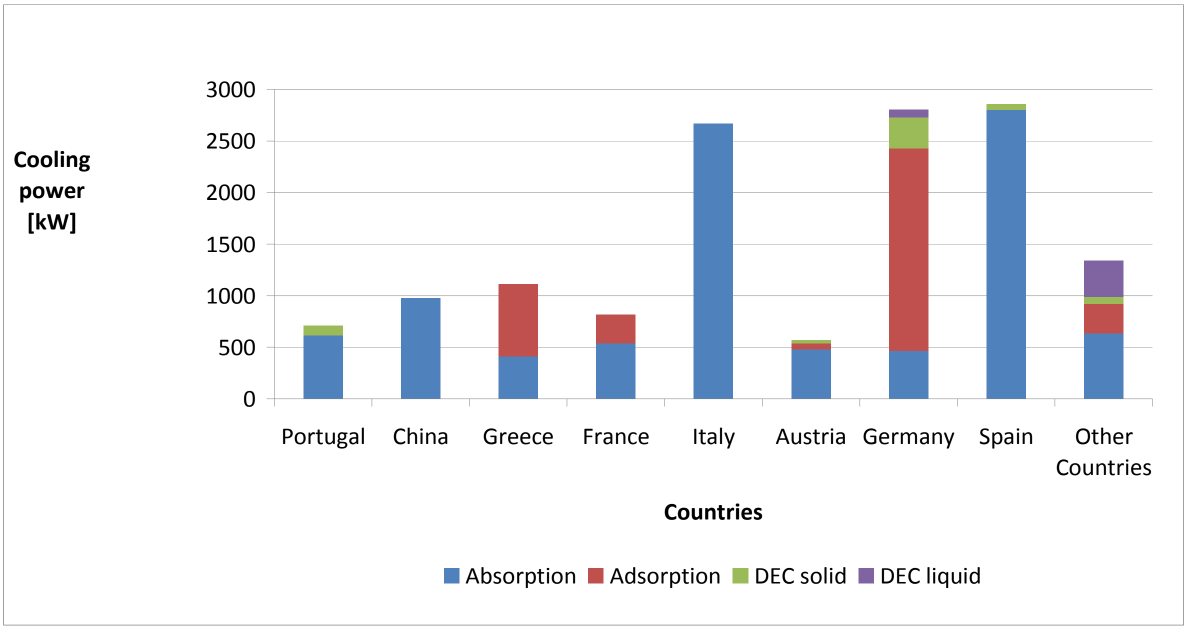


Figure 4.1 World wide distribution of the cooling power are assisted by solar energy. The type of thermally driven chillers applied in the different countries is highlighted. "Other Countries" include:, Australia, Belgium, Denmark, Kosovo, Mexico, Netherlands, Singapore, South Africa, Switzerland, Turkey and one US installations. Data on kW installed in 2 "DEC solid" systems in Austria and two in Italy are missing.

Flat Plate (FP) collectors and Evacuated Tube Collectors (ETC) are the technology mostly used in large scale solar cooling assisted systems, as shown in Figure 4.2, whereas Compound Parabolic Collectors (CPC), Air Collectors (Air) and Parabolic Through Collectors (PTC) are respectively present in 9, 5 and 3 installations (PTC are used in Australia, in Turkey and in USA).

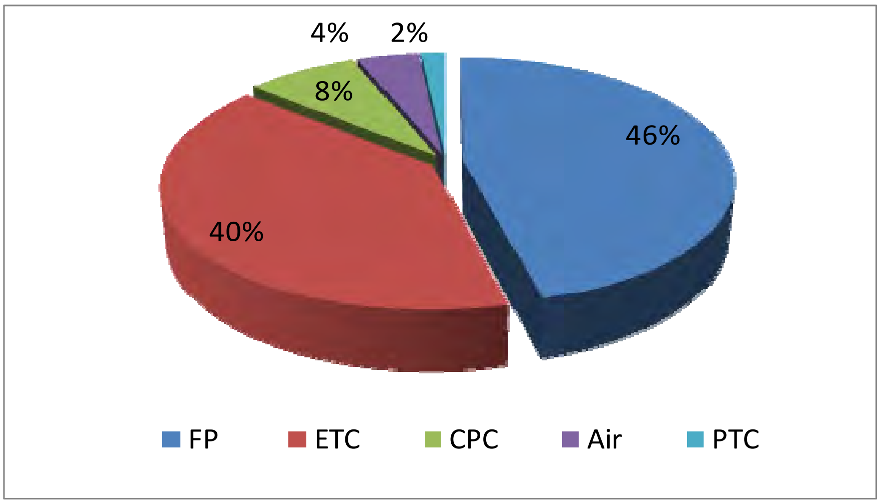


Figure 4.2 Percentage of use of different technologies for solar thermal collectors within 112 large scale systems.

The overall solar surface installed in 108 systems amounts to 35,530 m². The largest surfaces installed for cooling purposes are located in Germany, Italy and Spain as shown in Figure 4.3.

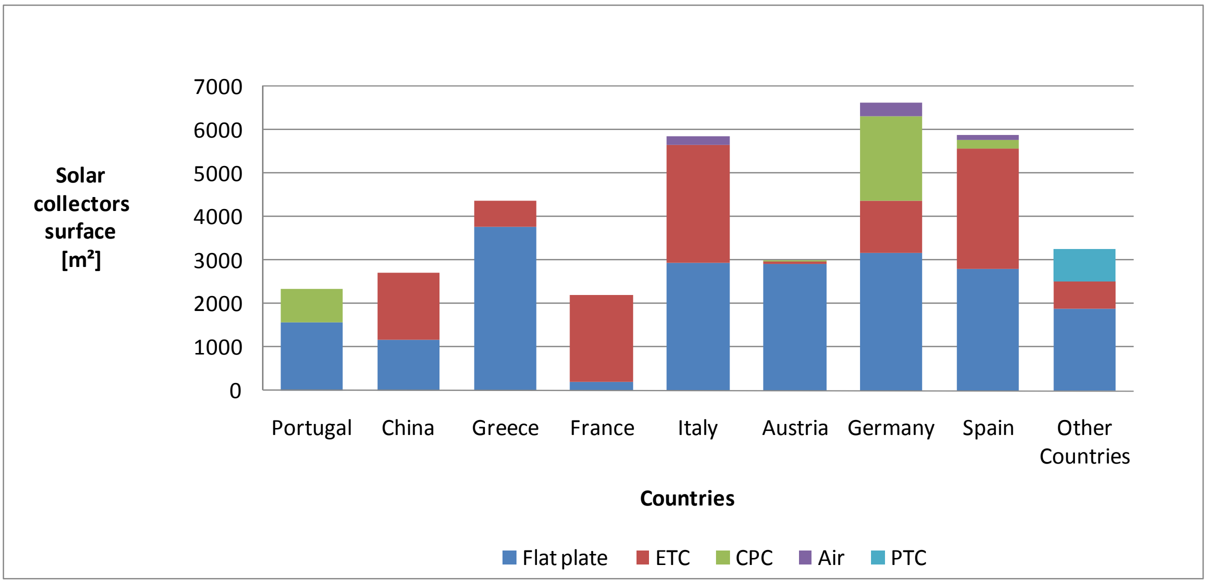


Figure 4.3: World wide installed solar collectors' surface for cooling purpose. The type of solar collectors used in each country is highlighted.

According to Figure 4.3, the technologies used for solar collectors can vary country by country. For instance, 20% of the total solar surface installed in Germany is made of CPC collectors even though they mostly belong to one installation, i.e. the FESTO Company with over 1,200 m² solar collector's surface. Still Italy and Spain feature similar characteristics as their solar surfaces for cooling purposes are nearly 50% made of FP and 50% of ETC. However, 9/12 Italian installations feature ETC fields and the surface distribution similar to the Spanish one is only due to the largest solar cooling plant which has been built in Rome on the roof of the METRO Cash & Carry building (2700 m² driving 700 kW absorption chiller).

Correlating the different technologies of solar collectors used per installed thermal driven technology in 112 installations, it results that not all possible combinations (20) are used (Tab. 2).

Tab. 2 Number of installations counted for each possible technology combination (hybrid systems are not included in this table).

Devices combinations	Absorption	Adsorption	DEC solid	DEC liquid
FP	32	8	6	2
ECT	35	5	1	0
CPC	4	2	3	0
AIR	0	0	5	0
PTC	3	0	0	0

At the same time, 6 hybrid systems also occur, coupling different types of solar collectors or heat driven chillers. E.g. the first installation of the Technology Center Cartiff in Valladolid (ES) is made of 37.5 m² of FP and 40 m² of VT collectors which feed solar energy into an absorption machine of 35 kW. Moreover, the collector's surface rated to the cooling power does not assume a recurrent value, even if the same technologies are applied.

The average rates used within 95 large scale installations for different combinations of solar thermal surface (STC) and thermally driven chillers (TDC) are shown in Figure 4.4. Actually, such average values are the results of largely different rates used in the examined installations, as represented in Figure 4.5.

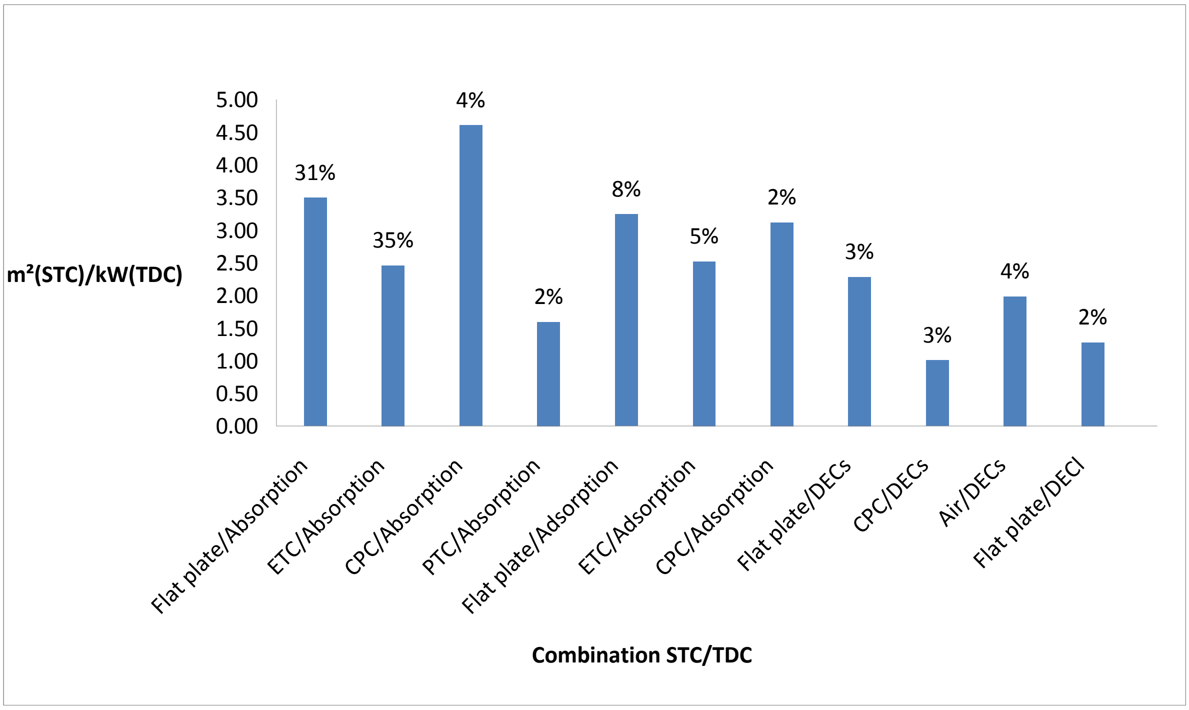


Figure 4.4 Mean of the rates used for each combination taking into account the rate between the used combination and the overall number of cases (95)

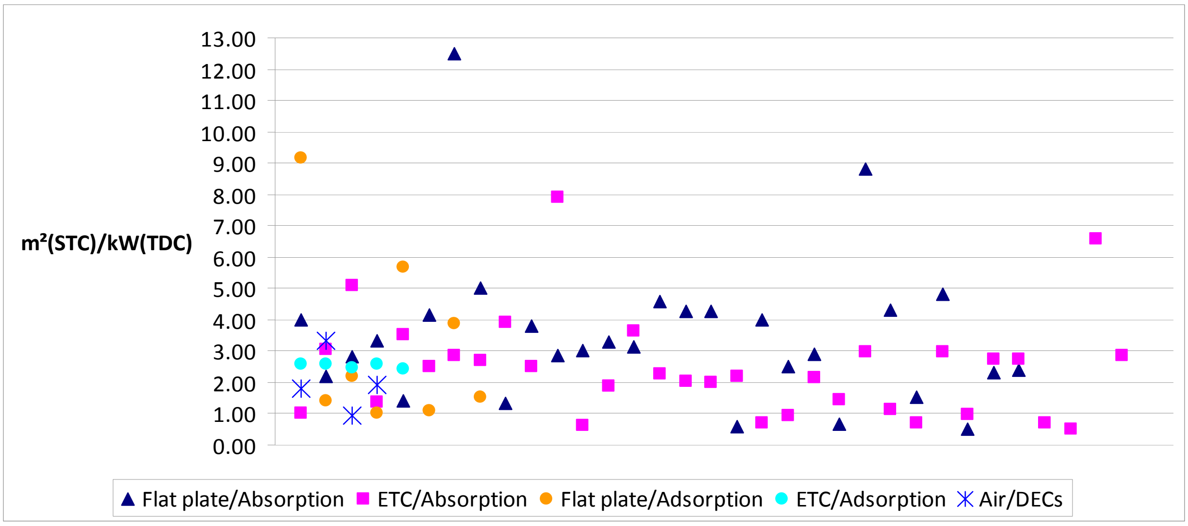


Figure 4.5 Rates between the solar surface collectors installed per kW of cooling capacity for the most used technologies combinations

Heat and cold storage tanks

43 large scale installations are equipped with hot water storage tanks. The rates between the volumes of the tanks and the installed solar surfaces vary case by case, even for those ones which appear similar as shown in Tab. 3.

Tab. 3 Collectors' surface, cooling power, technical data and the rates between them, of two selected systems using the most applied technology combinations.

Installations	**Press and Information Centre of the German Government**	**Ott & Spiess Headquarter**
Locations	Berlin (DE)	Langenau (DE)
Final Use	Offices	Offices
Cooling capacity	44 kW	35 kW
Collectors' surface	348 m² of ETC	22 m² of ETC
Solar Tank	1,600 l	2,000 l hot water
Storage/collectors	4.60 l/m²	90.9 l/m²

Figure 4.6 shows an average calculated on 40 cases (hybrid systems are excluded). Nevertheless, such average values are still based on quite different rates as shown in Figure 4.7. In these cases, any average could not reflect the reality.

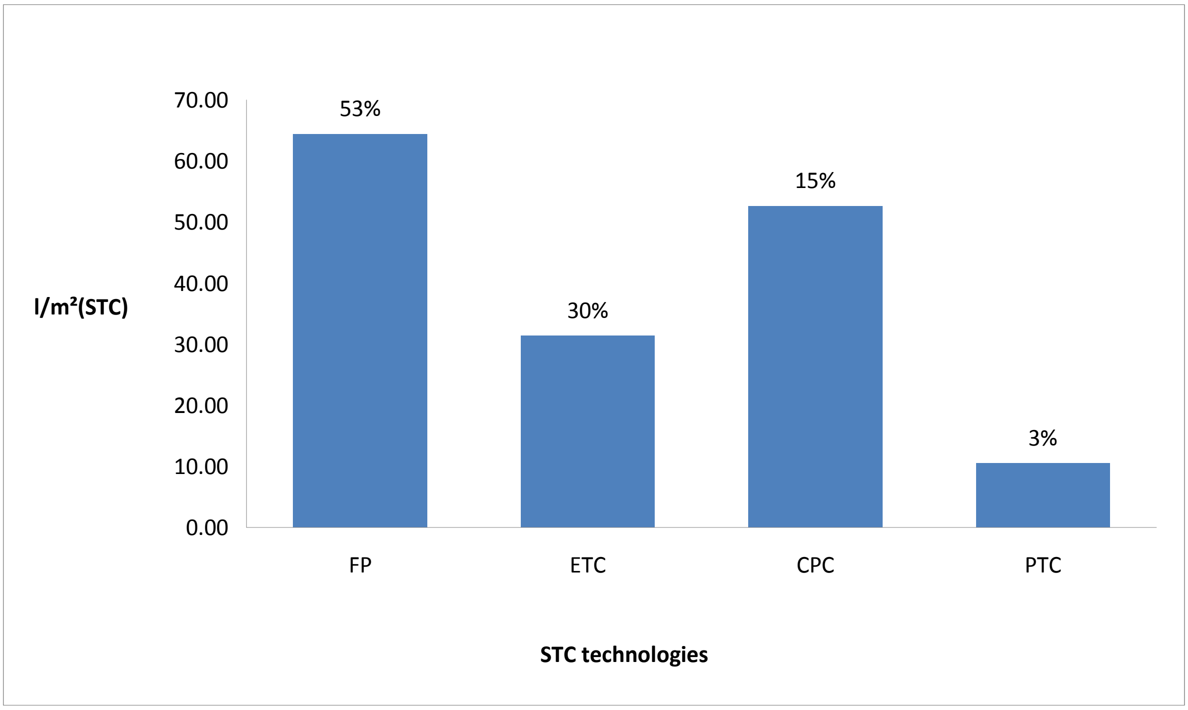


Figure 4.6 Mean of the litres of solar tanks used for different technologies of solar collectors taking into account the rate between the used combination and the overall number of examined cases (40)

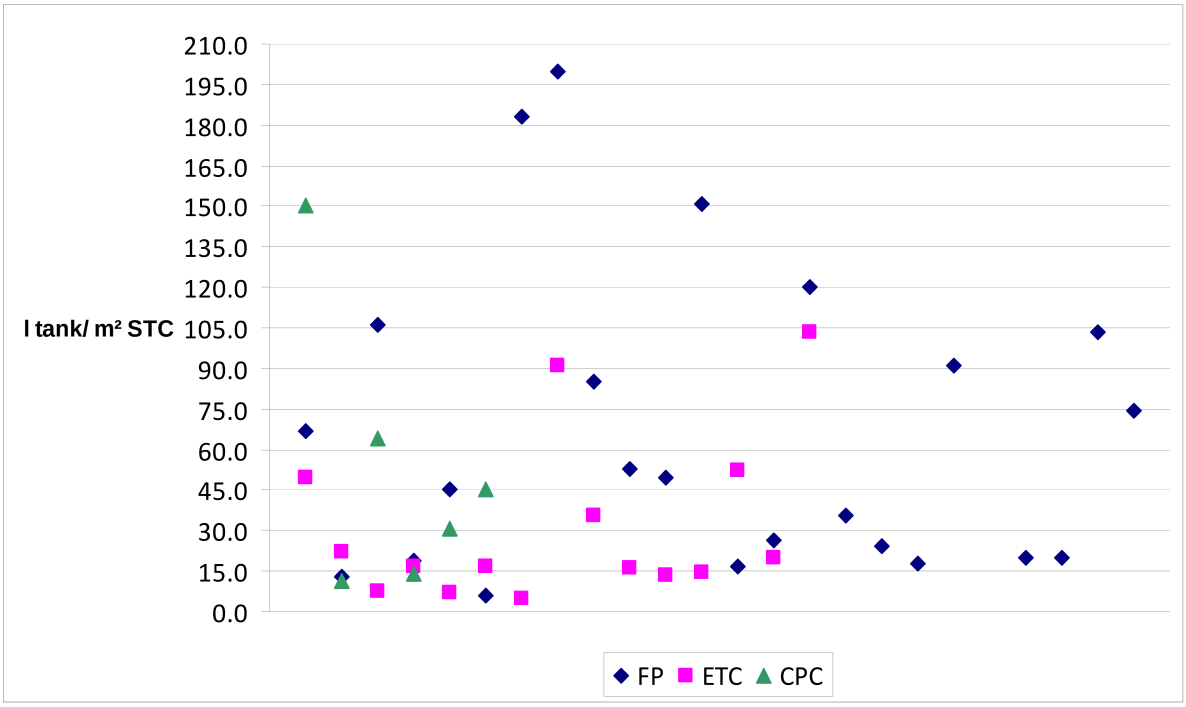


Figure 4.7 Rate between the litres of hot water storage tanks used in each of 46 installations and classified according to the technology used for solar thermal collectors

Cold water storage tanks are not spread as the hot water ones: within 33 installations, 19 have a cold water tank whereas the rest do not (especially DEC systems). Also in this case the rate between the volume of the tank and the kW of cooling installed varies case by case as shown in Figure 4.8.

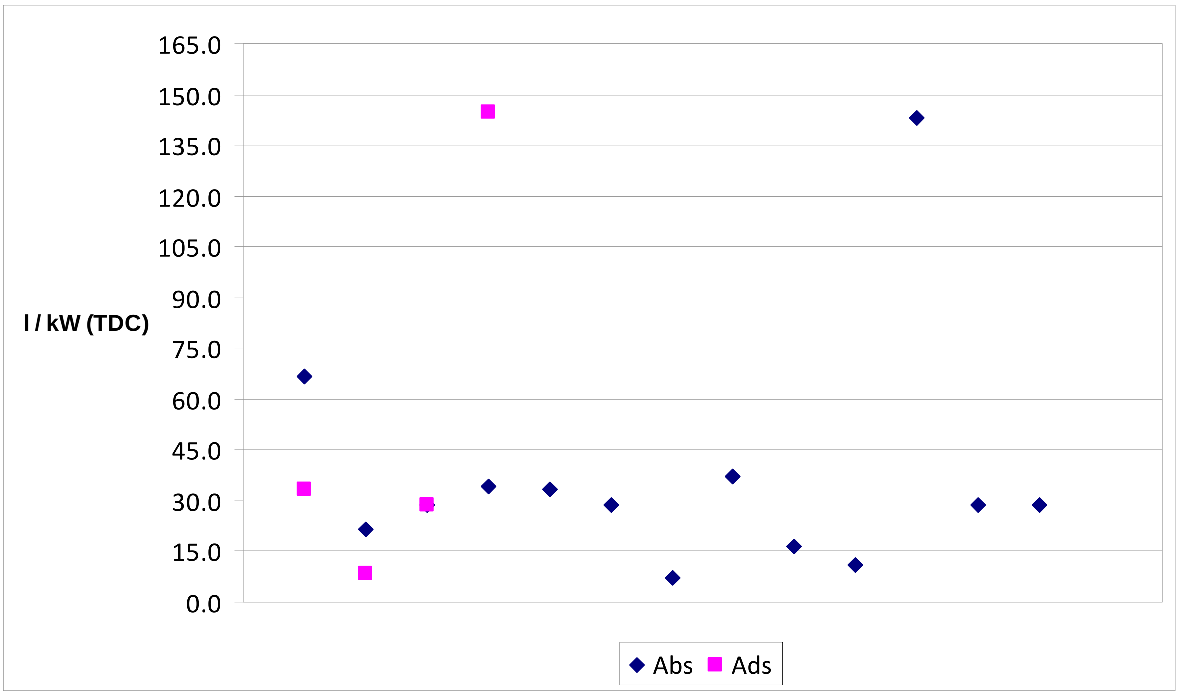


Figure 4.8 Rate between the volume of cold water tanks and the corresponding installed cooling capacity

Heat and cold back up systems

Within 45 systems, 40 are equipped with heat back up systems, based on gas/oil boilers, or heat recovery from cogeneration unit, district heating and heat pumps as shown in Figure 4.9.

Within 36 installations, 25 do not have any cold backup. Within the left systems, 58% employs vapor compression chillers whereas 32% employs heat pumps (NB, systems for which data are available can be different from the ones where data on heat backup systems are available).

As example of solar cooling systems with no heat neither cold backup system is the plant installed in La Reunion (F), where 30 kW absorption chiller is exclusively driven by the heat deriving from 90 m² flat plate solar collectors and thanks to the management of two storage tanks, one for solar heat of 1,500 l tank and one for cold water (1,000 l).

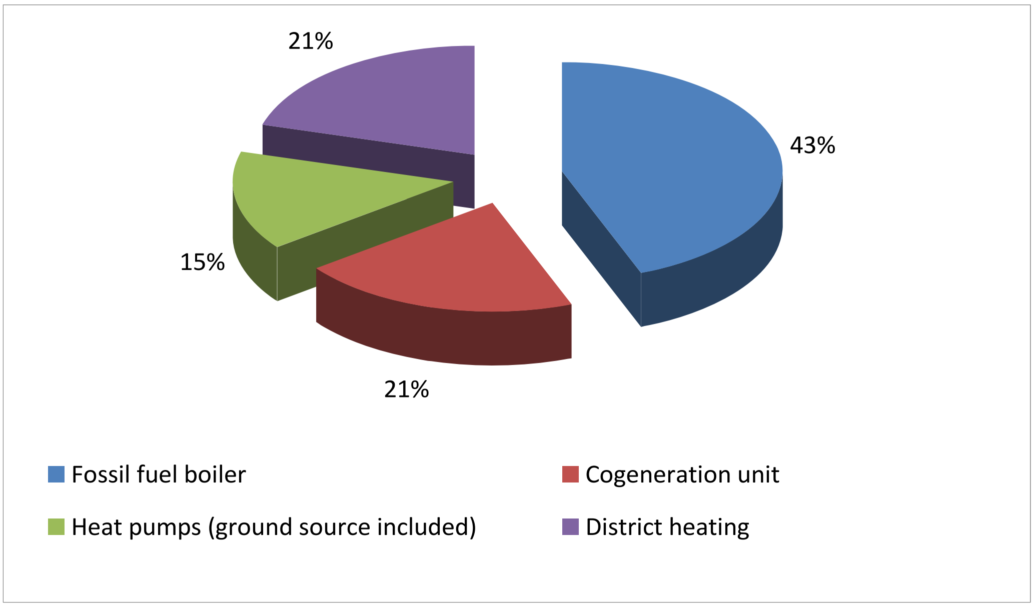


Figure 4.9 Used heat back up in 34 installations

5. Conclusions

Considering the total number of existing solar cooling installations known to the authors in 2009, 10% growth in the market has been registered with respect to 2007, as shown in Figure 5.1

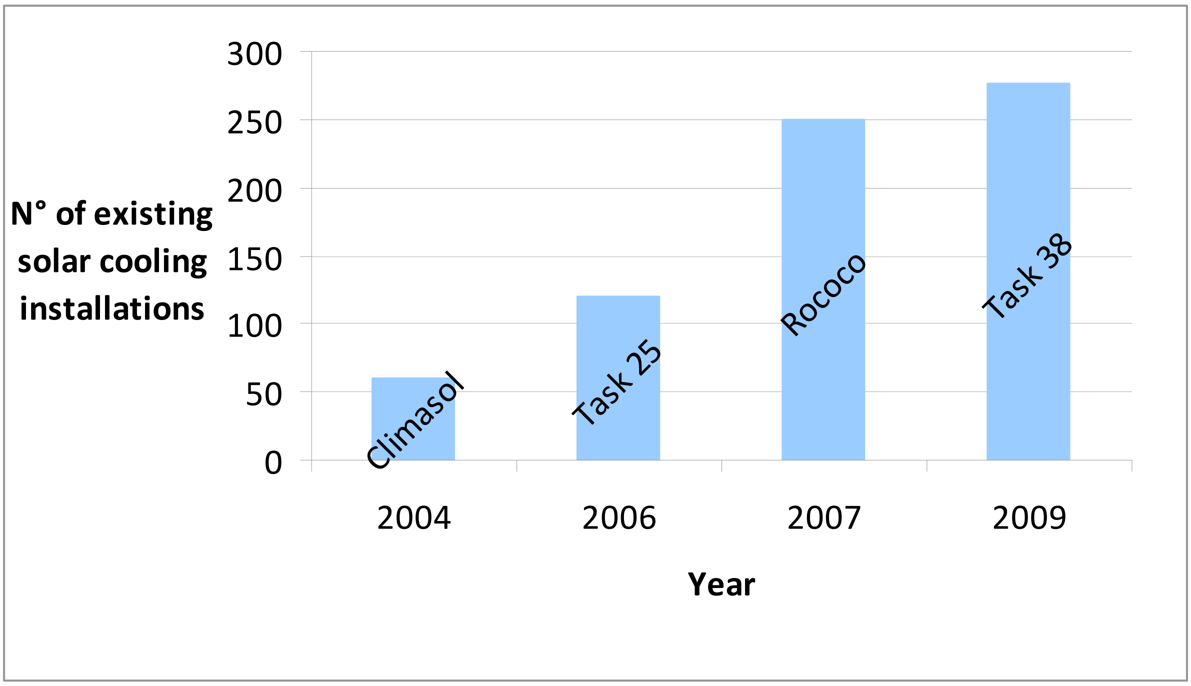


Figure 5.1 Number of total solar cooling installations per year according to different sources as collected by SolarNext

Such growth has been more significant in some Countries, as Spain, Italy and France leading to an overturning of the situation registered in 2005 as shown in Figure 5.2 and Figure 5.3.

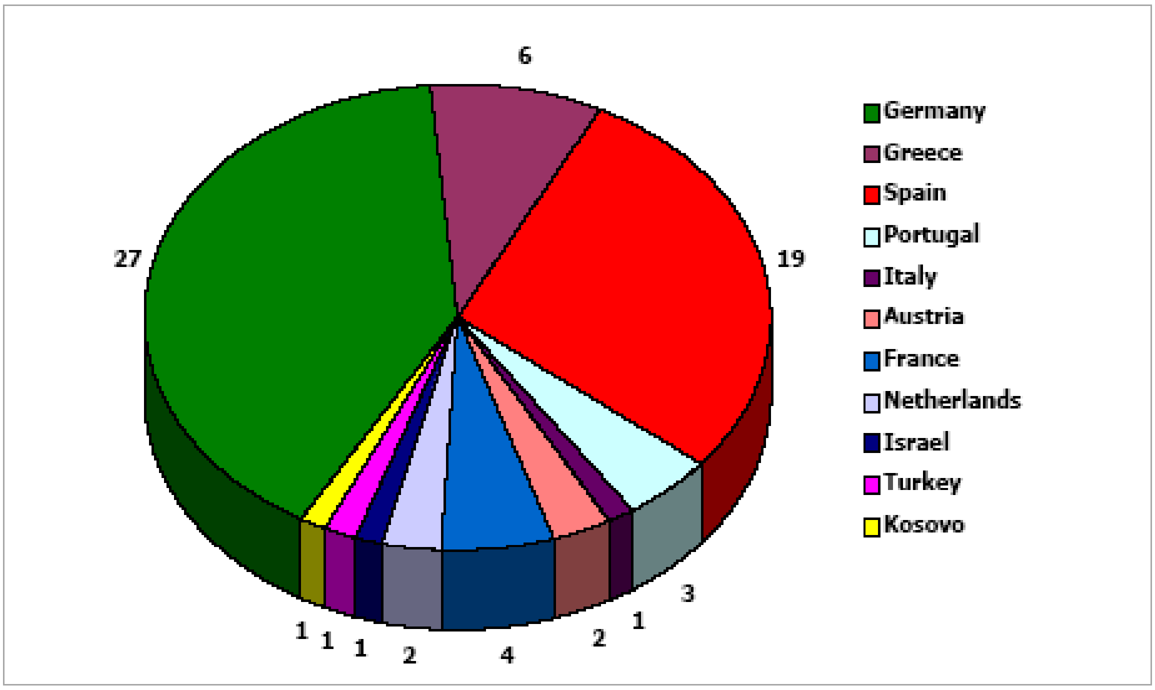


Figure 5.2 Distribution of existing solar cooling installations listed in 2005 in the final report of the IEA Task 25 [1] within different Countries.

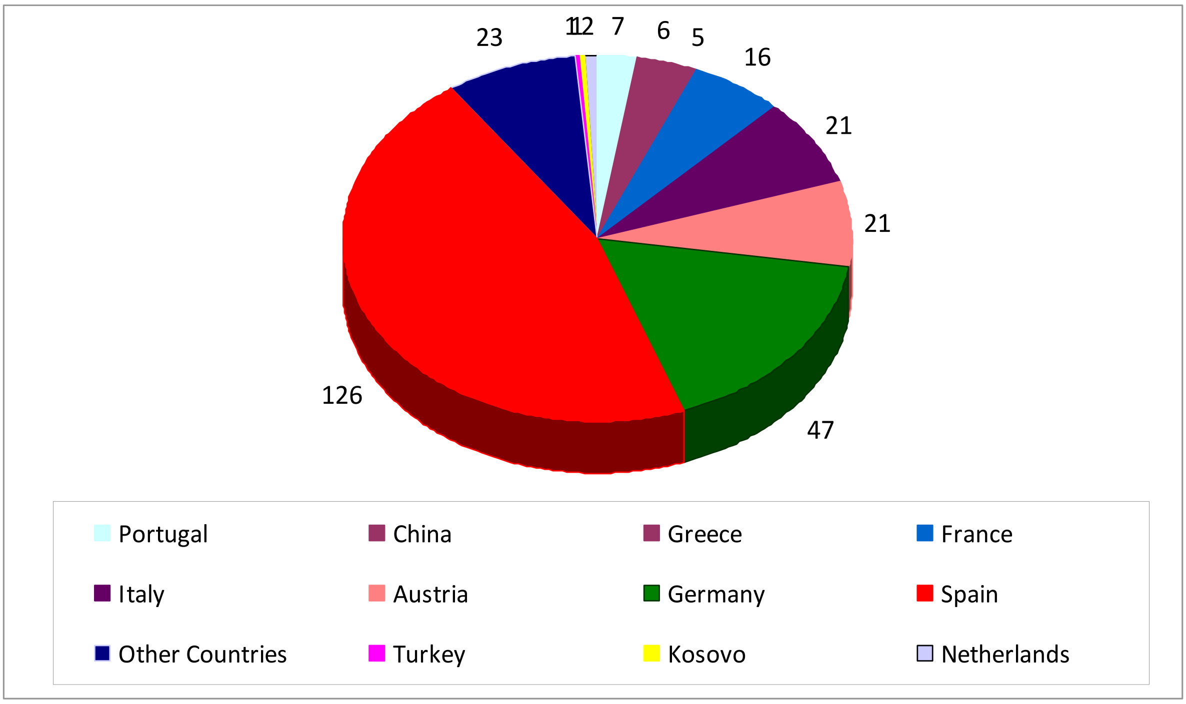


Figure 5.3 Distribution of existing plants listed in 2009 in Task 38 within different Countries. "Other Countries" include: Armenia, Australia, Belgium, Denmark, Egypt, Japan, Lichtenstein, Malta, Mexico, Singapore, South Africa, Switzerland, Syria, UK, United Arab Emirates and USA

Absorption chillers are the thermally driven chillers most present on the market in both small and large applications. Their combination with solar flat plates or evacuated tube collectors is quite well experienced in large systems.

DEC systems have not penetrated the market as absorption and adsorption chiller, especially the DEC liquid technology. To the knowledge of the authors, only 2 DEC liquid systems exist, one in Germany (75 kW) and another one in China (350 kW). Nevertheless, the combination of DEC solid systems with air collectors seems to be competitive with respect to other kind of combinations (mostly DEC solid plus flat plate collectors).

Concerning the solar thermal technology, PTC are present in only 3 solar cooling installation over 113, wherever flat plates and evacuated tubes are the most used technologies.

Whereas the selection of components and their combinations clearly outline the solar cooling market, sizing criteria for the solar thermal surface compared to the installed cooling power, or of the heat and cold storage respectively compared to the installed solar surface and cooling capacity can not be easily derived. In fact, first of all, detailed and reliable statistical analysis is difficult to be carried out as information is not available for all the systems listed. Then, well known installed systems differ so much that possible average values cannot be considered representative of the sample. Finally, the large difference between existing and documented plants do not enable for comparing different design solutions and for deriving any criteria.

To address planners towards optimal design, monitoring results showing advantages and disadvantages of single solutions seem to be necessary. On the other hand, a detailed investigation on the approach used by planners to size components in analyzed projects could be useful to understand the reason the rates between the solar surface and the cooling capacity installed or between the storage capacity and the solar surface or the cooling power installed are so different.

At the current state of the art, the large difference between the systems highlights that solar cooling is still in an early phase of technology development. By addressing planners towards selected configurations and sizes, a clear learning curve could be defined and standardization could be achieved.

REFERENCES

[1] www.iea-shc-task25.org
[2] ec.europa.eu/energy/res/fp6_projects/solar_heating_cooling_en.htm
[3] www.ocp.tudelft.nl/ev/res/sace
[4] Sparber W., Napolitano A., Melograno P., 2007. *Overview on world wide installed solar cooling systems.* In: Otti, ed., 2007. 2nd International Conference on Solar Air Conditioning. Spain, October 2007.

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning
and Refrigeration

C1: State of the art – Survey on new solar cooling developments

A technical report of subtask C

Date: first draft: 26th October 2009, revised 31st October 2010

Subtask C1 coordinator: Michael Krause, Kassel University

Report edited by: Robert Ghirlando, University of Malta

Contributions from:

Erich Podesser, AEE-INTEC, Austria

Robert Ghirlando, University of Malta

Tomas Núñez, Fraunhofer, ISE, Germany

Mustafa Jaradat, Kassel University

Michael Krause, Fraunhofer, IBP, Germany

Roland Heinzen, Kassel University, Germany

Lucio Mesquita, Enerworks Inc. Canada

Paul Bourdoukan, LOCIE-INES/CNRS, France

Etienne Wurtz, LOCIE-INES/CNRS, France

Klaus Ellehauge, Ellehauge and Kildermoos, Denmark

Constanze Bongs, Fraunhofer ISE, Germany

Clemens Pollerberg, Fraunhofer UMSICHT, Germany

Pietro Finocchiaro, DREAM, University of Palermo, Italy

Marcelo Izquierdo Millán, CSIC, Spain

José Daniel Marcos del Cano, UNED, Spain

Note: Contact details are available at the end of the report in Appendix 1, page 75 .

Contents

1 Introduction

This report is the result of the work undertaken by one of the working groups set up as part of Task 38 of the Solar Heating and Cooling Programme of the International Energy Agency. The remit of Task 38 was to study Solar Air-conditioning and Refrigeration. The work was split into four sub-tasks and these were further sub-divided into smaller sub-tasks. One such sub-task CI, forming part of the sub-task C on *Modelling and Fundamental Analysis*, was set up to carry out a survey on new solar cooling developments, a *State-of-the-Art* report. This report documents the results of that work.

At the start of Task 38, it was decided that the work would be restricted to solar cooling systems that use solar energy in the form of heat, i.e. of systems that consist of solar heat collectors and thermally driven devices that produce the cooling effect using the heat from the sun. Systems consisting of photovoltaic panels producing electricity to drive conventional chillers were excluded *a priori* from the work of Task 38. It was also decided to restrict the work to the thermally driven devices, i.e. the technologies to capture the solar radiation (flat plate collectors, vacuum tube collectors, solar troughs, etc.) were not considered.

A number of participants in Task 38 were each assigned to research a particular technology and to report back. The technologies to be surveyed were:

Absorption chillers (ammonia/water, water/lithium bromide)
Adsorption
Solid desiccant
Liquid desiccant
Thermo-mechanical chillers
Steam jet chillers

During the half-yearly meeting of April 2009, it was decided that the report would include:

- Brief description of the technology
  - Thermodynamic cycle, characteristics, advantages and disadvantages, problematic areas
- What is on the market?
  - Names of companies selling machines and systems (including websites and contact details), details of products sold, sizes and other important parameters, new products on the market.
- Research and Development
  - Names of institutes, universities and companies involved in R&D; (including websites and contact details), names of researchers, what is being researched and developed.
- Relevant reports and publications

Thanks are due to all who have contributed to this report. All contributors have dedicated time and effort to record the state-of-the art of these technologies, some of which are quite old, but all of which require more research and development in order for them to reach a state where they can compete realistically with the conventional vapour compression machine driven by electricity. It is encouraging to note the increased interest in these technologies, not only in academia but also in the market. This bodes well for the future.

2 ABSORPTION REFRIGERATION

Erich Podesser (on behalf of AEE-INTEC)

2.1 Introduction and historical review

The large field of applications of the sorption cooling technology, which uses heat as the driving energy for the cooling process, is the oldest cooling technique. It is worth mentioning that the first artificial cooling device of Edmond Carré follows the periodical absorption cooling technique and was put in operation around 1850.

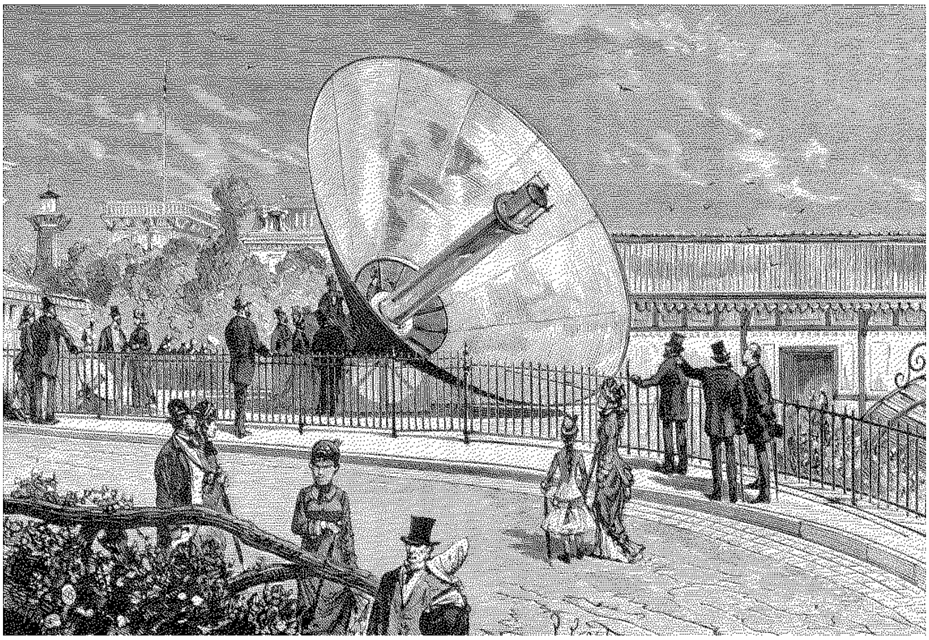


Figure 1: World exhibition 1878 in Paris /1/. On September 29 Augustin Mouchot produces the first ice block with solar energy using a periodical absorption machine of Edmund Carré (Courtesy of Olynthus).

The period from 1850 to 1880 in France was marked by an energy crisis caused by a tremendous increase in costs of conventional fuel (wood, coal). The reason for that cost increase was the industrial development and an insufficient extension of coal mining in France. For one ton of steel about seven to ten tons of charcoal were needed and the price for energy rose about 10% per year. The government of France decided to subsidize the development of solar applications. It is only little known and should therefore be mentioned, that at that time (1850 to 1880) in France all solar technologies, except photovoltaic applications, were carried out using the available techniques [1].

The continuous increase of man-made CO_2 concentration in the 20^{th} century and the environmental pollution from energy conversion promote the utilization of renewable energy like solar and biomass to meet air-conditioning and refrigeration needs.

2. 2 Description of the technology

2.2.1 Sorption refrigeration systems

A large variety of sorption processes are well known and tested. Most of them can be assigned to the structure in figure 2, which shows a compilation of possible processes.

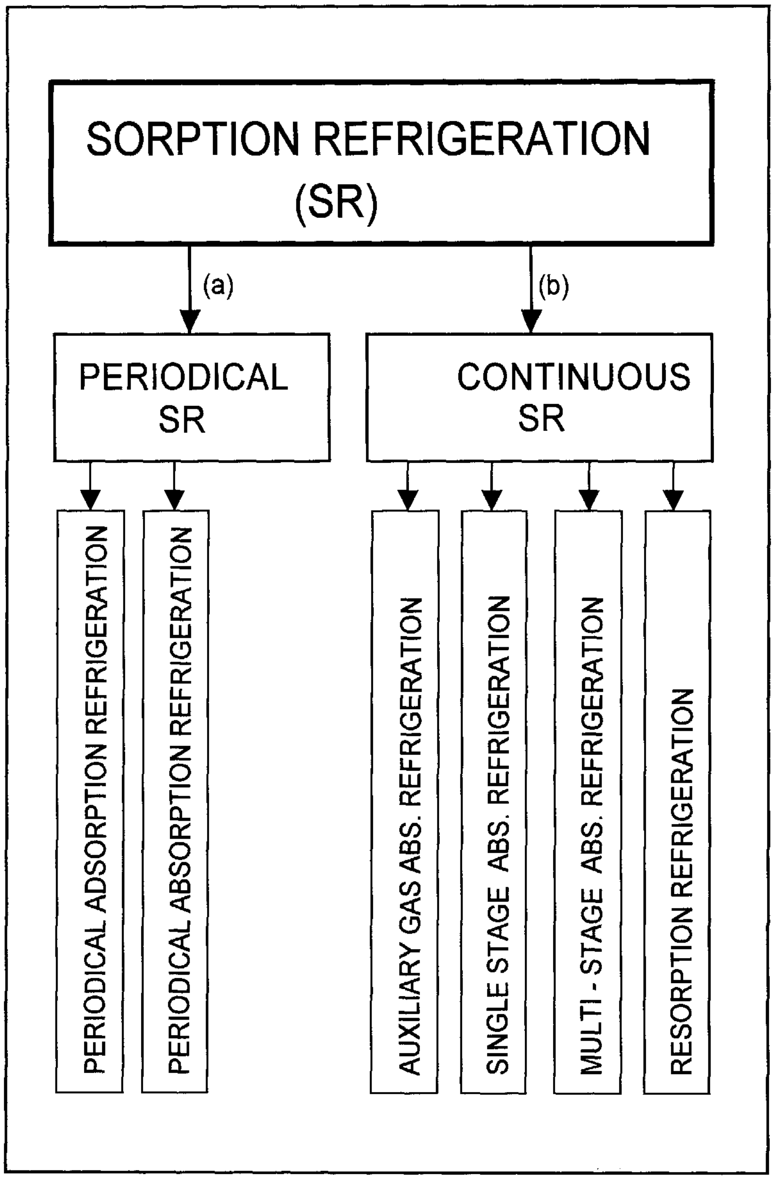


Figure 2: Arrangement of different sorption processes

Sorption processes are divided at first into the two branches, the periodical sorption processes (a) and the continuous sorption processes (b).

2.2.2 Periodical sorption refrigeration processes

Processes of this category (a) use either liquid refrigerants, like ammonia, and a suitable absorptive substance like water or calcium chloride, or water as a refrigerant and a solid adsorptive material, such as silica gel or zeolite.

The *absorptive* substances (water, calcium chloride) are capable of chemical bonding with refrigerant ammonia and in most cases bonding energy is released. Therefore the absorbent has to be cooled continuously in a period of the cooling process. Figure 3 shows the principle of a periodical refrigeration apparatus, such as it was invented by Edmund Carré in the Middle of the 19th century.

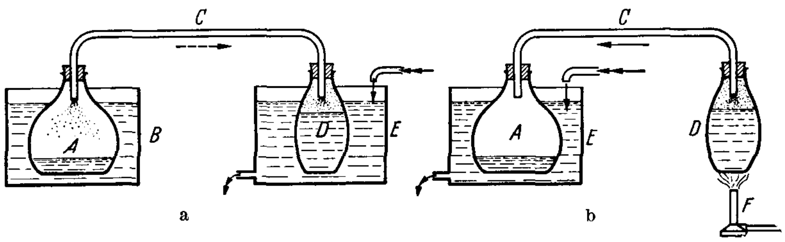


Figure 3: Principle of the periodical absorption refrigeration process.

a: first period, cooling of A and B: evaporation of the refrigerant in A, absorption of the refrigerant by an absorptive substance in D; b: second period, generation of refrigerant in A: boiling of a mixture refrigerant and absorptive substance in D and condensation in A.

The *adsorptive* substance (silica gel, zeolite) adsorbs the liquid refrigerant water with a physical bond onto its extremely large surface. It is worth mentioning that the surface of 1 gram of silica gel lies in the range of 300 to 400 m^2. So in the case of adsorption, which is normally an exothermal process, bonding energy has to be rejected out of the system.

2.2.3 Continuous (ab)sorption refrigeration processes

2.2.3.1 Components and definitions

These sorption processes use liquid absorbents and have one common characteristic feature, the continuously working thermal compressor. It replaces the mechanical compressor of a vapour compression cooling system. See the important components of the thermal compressor in Figure 4.

- Generator G: gaseous refrigerant is generated by boiling the working fluid
- Absorber A: cool, gaseous refrigerant coming out of the evaporator is absorbed
- Working fluid heat exchanger (WFHE): heat recovery from the hot and weak to the cold and strong working fluid
- Working fluid pump: in the case of ammonia/water, the pressure difference is in the range of 8 to 11 bar
- Working fluid control valve (WFCV): controls the level of the working fluid in the generator

The single stage, continuous absorption refrigeration machine needs, in addition to the thermal compressor, a condenser C, an evaporator EV, a refrigeration control valve RCV and a working fluid control valve WFCV. This process is the basic process of all systems in group (b) of Figure 2.

Description of the thermal compressor: The working fluid, mainly ammonia and water (or another working fluid pair like water/lithium bromide) is boiled in the generator by supplying heat at a suitable temperature, e.g. at 70 …100°C. Mainly ammonia leaves the generator as vapour and is condensed at the cool condenser heat exchanger, e.g. at 25 … 35°C. The ammonia which leaves the generator leads to a decrease of the ammonia concentration in the generator; therefore the boiling working fluid in the generator has to be renewed continuously. This is managed by the working fluid pump, which delivers the strong working fluid with a concentration of, e.g. 40% ammonia, from the absorber via the working fluid heat exchanger into the generator. The working fluid heat exchanger heats up the strong WF from the absorber temperature, e.g. 30 … 35°C, by heat recovery to about 65°C. So, the WF enters the generator at 65°C and starts boiling after additional heating. After a certain time of boiling, the concentration of the WF is decreased from, e.g. 40% to 35%, and is led back as weak WF via the WFHE to the absorber. The weak WF leaves the generator with an outlet temperature of about 80°C and enters the WFHE at the same temperature. The weak WF is cooled down in the WFHE by the cold, strong WF, which comes from the absorber with a temperature of about 35°C. The WFCV is controlled by the WF level in the generator and ensures that the same quantity of WF leaves the generator as it is delivered by the WF pump in the generator. That is why there is no danger that the generator becomes empty.

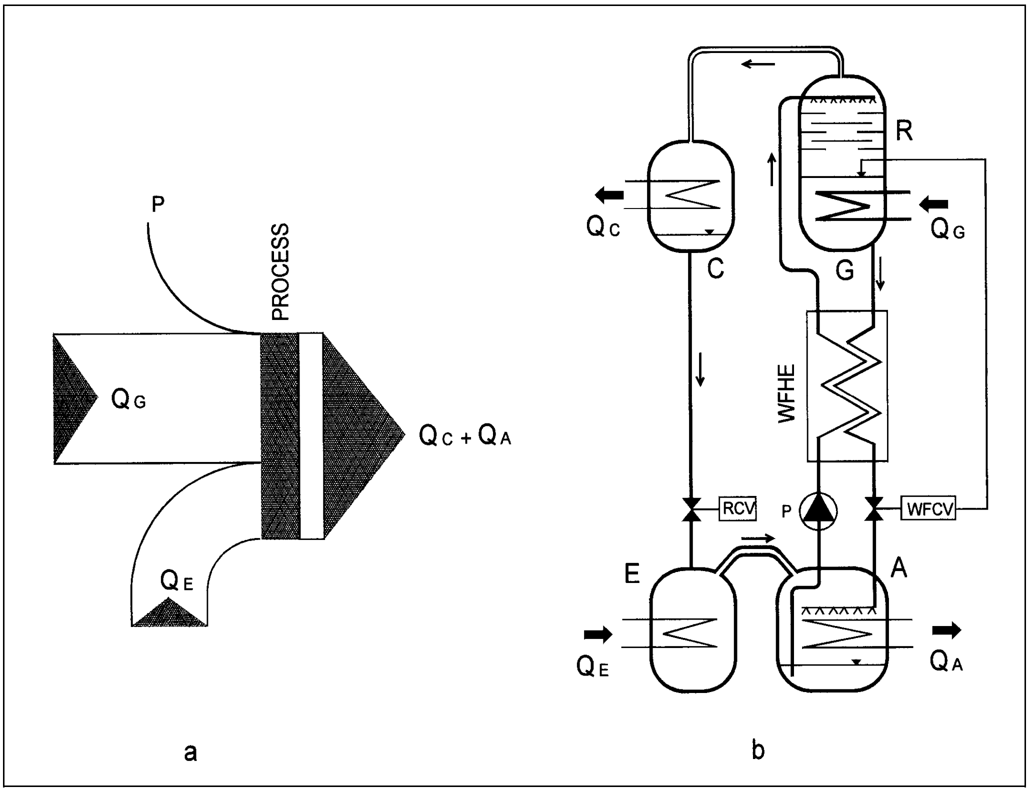


Figure 4: Single stage, continuous working absorption refrigeration system

A … absorber, G … generator, E…. evaporator, C … condenser, WFHE …working fluid heat exchanger, RCV ... refrigeration control valve, WFCV ...working fluid control valve, P... pump, R ... rectification

Description of the refrigeration process: The condensed refrigerant leaves the condenser C and is injected in the evaporator E via the RCV. The evaporator E works at the low pressure level, e.g. 2 bars, and the refrigerant boils and evaporates at temperatures of about 0 … to 5°C. The cold ammonia vapour leaves the evaporator E and flows to the absorber A, which absorbs the refrigerant vapour at an absorber working fluid temperature of about 35 to 40°C.

Heat ratio: Of high interest is the answer to the question: “What is the relation of the heat delivered to the cooling capacity of the system including the electric energy for the WF pump?” This question could be answered by the heat ratio, which is defined in the German literature [2] by the following equation.

$$\zeta = \frac{Q_0}{Q_H + P_{el}}$$

ζ … heat ratio

Q_0 …cooling capacity (kW)

Q_H …heat for the generator (kW)

P_{el} ….electric energy for WF pump (kW)

Also the thermal coefficient of performance ($COP_{thermal}$) is sometimes used

$$COP_{th} = \frac{Q_0}{Q_H}$$

The heat ratio and the COP $_{thermal}$ for the various applications of an absorption refrigeration process depend on the kind of working fluid and the application of the process.

2.2.3.2 Design of an absorption refrigeration process

The design of an absorption process will be demonstrated for the working fluid pair ammonia as refrigerant and water as the absorbent. The thermodynamic properties for the ammonia water mixture are well known. The tool for process configuration is the lgp,1/T-diagram for the basic process definition and the table of Merkel-Bosnjakovic for process design at a higher accuracy. A basic process definition is shown with the help of the lgp,1/T-diagram in Figure 5.

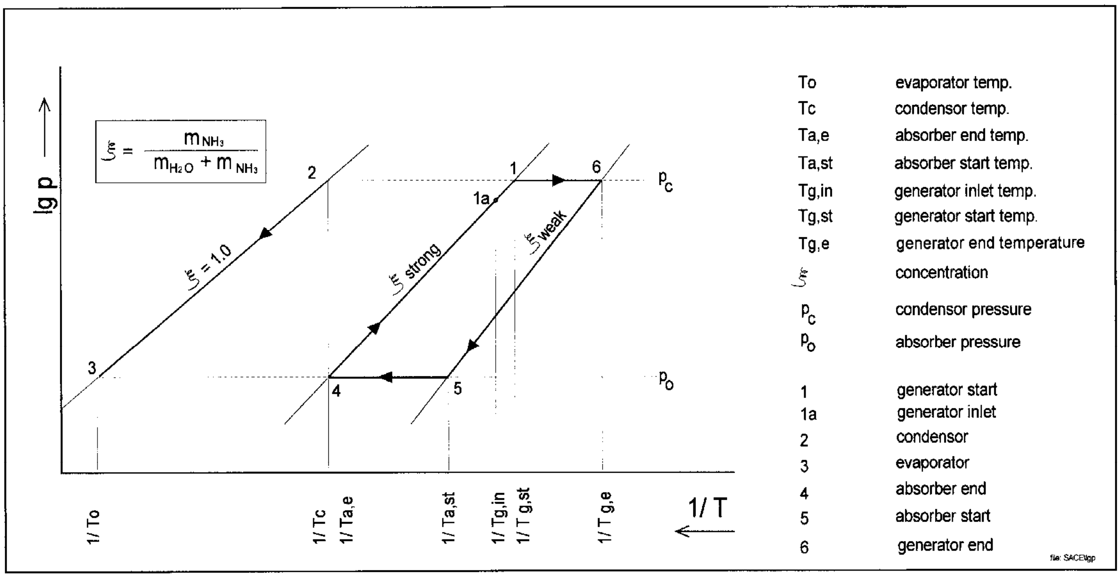


Figure 5: Basic design of a single stage, continuous absorption refrigeration process in the lgp,1/T-diagram for working fluids with positive heat of the chemical reaction.

The following items have to be determined for the basic design of a refrigeration process:

- Working pair (NH3/H2O; H2O/LiBr)
- Evaporation temperature (To)
- Temperature of cooling water (heat rejected at Ta,Tc)

The process starts in Figure 5 at point 4. The working fluid, an ammonia/water mixture, is pumped out of the absorber (30 … 40°C) at a pressure of about 3 to 4 bars to a pressure of 10 to 13 bars into the generator with the temperature $T_{g,in}$. In the generator the temperature rises to $T_{g,st}$ and the working fluid starts boiling. The refrigerant (NH3) is separated from the working fluid by boiling from point 1 ($T_{g,st}$) to 6 ($T_{g,e}$). At point 6 the WF boils at the highest temperature $T_{g,e}$ and the concentration reaches ξ_w. With the help of a pressure reduction valve (WFCV) the working fluid leaves the generator at point 6 and is reduced to the low pressure stage at point 5. On its way from 5 to 4 the weak WF absorbs the refrigerant, which comes from the evaporator 3 in the absorber 4. The absorber has to be cooled continuously.

The refrigerant (NH3) is separated from the working fluid in the generator between point 1 and 6, and is condensed in the condenser (point 2) on cooled surfaces at the high pressure stage. After condensing, a pressure reduction valve reduces the pressure of the liquid refrigerant before entering the evaporator at 3. The refrigerant receives heat at the low temperature (+5 … -30 °C) in the evaporator 2 and can now boil and evaporate at low temperature. The vapour of the refrigerant enters the absorber at point 4 and meets the weak working fluid. Further information on that matter is offered in Niebergall [2] and Bogard [3].

With the help of these definitions and the lgp,1/T-diagram the expected temperature of the boiling working fluid in the generator (expeller) can be found and a theoretical heat ratio can be calculated too. This first determination of the refrigeration process delivers rough numbers without the consideration of losses and limitations of the thermodynamic properties of the working fluid.

For higher accuracy of the process definition, the table of Merkel-Bosnjakovic or special computerized data should be used, which give reliable process data of temperatures, pressures, concentrations of the working fluid and the enthalpies of the various process states. A model machine, in which 1 kg of refrigerant circulates and all parameters of the outside conditions are involved, can be calculated with this thermodynamic tool. The enthalpy differences of the working fluid and the refrigerant at the inlet and at outlet of the apparatuses in Figure 4 allow the determination of the mass flow and the heat rejected or received by the heat exchangers.

The procedure of the process design is demonstrated by the sketch (not to scale) of the table of Merkel-Bosnjakovic in Figure 6, to explain and prepare the practical work with this graph.

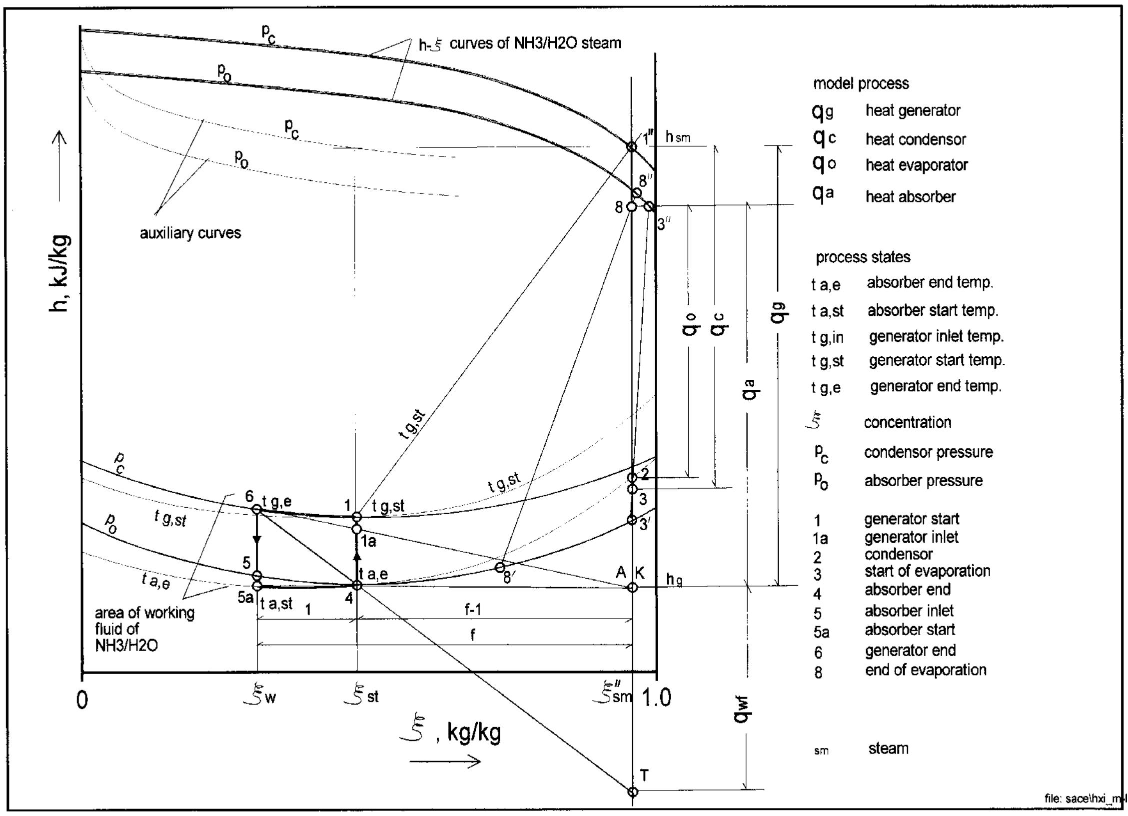


Figure 6: Principle of a one stage, continuous absorption refrigeration process in the graph of Merkel-Bosnjakovic. h … enthalpy, ξ … concentration, ξst … c. of strong WF, ξw … c. of weak WF, ξsm … steam concentration, numbers 1a,1,6,5,5a,4 to 8… process stages corresponding with fig. 5, t … temperatures, p_0 … absorber (evaporator) pressure, p_c generator and condenser pressure, q_g … spec. generator heat received, q_a … spec. absorber heat rejected, q_c … spec. condenser heat rejected, q_{wf} … spec. WFHE heat recovered. T, A, K design help

The horizontal axis of the diagram in Figure 6 indicates the concentration ξ of the working fluid (ξ = 0 only water, left side; ξ = 1 only refrigerant, right side). From the vertical axis the enthalpy of the various thermodynamic stages can be read out. The enthalpy at the vertical axis is related to the circulation of 1 kg refrigerant in the process. “f” shows the necessary kg of working fluid in order to adjust the concentration difference of the strong and the weak working fluid (ξ_{st} - ξ_a). The higher curve package in the graph is valid for the NH3/H2O-vapour and the lower curves for the NH3/H2O fluid.

The process stages (1a)–1–6–5–(5a)–4 indicate the circulation of the WF and can be compared with the numbers in the diagram in figure 5. Starting at the end of the absorber with point 4 the WF is pumped from the low pressure p_0 out of the absorber to the pressure p_C in the generator. With the help of the WFHEX (see figure 4) the temperature t $_{a,e}$ is raised to $t_{g,in}$ (generator inlet) and heated up to reach temperature of boiling $t_{g,st}$ (point 1). The concentration is lowered and point 6 is reached by boiling the working fluid with a small temperature rise from $t_{g,in}$ to $t_{g,e}$. Refrigerant vapour leaves the generator with $t_{g,st}$ and appears in the condenser at the same temperature and with an enthalpy given by point 1” (h_{sm}).

The refrigerant steam is cooled down in the condenser from 1” to 3´ by rejecting sensible heat. At 2 the condensation of 1 kg of NH3 is finished, the enthalpy stage of 3 is reached by reducing the temperature by some degrees centigrade. The enthalpy difference given by the distance q_C is the sum of the heat which has to be rejected out of the condenser as the sum of sensible and latent heat. After the pressure reduction at the throttle valve (RCV) the evaporation starts in the evaporator at 3´. The specific heat for the evaporation of 1 kg refrigerant is q_C minus the sensible heat given by the corresponding enthalpy differences. For the model process, which is characterized by circulating 1 kg of refrigerant and f kg of working fluid the corresponding heat can be read out of the diagram in figure 6.

Table 1: Characteristic data of the single stage, continuous working absorption refrigeration model machine for a circulation of 1 kg of refrigerant (NH3); Evaporation at minus 10 °C and cooling water temperature of 25 °C.,

	specifics		unit	comments
generator	q_K	597	Wh/kg	received
evaporator	q_0	294	Wh/kg	received
absorber	q_A	486	Wh/kg	rejected
WF-HEX	q_{TW}	561	Wh/kg	recovered
condenser	q_C	406	Wh/kg	rejected
WF pump	q_P	1,74	Wh/kg	received
high pressure	$p_{C,}$ p_g	10	bar	
low pressure	$p_{A,}$, p_o	1,85	bar	
concentration weak WF	ξw	26	%	
concentration strong WF	ξst	36	%	
concentration difference	$\Delta\xi$	10	%	
specific WF circulation	f	7,4	kg/kg	
heat ratio	ζ	0,492	-	$q_0/(q_E + q_p)$

WF … working fluid, HEX … heat exchanger

Table 1 shows the results of the calculation of the model process following the procedure of Figure 6 for a continuous, single stage absorption refrigeration machine. The cooling tower works at 25/30 °C and the ammonia evaporates at minus 10 °C for brine cooling.

- q_g specific heat received at the generator
- q_C specific heat rejected at the condenser
- q_a specific heat rejected at the absorber
- q_{wf} specific heat recovered from the weak hot WF
- q_0 specific heat received at the evaporator
- ζ_I heat ratio, which can now be calculated

The basic data in Table 1 allows the design of a real absorption refrigeration machine. The factor of enlargement can be determined by dividing the desired evaporation capacity of the real machine (Q_o) by the specific evaporation heat (q_o) in table 1. With the knowledge of the heat and mass flows in the apparatuses the necessary surfaces for heat exchange and mass transfer, the diameters and length of the tubes, and the capacity and power of the working fluid pump of the absorption refrigeration machine can be calculated.

The specific data for every refrigeration application, like air-conditioning (+5 °C), brine cooling (-5 to –15 °C) and deep freezing (-30 to - 50 °C) can be taken from the Merkel-Bosnjakovic enthalpy-concentration graph. Further information on that matter is offered in Niebergall 2] and Bogard [3].

2.3 Market situation – Ammonia/Water Systems

The small and medium size absorption refrigeration machines are generally able to fulfil all cooling demands. But at the moment only a modest market exists for these absorption refrigeration machines. A lot of interest for heat driven refrigeration technology could be received from the persons responsible in the food processing companies, hotels, office building and similar enterprises. The main reason for the modest turnover is the investment cost for absorption technology in comparison to the cost of conventional vapour compression technology. All components for the vapour compression technology are really mass-produced products. Every component of the vapour compression machine is produced in a separate company. By contrast the absorption refrigeration machines with the working fluid ammonia and water are manufactured in one company piece by piece.

In the air-conditioning sector the absorption refrigeration machines are able to cover the cooling load including the dehumidification of the air, due to the possible evaporation temperatures below zero.

An important area of implementation could be expected for cold storages especially of vegetables and all kind of food. The customers in this sector of activities could be companies in the food processing industry and chain stores.

In the cooling capacity sector below 100 kW, at present the companies Robur in Italy, Pink and ECONICsystems in Austria are producing small absorption units, using Ammonia as a refrigerant.

Company Robur was founded in 1956 in Verdelino/Zingonia (BG), Italy and produces equipment for heating and cooling, which uses natural gas as an energy source. Also the well known 16 kW absorption refrigeration unit is directly fired with natural gas and the heat rejection works with ambient air. A growing market could also be expected with the Robur absorption heat pump, which is used in some cases with heat from bore hole. More information is available at the website www.robur.com.

Company Pink is located in Langenwang, Austria. The main products of the company Pink are customer specific designed water tanks with volumes of more than 1.000 Litres. Since 2006 the company produces also ammonia absorption refrigeration units in the range of 10 kW. The Pink absorption refrigeration units operate normally with hot water from solar collectors or other hot water sources. The heat exchangers of the Pink absorbers are designed for air-conditioning of rooms with thermal solar flat plate collectors in the temperature range of 70 to 80 °C under normal heat rejection conditions. But also other heat sources can be used. Brine temperatures below zero up to – 10 °C can be offered, if higher temperatures as mentioned above at the generator are available. Therefore Pink absorbers can also be used for providing process cooling for the production line in small companies. Company Pink plans to enlarge their cooling capacities of the units produced up to 20 kW. Additional information is available at Pink´s home page (www.pink.co.at).

Company ECONICsystems is located in Gars am Kamp, Austria. The company started in 2008 with very small refrigeration capacities of 3 kW for container cooling. Afterwards larger cooling units around 10 kW were produced. The products are designed both for air-conditioning including the dehumidification of the air and also for applications with temperatures below zero. The absorption refrigeration activities of ECONICsystems are now focused above all on commercial and industrial applications in the refrigeration capacity range of 20 to 200 kW. The absorption refrigeration equipment is customer specific designed for production lines and also for cold storages in the food and luxury food industry. ECONICsystems makes also some effort and invests money to reduce the operational cost of the small wet cooling towers by lowering the electric energy consumption and the water consumption by electric conductivity control. In addition to these measures, metal ion disinfection equipment for hygienic control of the open cooling water cycle has been developed and successfully tested. Furthermore the company is intensively working on “Reduction of the necessary electric energy for auxiliary drives in absorption refrigeration units”. Respective patents protect these efforts. Additional information is available at ECONICsystems´s home page www.econicsystems.com.

Company AOSOL

As part of the Polysmart project, AOSOL, a Portuguese company, developed an ammonia-water vapour absorption machine for domestic cooling to prototype stage. This machine had a cooling capacity of 6kW and a thermal COP of 0.55, required a hot water temperature above 85^{0}C, produced chilled water at 8-18^{0}C and was air-cooled.

2.4 Research and development – Ammonia/Water Systems

The ammonia absorption refrigeration technology was developed already at the beginning of the 20th century for commercial applications. Above all, large refrigeration capacities (MW) were realized for industrial applications. The downscaling of this technology, developed to the low capacity range (10 kW), leads to big and heavy constructions. Only when the falling film technology for heat and mass transfer at the inner side of narrow tubes (12 mm) was successfully developed, could relatively small and light apparatus for this technology be designed and constructed. The above mentioned companies are able to deliver these absorption refrigeration units in the low capacity range. Both the technical realization by suitable process components and the control of the absorption refrigeration unit itself and also plant control systems have reached technical and market maturity. But the following improvements are desirable and recommendable.

➢ *Reduction of the production cost of absorption refrigeration units*

The main problem for the market penetration of thermally driven cooling systems is the first cost. All imaginable measures have to be done to reduce these first costs for the customer. In Austria a 100 plant program with support for the cost difference between conventional vapor compression cooling plant and solar thermal driven cooling systems is suggested. Participating countries should also consider a similar proposal in their own countries.

➢ *Reduction of the electric consumption of the auxiliary drives*

A 10 kW market available absorption refrigeration plant needs the following electrical power for the auxiliary drives:

- 450 W for the working fluid pump

- o 60 W for the pump in the generator circuit
- o 60 W for the pump in the brine circuit
- o 500 W for cooling water in the cooling tower water circuit
- o 350 W for the fan of the cooling tower
- o 50 W for the control unit

This is in sum total 1970 W. The potential for reducing this amount lies in the electrical energy for the pumping of fluids. The power for pumping fluids in the internal and external circuits could be generated by a suitable thermodynamic process out of the driving heat of the absorption refrigeration unit. This means that 1070 W have to be generated with an efficiency of about 8% by using the driving heat with a temperature of around 80°C. Suitable concepts for those processes are already available at ECONICsystems.

The reduction of the electric energy consumption of the cooling tower fan could be realized by fan speed control led by the cooling water temperature. For solar cooling applications PV panels with a surface of about 12 to 15 m2 could also be used for a grid independent 10 kW solar cooling plant. These proposals have to put into practice for doing the first steps towards a CO_2-free cold production.

<u>*Heat rejection*</u>

New possibilities for heat rejection from thermally driven sorption machines are limited. Despite the well known disadvantages (water consumption, electric power consumption, winter operation, hygienic problems) the wet cooling tower is from the thermodynamic point of view one of the best technical options.

Electrical consumption: Proposals for the reduction of the electrical consumption have already been discussed above.

Hygienic measures: A lot of measures for a hygienic cooling tower operation are technically proven and state-of-the-commerce. Beside chemical measures, like injection of biocides, also physical methods, like water treatment by UV-light or metal ion injection could be implemented. In particular, the metal ion disinfection was tested recently at ECONICsystems. Interesting results are already available.

Minimization of the water consumption of cooling towers: A wet cooling tower needs water to transfer the thermal load by evaporation to the ambient air. The evaporated water is replaced by fresh water and the content of calcium and magnesium rises constantly. If the content of magnesium and calcium rises above the maximum concentration, they fall out as a solid powder at the packing material in the cooling tower or as solid surface in heat exchangers or tubes. So, to avoid this effect, additional water has to be replaced during the operation of the cooling tower. The best technical solution for an economic water replacement in a wet cooling tower will be the measurement and the automatic control and limitation of the conductivity (µS/cm) of the water in the wet cooling tower.

<u>*Development of an air cooled absorption refrigeration unit*</u>*:* An interesting aim of a technical development is the replacement of water for the heat rejection. Hot and dry locations do not have in any case water in the necessary quality and quantity for wet heat rejection. Table 2 shows very generally the typical key data for four locations for a dry heat rejection operation.

All cases in table 2 have an evaporation start temperature of 0°C.

The process data of these four applications of absorption refrigeration units show that dry heat rejection needs higher temperatures of the heating medium and works at significantly higher process pressures. New components of the refrigeration unit and adaptation of the plant design would be necessary.

Table 2: Process data in case of dry heat rejection for four locations (single stage, continuous operation, evaporation at 0°C, NH3/H2O)

Location	Outside air temp.	Low process temp.	High process temp.	Conden-sation pressure	Heating medium temp.	WF concen-tration
	°C	°C	°C	bar	°C	%
Naples	30	40	98	16,0	105/95	47/41
Athen	33	43	112	18,0	117/109	43/37
Johannesburg	28	38	96	15,5	101/93	48/42
Abu Dhabi	41	51	118	22	123/115	40/34

2.5 Relevant references – Ammonia/Water Systems

In particular, references [2] and [3] give the reader the answers to the key questions of absorption refrigeration technology.

[1] Mouchot, Augustin, La chaleur Solaire et ses Applications Industrielles, (German translation 1987, published in Olynthus Verlag, 1987)
[2] Niebergall, W., Sorptionskältemaschinen, Handbuch der Kältetechnik, Bd 7, Springer Verlag, 1959.
[3] Bogard, M., Ammonia Absorption Refrigeration in Industrial Processes, Gulf Publishing Company, Book Division, Housten, Paris, London, Tokyo
[4] Grosman, G., Bourne, J.R., Ben-Dror, J., Kimichi, Y., Vardi, I., Design improvements in LiBr absorption chillers for solar applications, Transactions ASME, Journal of Solar Energy Engineering, 103 56-61 (1981)
[5] Podesser, E., Solare Kühlung, Dissertation an der Technischen Universität Graz, Fakultät für Maschinenbau, Institut für Wärmetechnik (Prof. P. V. Gilli), 1984.
[6] Podesser, E., Enzinger, P., Gossar, H., Monschein, W., Taferner, I., Proceedings of the XVII International Congress of refrigeration, Vienna 1987.

2.6 Description of the technology – Water/Lithium Bromide Systems

Robert Ghirlando (University of Malta)

The principle of operation of absorption cycles has already been described in detailed in the preceding sections. Hence this section will limit itself to pointing out some important differences between the two working fluid pairs, i.e. ammonia/water and water/lithium bromide.

The most common combinations of refrigerant and absorbent are ammonia/water and water/lithium bromide. Since in the water/lithium combination, the refrigerant is water, the condensing temperature cannot go below 0^0C; in fact the minimum temperature that can be reached is 3^0C. This is a problem in refrigeration, but not in air-conditioning. On the other hand, in the ammonia/water pair, the refrigerant is ammonia which can condense at temperatures below zero and can therefore be used in refrigeration applications. Water has a high latent heat and its use as a refrigerant in H_2O/LiBr systems is an advantage. It is important to point out that in the H_2O/LiBr system, for the water to evaporate at the very low temperatures required to produce the desired cooling effect, it needs to be at a very low pressure. At 4^0C, for example, water vapour pressure is only 0.8kPa, i.e. 8millibar. On the other hand, ammonia-water systems operate above atmospheric pressure. Also LiBr tends to crystallize, and methods have been devised to prevent this from happening. These characteristics can be clearly seen from the figure 6 below which shows the temperature-pressure-concentration diagram of saturated LiBr-H_2O solutions.

In a paper in which he compares the two working fluid pairs, Horuz [7] points out that the water/lithium bromide solution has "high safety, high volatility ratio, high stability and high latent heat". He also shows that, based on calculations, the COP is slightly better for a system using H_2O/LiBr than for one use NH_3/H_2O. Typically, the COP of a single effect H_2O/LiBr machine would be in the range 0.65-0.75 going up to 0.9-1.2 for a double effect chiller [8]. Higher COPs can be obtained by using double or triple effect machines, but then the generator (driving) temperatures need to be higher.

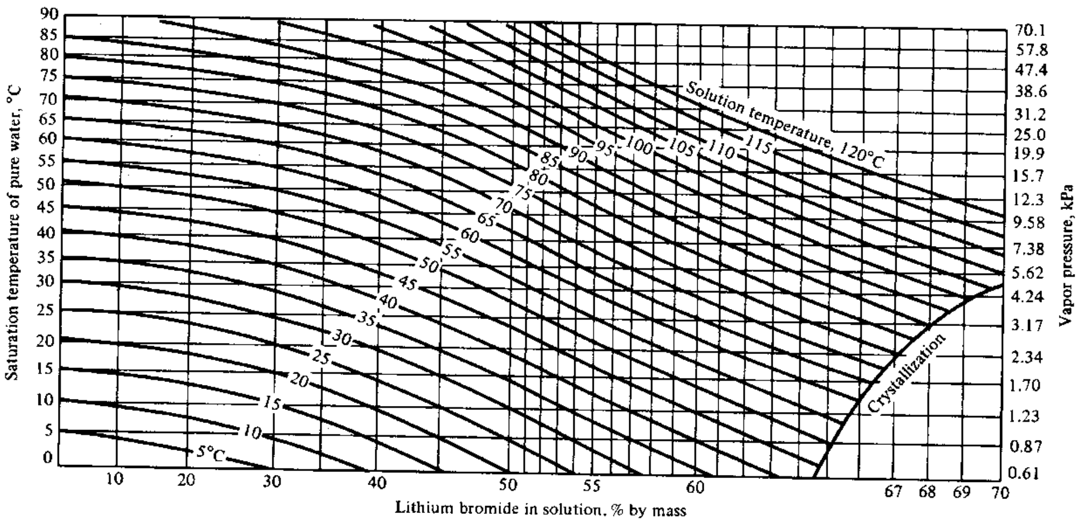


Figure 6: Temperature-pressure-concentration diagram of saturated LiBr-H_2O solutions (source: Stoecker W.F. & Jones J.W. Refrigeration and Air Conditioning, 2nd Edition, McGraw-Hill, ISBN 0-07-066591-5)

2.7 Market situation for Water/Lithium Bromide Systems

The following companies are producing water/LiBr absorption chillers.

Yazaki, Japan, produced the first solar-powered air-conditioning system in 1974. In 1970, they started production of absorption chiller/heaters of 12.25 & 17.5 kW capacity. In 1977, production of first solar collector with selective absorption surfaces. In 1980 start of production of double-effect steam fired chiller/heater (70 & 105 kW). In 1984, start of production of high-performance chiller with COP of 0.95. In 1996, start of production of excess heat fired chiller of 175kW. In 1998, start of production of gas driven chillers ranging from 455 to 700kW. (ref: www.yazaki-airconditioning.com/aircondition/history/html). Yazaki produces single-effect and double-effect chillers in the range of 17kW to 700 kW. (ref: www.yazaki-airconditioning.com/index.php?id=106)

Dalian Sanyo Refrigeration Co Ltd, Japan, started the manufacture of steam-fired LiBr absorption chillers in 1992. Their chillers feature a crystallization prevention control system. (http://www.dl-sanyo.com/beifen/neweng/production.asp). Both the steam and direct-fired SANYO chillers utilize a double-effect absorption cycle resulting in unit COPs of 1.0 for the direct-fired chiller/ heaters and 1.2 for the steam-fired chillers. Capacities range from 100 to 1500 tons (ref: www.overseas.sanyo.com/airconditioners/products/absorp/detail.html).

EAW-Energieanlagenbau Westenfield GmbH produce solar-driven absorption chillers with capacity 15kW, 30kW and 50kW (ref: www.eaw-energieanlagenbau.de).

Trane, USA, produce hot-water absorption chillers, single stage from 112 to 1350 tons and two-stage from 380 to 1650 tons.
(ref: www.trane.com/commercial/literaturesearchresults.aspx)

Entropie, France, is an engineering company with expertise in the design of absorption chillers using Li Br. (www.entropie.com/en/about/).

York, USA, produce single-stage absorption chillers from 120 to 1380 tons two-stage chillers from 200 to 700 tons. (ref: www.johnsoncontrols.com/publish/us/en/products/building_efficiency/integrated_hvac_systems/Industrial___Commercial_HVAC_Equipment/chiller_products.html)

ROTARTICA, Spain, are using rotation techniques to increase the efficiency of the absorption cycle, thereby reducing the size of the appliance and allowing it to be installed without the need for a cooling tower. They build a single-effect system specifically for solar application that gives a cooling power of 4.5 kW (in terms of solar cooling) with a COP of 0.7. (ref: http://www.rotartica.com)

Broad, China, make two ranges of LiBr chillers:

(i) with a cooling capacity ranging from 233 to 11630 kW, with direct firing, as well as steam, hot water and waste heat firing.
(ii) With a cooling capacity from 23 to 115kW, direct firing.

(ref: www.broad.com/english/include/en_index_pro.htm)

Carrier, USA, make LiBr chillers in the following configurations and sizes:

single-effect, steam fired	100 to 700 tons
single-effect hot water	75 to 525 tons
double-effect, direct fired	100 to 1500 tons
double-effect steam fired	98 to 1323 tons

(ref:www.commercial.carrier.com/commercial/hvac/general/0,,11_CLI1_DIV12_ETI1508_MID4369,00.html)

Century, South Korea, produce hot-water driven absorption chillers from 28 to 650 tons, COP of 0.725; direct-fired two stage absorption chiller from 20 to 1500 tons, with crystallization prevention function; two-stage steam absorption chiller with crystallization prevention function from 70 to 1650 tons. (ref: www.century.co.kr:8080/product/in_wa_absorption.asp)

SolarNext have a 17.5kW LiBr chiller.
(ref: www.solair-project.eu/uploads/media/10_Jakob_SolarNext.pdf)

Ebara Refrigeration Equipment and Systems Co Ltd, Japan, make double-effect LiBr absorption chiller from 528 to 2462, steam generated and single-effect hot water chillers from 158 to 1266 and single-effect steam or hot water from 334 to 5134 kW.
(ref: www.ers.ebara.com/en/product/tantai/1.html)

Thermax, India, produce steam driven single and double-effect LiBr chillers from 50 to 4000 tons, low temperature (75ºC-110ºC) hot water driven chillers from 10-1159 tons, medium temperature (110ºC-150ºC) hot water driven from 100-2000 tons, high temperature (150ºC-200ºC) hot water driven from 100-2000 tons, gaseous or liquid fuel driven from 40-1500 tons, multi-energy (any combination of hot water, exhaust gases, fuel or steam) driven Vapor Absorption Machine from 50-4000 tons.
(ref: www.thermxindia.com/v2/index.asp)

Hitachi, Japan, produce steam double-effect LiBr machines from 422kW to 2708 kW; direct gas-fired double-effect from 527kW to 3165kW. (ref: www.hitachi-ap.com/acc/ref_v.html)

Sonnenklima, Germany, make a small LiBr m/c (10kW) that operates with low temperature heat (55^{0}C). (ref: www.sonnenklima.de)

A report on Absorption Chillers (dated May 2001) prepared under the SAVE II Programme called CHOSE, “Energy savings by CHCP plants in the Hotel Sector” includes a survey of Lithium Bromide machines. The report includes a list of suppliers, as well as a short discussion on single-effect machines, double-effect machines and small capacity low temperate machines. (ref: www.inescc.pt/urepe/chose/reports/B-absorption_chillers.pdf). The list includes many of the suppliers mentioned above as well as the following about which no information was found on the internet:
LG Machinery, Korea
McQuay, USA
Mitsubishi, Japan
Toshiba, Japan
Kawasaki, Japan
Kyung Won, Korea

2.8 Research and Development – Water/Lithium Bromide Systems

This section lists organisations and individuals working in this field and projects relevant to LiBr, as well as some comments that were received in the course of this work.

1.The Department of Engineering Science of Martin-Luther University of Halle-Wittenberg have installed a 4.5 kW single-effect LiBr chiller with solar thermal collector.

2. Fundacion Cartif, Valladolid, Spain has been working on solar cooling since 1998, firstly through an ERDF (European Regional Development Funds) project, by means of which a solar air conditioning system was installed in its building and it has been kept working ever since 1999. The system is composed of a 40 m^2 vacuum collector field (mounted on a solar tracking platform), and a 37.5 m^2 flat collector field and a 35 kW absorption chiller for conditioning the administration area (200 m^2).

3. ALONE Project, Small Scale Solar Cooling Device: University of Florence – CREAR (Italy, Coordinator), EURAC Research (Italy), DLR (Germany), Solitem (Turkey), AOSOL (Portugal), Ikerlan (Spain), Climatewell (Sweden) and Riello (Italy) are partners in a 7th Framework Projct ALONE, Small Scale Solar Cooling Device. The main aim of ALONE project is to overcome the lack of small scale units, developing fully automated and autonomous package-solutions for residential and small commercial or industrial solar cooling applications based on systems able to cope with low temperature cooling applications. Main efforts are on absorption chiller optimization for providing both heating and cooling in solar systems: in fact, adaptation of components and control logic optimization is a necessary step towards higher conversion performances and reduced costs. At four test sites three different solar cooling technologies will be applied (Ammonia system, Lithium bromide system, Lithium chloride system). The test sites are located in: Firenze (Italy) NH3 with MT, Alentejo (Portugal) NH3 with LT, Vitoria-Gasteiz (Spain) LiBr, and Bolzano (Italy) LiCl.

4. Ahmed Hamza H. Ali, PhD, Ibrahim M. Ismail PhD, M. G. Morsy PhD, and I. S. Taha PhD, Department of Mechanical Engineering, Faculty of Engineering, Assiut University, Egypt.

5.Prof. Felix Ziegler, Technische Universität Berlin, Institut für Energietechnik.
http://www.eta.tu-berlin.de/felixziegler.html

6. Dr. Schweigler, ZAE Bayern, Dept I.
http://www.zae.physik.tu-muenchen.de/deutsch/abteilung-1/arbeitsgebiete/kaeltemaschinen-und-waermepumpen.html

7. L. Richter, M. Kuhn, M. Safarik, ILK Dresden, www.ilk.de have worked on chilled water generation below 0 °C by a water-LiBr resorption cycle.

8. According to Peter Noeres (Fraunhofer UMSICHT), future R&D-perspectives are in the further development of small units to get better values of COP, better performance; air-cooled absorber/condenser, chiller with integrated recooling system; size/cost reduction; modification of the working pair to manage/avoid crystallization.

9. Report [9] by CHOSE for EU Directorate General for Energy (31/05/01) – Energy Savings by CHCP plants in the Hotel Sector – Absorption Chillers, already mentioned above.

10. Research group belonging to the Department of Habitability, Energy and Environmental Impact in Buildings of the Instituto de Ciencias de la Construcción Eduardo Torroja (CSIC) located in Madrid, Spain. They are working on three research projects on air cooled LiBr prototypes of small power and small size. The main results obtained are:
Two air cooled double effect prototypes: one gas fired (2006-2007) and the other one driven by residual heat (2003-2006).
One air cooled single effect prototype designed using recooling (1996).
One Spanish patent (1999).
One International Patent in edition phase (2007).
1982-1985. Solar cooling systems constructed in the Solar Energy Experimental Plant of CSIC in La Poveda, Arganda del Rey, Madrid.
1997-2004. Solar cooling system constructed in the Madrid Carlos III University
R&D themes

- Development of absorbers of small size able to work with outdoor temperatures of around 40-42ºC.
- Simulation, design, construction and experimental evaluation of double effect lithium bromide-water, air cooled prototypes of small cooling power (below 10 kW).
- Simulation, design, construction and experimental evaluation of single effect lithium bromide-water, air cooled prototypes of small cooling power (below 10 kW).
- Air cooling lithium bromide systems driven by thermal solar energy.

11. Consortium for Alternative Cooling Technologies & Applications (ACTA), Center for Environmental Energy Engineering, University of Maryland.

12. Christopher Kren, Technical University of Munich – list of publications (1998-2007) indicates he is working on LiBr.

13. Powerpoint report by CSIRO Perspective on Renewable CHP & Absorption Refrigeration Laboratory, Dr Dong Chen and Dr Stephen White, National Energy Transformed Flagship, CSIRO, 14 May 2007, Renewable CHP. (ref: www.sustainability.vic.gov.au/resources/documents/CSIRO_Solar_Cooling.pdf).

14. Infante Ferreira, C.A. [C.A.InfanteFerreira@tudelft.nl], together with Dong-Seon Kim (now Arsenal Research) have been exploiting the advantages of half-effect LiBr-H2O absorption systems in combination with flat plate solar collectors to take advantage of the higher efficiencies at lower heating medium temperatures.

15. Egilegor Bakartxo [BEgilegor@ikerlan.es], member of Ikerlan, which is an applied research center. Part of its Energy Group collaborates intensively with Rotartica in its products development. www.ikerlan.es. Latest innovations: The reliability of the systems

and control has been improved. Future R&D perspectives: the main R&D effort now is focused on the following topics:
* Improved heat rejection systems in order to avoid cooling towers and improve the efficiency
* Reduce the cost of the absorption units and the global installation
* Definition of kits in order to make the installations easier and at lower cost

16. Polysmart, An Integrated Project partly funded by the European Commission under FP6, DG 'Energy and Transport' on POLYgeneration with advanced Small and Medium scale thermally driven Air-conditioning and Refrigeration Technology. The main purpose of the project was to support the development of the market for such technology. The work included a market analysis, the development of design models and tools, training and dissemination material, as well as a number of demonstration projects in various countries using different combinations of CHP and thermally driven chillers. http://www.polysmart.org

2.9 Relevant reports and publications – Water/Lithium Bromide Systems

[7] Horuz I., A Comparison Between Ammonia-Water And Water-Lithium Bromide Solutions in Vapour Absorption Refrigeration Systems, International Communications in Heat and Mass Transfer, vol. 25, No. 5, pp. 711-721, Elsevier, 1998.
[8] Kalogirou S., Florides G., Tassou S., Wrobel L., Design And Construction of a Lithium Bromide Water Absorption Refrigerator, 7th World Congress CLIMA 2000, Naples, 2001.
[9] Absorption Chillers, report prepared as part of the CHOSE (Energy Savings by CHCP plants in the Hotel Sector) project, SAVE II Progamme, Directorate-General for Energy, European Commission, May 2001.
[10] Estiot, E., Natzer, S., Harm, M., Kren, C., Schweigler, C., Heat Exchanger Development for Compact Water/Libr Sorption Systems, Proceedings of the International Sorption Heat Pump Conference, 2005, Denver, USA.
[11] Zogg R.A., Feng M.Y., Westphalen D., Guide to Developing Air-Cooled LiBr Absorption for Combined Heat and Power Applications, Distributed Energy Report, US Department of Energy, April 2005.
[12] Zogg, R. A., Feng, M., Westphalen, D., Developing Air-Cooled LiBr Absorption for Light Commercial Combined Heat and Power, Proceedings of the International Sorption Heat Pump Conference, 2005, Denver, USA.
[13] Killion, J., Garimella, S., Measurement of Local Absorption Rates in Films and Droplets in Lithium-Bromide/Water Absorbers, Proceedings of the International Sorption Heat Pump Conference, 2005, Denver, USA.
[14] Harm, M., Kren, C., Storkenmaier, F., Nogues, M., Schweigler, C., Experimental Evidence of the Vapor Surfactant Theory for Heat and Mass Transfer Enhancement in Water/Libr Absorption Chillers. Proceedings of the International Sorption Heat Pump Conference, 2005, Denver, USA.
[15] Estiot, E., S. Natzer, M. Harm, C. Kren, and S. Schweigler, Heat Exchanger Development for Compact Water/Libr Sorption Systems, American Society of Mechanical Engineers, Advanced Energy Systems Division (Publication) AES, 2006.
[16] Miyara, A., Islam, M.A, A Numerical Simulation of Steam Absorption by Wavy Falling Film of Libr Aqueous Solution, Proceedings of the 22nd International Congress of Refrigeration, 2007, Beijing.
[17] Takahashi, H., Koyama, S., Experimental Study of Falling Film Absorption Heat and Mass Transfer on the Horizontal Enhanced Heat Transfer Tube of Double Fluted Type with Libr Solution. Proceedings of the 22nd International Congress of Refrigeration 2007, Beijing.
[18] Clauß, V., Field Testing of a Compact 10 kW Water/LiBr Absorption Chiller, Proceedings of the 2nd International Conference on Solar Air-Conditioning, Tarragona, Spain, 2007.
[19] Macia A., Bujedo L.A., Vicente J., De Torre C., Development of a Model for the Simulation of an Absorption Chiller Air-Cooled “Rotartica” By TRNSYS, 3rd International Conference on Solar Air-conditioning, Palermo, 2009.

[20] Ajib S., Safarik M., Richter L., Kuhn M., Guenther W., Weidner G., Development of a 5 kW Absorption Chiller for Solar Installations, 3rd International Conference on Solar Air-conditioning, Palermo, 2009.
[21] El May S., Sayadi S., Bellagi A., Feasibility of Air-cooled Solar Air-conditioning in Hot Arid Climate Regions, 3rd International Conference on Solar Air-conditioning, Palermo, 2009.
[22] Marcos J.D., Izquiedo M., Lizarte R., Palacios E., Performance Optimization of a Water-Cooled Single-Effect Libr/Water Low Power Absorption Machine, 3rd International Conference on Solar Air-conditioning, Palermo, 2009.
[23] Rezai S.H., Witzig A., Wolf A., Pfeiffer M., Lacoste B., Modelling and Analyzing Solar Cooling Systems in Polysun, 3rd International Conference On Solar Air-Conditioning, Palermo, 2009.
[24] Jung S., Cagni A., Solar Cooling Application in Valle Susa Italy, 3rd International Conference on Solar Air-conditioning, Palermo, 2009.
[25] Asdrubali F., Baldinelli G., Presciutti A., An Experimental Solar Cooling System with a Small Size Absorption Chiller: Design and First Measurements, 3rd International Conference on Solar Air-conditioning, Palermo, 2009.
[26] Pospisil J., Chroboczek L., Skala Z., Small-Scale One Stage Libr-H_2O Absorption Chiller with Identical Design of Desorber and Absorber, 3rd International Conference on Solar Air-conditioning, Palermo, 2009.
[27] Onda N., Yokoyama T., Oka M., Homma R., Kajiyama K., Demonstration and Field Test of a Solar Air Conditioning System with an Absorption Chiller/Heater Operated with Solar Thermal Energy and/or Gas Fuel for Commercial Buildings, 3rd International Conference on Solar Air-conditioning, Palermo, 2009.

2.10 New Advances In Absorption Lithium Bromide Technologies For Solar Refrigeration

Marcelo Izquierdo Millán (Instituto de Ciencias de la Construcción Eduardo Torroja (CSIC), Universidad Carlos III de Madrid (UC3M)) and José Daniel Marcos del Cano (Universidad Nacional de Educación a Distancia (UNED)

2.10.1 Introduction

The Eduardo Torroja Institute for Construction Science, a Spanish National Research Council (CSIC) body, is the sponsor of an "Energy Saving an Emissions Reduction in Buildings" research group. In 2006 the group set out to build several prototypes of low-power lithium bromide-water absorption chillers, capable of competing economically with mechanical compression chillers, an endeavour funded by Spain's Ministry of Science and Innovation under the INVISO (Industrialization of Sustainable Housing) sub-project "Generación Sostenible de Energía en Viviendas" (sustainable energy generation in housing). The first prototype was an air-cooled, direct-fired, double effect absorption chiller; the second a solar powered single effect air-cooled absorption chiller and the third a combination air-cooled, single-double-effect chiller, driven by solar power (or waste heat) while operating as a single-effect apparatus or by burning fuel in double-effect mode.

The development of such prototypes was contingent upon designing, building and testing a new generation absorber, smaller, more readily assembled and with higher mass and heat transfer coefficients than in place to date. This absorber was developed between 2003 and 2006 under research projects DPI 2002-02439 and ENE 2005-08255-CO2-01 with funding from Spain's Ministry of Industry. Once assembled and tested, the absorber was built into the 4.5kW; 7 kW and 10 kW power cooling prototypes.

2.10.2 Flat Sheet Adiabatic Absorber

The falling film type absorbers presently used in lithium bromide absorption chillers are characterized by a number of problems: low mass transfer, low heat transfer and large volume.

The adiabatic spray absorber developed and patented by Ryan [28] as a solution, is not itself free of drawbacks. According to its inventor [29], these include:

a) Low liquid flow rate, calling for many spraying units. As a result, the absorber occupies a great deal of space.
b) Variable droplet diameter and a significant proportion with diameters under 150 µ.
c) Droplets with diameters under 150 µ and, consequently, a substantial portion of the pumped mass flow is of no use for absorption purposes [29].
d) Considerable head loss. High power demand in the pump that drives the solution.

In an attempt to solve these problems, Warnakulasuriya and Worek [30] developed a spray absorber with 400-micron diameter droplets. This device delivered higher mass transfer than Ryan's spray and multiplied the performance of commercial falling film absorbers by about four-fold.

Tackling the problem from a different angle, the Research Group modified the spray absorber, first by building a parallel droplet spray absorber, patented in Spain, able to generate droplets with no pressure loss and diameters of over 300 µ [31]. This absorber was not commercially viable because the mass of the solution was difficult to control and the mass transfer coefficient was similar to Ryan's. A jet absorber subsequently designed and built, also featuring no pressure loss, improved mass transfer but solution mass remained uncontrolled [32]. The next step was to develop a flat sheet absorber resting on a slanted surface [33] able to operate at a higher flow rate and therefore greater mass transfer. While it was also readily controlled, its need for material support constituted yet another difficulty. To solve these problems, two new adiabatic absorbers were studied: one generating a cone-shaped sheet [34 and 38] and the other a flat fan-shaped sheet with an oval nozzle [39].

This flat sheet adiabatic absorber, which consists of a bank of sprayers with an oval cross-section spaced at a minimum distance from one another, has been patented by Izquierdo et al. [37]. This arrangement increases both the mass transfer coefficient and the area available for mass transfer, significantly reducing volume. Moreover, since the flow rate is higher in sheet than in droplet sprays, absorber volume can be reduced even further. The process is readily controlled and pressure loss is small [39]. As an adiabatic unit, it transfers mass and heat separately [29].

Its operation involves pumping the solution from the absorber to the single generator in the single effect version or to the high and low temperature generators in the double-effect prototype. Similarly, the solution is pumped from the absorber and recirculated in a heat exchanger, where the absorption heat is transferred either to the outside air or to the water in a cooling tower. This characteristic affords adiabatic absorbers greater heat transfer capacity in a smaller exchange area than falling film absorbers [29].

After the device was tested, the results were published in the *International Journal of Refrigeration* [36], *Energy Conversion and Management* [38] and *Applied Energy* [39]. The paper "Evaluation of mass absorption in LiBr flat-fan sheets" [39] compared the mass transfer of a flat-sheet spray generated by an oval spray nozzle to Ryan's and Warnakulasuriya and Worek's sprays. The conclusion drawn was that the mass transfer coefficient delivered by the flat-sheet spray was five times greater than by the Warnakulasuriya and Worek absorber. The improvement in mass transfer obtained with the flat-sheet absorber constitutes a significant development in absorber technology. This new generation of absorber is more efficient than Ryan's droplet spray and Warnakulasuriya and Worek's absorbers and than any commercial absorber in use today. The patent granted in 2009 in the European Union (PCT/EP2009/057061 and European Patent EP09162208.4) was subsequently extended to the USA, Japan, China, Canada, Brazil, Argentina and Venezuela [37].

2.10.3 Single-Double Effect Chiller Prototype

To the best of the authors' knowledge, the first reference to a combination lithium bromide single-double effect absorption chiller appeared in US patent 4 439 999 entitled "Absorption Type Refrigeration System" and granted to Akio Mori, Shozo Watanabe, Mitsunubu Matshunada, Kenzy Machizawa and Ryohei Minowa. The patent protects a combination water-cooled single-double effect chiller in which the high-temperature double-effect chiller is powered by the heat released by internal combustion engine exhaust gases and the single-effect generator by the cooling heat from the same engine. Its claims include the use of exhaust gas in the double-effect high-temperature generator, cooling engine heat from the engine in the single-effect generator and spray in the absorber.

The new single-double effect prototype developed by this Research Group, patented in the European Union PCT/EP2009/057061 under European Patent EP09162208.4, features the following innovations:

Flat sheet adiabatic absorber able to operate in extreme outdoor temperatures (tested up to t_{ex} = 44 ºC), at which the solution does not crystallize.

Air-cooled (usable also for water-cooled systems) condensing to 54 ºC.

System for direct absorber and condenser cooling.

This combination single-double effect prototype, integrated into a solar cooling system, can use the heat from that renewable source (or the exhaust heat from an internal or external (turbine) combustion engine) to power the single-effect generator. When the renewable or exhaust heat is depleted, the system operates like a direct-fired double-effect facility in which fuel is burnt with maximum efficiency to supply heat to the high-temperature double-effect generator. Where a building's demand is partially covered by renewable, solar or biomass energy or waste heat, the single-double-effect systems can be used simultaneously, burning fossil fuel in the double-effect system to meet all the building's demand.

2.10.3.1 Description

Figure 9 shows the absorption chiller prototype that alternates between single-double effect operation, in which a series of valves enables one system or the other. The prototype has three operating modes: single-effect, double-effect and simultaneous. The bold line represents fluid circulation in single-effect operation, in which only the valveS V1S, LEV, V3 AND V5, while the valves 1B, V2LG, V1, V2, V4 and HEV remain closed.

In double-effect mode, only the valves VIS, V3 and V5 are closed, while the valves 1B, V2LG, LEV, HEV, V1, V2 AND V4 remain closes. In simultaneous mode the solution pump (SP) feeds to all three generators.

The following components are common for the three operation modes: absorber (A), evaporator (E), condenser (C), solution pump (SP), recirculation solution pump (RSP), heat exchanger (HEX), fan (F). In the figure 4 we can see a photograph of the single-double effect.

Nominal cooling capacity of this prototype is 4,5kW in single effect mode operation; 7 kW working as double effect mode operation and 11 kW if the operation mode is single and double effect simultaneously.

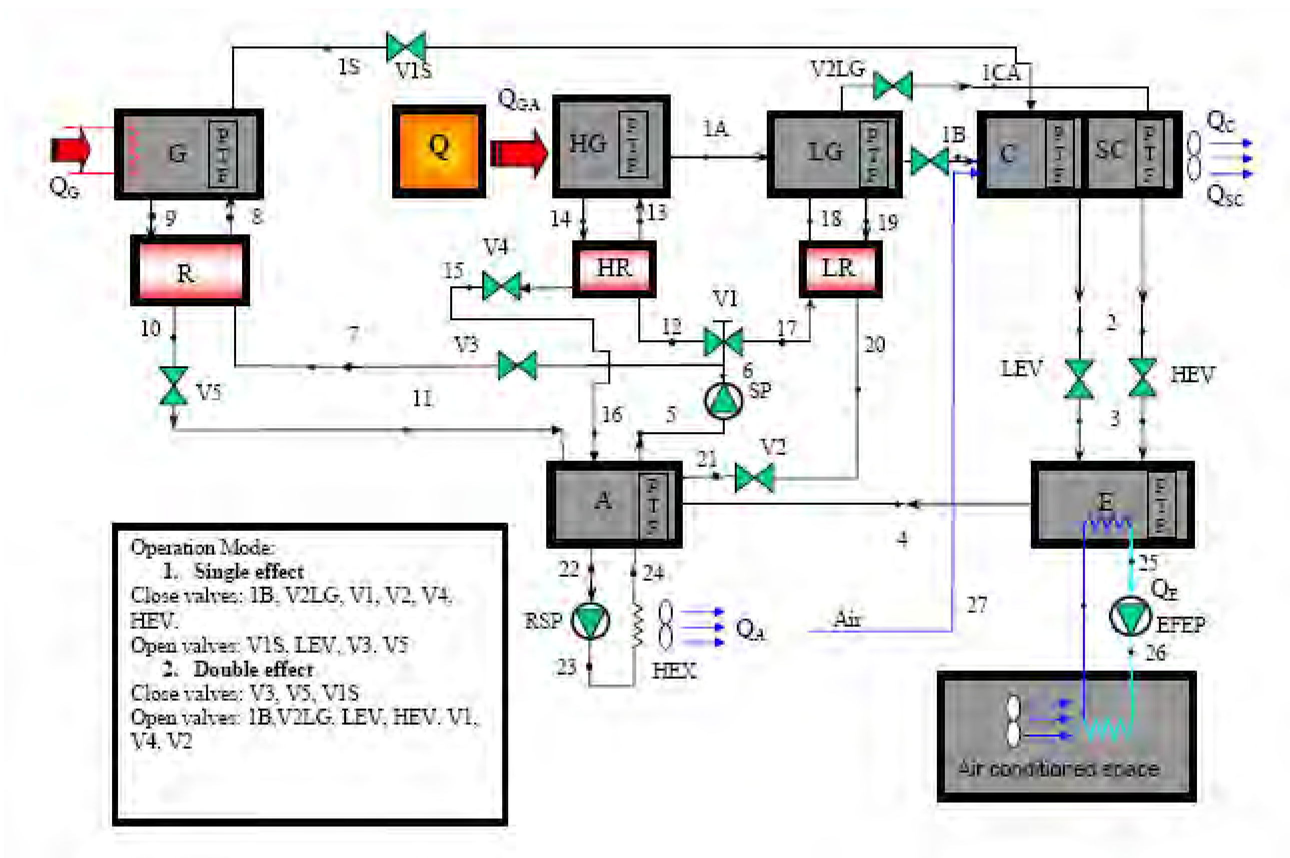


Figure 9: Flow chart for the air-cooled single-double effect prototype

2.10.3.2 Experimental Results

The prototype has been tested in single-effect operation mode during several days powered with thermal solar energy under environmental conditions of Madrid. The daily thermal COP obtained was between 0.55 and 0.7. The outlet evaporator chilled water temperature was between 15ºC and 17ºC when maximum outdoor temperature reached 38ºC. The maximum cooling power was 4.0 kW. No crystallization of the solution was observed.

Figure 8: Air-cooled, single-double effect lithium bromide absorption chiller prototype

2.10.4 Conclusions

Today's technology for thermally-driven domestic air-conditioning cannot compete with the cooling capacity of electric-powered air-cooled mechanical compression systems. Solar cooling systems use renewable energy, with a concomitant savings in fossil fuel. The initial investment is sizeable, however, while renewable energy is not always able to cover a building's total demand and the power consumed by the ancillary equipment is excessive. As a result, cooling costs are higher than with the conventional system. Nonetheless, inasmuch as these technologies are environmentally friendly, a research effort should be made to solve the problems associated with their current state of development. Against this backdrop, the Subproject SP3, "Sustainable power generation in homes", part of a broader INVISO Project, proposes to modernize absorption cooling technology as a first move toward the development of viable solar-powered cooling systems, or systems fired by renewable energy in general, including waste heat.

The first step was to design and build a new adiabatic absorber that increases the mass transfer coefficient of falling film absorbers 18-fold and the Warnakulasuriya and Worek apparatus (an improved version of the Ryan absorber) five-fold. This absorber was then built into a prototype for an air-cooled, direct-fired, lithium bromide double effect absorption chiller of 7 kW nominal cooling capacity that has been experimentally proven to be competitive with the air-cooled mechanical compression chillers used for domestic air-conditioning. A second prototype was also developed, consisting of a similarly air-cooled, combination single-double effect chiller. In this prototype the heat from a renewable (solar) source or waste heat is supplied to the single-effect chiller. When this heat source is depleted, double-effect, fossil fuel-fired operation is enabled. The high efficiency of this facility reduces cooling costs. The prototype can also operate simultaneously as a single and double effect chiller, for the two share essential components such as the absorber, condenser, evaporator, pumps and so on. The research group presently plans to experiment with this mode of operation. The experimental results set out above show that lithium bromide solutions are effective in very high outdoor temperatures, up to 44 ºC, at which temperature they do not crystallize.

The conclusions that can be drawn from these results are:

Air-cooled, natural gas-fired double-effect facilities can compete with electric-powered air-cooled mechanical compression chillers.

Single-double-effect chillers fired by waste heat from trigeneration systems are competitive with mechanical compression chillers.

Integrated in a solar heating-cooling system, the present single-double effect chiller can significantly reduce cooling costs and is coming closer to being competitive with electric chillers.

2.10.5 Future Work

The group's future research will aim to:

Reduce the cost of collector fields, increasing their performance to reduce the area needed and raising energy generation.

Reduce the cost of auxiliary equipmentet used by absorption machines.

Increase absorption chiller efficiency.

2.10.6 Acknowledgements

This research was funded by Spain's Ministry of Science and Innovation under the following projects: DPI 2002-02439; ENE 2005-08255-CO2-01 and the Subprogram SP3 of PSE INVISO. The authors wish to thank Emilio Martin, R&D Technician, for his enthusiastic contribution to this study.

2.10.7 References

[28] William A. Ryan, System and process for operating an absorption cycle around a crystallization curve of the solution Patent 5,255,534 (US), 1992.

[29] William A. Ryan, Water absorption in an adiabatic spray of aqueous lithium bromide solution, AES Vol. 31 International Absorption Heat Pump Conference, pp 155-162, (ASME 1993).

[30] Warnakulasuriya and Worek, International Journal of Heat and Mass Transfer 51, 3362–3368, 2008.

[31] M. Izquierdo, E. Martín, A. Lecuona, Absorbedor de gotas iguales de flujo paralelo. Spanish patent P9802674, Universidad Carlos III de Madrid (UC3M), 1998.

[32] Marcelo Izquierdo Millán, Emilio Martín Lázaro y Antonio Lecuona Neuman, Máquina de absorción de doble efecto enfriada por aire y alimentada por calor de los gases de escape de motores térmicos y su procedimiento de aplicación para climatización N. de solicitud: 9901979, Consejo Superior de Investigaciones Científicas (CSIC), 1999.

[33] D. Arzoz, P. Rodríguez and M. Izquierdo, Experimental study on the adiabatic absorption of water vapour into LiBr-H2O solutions, Applied Thermal Engineering, Vol 25/5-6, pp 797, 811, UK, 2005.

[34] E. Palacios, M. Izquierdo, R. Lizarte, J. D. Marcos, Water vapour absorption in lithium bromide aqueous solutions conical sheets, 8th IIR Gustav Lorentzen Conference, Copenhagen, September 2008, ISBN: 978-2-913149-63-2; ISSN: 0151.1637, 7-10.

[35] J.D. Marcos, M.Izquierdo, R.Lizarte, M.E Palacios, C.A. Infante-Ferreira, Experimental boiling heat transfer coefficients in the high temperature generator of a double effect absorption machine for the lithium bromide/water mixture Internacional Sorption Heat Pump Conference, Seoul (Korea), 23-26 September 2008.

[36] J. D. Marcos, M. Izquierdo, R. Lizarte, E. Palacios, C.A. Infante Ferreira, Experimental boiling heat transfer coefficients in the high temperature generator of a double effect absorption machine for the lithium bromide/water mixture, International Journal of Refrigeration, Vol. 32 pp. 627-637, 2009.

[37] M. Izquierdo, E. Martín and M.E. Palacios, Absorber and absorber-evaporator assembly for absorption machines and lithium bromide - water absorption machines that integrate said absorber and absorber-evaporator assembly, Patent PCT/EP2009/057061 y European Patent EP09162208.4 (extended to USA, Japan, China, Canadá, Brasil, Argentina, Venezuela), 2009.

[38] E. Palacios, M. Izquierdo, R. Lizarte, J.D. Marcos, Lithium bromide absorption machines: Pressure drop and mass transfer in solutions conical sheets, Energy Conversion and Management, 50 (2009) 1802–1809, 2009.

[39] E. Palacios, M. Izquierdo, J.D. Marcos, R. Lizarte, Evaluation of mass absorption in LiBr flat-fan sheets, Applied Energy 86 (2009) 2574–2582, 2009.

[40] M. Izquierdo; J. D. Marcos; M. E. Palacios; A. González; E. Martín, New advances in solar cooling and trigeneration systems, 8th expert meeting of Task 38 (IEA), Solar Air Conditioning and Refrigeration, April 26-27 Aarhus, Denmark, 2010.

3 Adsorption Chillers

Tomas Núñez (Fraunhofer ISE)

3.1 General description of the technology

Adsorption systems are very similar to absorption systems. The main difference is that the sorbent is a solid and not a liquid. The adsorption process is a physical process where the molecules of the working fluid or refrigerant are bound to the surface of the adsorbent by Van-der-Waals forces. Adsorbents used in adsorption chillers are very porous technical adsorbents like silica-gel, zeolites and activated carbons. The high porosity and thus extremely large internal surface of the adsorbents (several hundreds of m² per gram of material) allows the adsorption of a significant amount of refrigerant.

The main characteristic of the adsorbent-refrigerant pair is the amount of adsorbed refrigerant per unit of dry adsorbent.

$$x = \frac{m_{refrierant}}{m_{adsorbent}} = x(T, p)$$

The loading x is a function of refrigerant pressure and temperature and is represented in isosteric diagrams.

Figure 9 shows a typical isosteric diagram of the pair silica-gel / water.

The chilling cycle is carried out through four processes (Figure):

1. *isosteric heating*: in section 1, the loaded adsorbent is heated at constant maximum loading x_{max}. The equilibrium pressure in the system increases until it reaches the condensation pressure of the refrigerant at condensation temperature T_{cond}.
2. *isobaric desorption*: the adsorbent is further heated up und thus desorbed at constant pressure. The desorption ends as soon as the maximum temperature T_{max} provided by the external heat source is reached. This process is endothermic taking up the desorption heat. The released refrigerant is condensed in the condenser.
3. *isosteric cooling*: the desorbed material is cooled down until the equilibrium pressure reaches the evaporation pressure of the refrigerant in the evaporator.
4. *isobaric adsorption*: further cooling down of the adsorbent results in the adsorption process; refrigerant is evaporated in the evaporator, thus producing the cooling effect, and taken up by the adsorbent. The process ends as soon as the temperature of the adsorbent reaches the heat rejection temperature and closing the cycle.

The choice of the adsorbent depends on the affinity to the refrigerant and the envisioned application. The cooling energy Q_{cold} in one cycle is proportional to the evaporation enthalpy h_{evap} of the refrigerant and the loading difference $x_{max} – x_{min}$. These loading limits are given by the operation conditions: the minimum loading is given by the maximum desorption temperature T_{Des} and the condensation temperature T_{cond}, the maximum loading by the evaporation temperature T_{evap} and the heat rejection temperature which is identical to the condenser temperature T_{cond}. The driving heat is given by the heat necessary for the isosteric heating Q1 and isobaric desorption Qdes. Thus, the thermal COP of an adsorption chiller is calculated as a first approximation through:

$$COP_{th} = \frac{Q_{cold}}{Q_{drive}} = \frac{h_{evap} \cdot (x_{\max} - x_{\min})}{Q_1 + Q_{Des}}$$

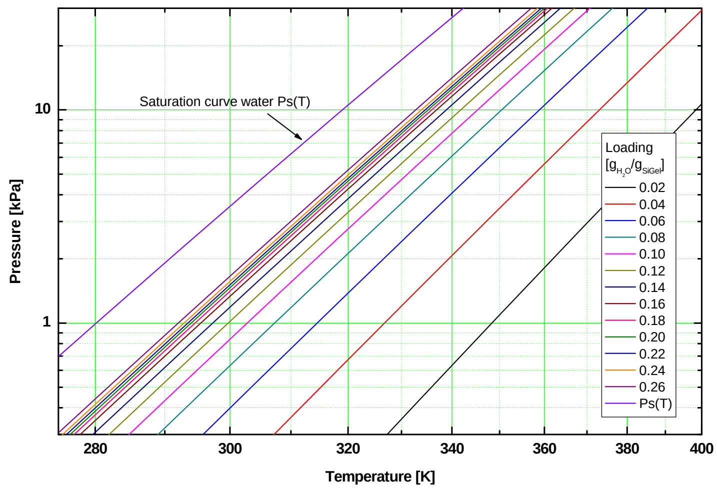


Figure 9: Isosteric diagram of the adsorption pair silica-gel / water. The isosteres represent lines of constant loading x.

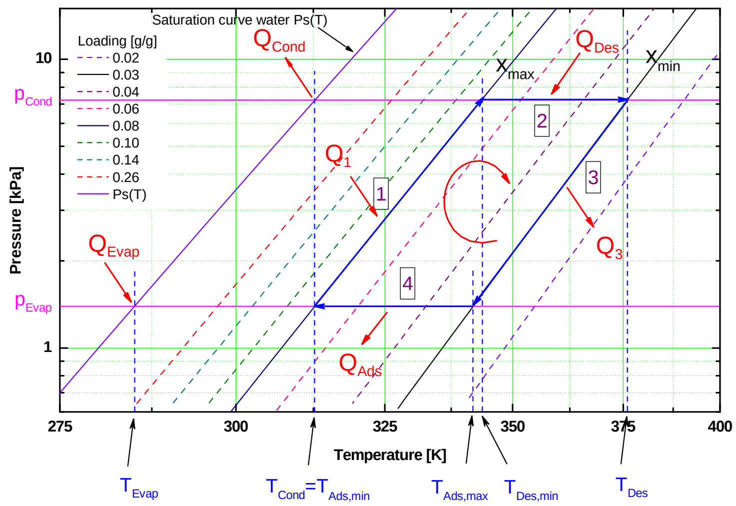


Figure 10: The adsorption cycle in the isosteric diagram of the silica gel / water adsorption pair.

In order to maximize the COP of an adsorption chiller the challenge is to increase the loading difference at the given boundary conditions and reduce the driving heat required from the external heat source. The first measure is achieved by specifically designed adsorption materials and adsorption pairs. This is still a research topic requiring basic material research. The second measure requires the reduction of the thermal masses for the adsorber and/or the implementation of an effective heat recovery in system designs with more than one adsorber. The typical design of an adsorption chiller is a two adsorber design. In this design two adsorbers are connected to a common evaporator and condenser via self-actuating flaps.

Figure shows a picture of a typical adsorption chiller.

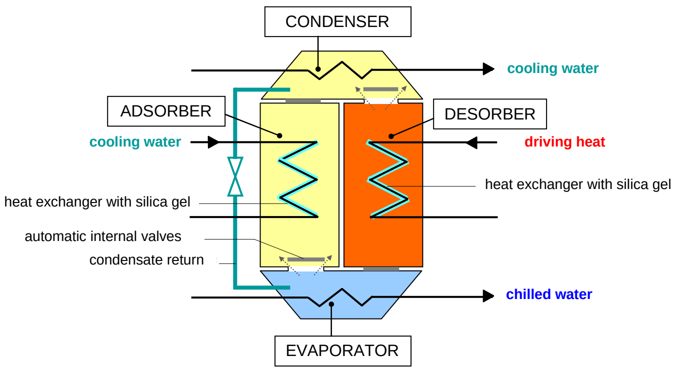


Figure 11: Scheme of a typical adsorption chiller with two adsorption chambers, one evaporator and one condenser.

The two adsorber adsorption chiller is operated in four phases:

1. Phase 1: the left reactor is regenerated, the right reactor is in the adsorption phase.
2. Phase 2: heat from the left, hot adsorber is used to pre-heat the cold adsorber on the right hand side
3. Phase 3: is similar to phase 1 with exchanged adsorbers
4. Phase 4: similar to phase 2 with exchanged adsorbers.

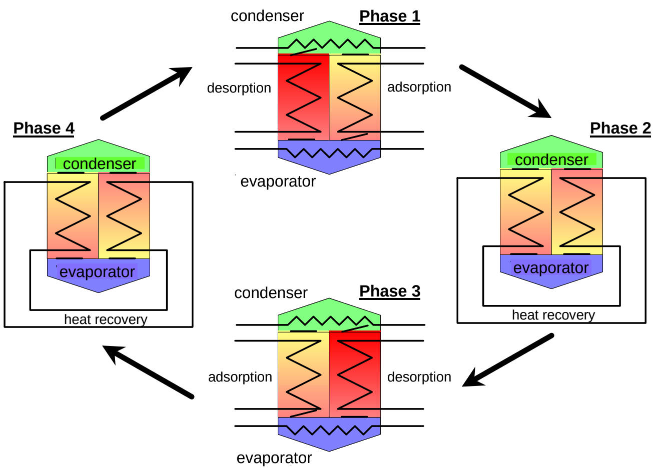


Figure 12: The four phases of an adsorption chiller with two reactors.

3.2 Main characteristics

The time for the heat recovery is critical: on the one hand a very good heat recovery is necessary to achieve reasonable COPs, on the other hand during the heat recovery process normally no cold is produced, thus reducing the cooling power.

Due to the intermittent, quasi continuous operation of such an adsorption chiller, the temperatures and thus the power in the three hydraulic circuits are fluctuating. Figure 13 shows a typical temperature curve of an adsorption chiller.

Depending on the construction type, the hydraulic circuits may be interrupted during the heat recovery phases.

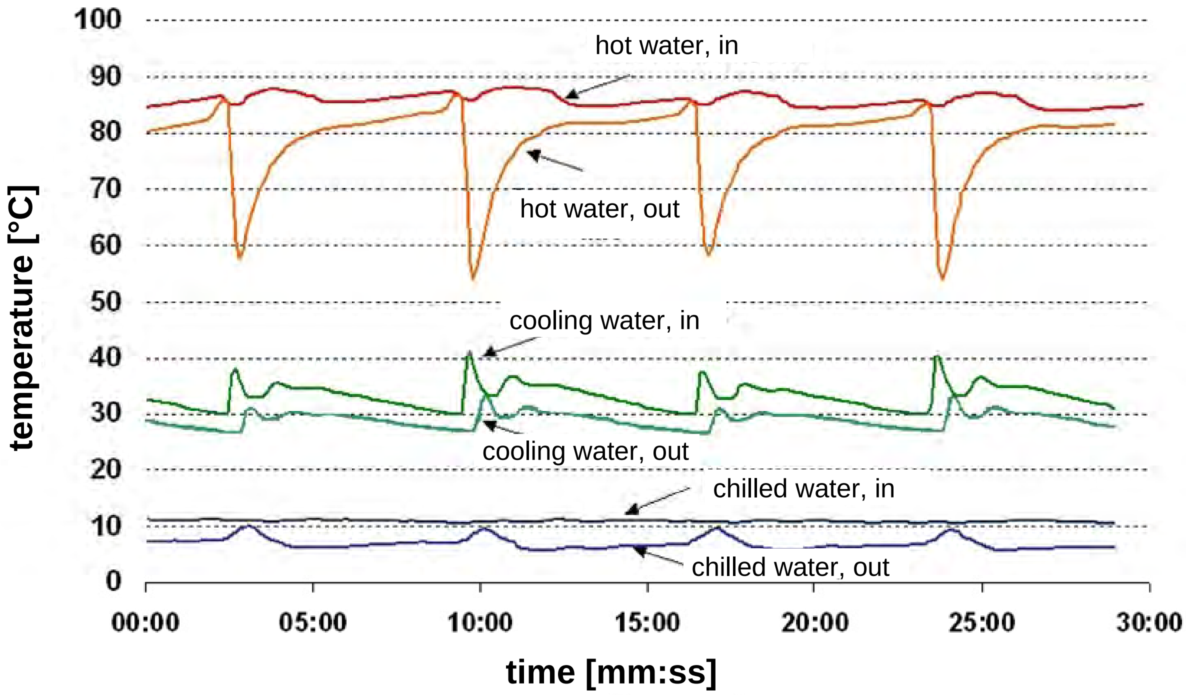


Figure 13: Typical temperature evolution in the forward and return lines of an adsorption chiller during operation.

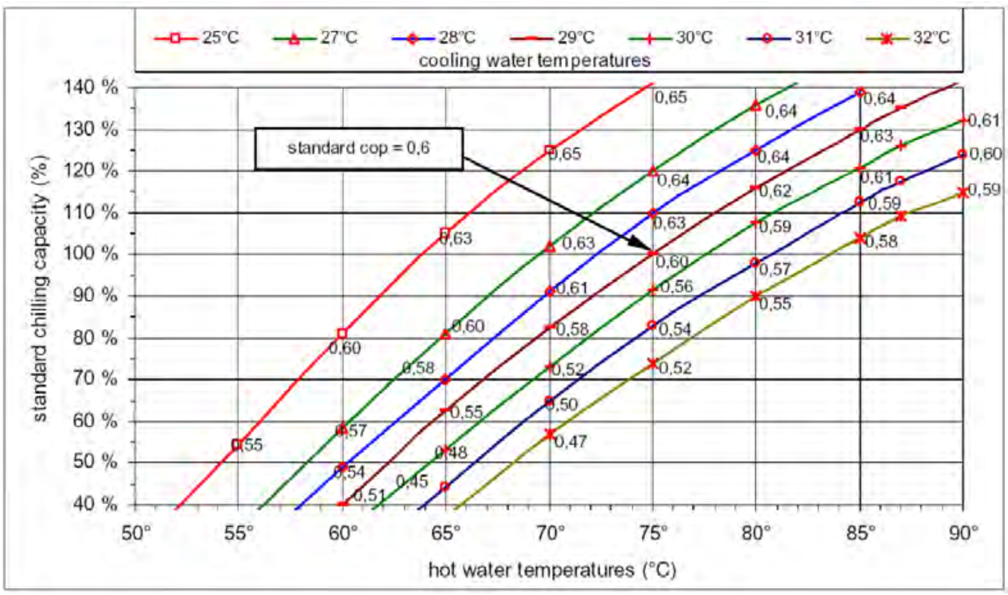


Figure 14: Cooling capacity and COP of an adsorption chiller (Mycom ADR) as a function of hot water temperature for a chilled water temperature of 14°C / 9°C and different heat rejection temperatures.

The performance of an adsorption chiller depends on the operating conditions. High heat rejection temperatures reduce the COP and power of the machines. Typical COP values are around 0.6.

As normally there is no pump within the machine, the operation is almost noiseless. It is possible to modulate the power within a certain range but this affects the COP. Normally efficiencies are higher under part load operation.

The main advantages are:

- It is a robust technology with no risk of crystallization, no danger of damage due to temperatures.
- The materials used today (zeolite, silica gel) are environmentally friendly.
- Very low intrinsic electricity consumption due to the lack of a pump. Electricity is only required for the switching valves and the control unit.
- Very little moving parts with the potential of low maintenance effort and costs.
- High potential of cost reduction in series production due to the small amount of individual parts.

The main disadvantages are:

- High requirements of vacuum tightness of the container.
- Slightly lower COP than for comparable absorption technology.
- Cyclic temperature variation in the hydraulic circuits requires careful design of the external hydraulic circuits.
- Commercially available machines are expensive and only few suppliers on the market.

3.3 State of the art and present R&D topics

The use of adsorption technology for cooling is a new technology compared to absorption technology. Thus, a lot of R&D is still going on in the field of material research, heat & mass transfer and component & machine development.

The next paragraphs describe the resent developments in the field of heat & mass transfer, working pairs, component development and projects involving state of the art chillers. A good overview of the techniques is also given in “Klimatisierung, Kühlung und Klimaschutz: Technologien, Wirtschaftlichkeit und C02 -Reduktionspotentiale: Materialsammlung“ [41] in German language. For a short English review see Wu et al. [41].

3.3.1 Heat & Mass Transfer

A major focus of research in the field of adsorption chillers is on the topic of heat and mass transfer. Due to the fact that adsorption chillers are periodic working chillers, the COP is limited by the sensitive heat capacity of the adsorber. Therefore, the scope of the research is to reduce the weight of the chillers. This will lead to higher COPs. At the same time also the size of the chillers is in the focus of scientists. There is an urgent need of smaller and lighter adsorption chillers and therefore to increasing the specific cooling power per volume or mass. A good overview about the current state of the art in adsorber development can be found in Schnabel [47]. Research is now going on in the field of foams and sponges for compact adsorber development in order to increase the fraction of adsorbent coated surfaces to the required substrate mass; Bonaccorsi [49], Berg [48].

In order to increase the heat transfer coefficient between the substrate and the adsorbent two techniques are in the focus of research. One option is to glue the adsorbent onto the heat exchanger (see Zhu [58], and the patent of SorTech [76]. Moreover, several working groups work on the principle to directly crystallize adsorbent onto the heat exchanger substrate (Coronas [50] Beving [51], Yang [52], Scheffler [54], Tatlier [55] and SorTech [77]).

3.3.2 Working pairs

A wide variety of working pairs are possible solutions for application in adsorption chillers. Wongsuwan [56], Srivastava [57] and Henninger [60] give an overview of possible working pairs. Most working pairs use water as the refrigerant due to the highest evaporation enthalpy of water compared to other possible working fluids. Furthermore, water is not toxic and easy to handle. The disadvantage of water is its limitation to applications above 0°C. For ice making or refrigeration below 0°C, methanol or ammonia as refrigerant is used with a variety of solid adsorbents; the main focus is on activated carbons.

The following section gives an overview of classical and modern adsorption materials as well as the latest developments. Here the first mentioned substance is the refrigerant, while the second mentioned is the adsorbent. With regards to the application in heat transformation processes, the main scope of research is the stability of materials – which means hydrothermal and mechanical stability of the pure material and possible composites and new synthesis routes, with regards to production costs. The focus is on template free syntheses and general investigations on the water vapour adsorption characteristics.

3.3.3 Classical working pairs

Classical working pairs are water/zeolites, water/silica-gel and methanol/activated-carbon. A comparison between water/zeolite, water/silica-gel and methanol/activated-carbon can be found in San [72] or in the review of Dieng [71].

Zeolites are hydrated alumino-silicates which consist basically of SiO_4 and AlO_4. The coordination of these tethradrones form secondary building units which composes a 3-dimensional framework. Within this framework water can be adsorbed. In the “Atlas of Zeolite Framework Types” [62] 133 lattice types are known but the corresponding webpage now lists 179 different zeolite lattice types (June 2008). Hauer [61] and Henninger [60] give a good overview of zeolites and their properties.

Silica gels are solid, porous silicic acids produced by the synthetic dehydratisation of a hydro gel. Therefore, silica gel consists of more than 99% of SiO_2. The porosity can be varied by the control of temperature and ph-value. Material properties can e.g. be found in Núñez [63], Ng [67] and Aristov [59].

Using methanol as the refrigerant, the solid adsorbent is in most cases activated carbon. This offers the possibility of refrigeration below 0°C. Physical properties are given in Carrott [68]. Wang [69] focuses on the heat and mass transfer in methanol/carbon working pairs. Stoeckli [73] also gives data about water/activated carbon.

3.3.4 Modern working pairs

Modern working pairs are:

- water/selective-water-sorbents (SWS),
- water/aluminium phosphates (AlPOs),
- water/silica-aluminium phosphates (SAPOs) and
- water/metal-aluminium phosphates (MAPOs).

Selective-water-sorbents (SWS) and selective-water-sorbents-like materials are a composition of an adsorbent and a salt, which means in principle that adsorption and absorption occur at the same time, while the substrate is still solid. Aristov [59] introduced a salt impregnated silica gel. In Núñez [64] the potential of this material for heat storage was estimated.

AlPOs consist of alternating AlO_2^- and PO_4^+ units, forming a similar structure like zeolites. The share of Al- and P-atoms is equal (besides lattice defects). The anion framework is electrically neutral but high electro-negativity leads to good adsorption characteristics. Properties are given e.g. Henninger [60]. In Jänchen [65] next to water also data for methanol as refrigerant are given.

SAPOs have a similar structure like AlPOs. They are based on AlO_2^- and PO_4^+ units but some of the Al^{3+} ions are substituted by Si^{4+}. Also, the embedding of FeO_4 tetrahedrons is possible and already investigated (MeAPO). Information about these groups can be found in Jänchen [65] and Henninger [60]. A good overview of the embedding of metal ions into AlPOs is given by Rajic [66].

The most novel working pair is water/metal-organic-frameworks (MOFs). MOFs are based on an inorganic cluster, so to say, a coordinated network of inorganic compounds and connected together by a linker. The choice of the cluster and linker – the so called cluster/linker concept - allows varying the size of the cavity that is responsible for the uptake of refrigerant. MOFs are described by Henninger [60].

3.3.5 Component development

Several working groups are working on the development of different components in adsorption chillers or in the enclosed system like evaporators, new heat storages, etc. The focus is on the development of adsorbers and the concept of heat & mass recovery between multiple adsorbers. The relevant documents for the development of adsorbers have already been mentioned in the paragraph about heat & mass transfer. In the area of heat & mass recovery during the switching process for adsorbers, two papers should be mentioned: Pons [82] reports about mass recovery – the recycling of refrigerant vapour, which is easily possible in adsorption chillers with two or more beds, and Schmidt [81] describes a new method to recycle heat with the help of stratified storage.

3.3.6 State-of-the-art Chillers

Many projects demonstrate the working principle of adsorption chillers. The following is just a small list of applications.

The adsorption chiller described by Núñez [42] is now developed to the standard product of SorTech. Chwieduk et al [45] describe the SOCOOL project - two adsorption chillers working with heat sources at 200°C and 90°C. Working pairs are water/SWS1L and ammonia/activated-carbon.

De Boer [46] describes experiments with a 3,6 kW cooling power water/silica-gel adsorption chiller with hot water 80-90°C, cooling water 20-35°C and chilled water 6-10°C.

The group of Wang is working basically on two small scale adsorption chillers, a water/silica-gel adsorption chiller and an ammonia/activated-carbon adsorption chiller. A review is given in Wang [80]. Various reports (e.g. Wang [78] and Zhai [79]) show the development of the water/silica-gel adsorption chiller. More information about the ammonia/activated-carbon chiller, which is used for ice making can be found e.g. in Wang [70].

3.4 Suppliers

The main suppliers of adsorption machines can be classified in two groups: suppliers of large machines with chilling powers above 70kW and suppliers of small machines. Today the only provider of large adsorption chillers is the company Mayekawa from Japan offering Water-Zeolite machines with 108kW, 215kW and 429kW. For small machines two new players are offering their small-series products: SorTech from Germany (www.sortech.de) with an 8kW and 15kW water-silica gel chiller and the company Invensor also from Germany (www.invensor.de) with a 7kW and 10kW water-zeolite chillers. A further development has been carried out at the Shanghai Jiao Tong University in China but the status of the availability of a commercial machine is not known.

3.5 Literature list

[41] SIMADER, Günter, RAKOS, Christian, *Klimatisierung, Kühlung und Klimaschutz: Technologien, Wirtschaftlichkeit und C02–Reduktionspotentiale – Materialsammlung.* http://www.energyagency.at/(de)/publ/kkk_material.pdf - 7 March 2005 – Wien.

[42] WU, D.W., WANG, R.Z, Combined cooling, heating and power: A review. *Progress in Energy and Combustion Science* 32 (2006), p. 459-495.

[43] NÚÑEZ, T, MITTELBACH, W., HENNING, H-M., Development of a Small Capacity Adsorption System for Heating and Cooling Applications. *International Sorption Heat Pump Conference (2005),* June 22-24; Denver, Co, USA – proceedings.

[44] WANG, R.Z., KONG, X.Q., WU, J.Y., HUANGFU, Y., WU, D.W., Performance and Research of a Micro CCHP System with Adsorption Chiller. *International Sorption Heat Pump Conference (2005),* June 22-24, Denver, Co, USA – proceedings.

[45] CHWIEDUK, D., POMIERNY, W., CRITOPH, R.E., RESTUCCIA, G., FRENI, A., DE BOER, R., SMEDING, S.F., MALVICINO, C., Tri-generation in the tertiary sector. In: *World Renewable Energy Congress* VIII, 28.08.-03.09.2004, Denver, Colorado, USA – proceedings.

[46] DE BOER, R., SMEDING, S.F., GRISEL, R.J.H., Performance of silica-gel+water adsorption cooling system for use in small-scale tri-generation applications. *Heat Powered Cycles Conference 2006* – proceedings.

[47] SCHNABEL, Lena, *Experimentelle und numerische Untersuchung der Adsorptionskinetik von Wasser an Adsorbens-Metallverbundstrukturen*. Technische Universität Berlin, Fakultät für Prozesswissenschaften, 2008, PhD-Thesis.

[48] BERG, A., BAUMEISER, J., RAUSCH, G., BUSSE, M., SCHNABEL, L., BAUER, J., Application of Aluminium Sponges in the Adsorption Refrigeration Engineering, *5th international Conference on Porous Metals and Metallic Foams* (Montreal, Canada 2007) - conference proceedings.

[49] BONACCORSI, L., FRENI, A., PROVERBIO, E., RESTUCCIA, G., RUSSO, F., Zeolite coated copper foams for heat pumping applications. *Microporous and Mesoporous Materials* 91 (2006), p. 7-14.

[50] CORONAS, J., SANTAMARIA, J., The use of zeolite films in small-scale and micro-scale applications. Chemical Engineering Science 59 (2004), Vol. 22, p.4879-4885.

[51] BEVING, Derek E., O'NEILL, Cory R., YAN, Yushan, Hydrophilic and antimicrobial low-silica-zeolite LTA and high-silica-zeolite MFI hybrid coatings on aluminium alloys. *Microporous and Mesoporous Materials* 108 (Feb. 2008), p. 77-85.

[52] YANG, Guohui, ZHANG Xiongfu, LIU, Shuqin, YEUNG, King Lun, WANG, Jinqu, A novel method for the assembly of nano-zeolite crystals on porous stainless steel microchannel and then zeolite film growth. *Journal of Physics and Chemistry of Solids* 68 (2007), p. 26-31.

[53] BONACCORSI, Lucio, FRENI, Angelo, PROVERBIO, Edoardo, RESTUCCIA, Giovanni, RUSO, Fabio, Zeolite coated copper foams for heat pumping applications. *Microporous and Mesoporous Materials* 91 (2006), p.7-14.

[54] SCHEFFLER, Franziska, HERRMANN, Ralph, SCHWIEGER, Wilhelm, SCHEFFLER, Michael, Preparation and properties of an electrically heatable aluminium foam/zeolite composite. *Microporous and Mesoporous Materials* 67 (January 2004), pages 53-59.

[55] TATLIER, Melkon, DEMIR, Mesut, TOKAY, Begüm, ERDEM-ŞENATALAR, Ayşe, KIWI-MINSKER, Lioubov, Substrate heating method for coating metal surfaces with high-silica zeolites: ZSM-5 coatings on stainless steel plates. *Microporous and Mesoporous Materials* 101 (2007), p. 374-380.
[56] WONGSUWAN, W., KUMAR, S., NEVEU, P., MEUNIER, F., *A* review of chemical heat pump technology and applications. Applied Thermal Engineering 21 (2001), p. 1489-1519.
[57] SRIVASTAVA, N.C., EAMES, I.W., A review of adsorbents and adsorbates in solid-vapour adsorption heat pump systems. *Applied Thermal Engineering* 18 (1998), p. 707-717
[58] ZHU, Dongsheng, WANG, Shengwei, Experimental investigation of contact resistance in adsorber of solar adsorption refrigeration. *Solar Energy* 73 (2002), p. 177-185.
[59] ARISTOV, Yu. I., RESTUCCIA, G., CACCIOLA, G., PARMON, V.N., A family of new working materials for solid sorption air conditioning systems. *Applied Thermal Engineering* 22 (2002), p. 191-204.
[60] HENNINGER, Stefan, *Untersuchung von neuen hochporösen Sorptionsmaterialien für Wärmetransformationsanwendungen*. Albert-Ludwigs-Universität Freiburg, Fakultät für Mathematik und Physik, 2008, PhD thesis – http://www.freidok.uni-freiburg.de/volltexte/5264/
[61] HAUER, A., Thermal energy storage with zeolite for heating and cooling applications. *International Sorption heat pump conference* (2002), p. 385-390 – conference proceedings.
[62] BAERLOCHER, Ch., MEIER, W.M., OLSON, D.H., Atlas of Zeolite Framework Types. Elsevier, 5. edition, 2001. – http://www.iza-structures.org/databases
[63] NÚÑEZ, T., HENNING, H-M., MITTELBACH, W., Adsorption cycle modelling: Characterization and comparison of materials. *ISHPC '99, Proc. of the Int. Sorption Heat Pump Conference* (1999) – conference proceedings.
[64] NÚÑEZ, T., HENNING, H-M., MITTELBACH, W., High energy density heat storage system – achievements and future work. *9th International Conference on Thermal Energy Storage*, Futurestock 2003, Warsaw, Poland – conference proceedings.
[65] JÄNCHEN, Jochen, ACKERMANN, Dietrich, STACH, Helmut, Adsorption properties of aluminophosphate molecular sieves – potential applications for low temperature heat utilisation. *ISHPC '02 Proc. of Int. Sorption heat pump Conference* (2002), p. 385-390.
[66] RAJIC, N., Open-framework aluminophosphates: synthesis, characterization and transition metal modifications. *J. Serbian Chem. Soc.* 70 (2005), p. 371-391.
[67] NG, K.C., CHUA, H.T., CHUNG, C.Y., LOKE, C.H., KASHIWAGI, T., AKISAWA, A., SAHA, B.B., Experimental investigation of the silica gel-water adsorption isotherm characteristics. *Applied Thermal Engineering* 21 (2001), p.1631-1624
[68] CARROTT, P.J.M., RIBEIRO CARROTT, M.M.L., CANSADO, I.P.P., Reference data for the adsorption of methanol on carbon materials. In: *Carbon* 39 (2001), p.193-200.
[69] WANG, L.W., WU, J.Y., WANG, R.Z., XU, Y.X., WANG, S.G., LI, X.R., Study of the performance of activated carbon-methanol adsorption systems concerning heat and mass transfer. *Applied Thermal Engineering* 23 (2003), p. 1605-1617
[70] WANG, L.W., WANG, R.Z., WU, J.Y., WANG, K., WANG, S.G., Adsorption ice makers for fishing boats driven by the exhaust heat from diesel engine: choice of adsorption pair. *Energy Conversion and Management* 45 (2004), p. 2043-2057.
[71] DIENG, A.O., WANG, R.Z., Literature review on solar adsorption technologies for ice-making and air-conditioning purposes and recent developments in solar technology. *Renewable and Sustainable Energy Reviews* 5 (2001), p. 313-342
[72] SAN, Jung-Yang, LIN, Wei-Min, Comparison among three adsorption pairs for using as the working substances in a multi-bed adsorption heat pump. *Applied Thermal Engineering* 28 (2008), p. 988-997.
[73] STOECKLI, F., Water adsorption in activated carbons of various degrees of oxidation described by the Dubinin equation. *Carbon* 40 (2002), p. 955-971.
[74] Industrial property rights EP 1391665-A3 (2003-08-19): ZAE Bayern. Pr.: DE 10237850-A1 2004-02-04. - Multistage absorption refrigerating apparatus or heat pump and the use of the apparatus in an energy conversion system.

[75] Industrial property rights WO 2008/ 034561 A2 (2008-03-27): Fraunhofer Gesellschaft. Pr.: DE 102006043715 A1 2008-03-27 - Adsorption Heat Pump With Heat Accumulator.
[76] Industrial property rights WO 2006/097198 A1 (2006-09-21): SorTech AG. Pr.: - Method For Producing An Adsorption Heat Exchanger.
[77] Industrial property rights WO 2006/048211 (2006-05-11): A2 SorTech AG. – Method For Production Of A Substrate Coated With A Zeolite Layer.
[78] WANG, D.C., WU, J.Y., WANG, R.R., XIA, Z.Z., Research on a novel adsorption chiller driven by low grade heat source. *International Sorption Heat Pump Conference* (2005), Denver, CO, USA – conference proceedings.
[79] ZHAI, X.Q., WANG, R.Z., WU, J.Y., DAI, Y.J., MA, Q., Design and performance of a solar-powered air-conditioning system in a green building. *Applied Energy 85* (2008), p. 297-311.
[80] WANG, R.Z., Efficient adsorption refrigerators integrated with heat pipes. *Applied Thermal Engineering* 28 (2008), p. 317-326.
[81] SCHMIDT, Ferdinand P., FÜLDNER, Gerrit, SCHNABEL, Lena, HENNING, Hans-Martin, Novel cycle concept for adsorption chiller with advanced heat recovery utilising a stratified storage. *Solar Air-Conditioning*, Tarragona, Spain, 2007 – conference proceedings.
[82] PONS, M., POYELLE, F., Adsorptive machines with advanced cycles for heat pumping or cooling applications. *International Journal of Refrigeration* 22 (1999), p. 27-37.

4 Liquid Desiccant Systems

Mustafa Jaradat (Kassel University), Michael Krause (Fraunhofer IBP),
Roland Heinzen (Kassel University), Lucio Mesquita (EnerWorks Inc.)

4.1 Principles of Operation

In liquid desiccant systems (LDS), air is dehumidified by being brought into contact with a strong solution of a liquid desiccant. Because of the dehumidification process, the solution is weakened by the absorption of moisture. In order to reuse the solution for further operation, it is directed to a regenerator where heat drives out the moisture. After that, the strengthened solution is returned to the dehumidifier. The main components of a liquid desiccant system are the absorber (dehumidifier) and the desorber (regenerator) shown in Figure 15.

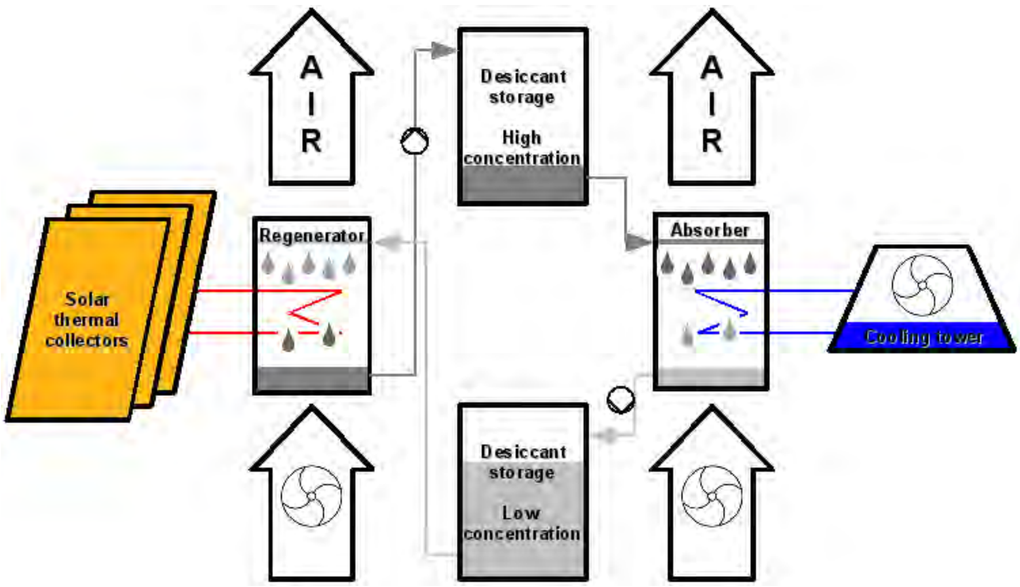


Figure 15: Schematic diagram of a solar driven liquid desiccant air-conditioning system.

In order to use liquid desiccant systems for comfort air-conditioning, the system is extended by evaporative cooler or (cooling towers), which are either operated in the supply air stream to a building or using an additional heat exchanger in the return air. Usually, solution storage tanks that offer energy storage without energy losses are integrated in the system, making the system a very promising approach for the combination with solar thermal collectors.

4.2 Liquid desiccant systems for HVAC and cooling applications

Liquid desiccant systems represent a particular configuration of a more generic class of cooling and air-conditioning systems called open-absorption systems. In such systems, the chemical solution used to absorb water is in direct contact with atmospheric air. This contact can occur at the regenerator/desorber, at the absorber/dehumidifier, or at both. The direct contact with atmospheric air is in contrast with closed absorption systems, which are referred to in the literature simply as absorption systems. Figure 16 shows a liquid desiccant system with desiccant storage operating as a ventilation cycle.

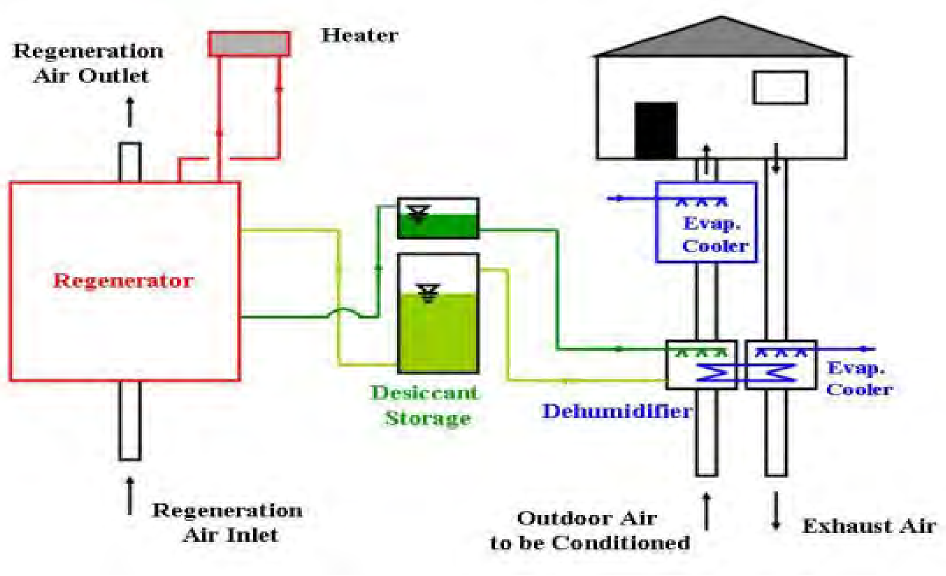


Figure 16: Liquid desiccant air-conditioning system, ventilation cycle. Source: Mesquita [83].

Herold et al. [84] presented the principles of operation and thermodynamic analysis of such systems. A schematic of a single-effect LiBr-H_2O cycle is shown in Figure 17.

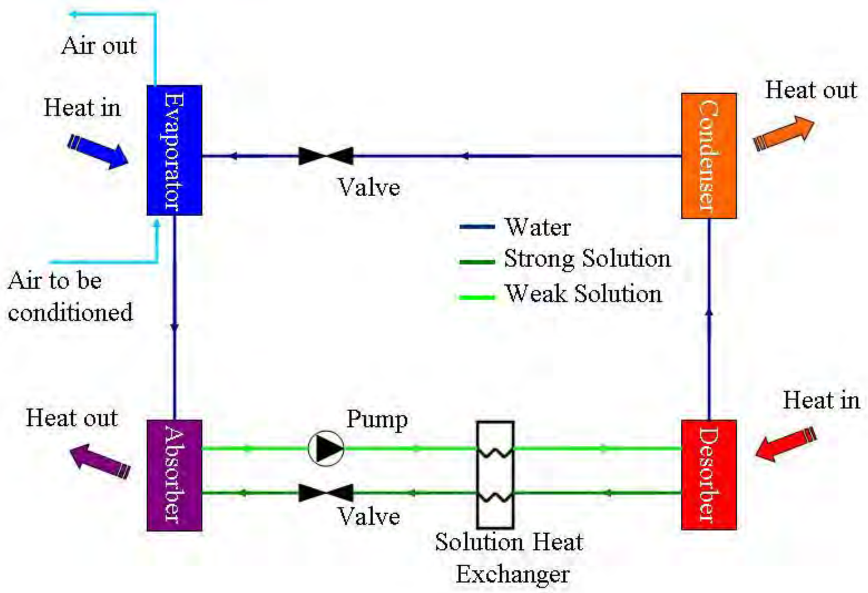


Figure 17: Schematic of a single-effect LiBr-H_2O closed absorption cycle. Source: Mesquita [83].

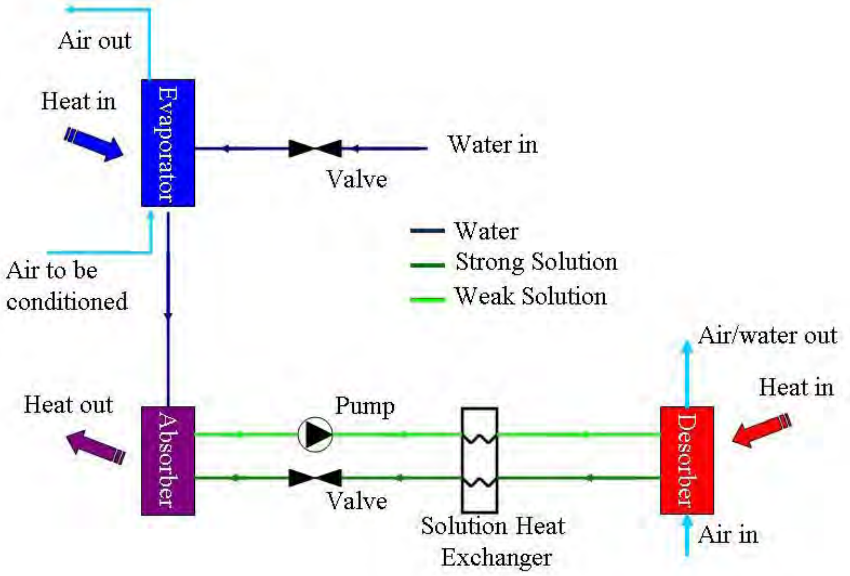


Figure 18: Schematic of an open-absorption system. Source: Mesquita [83].

Collier [85] analyzed what is commonly referred to as the open-absorption refrigeration system. In such a system, the closed desorber from a conventional absorption system is substituted for an open one, in this case an open solar collector. The author used a solution of LiCl-H_2O, with water being the refrigerant. Since the system is open, water is spent in the desorber/collector and has to be reintroduced into the system in the evaporator. Figure 18 presents a schematic of such a system. Lowenstein et al. [86] suggested an LDS with a closed boiler as the desorber/regenerator. Such a system, represented in Figure 19, could also be classified as an open-absorption system. One could argue that such a system represents more closely the definition of open-absorption, since the absorption is open and the desorption is closed.

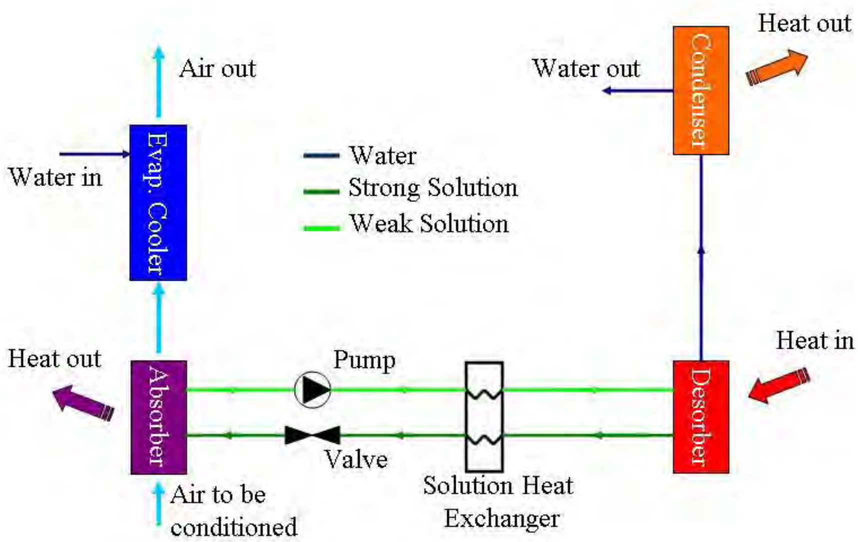


Figure 19: Schematic of a liquid-desiccant system with a closed desorber.

Source: Mesquita [83]

Perhaps a more appropriate nomenclature would be to call such systems semi-open absorption systems. Therefore, there would be a clear distinction between closed, semi-open and open-absorption, where both absorption and desorption are in direct contact with air. The fact that liquid-desiccant systems are usually used for dehumidification is not enough to clarify the nomenclature in the literature. For example, Grossman [87] proposed a system with an open regenerator and absorber, where the dehumidified air was directed to a cooling tower. This way, the system generated chilled water, as in a traditional vapour compression or closed absorption system. Bolzan and Lazzarin [88], Lazzarin et al. [89], and Johansson and Westerlund [90] presented systems that were liquid-desiccant cycles with open absorbers and regenerators, where the air running through the absorber was the air to be conditioned. Although this is an arrangement that has been mostly referred to as a desiccant system, the authors labelled the system as open-absorption.

One advantage of LDS over solid desiccant and closed absorption systems is the flexibility in terms of heat sources for regeneration and heat sinks for the dehumidification process. Heat can be supplied by any medium temperature source, usually above 60°C. Cooling can be accomplished by the use of heat sinks providing moderately low temperatures, usually between 5 and 25°C.

4.3 Applications

Liquid desiccant systems are growing in popularity because of their ability to independently control humidity levels (latent loads) moisture without cooling the air to saturation, the supply air relative humidity falls below 70%. This keeps supply ducts dry and helps avoid mold and bacterial growth. In addition, the scavenging action of liquid desiccant systems could improve indoor air quality by removing airborne contaminants. Heinzen et al. [91] presented a comprehensive overview for the liquid desiccant applications.

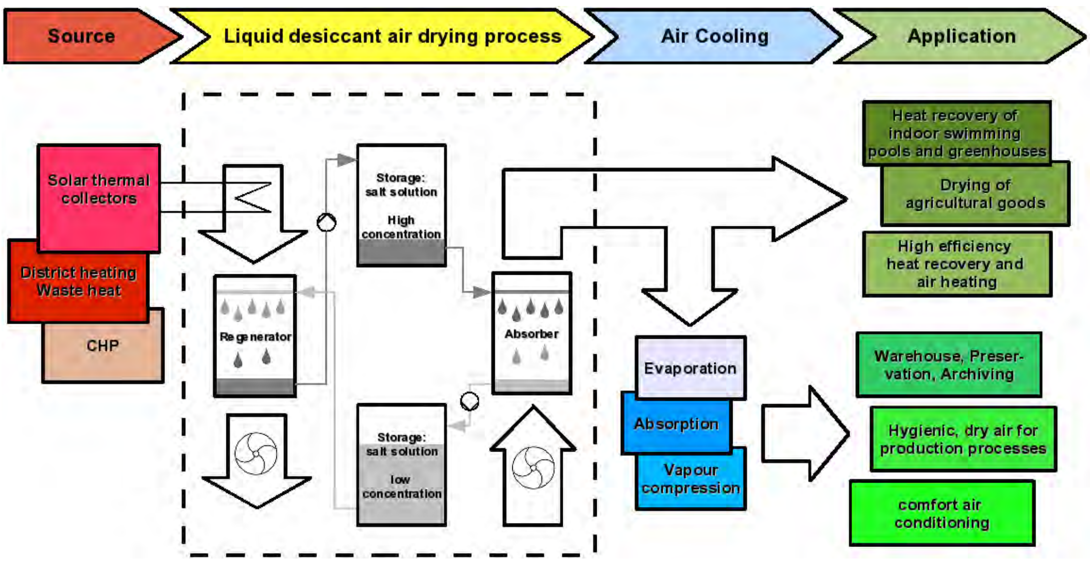


Figure 20: Liquid desiccant air drying process, from source to application. Source: Heinzen et al. [91].

Figure 20 shows a schematic of the liquid desiccant air drying process in the dashed area. It presents possible driving heat sources (left), the coupling to different cooling technologies (center right) and possible applications where LDS can be technically viable and economically and ecologically useful (right).

Lowenstein [86] presented the applications of LDS coupled with cooling processes that can be used for:

● comfort air-conditioning in offices, public and residential buildings

● warehouses and production halls for preservation and archiving purposes

● condensation protection to prevent mould and rust destruction from equipment

● production processes e.g. in the food production, pharmaceutical production, semi-conductor production, rubber industry, confectioneries.

The air stream from the absorber can be used directly for:

● high efficiency heat recovery and indirect air heating in low energy buildings, Kerskes [92]

● low temperature drying of agricultural goods and industrial products ("gentle drying"), Rane [93]

● high efficiency heat recovery and humidity control for indoor swimming pools and greenhouses, Waldenmaier [94].

4.4 Technology status

In the last years, significant progress has been made in the basic components of desiccant technology. These include:

4.4.1 Desiccant materials

The behaviour of all desiccant system components is profoundly influenced by the operating characteristics of the desiccant materials they contain. Recognizing this fact, research institutions and manufacturers have focused on material science to develop desiccants which are especially suited to air-conditioning applications.

These efforts have had two primary goals; to develop desiccants which:

1. Use less energy for reactivation and therefore need less energy for cooling

2. Are more stable and fault-tolerant and therefore require less maintenance

Al-Farayedhi et al. [95] listed several important considerations in choosing or designing the optimal liquid desiccant solution for a dehumidification application:

• High vapor pressure of water in solution

• Low vapor pressure of solute

• Performance of solution steady over large concentration range

• Non-corrosive and chemically stable

• Low viscosity

• High solubility

• Low regeneration temperature

• Non-toxic, harmless

• Low cost

Lithium chloride (LiCl), Calcium Chloride (CaCl2), Lithium Bromide (LiBr), and tri-ethylene glycol (TEG) are common liquid desiccant materials meeting the above performance characteristics to varying degrees. LiCl and CaCl2 dominate the most recent research efforts into liquid desiccant dehumidification systems. Wimby and Berntsson [96] investigated aqueous solutions of various desiccants, including LiCl and CaCl2, producing experimental data of density as a function of temperature and mass fraction. This data is of critical importance when experimenting with liquid desiccant materials, providing concentration as a function of density, which is relatively easy to measure in the laboratory. A mixture of LiCl and CaCl2 was the subject of a study by Ertas, Anderson, and Kiris [97]. LiCl has excellent regeneration performance and stability but high cost, while CaCl2 has lower performance but low cost. A mixture of the two at various ratios was analyzed to produce functions of vapor pressure for various temperatures. In an important study of aqueous LiCl and CaCl2, Conde [98] gathered data from 1850 and onwards and fitted empirical curves to selected data.

Functions of density, heat capacity, enthalpy of dilution, vapor pressure, solubility, and others, were presented. These correlations were used extensively in the present study. The concept of storage was explicitly investigated by Peng, Zhang, and Yin [99], with the conclusion that liquid desiccant regeneration equipment performance is improved with desiccant storage.

4.4.2 Desiccant-air contactors

In desiccant systems, the component which presents the desiccant to the air stream (the desiccant contactor) is one of the most critical elements of the system, and the one which most influences the net energy consumption of the system.

Ideally, the desiccant contactor would have an infinitely large surface area for desiccant-air interaction, but infinitely low mass, so no excess material must be heated and cooled along with the desiccant. Further, the contact media must be very durable, as it is repeatedly wetted and dried as the desiccant moves through the sorption-desorption cycle.

There are many different technologies available for the absorber and regenerator. The regeneration can be built as a packed bed, heated coil, spray chamber, falling-film parallel plates, boiler or open solar collector. As for the absorbers, they are typically built as packed beds, cooled coils, spray chambers and falling-film parallel plates.

4.4.2.1 Packing towers

A packed tower arrangement sprays liquid desiccant into the process air stream in counter flow, through a random or structured packing material. Figure 21 presents examples of random packings used in packed beds and columns. The bed or column is filled with such random packings, forming the contacting surface.

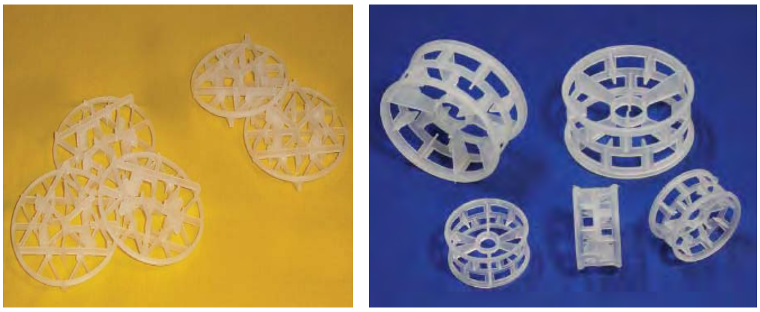


Figure 21: Commercially available random packings for packed beds. Source: Koch-Glitsch [100].

Packed bed design was first proposed by Lof [101] in 1955 using a triethylene glycol solution as the desiccant. This design incorporates a porous packing material, sprinklers, and a solution pump. The concentrated liquid desiccant is sprayed over the packing material, which forms a thin film along the bed. Similar to a cooling tower design, the inlet air is forced through the packed bed structure in a counter-flow direction, where the weight of the desiccant enables collection at the bottom of the bed. The packed bed structure has been extensively studied by Factor and Grossman [102] with exit condition predictions using a one-dimensional model. Their analysis included a set of differential equations derived from energy and mass balances along a differential slice of the packed column. A finite difference scheme was used to solve these sets of differential equations, revealing the temperature and humidity profiles along the bed.

Industrial dehumidification systems usually employ packed bed regenerators and absorbers, with high flow rates of desiccant. In a review of liquid desiccant systems, Öberg and Goswami [103] showed that typical systems have an MR of between 0.5 and 2. On the other hand, low desiccant flow and internally cooled absorbers can use an MR above 100, as shown by Keßling et al. [104].

As pointed out by Dudukovic et al. [105], packed beds, packed towers or packed columns are widely used in the pharmaceutical, chemical and petroleum industries. Therefore, extensive research has been conducted on the design and operation of such technology. Strigle [106] and Billet [107] present extensive information on packed towers. This type of equipment is relatively cheap and easy to build, and provides elevated interfacial surface area per unit volume (specific surface area).

Regarding the specific application of packed beds for liquid-desiccant dehumidifiers and regenerators, Öberg and Goswami [103] presented a review of mass transfer performance correlations available in the literature, most of them using random packings. More recently, a number of studies, both theoretical and experimental, have been published on structured packing dehumidifiers and regenerators, e.g., Al-Farayedhi et al. [108], Gandhidasan et al.[109], Abdul-Wahab et al. [110], Dai and Zhang [111], Elsarrag et al.[112], Elsarrag [113] and Liu et al. [114]. Longo and Gasparella [115] experimentally compared the performance of structured and random packings. Their results showed dehumidification rates and an efficiency that was 20% to 30% higher for random packings, but a 65 to 75% lower air pressure drop with structured packings. Besides the high pressure drop, packed beds have a number of disadvantages associated with high desiccant flow rates.

Lowestein et al. [86] listed the most important disadvantages associated with packed beds:

• air passages within the packed bed are fairly wide to prevent the desiccant from restricting the airflow (flooding). This increases the size and cost of equipment;

• desiccant pumps are large and therefore have high energy consumption;

• the change in concentration of the desiccant is slight, making the desiccant storage not viable and less effective;

• desiccant flow is high leading to more heat being dumped back into the dehumidifier cooler by the regenerator; and

• air velocities are low to avoid high pressure drops and entrainment of droplets in the air stream.

4.4.2.2 Plate Type / Falling Film

The problem of droplet entrainment and desiccant carry-over into building ducts can be a significant issue associated with packed beds. Mist eliminators require regular maintenance, which is usually not available in many buildings.

In order to avoid some of the shortcomings of packed beds, parallel-plate, internally-cooled or heated absorbers and regenerators have been proposed. In such systems, such as the one shown in Figure 22, the desiccant solution trickles down the surface of a plate, where it contacts the air to be dehumidified. Water flows inside a channel to remove the heat generated by the absorption process.

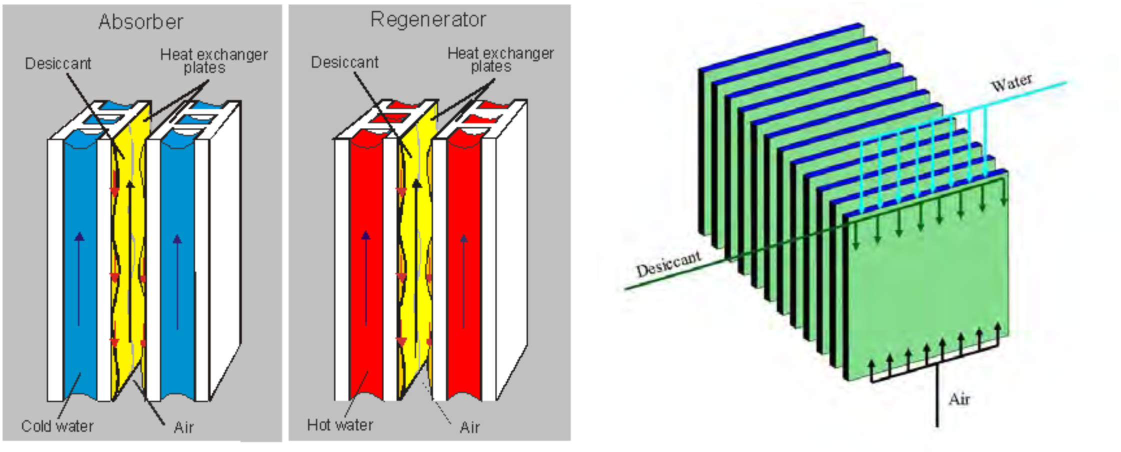


Figure 22: Parallel-plate, internally-cooled/heated absorber/regenerator, Krause et al. [116]

Different techniques, [117] Jaradat et al. and Hubliz [118], have been presented in order to introduce the liquid desiccant into the plates, the uneven horizontal distribution of the liquid desiccant as it enters the generator is undesirable because it reduces the effective area of contact between the liquid desiccant and air and thus decreases the mass transfer and heat exchange between the liquid and vapour. To ensure proper operation of the generator, it is also important to ensure that the ratio of liquid to vapour is constant over the cross-section of the plates. For this reason, it is important to have an even distribution of liquid as it enters the heat and mass exchanger. Figures 23 and 24 show different distribution techniques of the liquid desiccant over the heat and mass transfer plates.

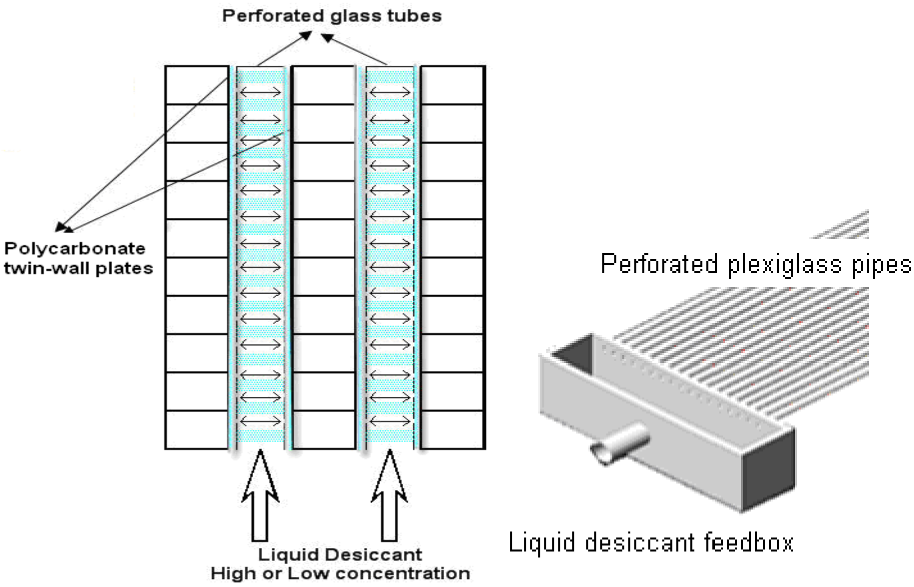


Figure 23: Schematic diagrams of a liquid desiccant distributor; perforated pipes concept: Jaradat et al. [117]

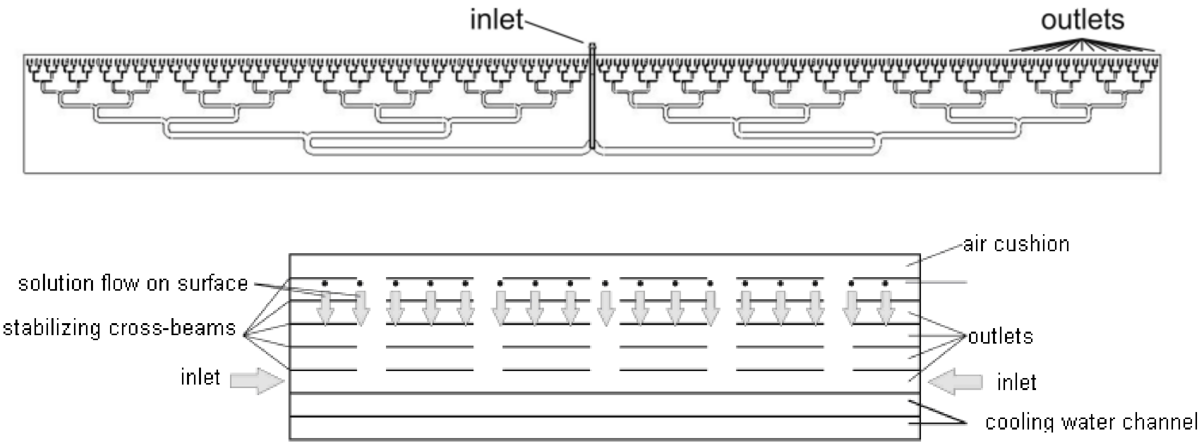


Figure 24: Schematic diagrams of different liquid desiccant distributors; channel concept (upper) and outlet concept integrated into the heat and mass transfer plate (lower). Hublitz [118].

Another option is to provide cooling through an internal evaporative cooler. In such systems, as proposed by Saman and Alizadeh [119] and shown in Figure 25, the desiccant and evaporating water flow on opposite sides of the same wall. The air to be conditioned flows on the desiccant side. The scavenging air, which can also be the exhaust air from the building, flows on the water side. These absorbers are potentially more economical and compact than the water channel cooled varieties. The additional stream of air and the coupled evaporative cooler, however, make its construction significantly more complex. The additional complexity comes from building an absorber with two streams of air, one stream of desiccant and one stream of water, and ensuring no leakages/contamination between the two air streams.

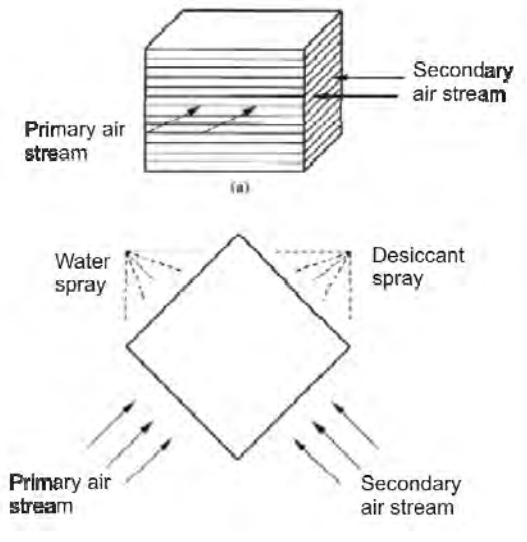


Figure 25: Schematic of an evaporatively cooled absorber. Source: Saman and Alizadeh [119].

The main disadvantages and challenges of the plate-type liquid desiccant heat and mass exchanger are:

1. Difficult to construct, especially when internally cooled designs are used in a “three-way heat exchanger”. Here, a mixing of the different fluids has to be prevented

2. Difficult flow operation to ensure an even flow distribution over plates
3. Wetting difficulties when using low flow rates of desiccant solution
4. High material requirements, e.g. when used plastic material for plates

4.5 New developments of desiccant cooling systems

- L-DCS (Germany, Singapore): Advanced open cooling system using liquid desiccant in a plate heat and mass exchanger approach. Due to low flow rates, concentrated solution offers storage with high energy density. Three systems are currently in operation.
- Other liquid sorption developments: Kassel University (Germany), Stuttgart (Germany), Queens University (Canada), University of South Australia (Adelaide), Technion Haifa (Israel), Florida (U.S.), Genius (U.S.). Different prototypes, Figures 26 and 27, have been built with different materials and ideas in order to overcome the obstacles that hinder the progress and wide acceptance of such systems in the HVAC industry.

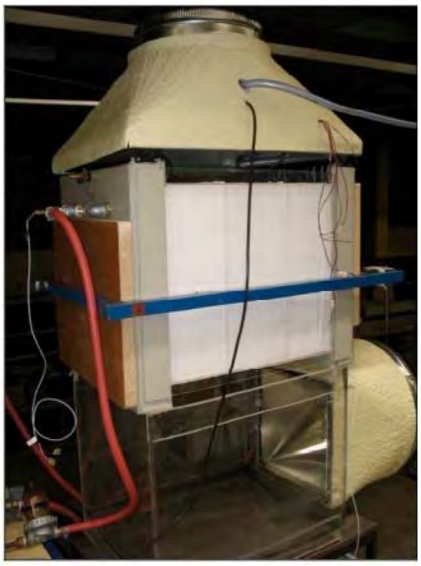


Figure 26: Non-adiabatic liquid desiccant absorber/regenerator made-out of polypropylene, University of South Australia, Krause [116]

Figure 27: Non-adiabatic liquid desiccant absorber/regenerator (left) and adiabatic liquid desiccant absorber (right) made-out of polycarbonate in the pilot plant stage in the laboratories of Kassel University. Jaradat et al [120].

4.6 Commercial products suppliers

- Kathabar (USA): Packed bed (Raschig rings)
 Kathabar Systems has manufactured desiccant dehumidification equipment for over 70 years. This company provides a wide range of dehumidifiers and regenerators. A total of 14 different size conditioners is available, each capable of handling either horizontal or vertical air flow arrangements. The smallest in the series is designed to handle airflows from 1274 to 2548 m3/h, the largest from 71,358 to 142,716 m3/h. Kathabar offers a total of eight different size regenerators which are matched with the conditioners required for a given installation, Figure 28.

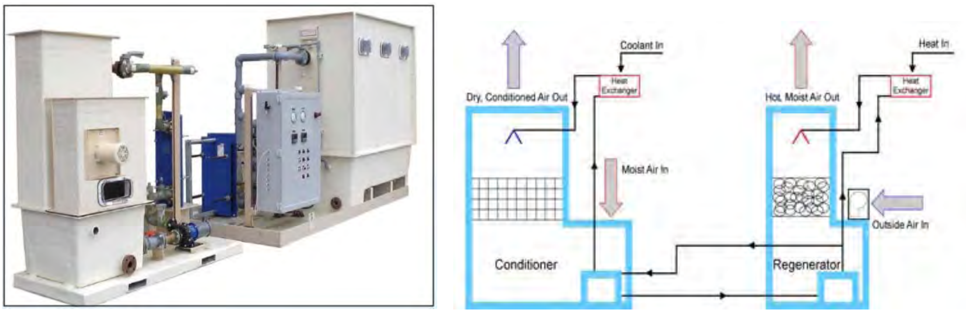


Figure 28: A commercially available packed bed liquid desiccant dehumidifier (Kathabar, 2007) [121].

- AIL Research: provide commercially liquid desiccant dehumidifiers and regenerators that are based on parallel-plate liquid-to-air heat exchanger. The dehumidifier is capable of handling airflows from 3398 to 10194 m3/h, Figure 29.

Figure 29: Rooftop Liquid-Desiccant Air Conditioner (left), schematic of an AIL Research Liquid Desiccant dehumidifier. Source: Lowenstein et al. [122].

- Liquid-Desiccant Air Conditioner Menerga (Germany): New air handling unit using liquid sorption, dehumidifier in combination with a standard indirect evaporator cooler. Four pilot plants are currently in operation, one is operated using solar thermal energy (Freiburg, Germany), Figure 30.

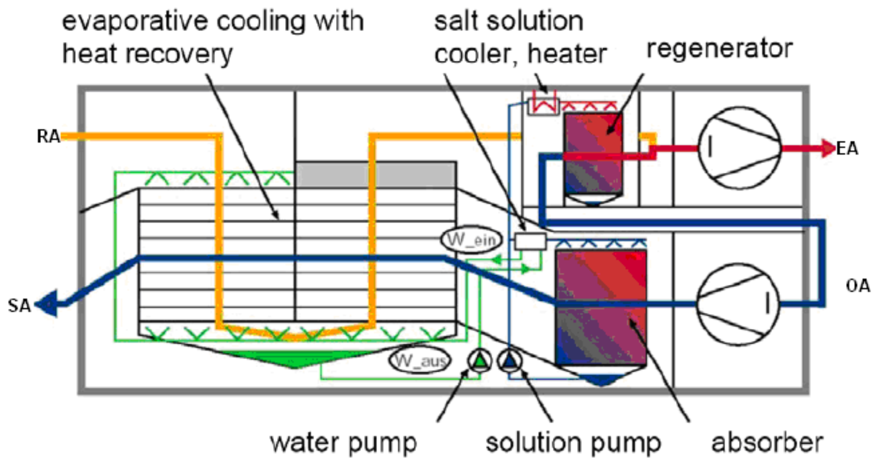


Figure 30: Menerga air handling unit using liquid sorption technology. Biel et al. [123].

4.7 References

[83] Lucio Cesar De Souza Mesquita. Analysis of a Flat-Plate, Liquid-Desiccant, Dehumidifier and Regenerator. PhD thesis, Queen's University, 2007.
[84] Herold, K.E., Radermacher, R., and Klein, S.A. (1996) Absorption Chillers and Heat Pumps, CRC Press.
[85] Collier, R.K. (1979) The Analysis and Simulation of an Open Cycle Absorption Refrigeration System, Solar Energy Journal, vol. 23 (4), pp. 357-366.
[86] Lowenstein, Slayzak, Ryan, Pesaran (1998), Advanced Commercial Liquid-Desiccant Technology Development Study, Report Proj.-No. NREL/TP-550-24688, National Technical Information Service (NTIS) U.S. Department of Commerce, 5285 Port Royal Road Springfield, VA 22161, 703-605-6000 or 800-553-6847.
[87] Grossman, G. (2002), Solar-Powered Systems for Cooling, Dehumidification and Air-Conditioning, Solar Energy, vol. 72 (1), pp. 53-62.
[88] Bolzan, M. and Lazzarin, R. (1979), Comparison between Two Absorption Cooling Systems of The 'Open' Type Under Different Climatic Conditions, International Journal of Refrigeration, vol. 2 (3), pp. 143-149.
[89] Lazzarin, R.M., Longo, G.A. and Gasparella, A. (1996), Theoretical Analysis of an Open-Cycle Absorption Heating and Cooling System, Int. J. Refrig., vol.19 (3), pp. 160-167.
[90] Johansson, L. and Westerlund, L. (2000), An Open Absorption System Installed at a Sawmill - Description of Pilot Plant Used for Timber and Bio-Fuel Drying, Energy, vol.25, pp. 1067–1079.
[91] Heinzen R., Krause M., Jordan U., Vajen K., Technical Potential of Solar Thermal Driven Open Cycle Absorption Processes for Industrial and Comfort Air Conditioning, Proc. ISES Solar World Congress, Beijing, 18.09. - 21.09.2007, pp. 809 – 813
[92] Kerskes H., Heidemann W., Müller-Steinhagen H. (2004), MonoSorp- Ein weiterer Schritt auf dem Weg zur vollständig solarthermischen Gebädeheizung, 14. Symposium Thermische Solarenergie OTTI Energie-Kolleg.
[93] Rane M. V., Reddy S. V. K., Easow R. R. (2005), Energy efficient liquid desiccant-based dryer, Applied Thermal Engineering, Vol. 25, pp. 769 – 781.
[94] Waldenmaier M., A Sorption Heat Storage System for Dehumidification of Indoor Swimming Pools, IEA Annex 10, Phase Change Materials and Chemical Reactions for Thermal Energy Storage, First Workshop, 16 - 17 April 1998, Adana, Turkey.
[95] Al-Farayedhi A.A., Gandhidasan P. M., Antar A., and Abdul Gaffar M.S., Experimental study of an aqueous desiccant mixture system: air dehumidification and desiccant regeneration. Proceedings of the Institution of Mechanical Engineers, Part A (Journal of Power and Energy), 219(A8):669 – 80, 2005.

[96] J. Martin Wimby and Thore S. Berntsson. Viscosity and density of aqueous solutions of LiBr, LiCl, ZnBr2, CaCl2, and LiNO3. 1. single-salt solutions. Journal of Chemical and Engineering Data, 39:68 – 72, 1994.

[97] Ertas, E.E. Anderson, and I. Kiris. Properties of a new liquid desiccant solution-lithium chloride and calcium chloride mixture. Solar Energy, 49(3):205 – 12, 1992.

[98] Manuel R. Conde. Properties of aqueous solutions of lithium and calcium chlorides: Formulations for use in air conditioning equipment design. International Journal of Thermal Sciences, 43(4):367 – 382, 2004.

[99] Donggen Peng, Xiaosong Zhang, and Yonggao Yin. Theoretical storage capacity for solar air pre-treatment liquid collector/regenerator. Applied Thermal Engineering, 28(11-12):1259 – 66, 2008.

[100] Koch-Glitsch (2003), Intalox Packed Tower System, Plastic Random Packing, Wichita, Kansas.

[101] Löf, House heating and cooling with solar energy. Solar Energy Research, pages 33–46, 1955.

[102] Factor, H. M., Grossman, G., A packed bed dehumidifier/regenerator for solar air conditioning with liquid desiccants. Sol. Energy 1980, 24, 541.

[103] Öberg, V. and Goswami, D. (1998), A Review of Liquid-Desiccant Cooling, in Advances in Solar Energy, Böer, K., Ed. American Solar Energy Society, Boulder, Colorado.

[104] Keßling W., Lävemann, E. and Kapfhammer, C. (1998), Energy Storage for Desiccant Cooling Systems Component Development, Solar Energy, vol. 64 (4-6), pp. 209-221.

[105] Dudukovic, M.P., Larachi, F. and Mills, P.L. (1999), Multiphase Reactors Revisited, Chemical Engineering Science, vol. 54, pp. 1975-1995.

[106] Strigle, R.F. (1987), Random Packings and Packed Towers: Design and Applications, Gulf Pub. Corporation.

[107] Billet, R. (1995), Packed Towers: in Processing and Environmental Technology, VCH.

[108] Al-Farayedhi, A.A., Gandhidasan, P. and Al-Mutairi, M.A. (2002), Evaluation of Heat and Mass Transfer Coefficients in a Gauze-Type Structured Packing Air Dehumidifier Operating with Liquid Desiccant, International Journal of Refrigeration, vol. 25, pp. 330– 39.

[109] Gandhidasan, P., Al-Farayedhi, A.A. and Antar, M. A. (2002), Investigation of Heat and Mass transfer in a Gauze-Type Structured Packing Liquid Desiccant Dehumidifier, International Journal of Energy Research, vol. 26, pp. 1035-1044.

[110] Abdul-Wahab, S.A., Zurigat, Y.H. and Abu-Arabi, M.K. (2004), Predictions of Moisture Removal Rate and Dehumidification Effectiveness for Structured Liquid Desiccant Air Dehumidifier, Energy, vol. 29, pp. 19–34.

[111] Dai, Y.J. and Zhang, H.F. (2004), Numerical Simulation and Theoretical Analysis of Heat and Mass Transfer in a Cross Flow Liquid Desiccant Air Dehumidifier Packed with Honeycomb Paper, Energy Conversion and Management, vol. 45, pp.1343–1356.

[112] Elsarrag, E., Magzoub, E.E.M. and Jain, S. (2004), Mass-Transfer Correlations for Dehumidification of Air by Triethylene Glycol in a Structured Packed Column, Ind. Eng. Chem. Res., American Chemical Society, vol. 43, pp. 7676-7681, 2004.

[113] Elsarrag, E. (2006), Performance Study on a Structured Packed Liquid Desiccant Regenerator, Solar Energy, vol. 80, pp. 1624–1631.

[114] Liu, X.H., Zhang, Y, Qu, K.Y. and Jiang, Y. (2006), Experimental Study on Mass Transfer Performances of Cross Flow Dehumidifier Using Liquid Desiccant, Energy Conversion and Management, vol. 47, pp. 2682–2692.

[115] Longo, G.A. and Gasparella, A. (2005), Experimental Analysis on Chemical Dehumidification of Air in a Packed Column by Hygroscopic Salt Solution: Comparison between Structured and Random Packings, International Sorption Heat Pump Conference, Denver, Colorado.

[116] Krause, M., Heinzen, R., Saman, W., Vajen, K., Jordan, U. (2007), Validation and Comparison of Absorber and Regenerator Models for Liquid Desiccant Air Conditioning Systems. 2nd. Intern. Conference Solar Air Conditioning, Taragona (Spain), 18.-19.10.2007

[117] Jaradat, M., Heinzen, R., Jordan, U., Vajen, K., A Novel Generator Design for a Liquid Desiccant Air Conditioning System, EuroSun 2008, Lisbon (PO), 07.10 - 10.10. 2008.

[118] Astrid Hublitz. Efficient Energy Storage in Liquid Desiccant Cooling Systems. PhD thesis, Technische Universität München.

[119] Saman, W.Y. and Alizadeh, S. (2002), An Experimental Study of a Cross-Flow Type Plate Heat Exchanger for Dehumidification/Cooling, Solar Energy,vol.73(1), pp.59-71.

[120] Jaradat, M., Heinzen, R., Jordan, U., Vajen, K., Initial Experiments of a Novel Liquid Desiccant Dehumidifier for Industrial and Comfort Air Conditioning Systems, proceeding 3rd International Conference Solar Air-Conditioning 2009, Palermo (IT), 30.09 - 02.10 2009.

[121] Kathabar Systems (2007), website accessed: 07 September 2009 (www.kathabar.com/products/dehumidifiers/liquiddesiccant.html).

[122] Lowenstein, A. I., A. S. Slayzak, E. Kozubal, A zero carryover liquid-desiccant air conditioner for solar applications, ISEC 2006.

[123] Biel, S.,et al. (1997), Sorption Entfeuchtung unde Temperaturabsenkung bei der klimatisierung, Final Report on the BMB+F Project 032 9151J.

5 Solid desiccant cooling systems

Paul Bourdoukan, Etienne Wurtz, (LOCIE-INES/CNRS)

5.1 Desiccant cooling principle

Using solar energy appears to be an interesting option for cooling applications, as the building load almost matches the availability of the source. In a solar desiccant cooling cycle, solar energy is used to regenerate a desiccant that dehumidifies moist air; the resulting dry air is cooled in a sensible heat regenerator and then in an evaporative cooler. By associating different elementary changes in moist air (dehumidification, sensible cooling and evaporative cooling) the technique uses water as a refrigerant and solar energy as a driving heat; while electricity is only used in the auxiliaries, so the technique is environmentally friendly. It is a thermally driven open cooling cycle based on evaporative cooling and adsorption.

With reference to figure 31, a desiccant cycle operates as follows: outside air (1) is dehumidified in a desiccant wheel (2); it is then cooled in the sensible heat regenerator (3) by the return cooled air before being further cooled in an evaporative process (4), finally, it is introduced into the building. The operating sequence for the return air (5) is as follows: it is cooled to its saturation temperature by evaporative cooling (6) and then it cools the fresh air in the rotary heat exchanger (7). It is then heated in the regeneration heat exchanger by solar energy (8) and finally regenerates the desiccant wheel (9) by removing the humidity before exiting the installation.

The desiccant wheel is used to remove moisture from the outside air. Its function is first, to reduce the humidity of the outside air in order to match indoor air standards and second, to provide extra dehumidification to increase the cooling potential of the supply humidifier.

Depending on load conditions this system can operate under different modes. Simple ventilation (fans only), indirect evaporative cooling for low cooling loads (fans, return evaporator cooler and the sensible heat regenerator) and desiccant mode (all components operational).

For moderate summer conditions, in the morning when the outside temperature is low, the indirect evaporative cooling mode is able to keep the room in a comfortable range; there is thus no need for regeneration and the solar heated water can be stored in the tank. During the day with the outside temperature rising and increasing solar gains the indirect evaporative cooling cannot provide the cooling and so the desiccant mode is then required, while solar energy is needed for the regeneration of the desiccant wheel. The minimum temperature required for regeneration depends on the nature of the desiccant material. It varies from 50°C for lithium chloride to 60°C for silica gel. We chose silica gel for its higher dehumidification performances in spite of the higher temperature needed for its regeneration.

The desiccant cooling process is well-suited to the requirements of non-residential buildings with high occupancy needing high air exchange rates, e.g. seminar rooms and banks. In these buildings the rooms are usually occupied during the day, so air-conditioning loads match solar energy availability; so coupling desiccant cooling with solar collectors would seem a very interesting option.

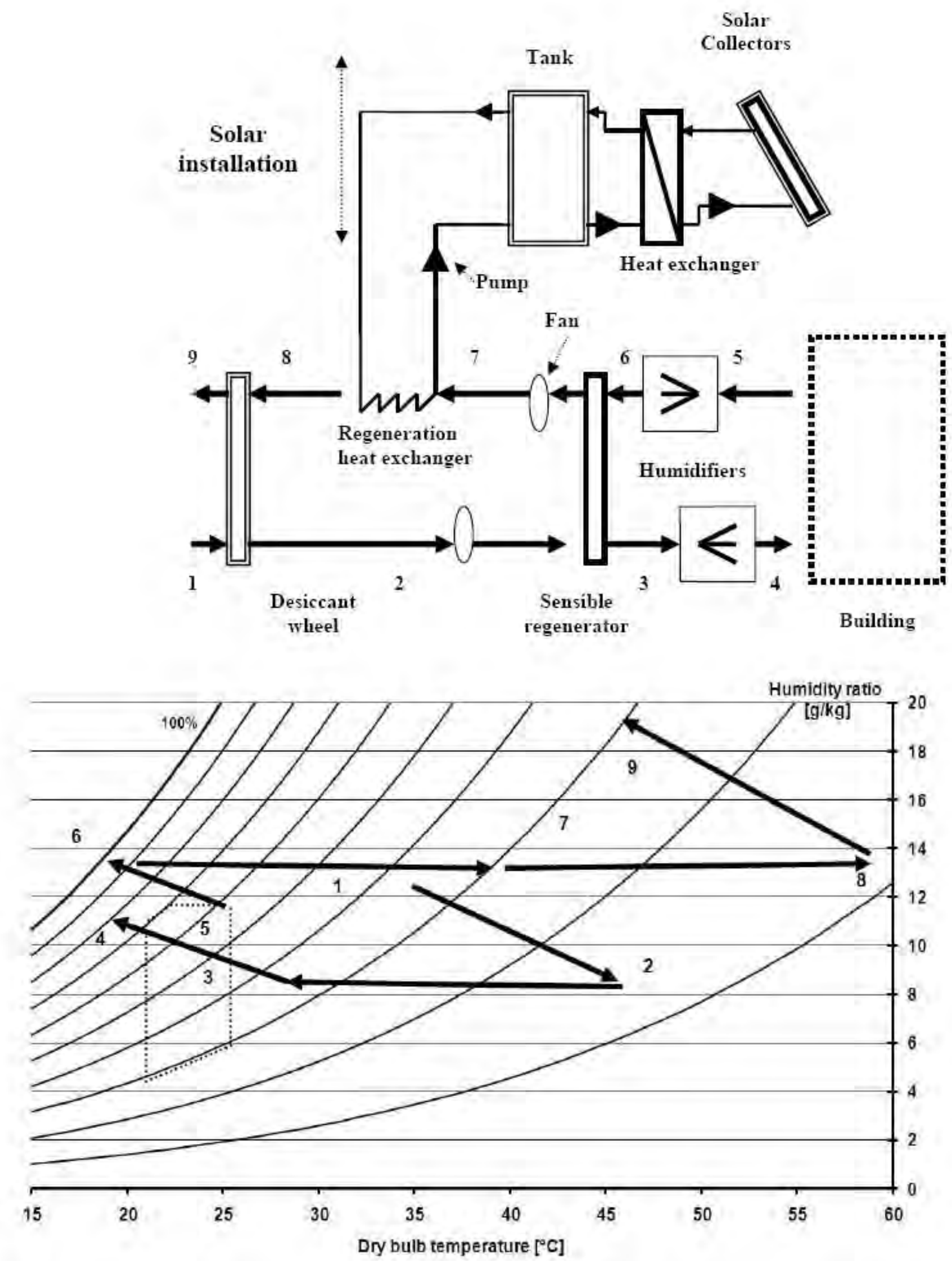


Figure 31: Desiccant cooling installation with corresponding evolution of moist air properties in the psychrometric chart.

5.1.1 Advantages

The main advantages of the desiccant cooling system are the following:

- The minimum required driving temperature is 50°C which makes the coupling to solar collector a very interesting option.
- Different operating modes can be used depending on load conditions. This allows operations on partial loads while storing solar energy for peak loads.
- The use of electricity is limited to the auxiliaries.
- The process uses water as a refrigerant.

5.1.2 Disadvantages

The disadvantages of a desiccant cooling system are:

- The cooling potential depends intrinsically on operating conditions (outside temperature, outside humidity ratio and regeneration temperature).
- If the humidifiers are not carefully designed, they may cause hygienic problems.
- Rotating elements (desiccant wheel, sensible heat regenerators) need maintenance.

5.2 What is on the market?

In this section the market development concerning desiccant cooling technologies will be developed. A conventional desiccant air-handling unit consists of a desiccant wheel, a sensible heat regenerator, and evaporator coolers. For the sensible regenerator and the evaporator coolers, these components are conventionally used in the HVAC air-handling units. This market development section will consider only desiccant wheels as these are not conventionally used and the manufacturers are limited worldwide.

This list is of course not exhaustive, it is the result of the author's queries and research.

USA

Novel Air
Address: 10132 Mammoth Ave., Baton Rouge, Louisiana 70815-8013, USA
website: http://www.novelaire.com/technologies/desiccant.htm
Tel: +1-(225)-924-0427
Fax: +1-(225)-930-0340

Germany

Klingenburg GmbH
Address: Boystraße 115, 45968 Gladbeck, Germany
Website: http://www.klingenburg.de/en/index.html
Tel.: +49 (0) 20 43 / 96 36-0
Fax: +49 (0) 20 43 / 7 23 62

Sweden

Munters Europe AB
Address: Östra Nygatan 104,931 35 Skellefteå, Sweden
Website: http://www.munters.se
Tel: +46 (0)10-451 54 69
Fax: +46 (0)910-521 99

Proflute
Address: 18732 Täby, Sweden
Website: www.proflute.se
Tel: +46 8 511 87 800
Fax: +46 8 511 87 700

Japan

Seibu Giken Co. Ltd.
Address: 3108-3 Aoyagi, Koga-shi, Fukuoka, 811-3134 Japan
Website: http://www.seibu-giken.co.jp/eg/products/dry_save.html
Tel: +81-92-942-3511
Fax: +81-92-942-3761

Earth Clean Tohoku Co. Ltd.
Address: Sendai City, Japan 984-0038
Website: http://www.earthclean.co.jp/products/desiccant/catalog.html

Tel: +81-22-288-2888
Fax: +81-22-288-2890

Amefrec Co. Ltd.
Address: 8-15, Toyosaki 2-Chome, Kita-Ku, Osaka 531-0072, Japan
Website: http://www.amefrec.co.jp/afceng/index.htm
Tel: +81-6-6373-3201
Fax: +81-6-6373-3290

Panasonic Ecology Systems Co. Ltd.
Address: 4017, Shimonakata, Takaki-cho, Kasugai-city, Aichi 486-8522 Japan
Website: http://www.panasonic.net/corporate/segments/pes/
Tel: +81-6-6908-1121

Mitsubishi Plastics Inc.
Address: 1-2-2, Nohonbashihongokucho, Chou-Ku, Tokyo, 103-0021, Japan
Website: http://www.mpi.co.jp
Tel: +81-3-3279-3700

China

The major desiccant system producers are mainly located in southeast of China, where is nearing the application market as shown in Figure 1. There are also some companies located in Beijing, whose products are distributed all over the whole country.

The information about the main companies is summarized and introduced below:

Munters Air Treatment Equipment (Beijing) Co. Ltd.
It is a subsidiary of world famous dehumidification products producer Munters group. Products of Munters entered China in 1970s. It is also one of the earliest foreign companies that established its manufacturing factory in China (in 1995), which is their biggest manufacturing base in Asia. They have branch offices in Beijing, Shanghai and Guangzhou to provide products and services in the most demanding markets of China. Their products are usually in the higher price and high quality bracket (better supply air accuracy and operational reliability). Their industry customers include ABB China, Shanghai Volkswagen, Towngas, Trane etc. Their commercial customers include Walmart, Nestle etc.
Website: http://www.munters.cn/cn/cn/Contact/
Tel: +86-10-8041-8000
Fax: +86-10-8048-3493

Dryer Air-handling Equipment Co. Ltd.
It was founded in 2001. It is a joint venture with the Swedish manufacturer and supplier of desiccant (adsorption) dehumidifiers and desiccant cooling equipment, DehuTech AB. The desiccant rotor, which is patented, is manufactured in Sweden. Their head office and manufacturing base are in Shanghai with branch offices located in Beijing, Guangzhou and Chongqing. Similar to Munters, the products of Dryer are in the higher price and high quality bracket. This company has been awarded dehumidification projects for some famous bridges in China such as Lupu Bridge in Shanghai, Jiangshu Runyang Bridge, Yichang Changjiang Bridge etc.
Website: http://www.dryer-inc.com/cn/contact.asp
Tel: +86-10-8225-1978
Fax: +86-10-8225-1978

Sat Air Treatment Co. Ltd.
It was founded in 2000 and is a joint venture with the Swedish company PROFLUTE. It is located in Jiangying, Jiangshu Province. In the beginning, they made evaporative cooling

equipment for henhouses as well as workshops. This company started to manufacture desiccant dehumidification equipment in 2004 with rotors imported from PROFLUTE.
Website: http://www.sat.net.cn/contact.asp
Tel: +86-510-8660-5998
Fax: +86-510-8660-5978

Wuxi Shamo Dehumidifying Equipment Co. Ltd
It was founded in 1990 and is one of the oldest dehumidifying equipment manufacturers. Their rotor is self made with paper as substrate. Their products are usually in the lower price and medium quality range. Hence, they have substantial market share among medium or low end users who do not require high accuracy/operational reliability but cost effective products.
Website: http://www.shamo.cn/index_e.asp
Tel: +86-510-8310-9612
Fax: + 86-510-8375-0058

Wuxi Aobo Dehumidifying Equipment Co. Ltd.
They make their own brand equipment with rotors imported from the Swedish company PROFLUTE.
Website: http://www.aobocs.com/english/
Tel: +86-510-8320-3593
Fax: +86-510-8320-3470

Hangzhou Dry Air Co. Ltd.
It was established from a paper research institute in Zhejiang Province. They make their own brand equipment with rotors imported from Swedish company PROFLUTE and Japanese company NICHIAS. Their products have been used in some defence applications.
Website: http://www.hzdryair.com/doce/index2_en.asp
Tel: +86-571-8990-5556
Fax: +86-571-8990-5538

Hangzhou Fuda Co. Ltd.
It was founded in 1995. Their employees include the first generation specialists in rotary desiccant equipment in China. This is a typical local desiccant dehumidification equipment producer with market mostly in east China. Their products are mainly for small and medium size industry application.
Website: http://www.hzfdcs.com/#
Tel: +86-571-8689-9396
Fax: +86-571-8689-9390

Shanghai Hanfu Air Treatment Equipment Co. Ltd.
It is also a joint venture. Their rotors are from Swedish Proflute, Japanese Nichias, Swedish Enuentus. They have produced moveable products that can provide heating, cooling, dehumidification according to requirements.
Website: http://www.hanfuair.com/
Tel: +86-21-5759-9990
Fax: +86-21-5759-6329

Guangzhou Huagongtai Dehumidification Equipment Co. Ltd.
It is a subsidiary company of the National Science Park of South China University of Technology. Its market is mainly in South China Guangdong Province. Their rotors are self made with patent technology.
Website: http://www.gdscw.com/company/com/index.php?newsid=19540
Tel: +86-20-8754-2392
Fax: +86-20- 8711-2101

5.3 Research and Development and References

In this section the research and development worldwide will be reported. It concerns institutions, universities and companies involved in the field of solar desiccant cooling. The enumeration of institutions etc. will be on country basis.

This list is of course not exhaustive; it is the result of the author's query and research.

France

LEPTIAB: Université de La Rochelle

Research: LEPTIAB carried out numerical and experimental investigation of a solar desiccant cooling system.

Corresponding researchers:

Paul Bourdoukan: Paul.Bourdoukna@univ-lr.fr
Patrice Joubert: Patrice.Joubert@univ-lr.fr

Relevant publications:

[124] Bourdoukan, P., Wurtz, E., Joubert, P., and Sperandio, M., (2008) "Potential of solar heat pipe vacuum collectors in the desiccant cooling process: modelling and experimental results." *Solar Energy*. 82(12), 1209-1219.

[125] Bourdoukan, P., Wurtz, E., Joubert, P., and Spérandio, M. (2008). "Overall cooling efficiency of a solar desiccant plant powered by direct flow vacuum tube collectors: simulation and experimental results" *Journal of Building Performance Simulation* 1(3), 149-162

[126] Bourdoukan, P., Wurtz, E., Joubert, P. (2009). "Comparison between the conventional and recirculation modes in desiccant cooling cycles and deriving critical efficiencies of components". *Energy an international journal, in press corrected proof.*

[127] Bourdoukan, P., Wurtz, E., Joubert, P. (2009). "Experimental investigation of a solar desiccant cooling system" *Solar Energy, accepted for publication*

[128] Bourdoukan, P., Wurtz, E., Joubert, P., and Spérandio, M. (2008). "A control strategy to prevent the impact of outside and load conditions on the hygro-thermal performance of a desiccant cooling system." *In proceedings of Indoor Air conference*, Copenhagen, Denmark. CD-Rom

[129] Bourdoukan, P., Wurtz, E., Joubert, P., and Sperandio, M. (2008). "Critical efficiencies of components in desiccant cooling cycles and a comparison between the conventional mode and the recirculation mode." *In proceedings of ECOS conference*, 1, 435-443. Krakow, Poland.

[130] Bourdoukan, P., Wurtz, E., Joubert, P., and Spérandio, M. (2008). "A sensitivity analysis of a desiccant wheel" *In proceedings of Eurosun conference*, Lisbon, Portugal. CD-Rom

[131] Bourdoukan, P., Wurtz, E., Sperandio, M., and Joubert, P. (2007). "Global efficiency of direct flow vacuum collectors in autonomous desiccant cooling: simulation and experimental results." *In proceedings of Building Simulation conference*, 1, 342-347. Beijing, China.

LOCIE-INES/CNRS

Research: In the field of desiccant cooling systems, LOCIE carries out numerical investigation of desiccant systems.

Corresponding researchers:

Paul Bourdoukan: Paul.Bourdoukan@univ-savoie.fr
Etienne Wurtz: Etienne.Wurtz@univ-savoie.fr

Relevant publications:

[132] Bourdoukan, P., Wurtz, E., Joubert, P. (2009). "Comparison between the conventional and recirculation modes in desiccant cooling cycles and deriving critical efficiencies of components". *Energy an international journal, in press Corrected proof.*

[133] Bourdoukan, P., Wurtz, E., Joubert, P. (2009). "Experimental investigation of a solar desiccant cooling system" *Solar Energy, accepted for publication*

CEA-INES

Research: CEA-INES is investigating a new desiccant exchanger cooled by water besides some model development for desiccant wheel.

Corresponding researchers:

Philippe Papillon: philippe.papillon@cea.fr
François Boudéhenn: francois.boudehenn@cea.fr

Relevant publications:

[134] Clausse M., Meunier F., Perigaud Y., Boudehenn F. & Demasles H., Experimental characterisation of a novel adsorber heat exchanger for desiccant cooling applications, Article, 22nd IIR International Congress of Refrigeration, Beijing (Chine), 21-26 August 2007.

[135] Demasles H., Boudehenn F. & Clausse M., Numerical and experimental studies of a novel adsorber heat exchanger for desiccant solar air conditioning, Article, 2nd International Conference Solar Air-Conditioning, Tarragone (Espagne), 18-19 October 2007.

[136] Boudehenn F. & Burgun F., Technical analysis of desiccant cooling systems, Article, 2nd Workshop of EMINENT project, Weszprem (Hongrie), 05-06 May 2008.

[137] Clausse M., Meunier F., Perigaud Y., Boudehenn F. & Demasles H., Experimental investigation and building integration of a novel adsorber heat exchanger for solar desiccant cooling applications, article, International Sorption Heat Pump Conference 2008, Seoul (Corée), 23-26 September 2008.

CETHIL-INSA Lyon

Research: CETHIL developed mainly numerical model for desiccant cooling systems

Corresponding researchers:

Jean Brau: jean.brau@insa-lyon.fr

Relevant publications:

[138] Thibaut Vitte, Jean Brau, Nadège Chatagnon and Monika Woloszyn, "Proposal for a new hybrid control strategy of a solar desiccant evaporative cooling air handling unit", Energy and Buildings, In Press, Corrected Proof , Available online 25 July 2007.

EDF, Electricité de France

Research: In association with CETHIL, EDF research domain is the simulation of desiccant cooling systems.

Corresponding researcher:

Nadège Chatagnon: nadege.chatagnon@edf.fr

China

In China, some research institutes have conducted extensive investigation work on solid desiccant cooling systems. Information about these institutes is listed below:

Shanghai Jiao Tong University
Research: carried out lots of investigations in the past years relating to desiccant cooling systems, such as desiccant material, construction of solid desiccant wheel and desiccant cooling system.

Corresponding researchers:
Prof. Ruzhu Wang (rzwang@sjtu.edu.cn)
Prof. Yanjun Dai (yjdai@sjtu.edu.cn).

Relevant publications:
[139] X.J. Zhang. Study on Dehumidification Performance of Silica Gel-Haloid Composite Desiccant Wheel. PhD Thesis, Shanghai Jiao Tong University, Shanghai, China; 2003 [in Chinese].
[140] C.X. Jia, Y.J. Dai, J.Y. Wu, R.Z. Wang. Experimental comparison of two honeycombed desiccant wheels fabricated with silica gel and composite desiccant material, Energy Conversion and Management 2006, 47(15-16): 2523-2534.
[141] C.X. Jia, Y.J. Dai, J.Y. Wu, R.Z. Wang. Use of compound desiccant to develop high performance desiccant cooling system, International Journal of Refrigeration 2007, 30(2): 345-353.
[142] C.X. Jia, Y.J. Dai, J.Y. Wu, R.Z. Wang, Analysis on a hybrid desiccant air-conditioning system, Applied Thermal Engineering 2006, 26(17-18):2393-2400.
[143] Ge TS, Li Y, Wang RZ, Dai YJ. Experimental study on a two-stage rotary desiccant cooling system. International Journal of Refrigeration 2009; 32 (3): 498-508.
[144] Ge TS, Dai YJ, Wang RZ, Li Y. Experimental investigation on a one-rotor two-stage desiccant cooling system. Energy 2008; 33 (12): 1807-15.
[145] K. Daou, R.Z. Wang, Z.Z. Xia. Desiccant cooling air conditioning: a review. Renewable and Sustainable Energy Reviews 2006, 10(2): 55-77.
[146] Y.J. Dai, R.Z. Wang, Y.X. Xu. Study of a solar powered solid adsorption desiccant cooling system used for grain storage, Renewable Energy 2002, 25(3): 417-430.

South China University of Technology
Research: investigated lots of different desiccant materials.

Corresponding researcher:
Prof. Jing Ding (cejding@scut.edu.cn).

Relevant publications:
[147] J. Ding, X.X. Yang, G.Q. Li, S.Z. Liang, Y.K. Tan. Sorption equilibrium of microporous active silica gel and the effect of sorption properties on the performance of the desiccant rotary dehumidifiers, Chemical Engineering 1998, 26:11-13. (in Chinese).

Beijing University of Aero & Astro
Research: conducted some work on solid desiccant cooling systems recently.

Corresponding researcher:
Weixing Yuan (yuanwx@buaa.edu.cn).

Relevant publications:
[148] W.X. Yuan, Y. Zheng, X.R. Liu, X.G. Yuan. Study of a new modified cross-cooled compact solid desiccant dehumidifier. Applied Thermal Engineering 2008, 28(17-18): 2257-2266.

[149] W.X. Yuan, Y. Zheng, H. Wang, X.G. Yuan, C.X. Yang. Study of transient dehumidifying performance of an internally cooling compact solid dehumidifier, Acta Energiae Solaris Sinica 2007, 28(1):7-11. (in Chinese).
[150] Y. Zheng, W.X. Yuan. Study of solar/waste heat driven solid desiccant cooling system. Refrigeration And Air-Conditioning 2006, 6(5):5-9. (in Chinese).
[151] Y. Zheng, W.X. Yuan, H. Wang, X.G. Yuan. Experiments on dynamic dehumidification of internally cooling compact solid dehumidifier. Journal of Beijing University of Aeronautics and Astronautics 2006, 32(9):1100-1103.

Tong Ji University
Research: studied the performance of solid desiccant cooling system.

Corresponding researcher:
Chaokui Qin (chkqin@mail.tongji.edu.cn).

Relevant publications:
[152] Y. Xu, C.K. Qin. The Solar-driven Solid Desiccant Dehumidification Air-conditioning System, Shanghai Gas 2007, 1:27-30. (in Chinese).

Japan
Tohoku University
Research: Numerical simulation of desiccant cooling systems

Corresponding researcher:
Hiroshi Yoshino: yoshino@sabine.pln.archi.tohoku.ac.jp

Relevant publications:
[153] N. Enteria, H. Yoshino, A. Mochida, R. Takaki, R. Yoshie, T. Mitamura, S. Baba, Construction and Initial Operation of the Combined Solar Thermal and Electric Desiccant Cooling System, Solar Energy, Vol. 83, Issue 8, pp. 1300-1311 (2009).
[154] N. Enteria, H. Yoshino, A. Mochida, R. Takaki, R. Yoshie, T. Mitamura, S. Baba, Synergization of Clean Energy Utilization, Clean Technology Development and Controlled Clean Environment Through Thermally Activated Desiccant Cooling System, 2008 ASME International Conference on Energy Sustainability "ASME ES" (August 10-14, 2008), Jacksonville, Florida, USA. Proceedings of ASME ES 2nd International Conference (ISBN 0-7918-3832-3), Paper No. ES2008-54103.

Tokyo University of Agriculture and Technology
Research:

Corresponding researcher
Prof. Atsushi Akisawa
Graduate School of Bio-Applications and Systems Engineering
Tokyo University of Agriculture and Technology
Naka-Cho 2-24-16, Koganei-shi, Tokyo, Japan
Tel: +81-42-388-7076
Fax: +81-42-388-7226
Webpage: http://www.tuat.ac.jp/~akilab/index.html

Relevant Publications:
[155] S. Elsayed, Y. Hamamoto, A. Akisawa, T. Kashiwagi, Using Air Cycle Refrigerator Integrated with Desiccant System for Simultaneous Air Conditioning and Domestic Hot

Water Heating, Journal of Environment and Engineering, Vo. 2, No. 3, pp. 514-524 (2007).

[156] Y. Hamamato, T. Tran, A. Akisawa, T. Kashiwagi, Experimental and Numerical Study of Desiccant Rotor with Direct Heating Regeneration by Solar Energy, 6th ASME-JSME Thermal Engineering Joint Conference, March 16-20, 2003. Paper No. TED-AJ03-273.

Tokyo University
Research:

Corresponding researcher:

Shinsuke Kato
Environmental Technology for Urban Architecture
Institute of Industrial Science
Tokyo University
4-6-1 Komaba, Meguro-Ku
Tokyo, 153-8505 Japan
Tel: +81-3-5452-6431
Fax: +81-3-5452-6432
Website: http://venus.iis.u-tokyo.ac.jp/english/default.htm

Relevant Publications:

[157] Y. Tsay, S. Kato, R. Ooka, M. Koganei, N. Shoda, Study on the Applicability of Combining Desiccant Cooling System with Heat Pump in Hot and Humid Climate, ASHRAE Transactions, Vol. 112, Part 1, pp. 189-194 (2006).

[158] Y. Tsay, S. Kato, R. Ooka, M. Koganei, N. Shoda, K. Nishida, K. kawamoto, Study on Non-Condensing Air-Conditioning System – Performance when Combining Desiccant Cooling System with CO2 Heat Pump, HVAC&R Research, Special Issue, Vol. 12, No. 3c, pp. 917-933 (2006).

Kanazawa University
Research:

Corresponding researcher:

Akio Kodama (kodama@t.kanazawa-u.ac.jp)
Technology and Human Communication Laboratory
Department of Human and Mechanical Systems Engineering
Kanazawa University
Kanazawa, Ishikawa, 920-1192 Japan
Tel: +81-76-234-4663
Fax: +81-76-234-4664
Website: http://www.hm.t.kanazawau.ac.jp/lab/people/kodama.html

Relevant Publications:

[159] Kodama, M., Ohkura, T. Hirose, M. Goto, H. Okano, An Energy Flow Analysis of a Solar Desiccant Cooling Equipped with a Honeycomb Adsorber, Adsorption Vol. 11, pp. 597-602 (2005).

[160] K. Ando, A. Kodama, T. Hirose, M. Goto, H. Okano, Experimental Study for Adsorption Desiccant Cooling Driven with a Low-Temperature Heat, Adsorption Vol. 11, pp. 631-636.

Germany
Fraunhofer ISE
Research: **The ECOS System Concept**
(Contribution from Constanze Bongs)

A new development of a system concept called ECOS (Evaporatively cooled sorptive heat exchanger) for small air-flow capacities up to 400 m^3/h is currently developed at Fraunhofer ISE. The ECOS system concept consists of two air-to-air plate heat exchangers which unite air dehumidification by adsorption and air cooling by indirect evaporative cooling in a single component. An operational scheme is given in Figure 32.

The plate heat exchanger is divided into desiccant-coated sorptive channels and cooling channels which are in thermal contact. Ambient air passing the sorptive channels is dehumidified and simultaneously cooled by indirect evaporative cooling. This is achieved through evaporation of water sprayed into the cooling channels. Therefore, the heat of adsorption is removed. This leads to the cooling of the sorptive matrix and to an enhanced sorption capacity in comparison to adiabatic processes such as in a dehumidifier wheel. In order to provide a continuous supply air flow to the building, the two heat exchangers are operated alternately – while one heat exchanger is in adsorption mode providing air dehumidification and cooling, the second heat exchanger is regenerated and pre-cooled. The main advantages of the ECOS cooled sorptive heat-exchanger concept are the enhanced dehumidification, the simultaneous air cooling and dehumidification and the strict separation of the supply and the return air flow, avoiding carry-over effects common in rotary DEC systems.

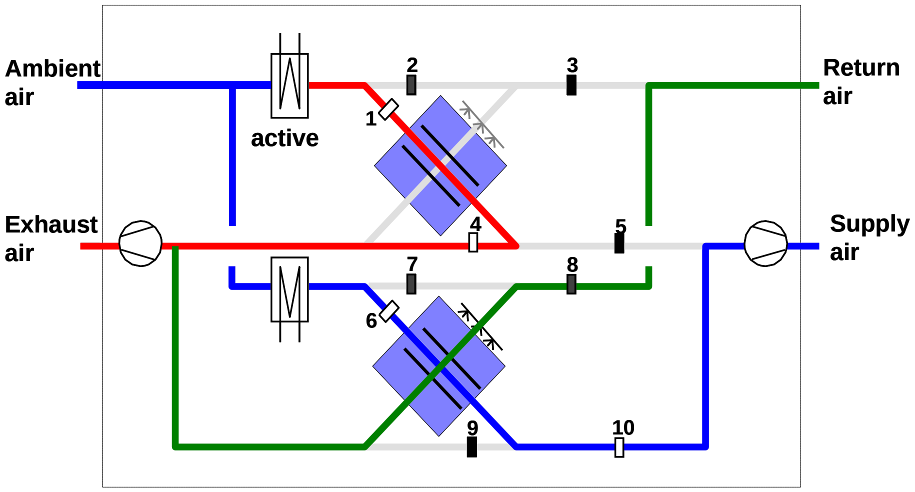


Figure 32: ECOS system concept: regeneration of the upper heat exchanger and ambient air dehumification and cooling in the lower heat exchanger.

Corresponding researchers:

Hans-Martin Henning: Hans-Martin.Henning@ise.fraunhofer.de

Alexander Morgenstern: Alexander.Morgenstern@ise.fraunhofer.de

Constanze Bongs: Constanze.Bongs@ise.fraunhofer.de

Relevant publications:

[161] Motta, M., Henning, H.-M., Kallwellis, V. (2004): Performance analysis of a novel desiccant and evaporative cooling cycle, Proc. HPC2004, Lacarna, October 11-13, Greece.

[162] Motta, M., Henning, H.-M. (2005): A novel high efficient sorption system for air dehumidification (ECOS). Proc. International Sorption Heat Pump Conference, Denver, June 22-24.

[163] Motta M., Henning, H.-M. (2006) : An Advanced Solar assisted sorption cycle for building air-conditioning: the ECOS potential and performance assessment, Proc. Eurosun 2006, Glasgow, June 27-39.
[164] Henning, H.-M. (2007): Solar assisted air-conditioning of buildings – an overview, Applied Thermal Engineering 27 (2007) 1734 – 1749.
[165] Morgenstern, A., Bongs, C., Henning, H.-M. (2007): ECOS- Ein neues, hocheffizientes Verfahren zur sorptiven Luftentfeuchtung im Vergleich mit sorptionsgestützten Klimatisierungsanlagen in Rotorbauweise. Proc. OTTI Solarenergie, Kloster Banz, Germany, April 23-25.
[166] Bongs, C., Henning, H.-M., Morgenstern, A. (2008): Modelling and exergetic assessment of a sorptive heat exchanger for the application in a novel desiccant evaporative cooling cycle. Proc. EuroSun 2008, Lisbon, October 7-10, Portugal.
[167] Bongs, C., Morgenstern, A., Henning, H.-M. (2009): Evaluation of Sorption Materials for the Application in an Evaporatively Cooled Sorptive Heat Exchanger, Proc. HPC2009, Berlin, September 7-9.
[168] Morgenstern, A., Bongs, C., Wagner, C., Henning, H.-M. (2009): Experimental evaluation of a sorptive-coated heat exchanger prototype for dehumidification purposes, Proc. OTTI Solar Air Conditioning, Palermo, September 30- October 2.

ZAE Bayern

Research: An open adsorption system using Zeolite was installed for the air-conditioning of a Jazz Club in Munich/Germany. The system was used as a thermal driven chiller and a thermal energy storage system simultaneously. The thermal COP reached was about 0.85. The open system reduced room temperature and humidity at the same time.

Corresponding researcher:

Dr. Andreas Hauer, hauer@muc.zae-bayern.de

Relevant publications

[169] Hauer, Thermal Energy Storage with Zeolite for Heating and Cooling, Proceedings of the 7th International Sorption Heat Pump Conference ISHPC ´02, Shanghai, China, 24.-27. September 2002.

Italy

Polimi: Politechnico di Milano

Research: Polimi is carrying on research in rotor based desiccant cooling. Both experimental and modelling activities are ongoing. One demonstrative AHU of 5,000 m3/h driven by CHCP is installed and monitored in the laboratory.

Corresponding researchers:

Mario Motta – mario.motta@polimi.it
Marcello Aprile – marcello.aprile@polimi.it
Lorenzo Pistocchini – Lorenzo.pistocchini@polimi.it

DREAM

Research: Research activities at DREAM on desiccant cooling systems are focused on simulation in TRNSYS and SIMULINK environment and monitoring of a real desiccant cooling system. Main development and results of the experimental work are related to the detailed characterization of single components as well as the whole system in terms of energy performance and reliability, the control strategy, the heat rejection in coupling with conventional cooling machine and the calibration of the simulation models.

Corresponding researchers:
Pietro Finocchiaro finocchiaro@dream.unipa.it
Bettina Nocke bettina@dream.unipa.it
Marco Beccali marco.beccali@dream.unipa.it

Relevant publications

[170] Beccali M., Finocchiaro P., Sorce M. – "Optimisation of Solar Desiccant Cooling Systems for applications in the Mediterranean Climate: design and control issues" - International Conference Solar Air-Conditioning – OTTI Kloster Banz - 05.10.2005

[171] Beccali M., Finocchiaro P., Nocke B. – "Solar desiccant cooling systems with hybrid Air-PV solar collectors for applications in the Mediterranean climate" - 61st ATI National Congress – International Session "Solar Heating and Cooling" – pagg. 75-79 - Perugia 14.09.2006

[172] Beccali M., Finocchiaro P., Nocke B., Gioria S., "Solar desiccant cooling AHU coupled with chilled ceiling: description of a new installation at DREAM in Palermo", Proceedings of the OTTI Conference Solar Air Conditioning, Tarragona (E), October 18th -19th., 2007, pp 389-394, ISBN 978-3-934681-61-3

[173] Beccali M., Finocchiaro P.; Luna M, Nocke B. "Un impianto di Solar Desiccant Cooling a Palermo. Programma di ricerca e primi risultati sperimentali", Convegno AICARR Padova "Riduzione dei fabbisogni, recupero di efficienza e fonti rinnovabili per il risparmio energetico nel settore residenziale", 5 Giugno 2008

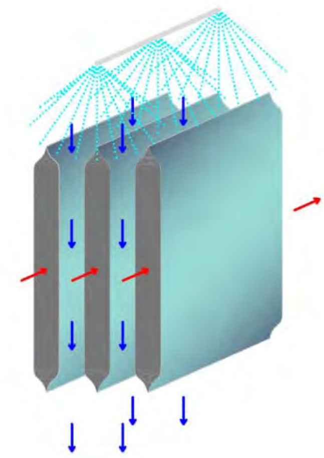
[174] Beccali M, Finocchiaro P., Luna M, Nocke B. "Monitoraggio di un impianto di solar desiccant cooling a Palermo. Primi risultati e progetto dei test". 63° Convegno ATI. Palermo, 23-26 Settembre 2008, pp. 07.024

[175] Beccali M., Finocchiaro P., Luna M., Nocke B. "Monitoring of a solar desiccant cooling system in Palermo (Italy). First results and test planning". Intern. Conference EUROSUN 2008. Lisbona. 7-10 Oct 2008. (pp. 316-317).

[176] Beccali M., Finocchiaro P., Nocke B. "Energy and economic assessment of desiccant cooling systems coupled with single glazed air and hybrid PV/thermal solar collectors for applications in hot and humid climate" Solar Energy Elsevier 2009

DEC system with wet heat exchanger

(contributed by Pietro Finocchiaro, DREAM, University of Palermo)

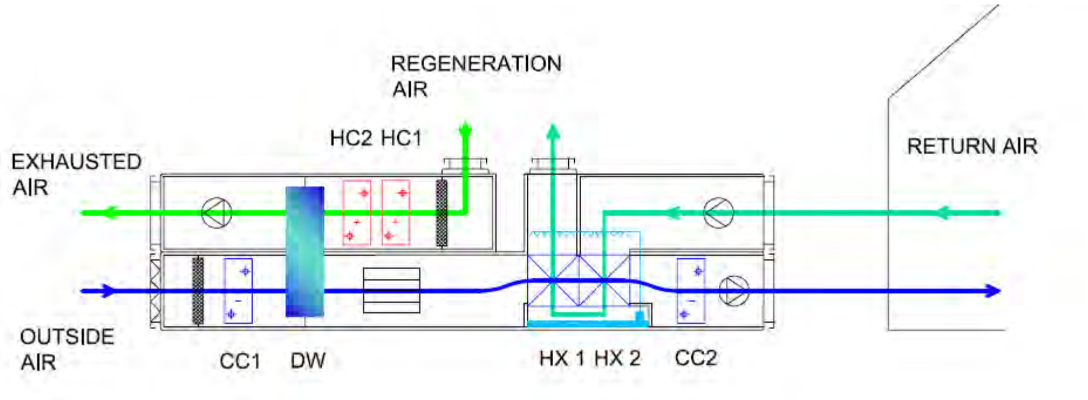


With the aim to increase the cooling effect due to water evaporation in the return air flow rate, standards DEC configuration can be modified replacing the heat recovery wheel (sensible heat exchanger) with one or more plate heat exchangers in series with continuous humidification of the secondary flow.

The wet heat exchanger used is similar to a closed loop wet cooling tower and consists of a cross flow flat plate heat exchanger, spray nozzles, basin and recirculation pump. Spray nozzles used operate with low water pressure and do not require special maintenance.

In the following figure a two-stage system is presented. In the configuration shown, return air is humidified in two steps and leaves the AHU after the heat exchange with supply air stream. In addition, desiccant wheel is regenerated by external air, which is heated before by the two heating coils. This means an additional fan, but at the same time, regeneration air flow can be reduced and no bypass is used (see following scheme). The exclusion of the by-pass across the desiccant rotor and the use of outside air for regenerating the desiccant rotor, can improve performances of the system also in terms of thermal COP and reduction on electricity consumption for the regeneration.

First experimental investigations confirm good performances of the system equipped with wet heat exchangers. Thanks to the optimization of the indirect evaporative cooling process, the contribution of auxiliary cooling coils can strongly reduced.



Furthermore, the use of the plate heat exchanger eliminate the moisture carryover that can occur in the rotative heat exchanger normally used in DEC systems.

The risk of fouling (limescale) in the wet heat exchangers has to be taken in consideration during the design phase.

6 Thermo-mechanical chillers

Klaus Ellehauge (Ellehauge and Kildermoos, Denmark)

6.1 Brief description of the technology

Thermo-mechanical chillers are mechanically driven heat-pumps where the mechanical driving forces are provided by conversion of low temperature heat to mechanical energy.

The idea of providing cooling by combining a mechanical heat pump with a machine for conversion of low temperature heat to mechanical power has only been tried by very few companies. To be successful it would seem desirable to integrate the mechanical structure and the thermodynamics of the two processes.

Of course in separate processes, the mechanical driven heat pump is a well known technology, while the applicable technology for the conversion of low temperature heat, e.g from a solar collector, to mechanical power is not so well developed.

This technology is available only from a small number of larger companies, while a larger amount of smaller companies are researching the technology.

The conversion of thermal power to mechanical power follows the Rankine cycle. In order to keep the driving temperatures low it is most common to operate the Rankine machine with an organic liquid as the working fluid. The technology making use of organic Rankine cycles is often referred to as ORC technology. The temperature ranges in which the ORC machines run efficiently are however not low enough to fully suit low temperature solar collectors.

However a Danish system under development that uses water at low pressure, operates at temperatures suited for solar collectors and provides cooling as well.

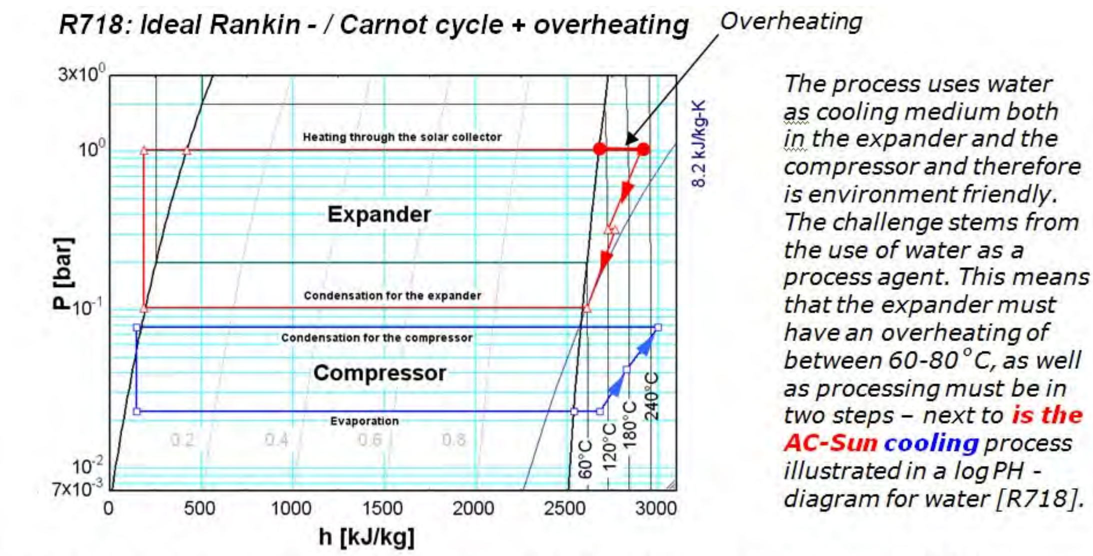


Diagram 1: process of AC-Sun illustrated in a log PH-diagram for water [R718]

6.2 Companies on the market

6.2.1 Designs based on the combined use of solar thermal and thermo-mechanical cooling

AC-Sun:

AC-Sun is a Danish company developing a turbo/expander, which works with water as a process agent at low pressure and temperatures, and which is driven by thermal solar panels. The conventional thermal solar collector produces low temperature steam on vacuum.

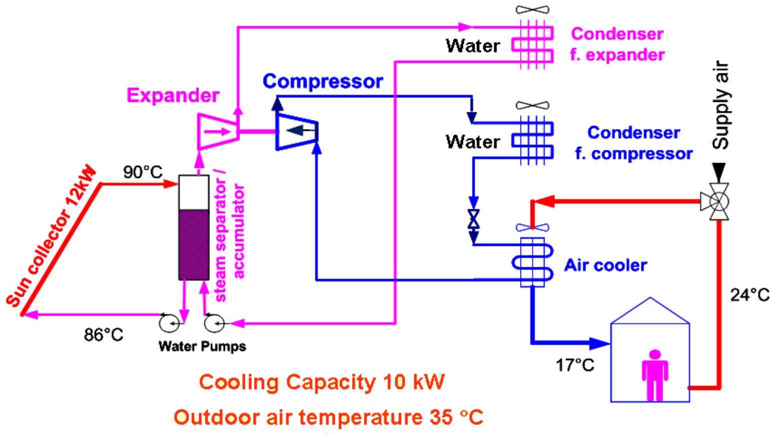


Figure 33: Principle sketch of AC-Sun

The plant is a closed self-contained unit, and only needs to be connected to the thermal solar collector, which consists of a traditional thermal solar panel using water with temperatures between 75-95˚C. District heating or waste heat can also be used as a heat source in the same temperature range. An expander, based on a Rankine process, uses the energy from solar panels to operate a compressor, which through a Carnot process cools air in traditional air coolers.

In the expander (turbine) the energy of the steam is converted into rotational energy which drives the compressor on the same shaft. The compressor - maintaining a vacuum in the cooling circuit - allows for the water to boil at low temperature and low pressure. As the water boils and evaporates, energy is absorbed. This energy is delivered by the circulating air in the air cooler, whereby the temperature in the surrounding areas is lowered.

The steam produced in the cooling process and the solar collector is finally condensed in two air cooled condensers. Electrical power consumption for driving the fans for air distribution and the small water pumps is only 10% of normal AC consumption.

<u>AC-Sun, design basis:</u>

For industrial air-conditioning plants for the Mediterranean area, the following conditions are assumed: an inlet temperature of 17°C and an outdoor temperature of 35°C. These are kept in the upcoming turbine construction with or without water supply.

The design capacity is based on:

Outdoor air temperature 35˚C (Range 25 - 40˚C)

Solar collector temperature 85˚C (Range 80 - 100˚C)

Air inlet temperature 17˚C

Solar collector capacity 12 kW (set to 12 m 2 , Equator)

Cooling capacity 10 kW

Power consumption 0.4 kW

Contact & further information: www.ac-sun.com

6.2.2 Designs using ORC in the thermo-mechanical cooling process.

Turboden

Turboden is an Italian based company specialized in the applications of Organic Rankine Cycle technology.

Main applications of the products of the company are stated as follows:

- Biomass cogeneration for district-heating or for sawmills and pellet manufacturing companies;
- Waste heat recovery: electric energy production from exhaust streams in industrial processes;
- Small combined cycles: electric energy production from residual heat of internal combustion engines, or gas turbines;
- Geothermal: mainly from low-temperature liquid-dominated wells (100-180 °C);
- Solar thermodynamic conversion: electric energy production from medium high temperature solar collectors.

Turboden ORC units are typically in the size of 2-10 MW (thermal input) and giving an electric power output with an efficiency around 20%.

Turboden units are chosen for the purpose and operate mainly with the following temperatures.

Biomass based cogeneration systems:

310/250°C (high temperature loop) 250/130°C (low temperature loop)

Plant in heat recovery process:

260/150°C

High efficiency units:

310/220°C (high temperature loop) 220/95°C (low temperature loop)

Website and contact: www.turboden.eu

Ormat Technologies, Inc.

Ormat Technologies, Inc, is based in USA and claims to be the world leader in Organic Rankine Cycle (ORC) power systems. Most of Ormat's products and business activities are based on its Ormat Energy Converter (OEC), for the utilization of low and medium temperature heat sources. The OEC is a Closed Cycle Vapor Turbogenerator (CCVT) using an organic motive fluid and operating on the Organic Rankine Cycle (ORC).

Main applications are: Geothermal Power, Recovered Energy, Remote Power Units (heated by a burner)

Ormat power plants range from 250 kW to 130 MW.

• Brine temperature: from 95°C (203°F) to 315°C (600°F)

Website and contact: www.ormat.com

Furthermore a number of minor companies delivering ORC technology products exist on the market. Examples are:

LTi ADATURB GmbH (Germany)

Plants for usage of energy in exhaust gasses or cooling energy of motors.

Range: 175 – 350 kW (thermal) – 30-60 kW (electricity)

Website and contact: http://adaturb.lt-i.com

6.3 Research and Development

The above mentioned larger companies Turboden and Ormat Technologies, Inc. as well as AC-Sun are active on the development side. It is not known which universities and other institutes are active in research.

6.4 Relevant reports and publications (only examples)

[177] Søren Minds and Klaus Ellehauge, The AC-Sun, a new concept for air-conditioning, Eurosun 2008, Lisbon, proceedings http://www.iea-shc.org/publications/downloads/task38-AC-Sun.pdf

[178] A number of papers on ORC are published on the Turboden website: http://www.turboden.eu/en/downloads/downloads.php

[179] and on the Ormat Technologies, Inc. website: http://www.ormat.com/media_center.php?did=147

[180] Other papers of interest:

GRC Annual Meeting 2005; Reno, NV, USA, Power Production from a Moderate - Temperature Geothermal Resource http://www.yourownpower.com/Power/grc%20paper.pdf

7 Steam jet chillers

Clemens Pollerberg (Fraunhofer UMSICHT, Germany)

7.1 Brief description of the technology

The Steam Jet Ejector Chiller (SJEC) is a thermo-mechanical chiller for cold generation. A refrigerant boils at a low pressure in the evaporator. The necessary heat of evaporation is provided by the water to be chilled. The vapour refrigerant is then compressed by a steam jet ejector to a higher pressure level and fed into a condenser. The driving energy of the steam jet ejector is supplied by motive steam, which can be generated by solar energy. The motive steam and vapour refrigerant are mixed in the steam jet ejector and both are liquefied in the condenser. The condensation takes place at a higher temperature than the ambient temperature, so that the heat from the condensation process can be rejected to the atmosphere. The condensate from the condenser is fed back to the evaporator and to the motive steam generator. Figure 33 depicts the principle of a solar driven SJEC.

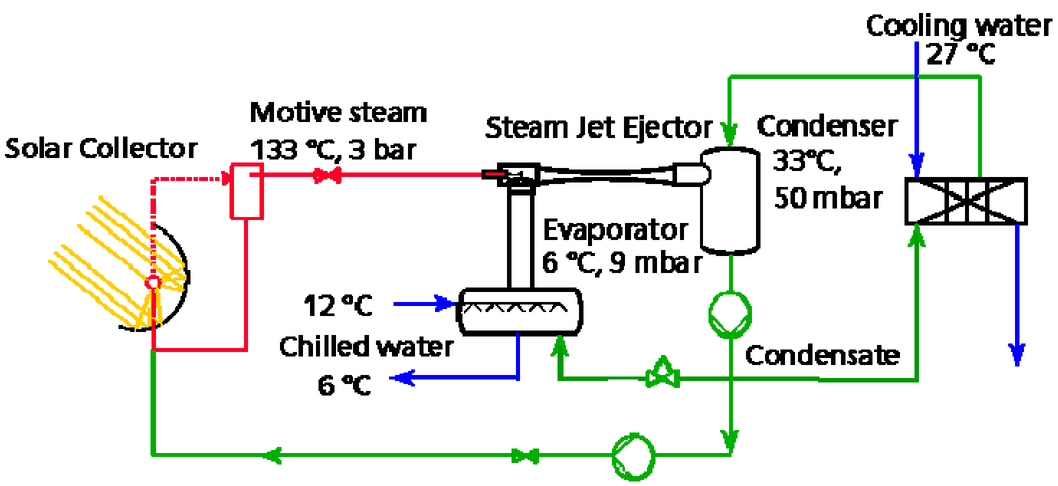


Figure 33: Principle sketch of a solar driven SJEC with water as working fluid and refrigerant.

A SJEC can use water as the working fluid, which is used for generating the motive steam and as a refrigerant. In that case, an open process can be constructed without hydraulic separation between the solar collector and the SJEC or the chilled water network and the SJEC. The evaporator can be designed as a flash evaporator and the condenser as a direct contact condenser, such that both devices are basically vessels. The steam jet ejector consists of a motive steam nozzle, a mixing chamber and a diffuser. Thus the whole solar cooling system is simple in design and a high reliability of operation can be expected.

Besides using water as both a working fluid and a refrigerant, different types of fluid have been discussed too. In particular, halocarbons, hydrocarbons and ammonia have been proposed for solar driven SJECs. The advantage of these proposed fluids is a reduction of the required temperature level of the driving heat, while still allowing adequate COP values to be reached. An overview of the possible types of fluid and an investigation of their applicability in a SJEC is given in [181].

SJECs in the higher cold capacity range are designed as two stage chillers and have several steam jet ejectors. In this case, the delivery rate of the ejectors could be different from each other in order to obtain a capacity control of the chiller by switching steam jet ejectors on/off. New approaches to increasing the efficiency of SJECs focus on the use of multiple ejectors in series and the use of two different types of fluid as working fluid and as refrigerant.

7.2 Main characteristics

SJECs have a specific operational behaviour. The comparison between the reversible COP_{rev} value and the real COP value of a SJEC points out this special operational behaviour, as shown in Figure 34. For this comparison, the COP is defined as the ratio of the cold energy produced in the evaporator Q_E to the motive heat energy Q_M required to operate the steam jet ejector, see equation 1. The reversible COP_{rev} value is calculated according to equation 2 with the motive steam temperature T_M (corresponding to the saturated steam pressure), the evaporator temperature T_E and the condenser temperature T_C. The evaporator temperature is 6 °C.

$$COP = \frac{\dot{Q}_E}{\dot{Q}_M}$$

Equation 1

$$COP_{rev} = \frac{(T_M - T_C)}{T_M} \cdot \frac{T_E}{(T_C - T_E)}$$

Equation 2

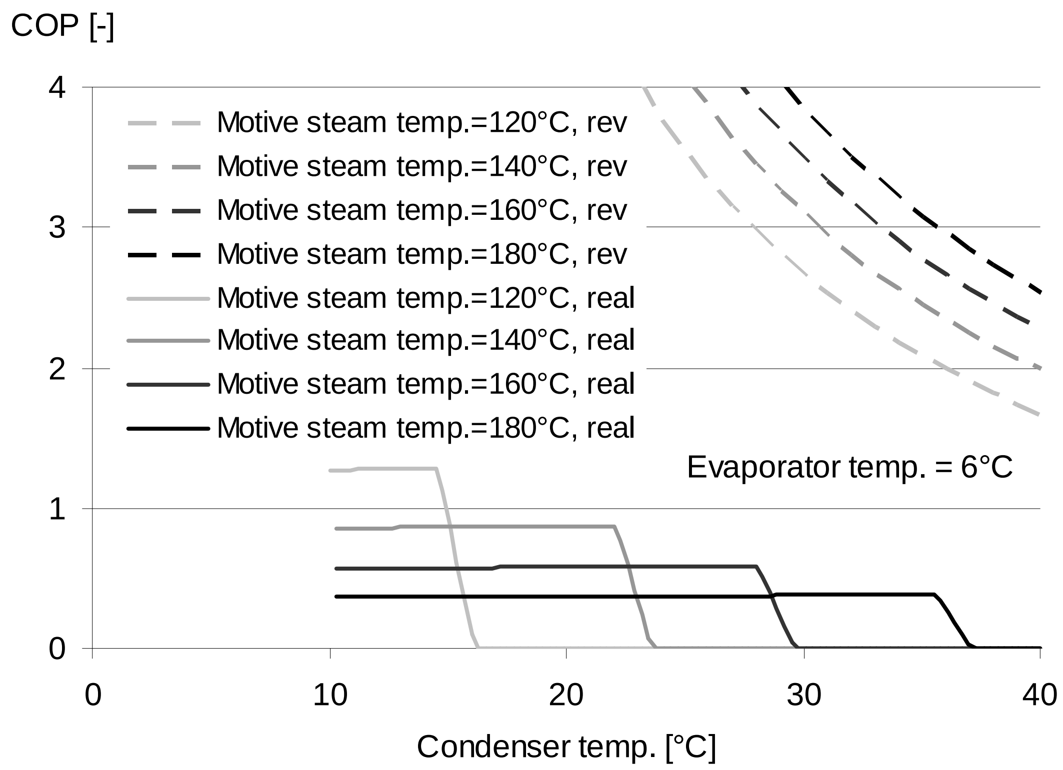


Figure 34: COP value of SJEC, reversible and real machine.

A higher motive steam temperature leads to a higher reversible COP_{rev} value. Furthermore the reversible COP_{rev} increases with decreasing condenser temperature. As a start, the COP value of a SJEC also increases with decreasing condenser temperature, but then remains constant below a certain condenser temperature. The reason for this behaviour is that the flow velocity in the steam jet ejector reaches supersonic speed and the mass flow through the ejector cannot be increased further. The comparison shows also that the motive steam temperature (corresponding to saturated steam pressure) can be reduced when the condenser temperature is low enough as the COP value increases. This correlation is used for the controlling concept of a SJEC and leads to the fact that the COP value of a SJEC depends predominantly on the condenser temperature. Furthermore, the COP value depends on the evaporator temperature as well. Figure 35 shows the influence of the condenser and evaporator temperatures on the COP value of a SJEC.

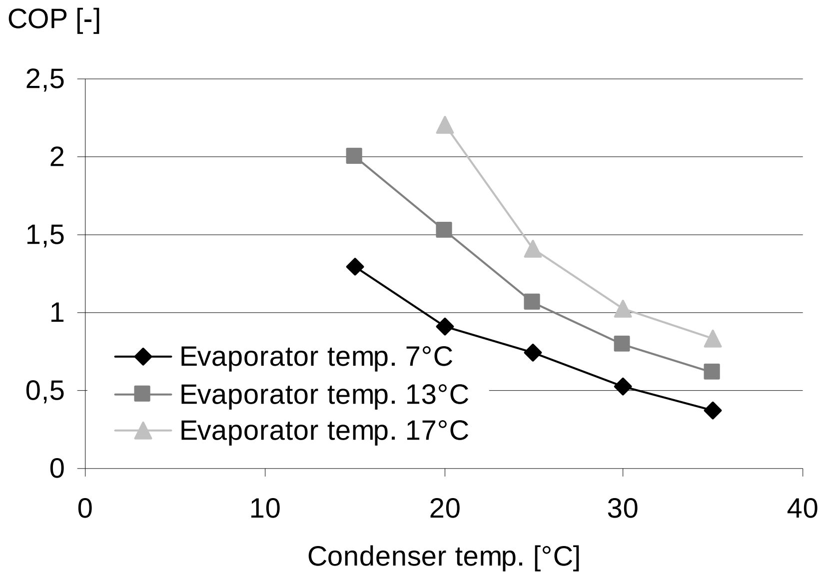


Figure 35: COP value as function of the condenser and evaporator temperature.

The COP value of the SJEC increases as the condenser temperature decreases or as the evaporator temperature increases. The condenser temperature is related to the cooling water temperature. The evaporator temperature is related to the chilled water temperature and may be raised when the cold load is lower. This means that at lower ambient temperatures or at part load the COP value of the SJEC increases. In view of a typical cold load curve for a comfort cooling system, which is characterised by long operation periods at low ambient temperatures and at part load of the system, a good mean COP value of SJEC can be expected.

7.3 Research and development

There are many companies producing steam jet ejector chillers for industrial use. This technology is currently used for large continuous flows of chilled water in industrial applications. As it is a well established product, there is no necessity in listing down the companies that manfucture these products in this report.

Until now, it is not known whether any solar driven steam jet chiller has come into the market as yet. However, there have been some test rigs and prototypes developed and tested. In 1966 Kakabaev and Davletov described in [182] a first test rig with a cold capacity of 1 kW_{th} with a total efficiency between 0.11 and 0.22. The motive heat is provided by a parabolic mirror sized 12 m². Freon is used as the working fluid and refrigerant. Huang, Petrenko and Chang describe in [183,184] a solar SJEC combined with flat plate collectors and R-141b as refrigerant. The temperature of the motive heat is 95 °C and the system reaches a total efficiency of 0.22 at a solar irradiation of 700 W/m². In [185] Wolpert and Riffat present a 7 kW test plant for air-conditioning of a hospital in Mexico. Water is used as the working fluid and refrigerant. In [186], a small test rig is described and its operational behaviour investigated. The SJEC has a cold capacity of 1 kW_{th} and is solar thermally driven by a parabolic trough collector with an aperture area of 10.5 m². Based on these results, a first completely automated prototype with a cold capacity of 5 kW_{th} was developed and is presented in [187]. Besides the practical work on solar SJECs, a lot of theoretical work has been done to investigate and to optimise the process, as shown for example in [188,189,190,191,192,193,194,195,196,197,198,199,200]. A more detailed review of solar driven steam jet ejector technologies is given in [201].

Research institutes that are active in this field are (note: list is not complete):

1. Fraunhofer Institute UMSICHT
   Osterfelder Strasse 3,
   46047 Oberhausen,
   Germany.

   Researcher: Clemens Pollerberg

   e-mail: clemens.pollerberg@umsicht.fraunhofer.de

2. AEE – Institute for Sustainable Technologies,
   Feldgasse 19,
   8200 Gleisdorf,
   Austria

   Researcher: D. Jaehnig

   e-mail: d.jaehnig@aee.at

3. University of Stellenbosch
   Department of Mechanical and Mechatronic Engineering
   PO Box x1, Matieland 7602
   South Africa

   Researcher: A. J. Meyer

   e-mail: meyer@sun.ac.za

4. University of Nottingham
   School of theBuilt Environment
   NG7 2RD
   United Kongdom

   Researcher: Saffa Riffat

   e-mail: saffat.riffat@nottingham.ac.uk

5. Université catholique de Louvain UCL
   Ecole Polytechnique de Louvain, Mechanical Engineering
   Department, TERM Division
   Place du Levant 2, B-1348, Louvain-la-Neuve, Belgium

   Researcher: A. Hemidi

   e-mail: amel.hemidi@uclouvain.be

6. Selcuk University – Alaeddin Campus
   Department of Mechanical Engineering,
   Falculty of Engineering and Architecture

42250 Konya
Turkey

Researcher: H. Kursad Ersoy

e-mail: kersoy@selcuk.edu.tr

7. National Taiwan University
Department of Mechanical Engineering
No.1 Roosevelt Rd. Sec 4
Taipei, Taiwan 106
Taiwan

Researcher: Bin-Juine Huang

e-mail: bjhuang@seed.net.tw

7.4 References

[181] Wimolsiri Pridasawas, Solar-Driven Refrigeration Systems with Focus on the Ejector Cycle, Doctoral Thesis, School of Industrial Engineering and Management – Royal Institute of Technology KTH, Stockholm, 2006.
[182] Kakabaev, A., Davletov, A., A Freon Ejector Solar Cooler, Geliotekhnika, Vol. 2, No. 5, pp. 42-48,. AS Turkmen SSR,1966.
[183] Huang, B.J., Petrenko, V.A., Chang, J.M., Zhuk, K.B., A High-Performance Solar Ejector Cooling System, Department of Mechanical Engineering, National Taiwan University, Taipei, Taiwan/Odessa State Academy of Refrigeration, Odessa, Ukraine, 2000.
[184] Huang, B.J., Petrenko, V.A., A combined Ejector Cooling and hot water supply system using solar and waste heat energy, Department of Mechanical Engineering, National Taiwan University, Taipei, Taiwan/Odessa State Academy of Refrigeration, Odessa, Ukraine, 2000.
[185] Wolpert, J.L., Riffat, S.B., Hybrid Solar/Gas Cooling Ejector Unit for a Hospital in Mexico, http://www.kenes.com/ises.abstracts/Htm/0171.htm, Institute of Building Technology, School of the Built Environment, University of Nottingham, Great Britain, 1999.
[186] Pollerberg, C. Ali, A.H.H., Dötsch, C: Solar driven steam jet ejector, Applied Thermal Engineering 29, pp. 1245–1252, 2009.
[187] Pollerberg C., Jaehnig D., Doetsch C., Prototype of a Solar Driven Steam Jet Ejector Chiller, Proceedings of the 3rd International Conference on Solar Air-Conditioning, Palermo, 30.09- 2.10.09, ISBN 978-3-941758-06-9, 2009.
[188] Anderson, H., Assessment of solar powered Vapor Jet Air-Conditioning Systems, Proc. Int. Solar Energy Congress and Exposition (ISES), UCLA, Los Angeles, Ca, pp. 408-409. USA, 1975.
[189] Zhadan, S., Z., Shchetinina, N., A., Selection of cycle Design Parameters for Solar Ejector Freon Refrigeration Machine (SEFRM), Geliotekhnika, Vol. 16, No.1, pp. 44-47, 1977.
[190] Sokolov, M., Hershgal, D., Solar-Powered Compression-enhanced Ejector Air Conditioner, Solar Energy Vol. 51, No. 6, pp. 183-194, 1993.
[191] Petrenko, V., A., Bulavin, I., V., Samofatov, I., YA., Investigation of the Methods increasing the efficiency of solar ejector cooling and refrigeration systems, Department of Mechanical Engineering, National Taiwan University, Taipei, Taiwan/Odessa State Academy of Refrigeration, Odessa, Ukraine, 2000.
[192] Vidal, H., Colle, S., Pereira, G. dos S., Modelling and hourly simulation of a solar ejector cooling system, Applied Thermal Engineering 26, pp. 663-672, 2006.

[193] Riffat, S., et al., Study of a gas-liquid ejector and its application to a solar-powered bi-ejector refrigeration system, Applied Thermal Engineering, Vol. 25, pp. 2891-2902, 2005.
[194] Khattab, N.M., Barakat, M.H., Modelling the design and performance characteristics of solar steam-jet cooling for comfort air conditioning, Solar Energy, Vol. 73, pp. 257-267, 2002.
[195] Hemidi, A., et al., Designing and Rating a Tritherm Solar Ejector System for Residential Cooling, Proceedings of the EUROSUN 2008, 1st International Conference on Solar Heating, Cooling and Buildings, 7th to 10th October Lisbon – Portugal, ISBN 978-972-95854-7-0, 2008.
[196] Ersoy, H.K., et al., Performance of a solar ejector cooling-system in the southern region of Turkey, Applied Energy, Vol. 84, pp. 971-983, 2007.
[197] Colle, S., et al., On the validity of a design method for a solar-assisted ejector cooling system, Solar Energy, Vol. 83, pp. 139-149, 2009.
[198] Vidal, H., et al., Modelling and hourly simulation of a solar ejector cooling system, Applied Thermal Engineering, Vol. 26, pp. 663-672, 2006.
[199] Guo, J., Shen, H.G., Modelling solar-driven ejector refrigeration system offering air conditioning for office buildings, Energy and Building, Vol. 41, pp. 175-181, 2009.
[200] Alexis, G.K., Karayiannis, E.K., A solar ejector cooling system using refrigerant R134a in the Athens area, Renewable Energy, Vol. 30, pp. 1457-1469, 2005.
[201] Abdulateef, J.M., Alghoul, M.A., et al., Review on solar-driven ejector refrigeration technologies, Renewable and Sustainable Energy Review, Vol. 13, pp. 1338-1349, 2009.

Appendix 1 Contact details of contributors

Name: Michael Krause
Institution: Fraunhofer Institute of Building Physics, Energy Systems
Address: Gottschalkstr. 28a
34125 Kassel,
Germany
e-mail: michael.krause@ibp.fraunhofer.de

Name: Robert Ghirlando
Institution: Faculty of Engineering
University of Malta
Address: Msida
Malta
e-mail: robert.ghirlando@um.edu.mt

Name: Erich Podesser
Institution: AEE-INTEC
Address: Feldgasse 19
A-8200 Gleisdorf,
Austria
e-mail: erich.podesser@utanet.at

Name: Tomas Núñez
Institution: Fraunhofer Institute of Solar Energy
Address: Heidenhofstr. 2
79110 Freiburg
Germany
e-mail: Tomas.Nunez@ise.fraunhofer.de

Name: Mustafa Jaradat
Institution: Kassel University, Solar and Systems Engineering
Address: Kurt-Wolters-Str. 3,
34125 Kassel
Germany
e-mail: jaradat@uni-kassel.de

Name: Roland Heinzen
Institution: Kassel University, Solar and Systems Engineering
Address: Kurt-Wolters-Str. 3,
34125 Kassel
Germany
e-mail: heinzen@uni-kassel.de

Name: Lucio Mesquita
Institution: Enerworks Inc., Applications Engineering
Address: 252, Hamilton Crescent
Dorchester
Canada
e-mail: l.mesquita@enerworks.com

Name: Paul Bourdoukan
Institution: LOCIE-INES/CNRS
Address: LOCIE FRE CNRS 3220
Université de Savoie
Institut National de l'Energie Solaire
Parc technologique de Savoie Technolac
50 av. Lac Léman - BP 332
F-73377 Le Bourget du Lac CEDEX
France
email: Paul.Bourdoukan@univ-savoie.fr

Name: Etienne Wurtz
Institution LOCIE-INES/CNRS
Address: LOCIE FRE CNRS 3220
Université de Savoie
Institut National de l'Energie Solaire
Parc technologique de Savoie Technolac
50 av. Lac Léman - BP 332
F-73377 Le Bourget du Lac CEDEX
France
e-mail Etienne.Wurtz@univ-savoie.fr

Name: Klaus Ellehauge
Institution: Ellehauge and Kildermoos
Address: Vestergade 48H, 2.tv.
8000 Århus C
Denmark
e-mail: klaus.ellehauge@elle-kilde.dk

Name: Constanze Bongs
Institution: Fraunhofer Institute of Solar Energy
Address: Heidenhofstr. 2
79110 Freiburg
Germany
e-mail: constanze.bongs@ise.fraunhofer.de

Name: Clemens Pollerberg
Insitution: Fraunhofer Institute UMSICHT
Address: Osterfelder Strasse 3,
46047 Oberhausen,
Germany.
e-mail: clemens.pollerberg@umsicht.fraunhofer.de

Name: Pietro Finocchiaro
Institution: Universita Degli Studi di Palermo
Dipartimento di Ricerche Energetiche ed Ambientali
Address: Viale delle Scienze
90128 Palermo,
Italy
e-mail: finocchiaro@dream.unipa.it

Name: Marcelo Izquierdo Millán (M. Izquierdo)
Institution: Instituto de Ciencias de la Construcción Eduardo Torroja (CSIC)
Address: C/ Serrano Galvache, 4,
28033 Madrid,
Spain
e-mail: mizquierdo@ietcc.csic.es

Name: José Daniel Marcos del Cano
Institution: Escuela Técnica Superior de Ingenieros Industriales (UNED)
Address: C/ Juan del Rosal 8,
28040 Madrid,
Spain
e-mail: jdmarcos@gmail.com

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning
and Refrigeration

Description of simulation tools used in solar cooling

New developments in simulation tools and models and their validation

Solid desiccant cooling

Absorption chiller

A technical report of subtask C
Deliverable C2-A

Date: Novembre 9, 2009

By

Paul Bourdoukan[1]

Contributions by

Edo Wiemken[2], Paul Kohlenbach[3], Etienne Wurtz[4], Harald Moser[5], Alexander Morgenstern[6], Antoine Dalibard[7], Felix Ziegler[8], Constanze Bongs[9]

[1] LOCIE FRE CNRS 3220, Institut National de l'Energie Solaire (INES),
Tel: +33 (0)4 79444545
Fax: +33 (0)4 79 253690
Email: Paul.Bourdoukan@univ-savoie.fr

[2] Fraunhofer Institute for Solar Energy Systems ISE, Freiburg, Germany
Tel: +49 761 4588 5412
Fax: +49 761 4588 9000
Email: edo.wiemken@ise.fraunhofer.de

[3] Solem Consulting
Postfach 2127, 71370 Weinstadt, Germany
Tel:+49 (0)174/ 41 30 92 1
Fax:+49 (0)7151/ 60 48 62 5
Email: info@solem-consulting.com

[4] LOCIE FRE CNRS 3220, Institut National de l'Energie Solaire (INES),
Tel: + 33 (0) 4 79444554
Fax:+ 33 (0) 4 79253690
Email: Etienne.wurtz@univ-savoie.fr

[5] Graz University of Technology, Institute of Thermal Engineering, Austria
Tel: +43 316 873 7304
Fax:+43 316 873 7305
Email: harald.moser@tugraz.at

[6] Fraunhofer Institute for Solar Energy Systems ISE, Freiburg, Germany
Tel: +49 761 4588 5107
Fax:+49 761 4588 9000
Email: alexander.morgenstern@ise.fraunhofer.de

[7] ZAFH.NET (Stuttgart University of Applied Sciences)
Tel: +49 (0)711/ 89 26-29 81
Fax: +49 (0)711/ 89 26-26 98
Email: antoine.dalibard@hft-stuttgart.de

[8] Technische Universität Berlin, Institute for Energy Engineering
Tel: 314-25624
Fax: 314-22253
Email: felix.ziegler@tu-berlin.de

[6] Fraunhofer Institute for Solar Energy Systems ISE, Freiburg, Germany
Tel: +49 761 4588 5487
Fax:+49 761 4588 9000
Email: constanze.bongs@ise.fraunhofer.de

Content

Introduction

Numerical simulation offers the possibility to study virtually physical systems and to test rapidly the proposed solutions. Simulation is then the most adapted method to understand the behaviour of a system in order to optimize it. The enhancement and the development of a technology are essentially based on the capacity to simulate accurately its behaviour in order to optimize it. This reality applies to solar cooling technologies.

In this report simulation tools used in the solar cooling domains are described. Then the recent development in the components and system models and validation are presented.

In part 1 of the report the simulation tools applicable in the domain of solar cooling are presented and the main advantages of each tool are highlighted.

Part 2 concerns the recent developments in modelling solid desiccant cooling technologies and the vacuum tube solar collectors with an experimental validation of an implementation in the SPARK simulation environment. Part 3 presents a recent transient model of the Li/Br absorption chiller with an experimental validation.

Each part has its reference section and conclusion.

This report and the authors don't pretend to include all the simulation tools used in solar cooling or all the recent models related to solid desiccant cooling and absorption chiller. The simulation tools described in this report are those identified and used by the participants of the IEA Task 38.

Chapter I: Description of simulation software, applicable for solar cooling

In this chapter a brief description of some simulation tools applicable in solar cooling is given. These tools are the most commonly used by IEA Task 38 participants. The range goes from predefined simple configuration software to low level equation solver. Some of the tools are developed by research groups and are dedicated for internal use and other are commercially available software. The simulation tools highlighted here are SPARK, EnergyPlus, EES, EasyCool, TRNSYS and INSEL.

SPARK

Paul Bourdoukan and Etienne Wurtz,
LOCIE FRE CNRS 3220, Institut National de l'Energie Solaire

SPARK is a general simulation environment that supports the definition of simulation models and solution of these models via a robust and efficient differential/algebraic equation solver. In SPARK, the modeller describes the set of equations defining a model as equation-based objects. At the lowest level, an atomic object characterizes one equation and its variables. Then, macroscopic objects can be created as an assembly of various atomic or macroscopic objects. The entire model is built by connecting the different necessary objects. If one class of objects needs to be reused, it can be instantiated as many times as required, without any additional effort.
At this stage, it is necessary to observe that the model is input/output free. The particular problem to be solved is then described by imposing the adequate input data (boundary and initial conditions) and by specifying the variables to be solved. So in this environment it is not necessary to order the equations or to express them as assignments statements (algorithms) in opposition to conventional modular environments such as Matlab/Simulink or TRNSYS.
SPARK uses a mathematical graph of the model to decompose it as strong components to be solved independently. Within each component, SPARK finds the appropriate function call sequence to get the solution. If no direct sequence is possible, as evidence by a cyclic problem graph, a small "cut set" is determined so as to minimize the number of variables involved in the Newton-like iterative numerical solution process. As a result, this decreases the size of the Jacobian matrix involved in the Newton iteration within the component. Consequently, the way SPARK handles the solution of coupled nonlinear equations makes it a fast solver for building energy simulation problems.
Since the "cut sets" of variables have been identified, the problem specification file is converted into a C++ program which is then compiled, linked and executed to solve the problem for given boundary and initial conditions.

SPARK is free and has a detailed reference manual. Using SPARK requires a 3 to 5 months learning period to be able to create models. A support service is available and provided by the simulation research group (SRG) of the Lawrence Berkeley National Laboratory (LBNL)

SPARK has it's own HVAC library based on some simple models. However various doctoral researches in the domain of building physics and HVAC systems were accomplished using SPARK as a simulation environment [1], [2], [3], [4], [5] and [6]. This allowed the creation of models library for the building envelope [1], [2] and [3], and a library for the systems [4], [5] and [6]. Recently works [4], [5] were developed on solar desiccant cooling beside an on going PhD thesis on absorption chillers powered by solar collectors. However all these models are dedicated for internal use but can be distributed by their developers.

Important links

http://gundog.lbl.gov/VS/spark.html (download)
http://simulationresearch.lbl.gov/dirpubs/spkall.html (reference manual)

References

[1]. Musy, M. « Génération automatique de modèles zonaux pour l'étude du comportement thermo-aéraulique des bâtiments ». PhD thesis, Université de La Rochelle, France. (1999).

[2]. Mora, L. « Prédiction des performances thermo-aérauliques des bâtiments par association de modèles de différents niveaux de finesse au sein d'un environnement ». PhD thesis, Université La Rochelle, France (2003).

[3]. Cordeiro Mendonça, K. « Modélisation thermo-hydro-aéraulique des locaux climatisés selon l'approche zonale (prise en compte des phénomènes de sorption d'humidité) ». PhD thesis, Université La Rochelle, France (2004).

[4]. Maalouf, C. « Étude du potentiel de rafraîchissement d'un système évaporatif àdésorption avec régénération solaire ». PhD thesis, Université de La Rochelle, France (2006)

[5]. Bourdoukan. P. « Etude numérique et expérimentale destinée à l'exploitation des techniques de rafraîchissement par dessiccation avec régénération par énergie solaire ». Ph.D thesis, Université La Rochelle, (2008) (Available in English)

[6]. Tittelin, P. « Environnements de simulation adaptés à l'étude du comportement énergétique des bâtiments basse consommation ». PhD thesis, Université de Savoie, France (2008)

EnergyPlus

Paul Bourdoukan and Etienne Wurtz
LOCIE FRE CNRS 3220, Institut National de l'Energie Solaire

EnergyPlus has its roots in both the BLAST and DOE–2 programs. BLAST (Building Loads Analysis and System Thermodynamics) and DOE–2 were both developed and released in the late 1970s and early 1980s as energy and load simulation tools. Their intended audience is a design engineer or architect that wishes to size appropriate HVAC equipment, develop retrofit studies for life cycling cost analyses, optimize energy performance, etc. Born out of concerns driven by the energy crisis of the early 1970s and recognition that building energy consumption is a major component of the American energy usage statistics, the two programs attempted to solve the same problem from two slightly different perspectives. Both programs had their merits and shortcomings, their supporters and detractors, and solid user bases both nationally and internationally.
Like its parent programs, EnergyPlus is an energy analysis and thermal load simulation program. Based on a user's description of a building from the perspective of the building's physical make-up, associated mechanical systems, etc., EnergyPlus will calculate the heating and cooling loads necessary to maintain thermal control setpoints, conditions throughout an secondary HVAC system and coil loads, and the energy consumption of primary plant equipment as well as many other simulation details that are necessary to verify that the simulation is performing as the actual building would. Many of the simulation characteristics have been inherited from the legacy programs of BLAST and DOE–2. Below is list of some of the features of the first release of EnergyPlus. While this list is not exhaustive, it is intended to give the reader and idea of the rigor and applicability of EnergyPlus to various simulation situations.

- **Integrated, simultaneous solution** where the building response and the primary and secondary systems are tightly coupled (iteration performed when necessary)
- **Sub-hourly, user-definable time steps** for the interaction between the thermal zones and the environment; variable time steps for interactions between the thermal zones and the HVAC systems (automatically varied to ensure solution stability)
- **ASCII text based weather, input, and output files** that include hourly or sub-hourly environmental conditions, and standard and user definable reports, respectively
- **Heat balance based solution** technique for building thermal loads that allows for simultaneous calculation of radiant and convective effects at both in the interior and exterior surface during each time step
- **Transient heat conduction** through building elements such as walls, roofs, floors, etc.using conduction transfer functions
- **Improved ground heat transfer modeling** through links to three-dimensional finite difference ground models and simplified analytical techniques
- **Combined heat and mass transfer** model that accounts for moisture adsorption/desorption either as a layer-by-layer integration into the conduction transfer functions or as an effective moisture penetration depth model (EMPD)
- **Thermal comfort models** based on activity, inside dry bulb, humidity, etc.
- **Anisotropic sky model** for improved calculation of diffuse solar on tilted surfaces
- **Advanced fenestration calculations** including controllable window blinds, electrochromic glazings, layer-by-layer heat balances that allow proper assignment of solar energy absorbed by window panes, and a performance library for numerous commercially available windows
- **Daylighting controls** including interior illuminance calculations, glare simulation and control, luminaire controls, and the effect of reduced artificial lighting on heating and cooling

- **Loop based configurable HVAC systems** (conventional and radiant) that allow users to model typical systems and slightly modified systems without recompiling the program source code
- **Atmospheric pollution calculations** that predict CO2, SOx, NOx, CO, particulate matter, and hydrocarbon production for both on site and remote energy conversion
- **Links to other popular simulation environments/components** such as WINDOW5, DElight and SPARK to allow more detailed analysis of building components More details on each of these features can be found in the various parts of the EnergyPlus documentation library.

No program is able to handle every simulation situation. However, it is the intent of EnergyPlus to handle as many building and HVAC design options either directly or indirectly through links to other programs in order to calculate thermal loads and/or energy consumption on for a design day or an extended period of time (up to, including, and beyond a year). While the first version of the program contains mainly features that are directly linked to the thermal aspects of buildings, future versions of the program will attempt to address other issues that are important to the built environment: water, electrical systems, etc.

References:

http://www.energyplus.gov/pdfs/gettingstarted.pdf
http://www.eere.energy.gov/buildings/energyplus/cfm/reg_form.cfm (Link for download)

EES

Harald Moser, Graz TU

Graz University of Technology, Institute of Thermal Engineering, Austria

EES is an acronym for Engineering Equation Solver. The basic function provided by EES is the numerical solution of a set of algebraic equations. EES can also be used to solve differential and integral equations, do optimization, provide uncertainty analyses and linear and non-linear regression, convert units and check unit consistency and generate publication-quality plots.

There are two major differences between EES and other equation-solving programs:

1. EES allows equations to be entered in any order with unknown variables placed anywhere in the equations; EES automatically reorders the equations for efficient solution.
2. EES provides many built-in mathematical and thermophysical property functions useful for engineering calculations. For example, the steam tables are implemented such that any thermodynamic property can be obtained from a built-in function call in terms of any two other properties. Similar capability is provided for most refrigerants, ammonia, methane, carbon dioxide and other fluids. Air tables are built-in, as are psychrometric functions and JANAF data for many common gases. Transport properties are also provided for all substances.

The available built-in thermophysical property functions are listed below in alphabetical order. Additional fluid property data can be added by the user.

EES built-in thermophysical property functions

ACENTRICFACTOR	CONDUCTIVITY	CP
CV	DENSITY	DEWPOINT
DIPOLE	ENTHALPY	ENTHALPY_FUSION
ENTROPY	FUGACITY	HUMRAT (humidity ratio)
INTENERGY	ISIDEALGAS	MOLARMASS
PRANDTL	PRESSURE	P_CRIT
P_SAT	QUALITY	RELHUM
SOUNDSPEED	SPECHEAT	SURFACETENSION
TEMPERATURE	T_CRIT	T_SAT
T_TRIPLE	VISCOSITY	WETBULB
VOLUME	V_CRIT	VOLEXPCOEF

The built-in thermophysical property data for the fluids listed below are provided by EES.

Fluids with EES built-in thermophysical properties

Ideal Gas	Real Fluid			
Air	Air_ha*	Methanol*	R11	R404A
AirH2O	Ammonia*	n-Butane*	R12	R407C
CH4	Argon*	n-Dodecane*	R13	R410A
C2H2	Carbondioxide*	n-Heptane*	R14	R500
C2H4	Carbonmonoxide	n-Hexane*	R22*	R502
C2H6	Cyclohexane*	n-Octane*	R23*	R507A
C2H5OH	Deuterium*	n-Pentane*	R32*	R508B
C3H8	Ethane*	Neon*	R114	R600*
C4H10	Ethanol*	Nitrogen*	R116*	R600a
C5H12	Ethylene*	Oxygen*	R123*	R717*
C6H14	Fluorine*	Propane*	R124*	R718*
C8H18	Helium*	Propylene*	R125*	R744*
CO	Hydrogen*	Steam*	R134a*	RC318*
CO2	HydrogenSulfide	Steam_IAPWS*	R141b*	
H2	Ice*	Steam_NBS*	R143a*	
H2O	Isobutane*	SulfurHexafluorid	R152a*	
N2	Isopentane*	Toluene*	R227ea	
NO	Krypton*	Water*	R290*	
NO2	Methane*	Xenon*		
O2				
SO2				

* These fluids are implemented using a high accuracy equation of state.

Further more the thermodynamic properties of mixtures of lithium bromide / water and ammonia / water are implemented as an externally compiled EES routine.

EES allows the user to enter his or her own functional relationships in three ways. First, a facility for entering and interpolating tabular data is provided so that tabular data can be directly used in the solution of the equation set. Second, the EES language supports user-written functions, procedures, modules and subprograms. Third, compiled functions and procedures, written in a high-level language such as Pascal, C, or FORTRAN, can be dynamically-linked with EES.

EES is particularly useful for design problems in which the effects of one or more parameters need to be determined. The program provides this capability with its Parametric Table, which is similar to a spreadsheet. The user identifies the variables which are independent by entering their values in the table cells. EES will calculate the values of the dependent variables in the table. Plotting capability is provided to display the relationship between any two variables in the table.

EES offers the advantages of a simple set of intuitive commands which a novice can quickly learn to use for solving complex problems. The large data bank of thermodynamic and transport properties built into EES is helpful in solving problems in thermodynamics, fluid mechanics, and heat transfer.

TRNSYS

Alexander Morgenstern

Fraunhofer Institute for Solar Energy Systems ISE, Freiburg, Germany

TRNSYS is a commercially available energy simulation tool designed for the transient simulation of thermal systems and multizone buildings.

The software is generally used by engineers and researchers for the validation of new energy concepts for simple applications like solar domestic hot water systems as well as complex systems for testing of control strategies like buildings with solar heating and cooling equipment (e.g. thermally activated slabs, capillary tube systems and chilled ceilings) with consideration of the occupant behavior. An extensive standard component library and a large number of AddOns like geothermal and heat pump libraries or air flow models allows the use for a wide field of application. The open, modular structure of TRNSYS afford the extension and the design of models in dependence of the user requirements. This provides a basis for the coupled simulation of the building and the active system.

TRNSYS offer dynamic building simulation with an arbitrary number of thermal zones, different walls and windows, also buffer zones like double facades, furthermore the consideration of convective and radiative internal loads, the influence of shading devices as well as natural and mechanical ventilation and interzonal air change. A special building editor is available to configure the building type of TRNSYS.

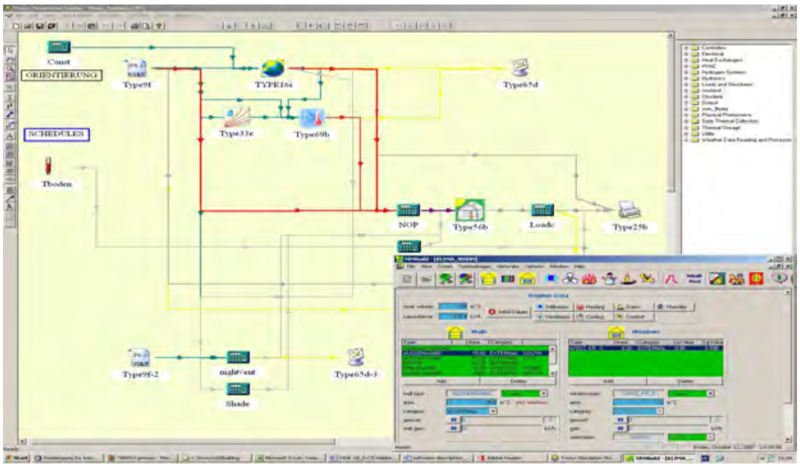
The support for the integration of control strategies for shading, ventilation, heating and air-conditioning as well as the import of weather data is implemented in the software. Time steps can be defined by user.

All common programming languages (C++, PASCAL, FORTRAN, …) can be used by developers to add custom component models.

TRNSYS allows e.g. simulations for the following applications:

- Solar systems (solar hot water, solar combi systems and electric systems with PV)
- Low energy buildings and HVAC systems
- Renewable energy systems
- Cogeneration and fuel cells

More information: http://sel.me.wisc.edu/trnsys/ ; www.trnsys.de



Example: graphical display of a building load simulation with TRNSYS. The building configuration editor is shown in the front.

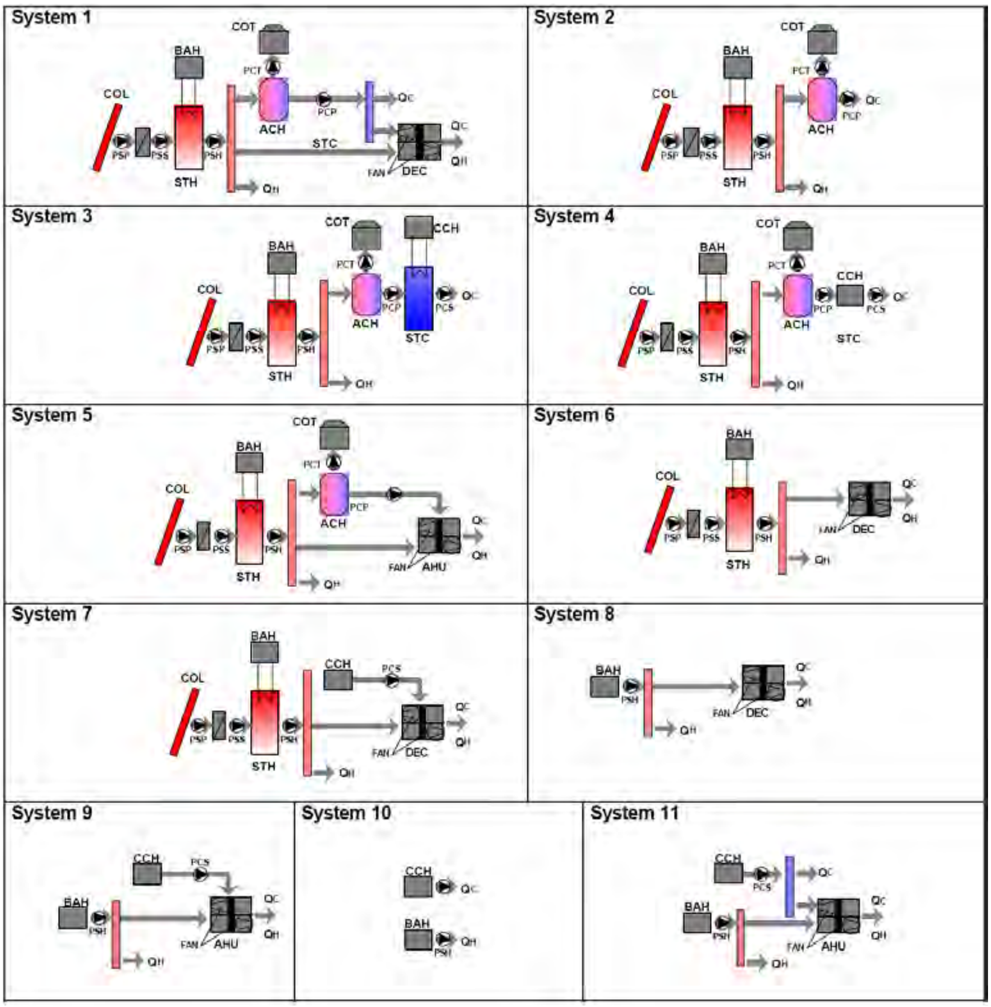
EASYCOOL

Edo Wiemken,
Fraunhofer Institute for Solar Energy Systems ISE, Freiburg, Germany

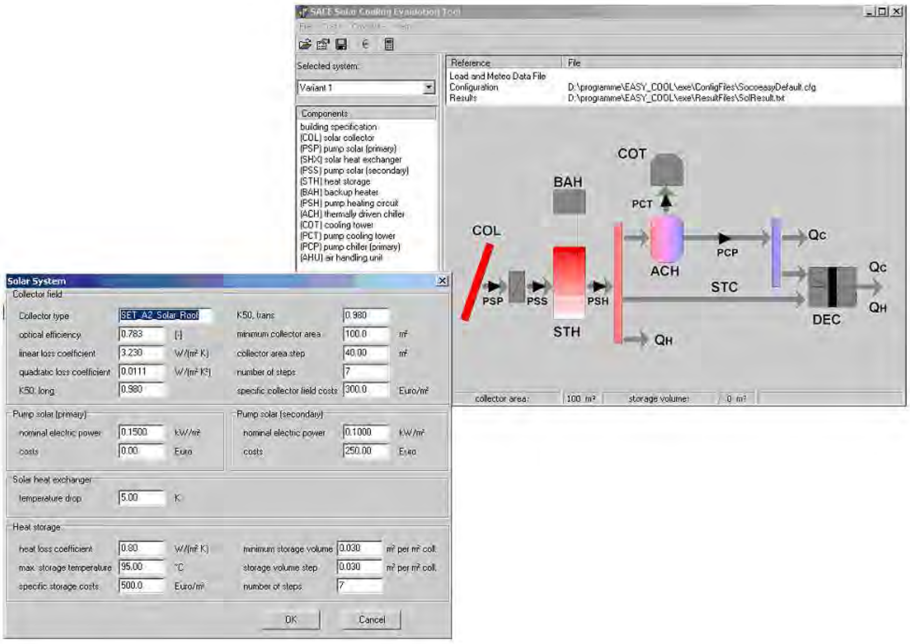
The simulation tool was developed in the frame of the project Solar Air-Conditioning in Europe (SACE), funded by the European Commission. It was originally designed to perform easy to programme applications and fast simulation runs for energetic and economic performance studies in SACE and in IEA Task 25.

EasyCool provides 11 pre-defined system configurations for solar thermally driven cooling applications, of which 4 configurations are foreseen for reference calculatios (non-solar, conventional system solutions). The program reads annual time series with hourly building load data and meteorological data of the respective site (these data set has to be prepared in advance) and calculates annual energetic and econonomic performance data as well as environmental figures such as CO_2 savings. With each simulation run, a range of system size (collector area and hot water storage size) is varied.

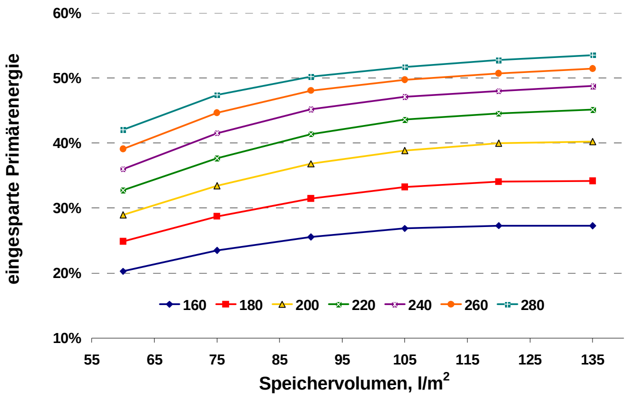
In input boxes, the most relevant average performance coefficients, power consumptions (pums, fans, ..) collector coeffictients and component costs are to be defined. A special economic input box requires cost data for installation, planning, maintenance, interest rate, fuel, water and electricity costs, etc.

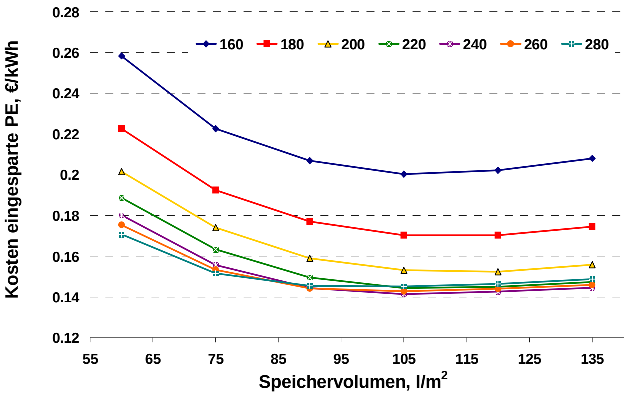


Pre-defined system configurations, accessible in EasyCool. System configurations 8 to 11 are for reference system calculations.



One of the pre-defined system configurations is selected and the input box for the solar thermal collector description is opened.





Example of simulation results: For a defined system configuration and load data file, the savings in primary energy consumption (compared to a reference system calculation) and specific cost per saved kWh of primary energy are displayed as a function of the system size (storage volume is shown at the x.-axis, the collector area is represented by the different curves)

EasyCool is used internally by Fraunhofer ISE and currently not disseminated for public use.

INSEL

Antoine Dalibard
ZAFH.NET (Stuttgart University of Applied Sciences)

The acronym INSEL stands for INtegrated Simulation Environment Language. This graphical programming language has been developed at the Faculty of Physics of Oldenburg University (Germany) in the early 1990's and was originally designed for the modelling of renewable electrical energy systems. Today INSEL covers the whole range of renewable energy systems, including building simulation and communication technologies. The software can be adapted by the user for completely own software developments and offers a transparent software solution from the planning phase to automation and simulation based control of any energy plants.

Basic idea of INSEL

The graphical programming language INSEL is based on the principle of "structured programming" on blocks diagrams. It consists of connecting blocks in order to obtain block diagrams that express a solution for a certain simulation task.

Application areas of INSEL

The main application areas of the program INSEL are:

- Energy meteorology
- Photovoltaic systems
- Solar thermal systems
- Solar thermal power plants
- Building simulation (under development)
- Building automation (under development)
- Facility management (under development)

Main uses of INSEL

INSEL is a block diagram simulator that can be used by researchers, planers, designers, operators and investors for:

- Design of complex energy systems
- Visualisation and Internet monitoring
- Monitoring and control of energy plants
- Highly precise yield prognoses and economic calculations of energy systems.

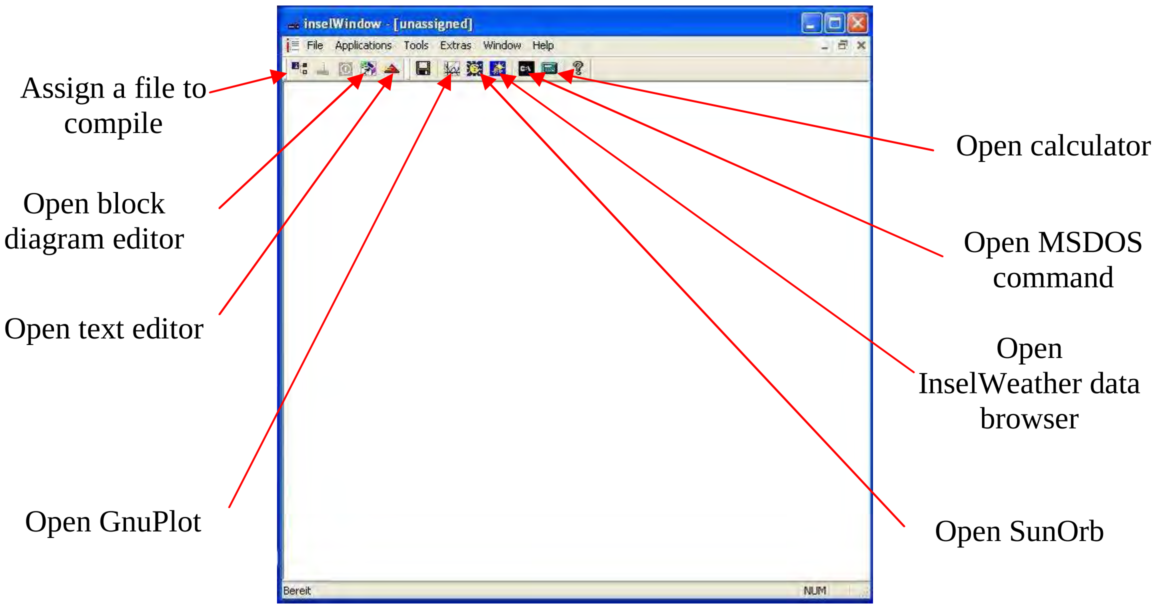
Structure of the program

The different layers of INSEL are:

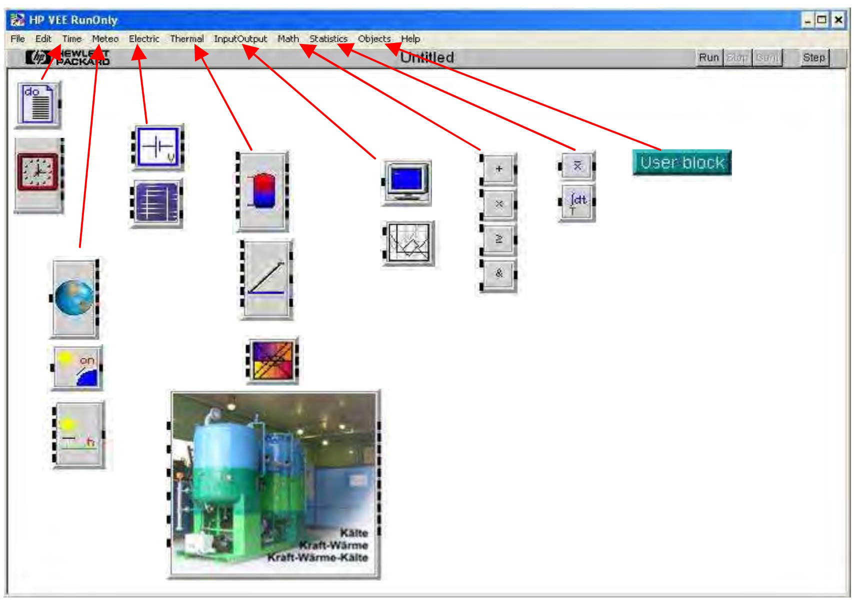
Graphical Interface : HP-VEE panel view	
Graphical simulation model (file .vee)	**Text simulation model (file .ins)**
Data bases (weather, components,…)	
INSEL compiler: inselEngine.dll	
Execution: inselDi.dll	
Interfaces: call of the conventions, header	
Blocks implementation	
INSEL block concept	

Additional Information:

- *INSEL main window*:



- *Graphical Interface*: HP-VEE is used as graphical interface



- *Source code of the models:* not available

- *Modularity*: since INSEL is a programming language, it is entirely modular. It is possible to define any energetic systems.
- *Time steps increment*: any time steps can be defined from less than 1sec to several years.
- *Components library extension*: It is possible to write source code in Fortran and C++ and to create new components in order to extend the components library by linking the code to a dynamic link library DLL.
- *Meteorological databases used*: INSEL database (2000 locations worldwide), European solar radiation Atlas, Meteonorm, S@tel light plus any other source available.

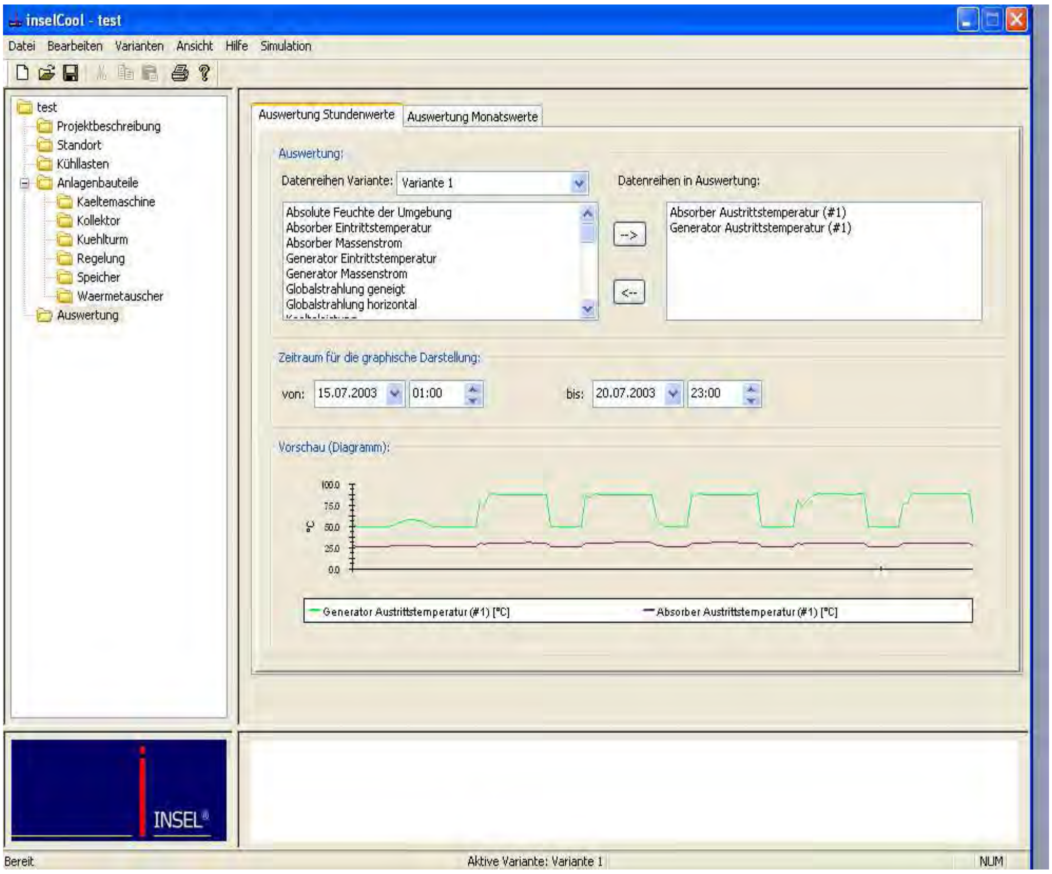
Components library

The main components of INSEL related to solar thermal application and solar cooling in particular are:

- *Collectors:* dynamic models of air and liquids collectors
- *Storage tanks:* fully mixed and stratified models
- *Heat exchangers:* simple models (parallel, counter and cross flows), cross flow heat exchanger with condensing and icing, earth heat exchanger (under development).
- *Desiccant wheel*
- *Absorption chillers:* models for Phönix (10 kW) and EAW (from 15 to 200 kW) chillers.
- *Adsorption chillers:* dynamic model under development.
- *Cooling towers*: open-wet model

A specific application for solar cooling: INSEL Cool

An application for solar cooling (INSEL Cool) has been implemented into INSEL in order to help planners during the pre-design phase of a solar cooling plant. Users have the possibility to perform simulation of solar cooling installing by selecting in a list the main components of the installation (cooling machines, solar collectors, cooling towers, storage tanks, heat exchangers), the location of the installation and the cooling load profile. It is also possible to define new components and new locations by introducing himself the required parameters and external cooling load files can be used for simulation. The analysis of the results can be done (either with hourly or monthly values) directly from the application as can be seen in the figure below.



Reference projects

INSEL has been used in several solar cooling projects, some examples are given below:

- 55 kW solar desiccant cooling system with 244 m² ventilated PV-Façade, 330 m² ventilated PV-Roof sheds and 155 m² of solar air collectors in the Library of Mátaro (Spain).
- 100 kW solar desiccant cooling system with 100 ² of solar air collectors and additional thermal energy delivered by a heat recovery system of the production process, installed in an production hall in Althengstett (Germany)
- 15 kW solar absorption cooling system with 37 m² flat plate and 29 m² vacuum tube collectors, 2 m³ hot and 1 m³ cold storage tank, installed in an office building of the SolarNext AG (Germany)
- Simulation based analysis of different control options for a 1.218 m² solar collector field which supplies the heating system of an office building in winter and three MYCOM adsorption chillers with a nominal cooling capacity of 353 kW, FESTO AG & C0. KG in Esslingen
- Performance and economic analysis of a 140 kW absorption cooling system for a production hall with 2.500 m² conditioned space of the ft-fertigungstechnik GmbH in Viernheim, Germany

Download

A demonstration version of INSEL and a detailed tutorial can be downloaded from the following website: www.insel.eu

Chapter II: New developments in simulations tools and models

A solar desiccant cooling installation model in SPARK,
A transient detailed model of the desiccant wheel.

Paul Bourdoukan
LOCIE FRE CNRS 3220, Institut National de l'Energie Solaire (INES),

Etienne Wurtz
LOCIE FRE CNRS 3220, Institut National de l'Energie Solaire (INES),

The present chapter presents a modelling approach of a solid desiccant air handling unit powered by vacuum tube solar collectors. It is divided into 3 sections.
In section (1) the model of the installation is described by separately considering each component (i.e. sensible heat regenerator, desiccant wheel, evaporative coolers, solar collectors, storage tank).
In the section (2) an experimental validation of the presented model is performed under different operating conditions on a component level and on a system level.
Section (3) is dedicated for a transient model of the desiccant wheel. Unlike conventional models that give a mean temperature and a mean humidity ratio at the outlet of the wheel this detailed transient model gives the temperature and humidity profile at the outlet of the wheel. The model is then validated experimentally.
The presented models with the experimental validation are a summary of different research work [1], [2], [3], [4] and [5] published recently and highlighted in the reference section.

1. Modelling of the solar desiccant cooling installation

A model implementation of a desiccant cooling installation powered by vacuum tube solar collectors in SPARK is described in this section. The Desiccant wheel, the sensible regenerator and the evaporative cooler are selected from bibliography. **However a new developed model of the vacuum heat pipe solar collectors is presented.**

The aim of this implementation is to predict accurately the supply conditions of a desiccant air handling for different operating conditions and to predict the real potential of solar energy in this process.

Nomenclature

Symbols

c_{pa}	heat capacity of air [$J.kg^{-1}.K^{-1}$]
c_{pv}	heat capacity of water vapor [$J.kg^{-1}.K^{-1}$]
c_{pm}	heat capacity of the regenerator matrix [$J.kg^{-1}.K^{-1}$]
C_{min}	min capacitance rate of the regenerator inlet fluids [$J.s^{-1}.K^{-1}$]
C_r	parameter used in the correlation of the regenerator [-]
e	thickness [m]
F_i	potential characteristic [-]
G	solar global radiation [$W. m^{-2}$]
h	heat transfer coefficient [$W.K^{-1}m^{-2}$]
h_a	enthalpy of moist air [$J.Kg^{-1}$]
h_{fg}	latent heat of vaporization [J.kg]
H	enthalpy of the sensible or desiccant regenerator matrix [$J.Kg^{-1}$]
J_t	lumped heat transfer coefficient [s^{-1}]
J_m	lumped mass transfer coefficient [s^{-1}]
K	heat loss constant [$W.K^{-1}m^{-2}$]
L	width of the desiccant wheel [m]
L_v	latent heat of vaporization of the heat pipe fluid [$J.kg^{-1}$]
Le	Lewis number [-]
m_a	air mass flow rate [$Kg.s^{-1}$]
M	mass [kg]
M_d	mass of the desiccant matrix [Kg]
M_m	mass of the aluminium matrix [Kg]
N	angular speed of the wheel [$rd.s^{-1}$]
Na	number of nodes in the angular direction of the desiccant wheel [-]
Np	number of nodes in the width direction of the desiccant wheel [-]

NTU	number of transfer unit [-]
S	area [m^2]
t	time [s]
T_a	air temperature [K]
T_{eq}	equilibrium temperature [K]
T_i	air temperature in the section i (i=1;9) [K]
T_m	matrix temperature [K]
u	fluid velocity ($m.s^{-1}$)
V	volume (m^3)
w_a	humidity ratio of moist air. [$Kg.Kg^{-1}_{dryair}$]
W_d	water content of desiccant [$Kg.Kg^{-1}_{desiccant}$]
x	variable [arbitrary]
z	coordinate in the fluid flow direction [m]

Greek letters

ε	emissivity [dimensionless]
γ_i	parameter used in the reduction of desiccant wheel equations [-]
$\Delta\lambda$	parameter in the desiccant enthalpy [-]
μ	ratio of matrix mass over air mass [-]
η_{cf}	efficiency of the counter flow heat exchanger [-]
$\tau\alpha$	transmission-absorptance coefficient
σ	Stefan Boltzmann constant [$W.K^{-4}m^{-2}$]
ρ	density [$kg.m^{-3}$]
λ	conductivity [$W.K^{-1}m^{-1}$]
τ_{ro}	half period of rotation of the desiccant wheel [s]
θ	cylindrical coordinate or angular position [rd]

Subscripts

a	air
b	buffer
c	condenser
d	desiccant
eq	equilibrium
f	fluid
g	glass
i	inlet
H	heat pipe

m	matrix (sensible regenerator)
p	plate absorber
o	outlet
reg	regeneration
sat	saturation

Desiccant cooling principle

A desiccant cooling installation operating under the conventional configuration (100% air-change rate), with corresponding changes in the air properties in the psychometric chart, is shown in Figure 1.

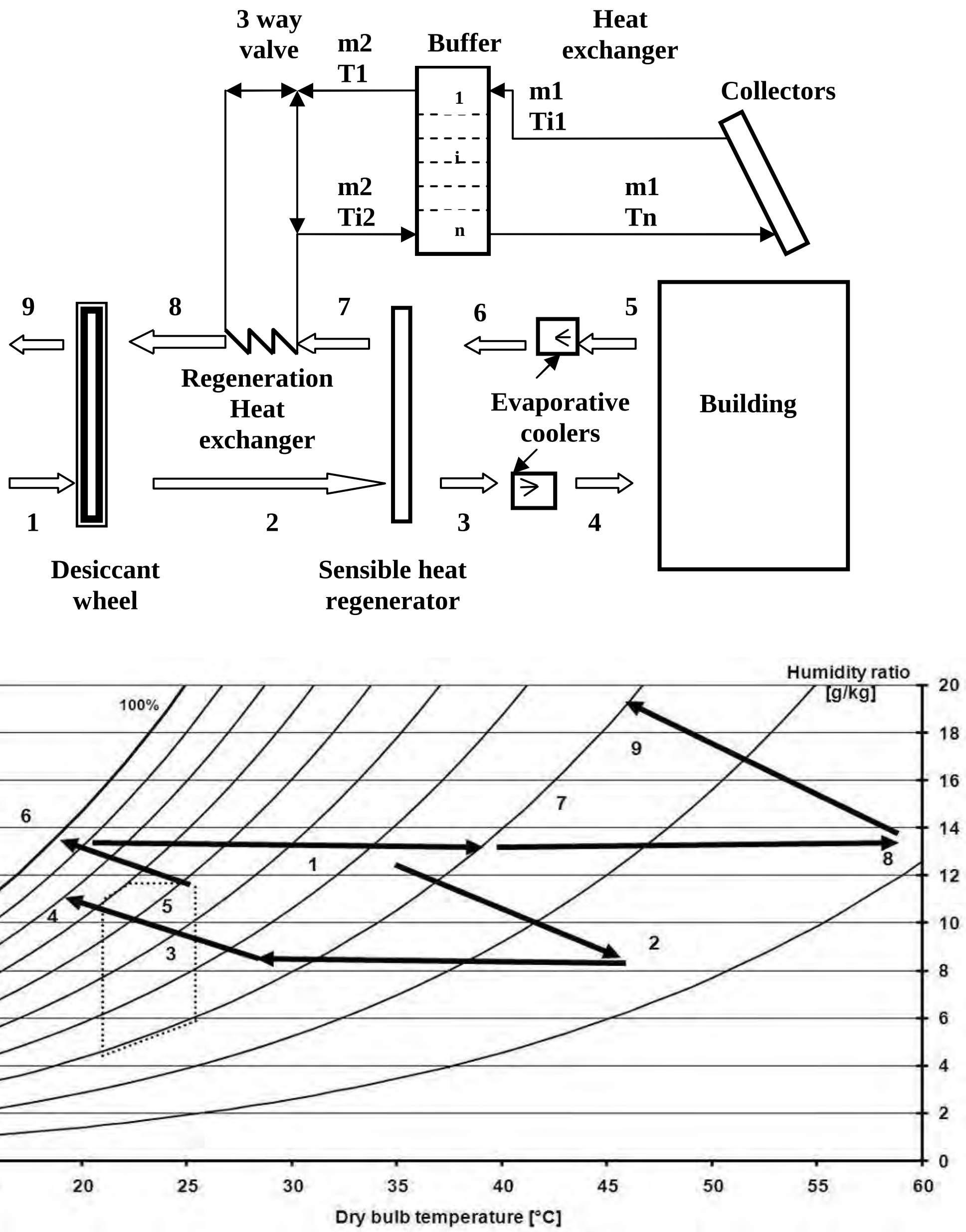


Figure 1: Desiccant cooling system with corresponding evolution of air properties in the psychometric chart

With reference to Figure 1, the conventional cycle operates as follows: outside air (1) is dehumidified in a desiccant wheel (2); it is then cooled in the sensible regenerator (3) by the return cooled air before undergoing another cooling stage by evaporation (4) and being introduced into the building. The operating sequence for the return air (5) is as follows: it is cooled to its saturation temperature by evaporative cooling (6) and then it cools the fresh air in the rotary heat exchanger (7). It is then heated in the regeneration heat exchanger by solar energy (8), it regenerates the desiccant wheel (9) by removing the humidity and exits the installation.

Modelling of the components

Sensible regenerator

A sensible regenerator is a porous matrix which passes periodically between a hot and a cold stream. The cellular matrix of the regenerator stores heat from the hot gas stream and releases it to the cold gas stream.

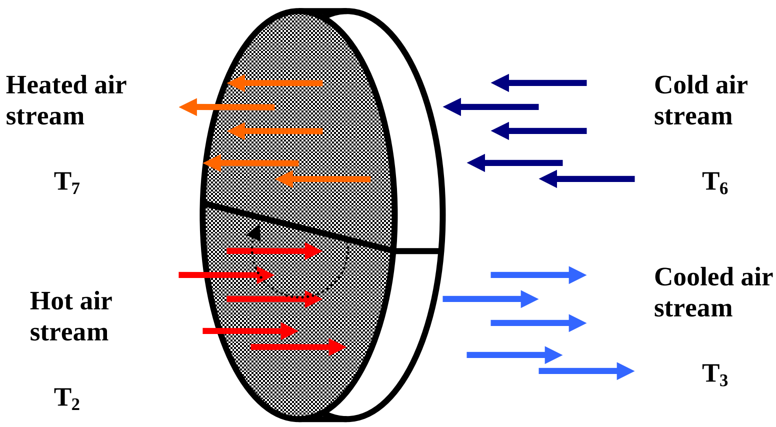


Figure 2: Sensible heat regenerator principle

For modelling purposes, the following assumptions are made:

- Heat transfer between air and the regenerator matrix is considered using a lumped transfer coefficient or a number of transfer units *(NTU)*
- The channels where the fluid flows are identical and parallel
- No leakage occurs between the air streams

The heat conservation and transfer governing equations, after Kays and London [6] and Maclaine-cross and Banks [7], are:

$$\frac{\partial Ta}{\partial t} + u\frac{\partial Ta}{\partial z} + \mu\frac{c_{pm}}{c_{pa} + w_a c_{pv}}\frac{\partial Tm}{\partial t} = 0 \tag{1}$$

$$\mu\frac{c_{pm}}{c_{pa} + w_a c_{pv}}\frac{\partial Tm}{\partial t} + J_t(T_{a,m} - T_a) = 0 \tag{2}$$

Kays and London [6] proposed the following correlation for regenerator efficiency (RE):

$$RE = \eta_{cf}\left(1 - \frac{1}{9\left(C_r^*\right)^{1.93}}\right) \tag{3}$$

Where:

$$C_r^* = \frac{M_m.c_{pm}.N}{C_{\min}} \tag{4}$$

$$\eta_{cf} = \frac{NTU_{cf}}{1 + NTU_{cf}} \tag{5}$$

is the efficiency of the counter-flow heat exchanger for balanced flow.

The outlet temperature (T_3) of the sensible regenerator can thus be calculated using:

$$RE = \frac{T_2 - T_3}{T_2 - T_6} \tag{6}$$

<u>Desiccant wheel</u>

The wheel is divided into two sectors the first is for dehumidification of moist air while the second is for the regeneration. It is a rotating porous matrix impregnated with a desiccant material that alternates periodically between a process air stream and a hot air stream. In contact with the dry desiccant, process air is dehumidified and heated by the heat of adsorption. The saturated desiccant then enters into contact with the hot air stream and is regenerated to again dehumidify the process air, perpetuating the dehumidification-regeneration cycle.

The wheel is driven by a small electrical motor, with a rotation speed that varies between 7 and 20 rph

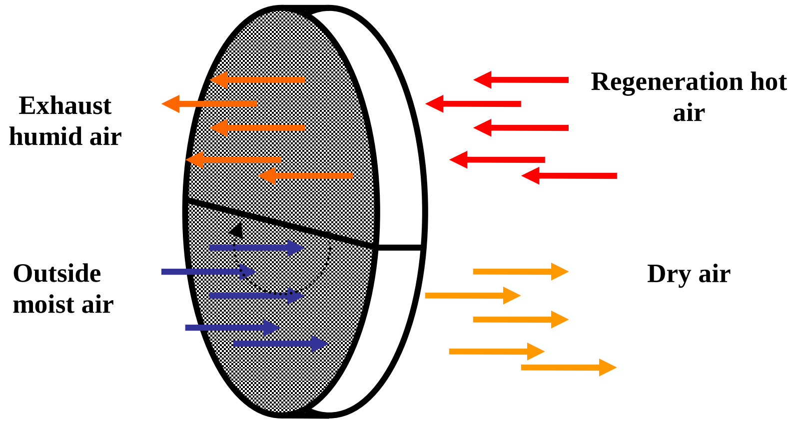


Figure 3: Desiccant wheel

The desiccant wheel model used below is that proposed first by Banks [8] and also by Maclaine-cross and Banks [7]. The following assumptions are made [3]:

- The state properties of the air streams are spatially uniform at the desiccant wheel inlet
- The interstices of the porous medium are straight and parallel
- There is no leakage or carry-over of streams
- The interstitial air velocity and pressure are constant
- Heat and mass transfer between the air and the porous desiccant matrix is considered using lumped transfer coefficients
- Diffusion and dispersion in the fluid flow direction are neglected
- No radial variation of the fluid or matrix states
- The sorption isotherm does not represent a hysteresis
- Air reaches equilibrium with the porous medium

Heat and mass conservation equations:

$$\frac{\partial h_a}{\partial t} + u\frac{\partial h_a}{\partial z} + \mu\frac{\partial H_d}{\partial t} = 0 \quad (7)$$

$$\frac{\partial w_a}{\partial t} + u\frac{\partial w_a}{\partial z} + \mu\frac{\partial W_d}{\partial t} = 0 \quad (8)$$

Heat and mass transfer equations:

$$\mu\frac{\partial H_d}{\partial t} + J_m(Le(T_{a,eq} - T_a)\left.\frac{\partial h_a}{\partial T_a}\right|_w) + (w_{a,eq} - w_a)\left.\frac{\partial h_a}{\partial w_a}\right|_T = 0 \quad (9)$$

$$\mu\frac{\partial W_d}{\partial t} + J_m(w_{a,eq} - w_a) = 0 \quad (10)$$

Equations (7), (8), (9) and (10) are coupled, hyperbolic and non-linear. With the assumption of the Lewis number *(Le),* equal unity and the desiccant matrix in equilibrium with air means that (T_d= T_{eq} and w_{eq}= w_a). Banks [8] used matrix algebra and demonstrated that these equations (7 to 10) can be reduced by applying the potential function *Fi(T,w)* to the following system:

$$\gamma_i \mu \frac{\partial F_{i,eq}}{\partial t} + u \frac{\partial F_{i,a}}{\partial t} + \frac{\partial F_{i,a}}{\partial z} = 0 \quad \text{i=1;2} \tag{11}$$

$$\gamma_i \mu \frac{\partial F_{i,d}}{\partial t} + J_m (F_{i,eq} - F_{i,a}) = 0 \quad \text{i=1;2} \tag{12}$$

These equations are similar to those for the sensible regenerator (Equations 1 and 2), with the potential function F_i replacing the temperature and the parameters γ_i replacing the specific heat ratio, and they can be solved using analogy with heat transfer alone as suggested by Maclaine-cross and Banks [7] and Close and Banks [9]. There are many expressions for the potential functions of moist air-silica gel; we chose those proposed by Jurinak [10] and Stabat [11].

$$F_1 = h \tag{13}$$

$$F_2 = \frac{(273.15 + T)^{1.5}}{6360} + 1.1 w^{0.8} \tag{14}$$

The solution sequence for the desiccant wheel is then:

$$C_j^i = \dot{m}_a^{\,j}$$

$$C_{ri}^* = \frac{M_d N}{\dot{m}_a^{\min}} \gamma_j^{i(moy)}$$

$$C_i^* = \frac{C_{\min}^i}{C_{\max}^i}$$

$$NUT_0 = \frac{1}{C_{\min}} \left(\frac{1}{(h_m A)_s} + \frac{1}{(h_m A)_r} \right)^{-1}$$

$\varepsilon_i = \varepsilon_{cc}\left(NUT_0^m, C_i\right)\left[1 - \frac{1}{9\left(C_{ri}^*\right)^{1.93}}\right]$ or $\varepsilon_i = C_{ri}^*$ ($C_{ri}^* \leq 0.4$) for very low rotation speed

For the desiccant wheel the efficiency is written with the potential functions instead of temperatures for the sensible regenerator:
(With $F_{i,k}$ where the subscript i indicates the potential and the subscript k indicates the position relatively to figure 1)

$$\varepsilon_1 = \frac{C_n^1}{C_{\min}^1}\frac{F_{1,2} - F_{1,1}}{F_{1,8} - F_{1,1}}$$

$$\varepsilon_2 = \frac{C_n^2}{C_{\min}^2}\frac{F_{2,2} - F_{2,1}}{F_{2,8} - F_{2,1}}$$

To calculate the outlet conditions of the regeneration stream the conservation equations are applied to the potential functions:

$$C_n^1\left(F_{1,2} - F_{1,1}\right) = C_r^1\left(F_{1,9} - F_{1,8}\right)$$

$$C_n^2\left(F_{2,2} - F_{2,1}\right) = C_r^2\left(F_{2,9} - F_{2,8}\right)$$

Having the potential functions calculated the temperature and humidity ratio at both outlet of the wheel can thus be calculated

<u>Evaporative coolers</u>

The humidification process occurs at a constant wet-bulb temperature. During the process, the energy needed for droplet evaporation is taken from the air which yields its temperature decrease and humidity increase. The enthalpy of the moist air at the evaporative cooler inlet is the sum of the enthalpy of the dry air and that of the water vapor. At the evaporator cooler outlet; the enthalpy of the dry air decreases due to the temperature decrease while that of water vapor increases due to the increased water vapor mass but the total enthalpy of moist air is approximately constant, with a slight increase due to the addition of the enthalpy of liquid water droplets.

$$h(T, w)_{a,inlet} = h(T, w)_{a,outlet} \quad (15)$$

The validity of this assumption can be verified on the psychometric chart where the lines of constant wet-bulb temperature are parallel to the constant enthalpy lines, especially when the temperature and humidity ratio are, respectively, between 19 and 25°C and 7 and 15g/Kg, which is in the application domain of desiccant evaporative cooling.

Heat pipe vacuum tube solar collectors

In an evacuated heat pipe collector (figure 4), a sealed copper pipe containing a vaporizable fluid is bonded to a copper fin plate absorber located inside a glass tube. A small copper condenser is attached from one side to the top of the heat pipe and from the other side to the storage working fluid. The heat pipe is an evaporating-condensing device. As the sun shines on the absorber, the pipe is heated and some of the liquid inside evaporates. The vapour rises toward the condenser at the top of the heat pipe and condenses on being cooled by the storage water circulating in the manifold. The liquid then returns to the heat pipe. The vacuum tube ensures minimization of the heat losses of the collector.

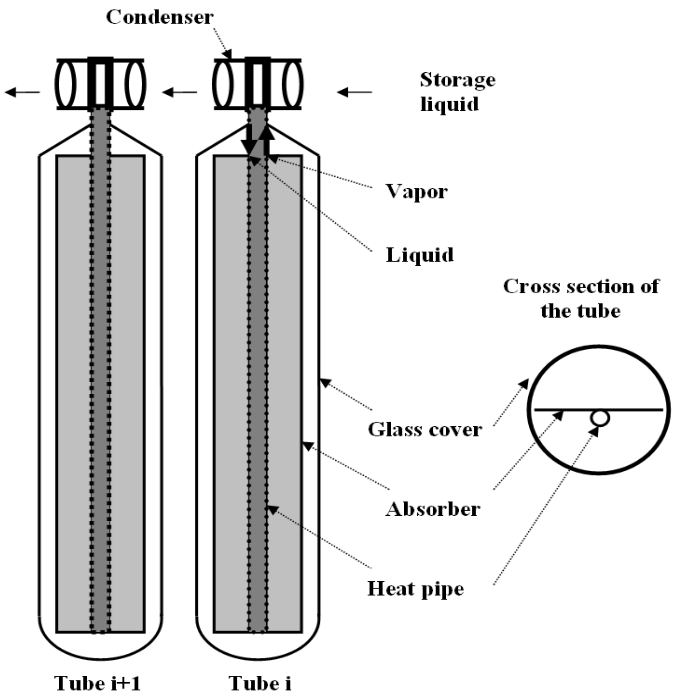


Figure 4: Heat pipe vacuum tube collectors

A model for the HPVT is proposed below, separately considering each component [12] and [13] of the tube, i.e. the glass cover, the absorber, the heat pipe fluid, the condenser and the storage fluid. The following assumptions are made regarding the model:

- The properties of the materials are independent of the temperature.
- The temperature gradient along the absorber and the condenser is negligible.
- Due to the sufficient quantity of fluid in the heat pipe (more than 90% of the heat pipe volume), the vapour is not superheated
- The liquid returning from the condenser to the heat pipe is saturated.
- The glass cover is clean and completely transparent to solar radiation.
- The part of radiation reflected by the absorber leaves the tube.
- Due to the vacuum, convection does not occur inside the tube.
- Conduction in the storage fluid direction (z) is considered to be negligible.

The glass cover exchanges heat by convection with the outside air and by radiation with the sky and the absorber. The absorber receives solar radiation and heats the fluid inside the tube. The vapour rising from the heat pipe enters the condenser and releases energy to the circulating storage fluid, then exits the condenser as a saturated liquid. The heat pipe liquid is heated to saturation before the evaporation process begins.
The governing equations for each component (characterized by heat capacity *C* and temperature *T*):

Glass cover:

$$M_g C_g \frac{dT_g}{dt} = \varepsilon_g \sigma S_g (T_{sky}^4 - T_g^4) + \varepsilon_p \sigma S_p (T_p^4 - T_g^4) + S_g h_g (T_a - T_g) \qquad (16)$$

Absorber:

$$M_p C_p \frac{dT_p}{dt} = \varepsilon_p \sigma S_p (T_g^4 - T_p^4) + G \tau \alpha S_p + S_H h_H (T_{sat} - T_p) \qquad (17)$$

Vapour flow rate:

$$m_v L = S_H h_H (T_p - T_{sat}) \qquad (18)$$

Condenser:

$$M_c C_c (\frac{dT_c}{dt}) = m_v L_v + S_c h_c (T_f - T_c) \qquad (19)$$

Storage fluid:

$$m_1 C_f (\frac{\partial T_f}{\partial t} + u \frac{\partial T_f}{\partial z}) = S_c h_c (T_c - T_f) \qquad (20)$$

<u>Storage tank</u>

In order to take into account thermal stratification in the storage tank, it is divided into *n* nodes [14]. The following equations apply to the first node, the last node (*n=12*) and the *ith* current node:

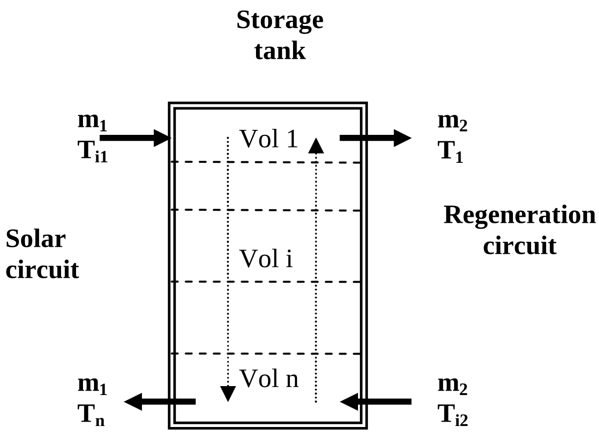


Figure5: Storage tank divided to n nodes

$$\rho_1 V_1 C_f \frac{dT_1}{dt} = m_1 C_f (T_{i1} - T_1) + \frac{\lambda S_i}{e_{12}} \left(T_2 - T_1\right) + m_2 C_f (T_2 - T_1) + S_1 K_1 (T_a - T_1) \tag{21}$$

$$\rho_n V_n C_f \frac{dT_n}{dt} = m_1 C_f (T_{n-1} - T_n) + \frac{\lambda S_i}{e_{n-1,n}} \left(T_{n-1} - T_n\right) + m_2 C_f (T_{i2} - T_n) + S_n K_n (T_a - T_n) \tag{22}$$

$$\rho_i V_i C_f \frac{dT_i}{dt} = m_1 C_f (T_{i-1} - T_i) + \frac{\lambda S_i}{e_i} \left(T_{i-1} - T_i\right) + m_2 C_f (T_{i+1} - T_i) + S_i K_i (T_a - T_i) - \frac{\lambda S_i}{e_i} \left(T_i - T_{i+1}\right) \tag{23}$$

These sets of equations describing the above-presented models are introduced into SPARK [15] - a general simulation environment that supports the definition of simulation models, providing the solution of these models via a robust and efficient differential/algebraic equation solver [16]. In SPARK, the modeler describes the set of equations defining a model as an equation-based object. At the lowest level, an atomic object characterizes, in SPARK language, a single equation and its variables. Macroscopic objects can then be created as an assembly of various atomic or macroscopic objects. The entire model is built by connecting the various required objects. One should note that the model is input/output-free. The particular problem to be solved is then described by imposing the adequate input data (boundary and initial conditions) and by specifying the variables to be solved. So in this environment it is not necessary to order the equations or to express them as assignment statements (algorithms), unlike conventional modular environments.

In next section the above presented model is experimentally validated.

2. Experimental validation

The experimental desiccant installation at La Rochelle [1] was used to validate the model presented above. The desiccant air handling unit consists of silica gel desiccant wheel, an aluminum sensible regenerator, two rotating humidifiers, high performance ventilators, a regeneration heat exchanger (water to air) and an electrical backup used for experimental purpose. The air handling unit is powered by 40 m^2 of heat pipe vacuum tube collectors and storage tank of 2350 liters.

Psychrometers are used to measure the moist air properties at each section of the air handling unit (position 1 to 9). Each psychrometer consists of 2 Pt100 sensors: the first delivers the dry bulb temperature and the second the wet bulb temperature. The local atmospheric pressure is measured too thus the humidity ratio and relative humidity can be accurately measured. The temperature of hydraulic system is measured by the means of Pt100 sensors integrated to the pipes by means of thermo wells. All the Pt100 sensors were tuned on a temperature range going from 0° to 100℃ . The water flow rates are measured using ultrasonic sensor. The air flow rates were measured using the gas tracer method.In the difficulty to tune a pyranometer the solar global radiation is measured using two different pyranometers.

Figure 1 shows a general scheme of the experimental installation. In order to test different outside operating conditions (to test the models under different conditions) an outside conditions simulator was coupled to the desiccant air handling unit. The outside conditions simulator consists of a pre-heater, a humidifier and a heater, this permits the test at high temperatures and high humidity ratio.

The cooling load on the installation was controlled using load simulator that consists of heater and replaces the building load. This permitted to control the return conditions during the tests.

For the model validation different operating were considered for all the components.

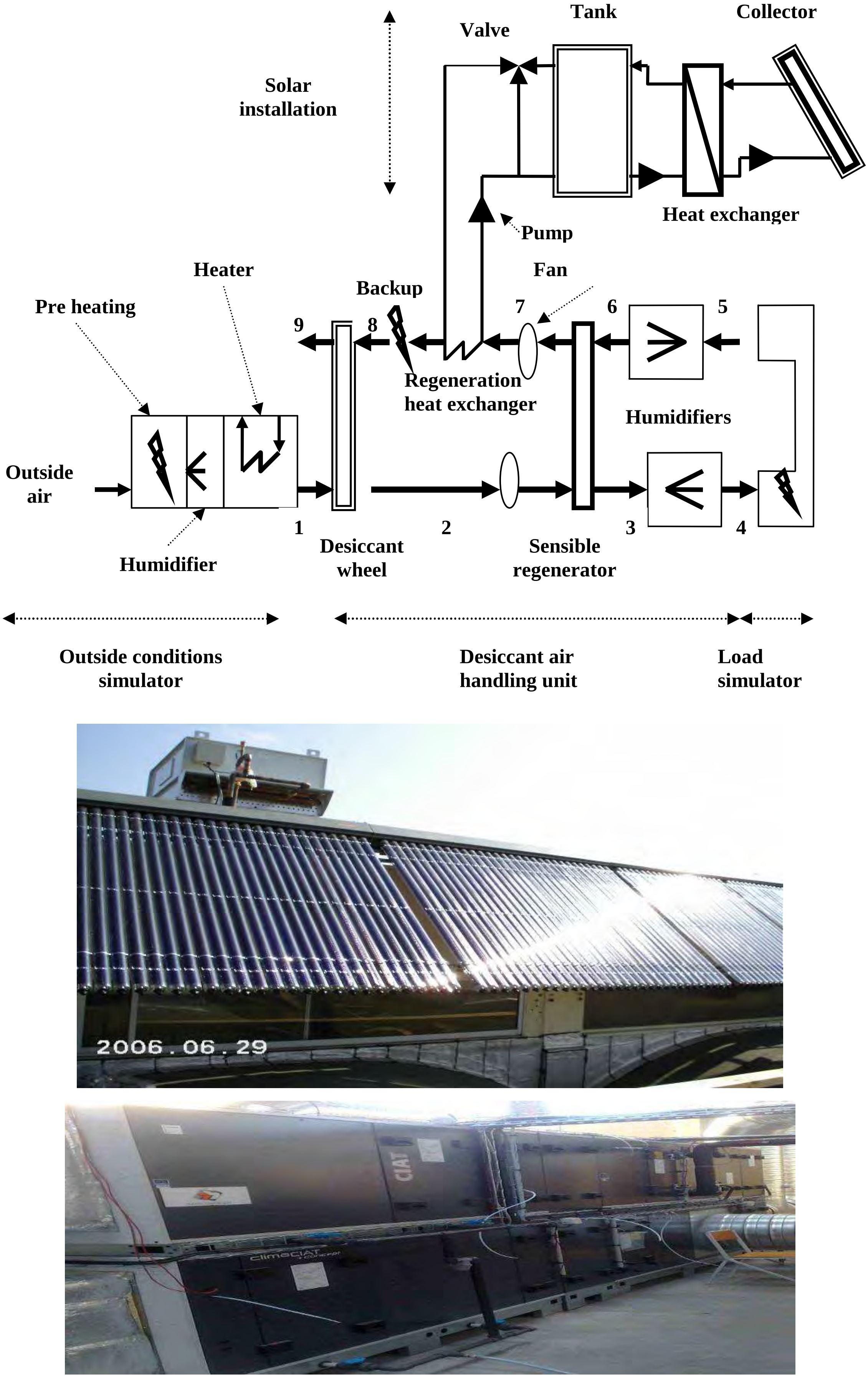


Figure 6: Experimental solar desiccant cooling installation of La Rochelle

Experimental validation of components' model

<u>The sensible regenerator</u>

The outlet temperature of the sensible regenerator depends on the temperature at both inlets of the regenerator. Various temperature differences between the inlets of the regenerator (position 2 and 6) were considered. Figure 7 below plots the experimental efficiency of the regenerator as well as the outlet temperature deviation of the model relative to the measurements.

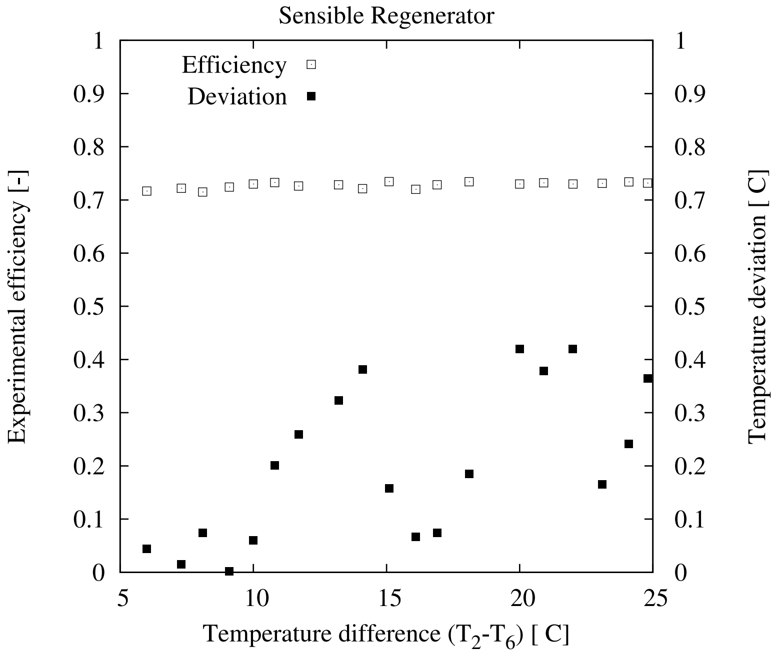


Figure 7: experimental efficiency of the sensible regenerator and deviation of the model prediction of the outlet temperature with respect to measurements

Figure 7 shows that the measured efficiency of the sensible regenerator is nearly constant and equal to 0.7 in all cases; it also shows that the error in predicting the outlet temperature (position 3 in Figure 1) of the regenerator using the model presented above is low, never exceeding 0.5℃. It should be noted that this devia tion is in the domain of the accuracy of the measurements. This accuracy can be predicted, as the efficiency is constant, independent of the operating temperature.

<u>The desiccant wheel</u>

For the desiccant wheel model, different inlet conditions (position 1 with reference to Figure 1) were considered with the temperature varying from 25℃ to 38℃ and the humidity ratio from 10 to 15g/kg, and for different regeneration temperatures (60℃, 65℃, 70℃ and 80℃). Figure 8 plots temperature error against humidity ratio error for the most significant points.

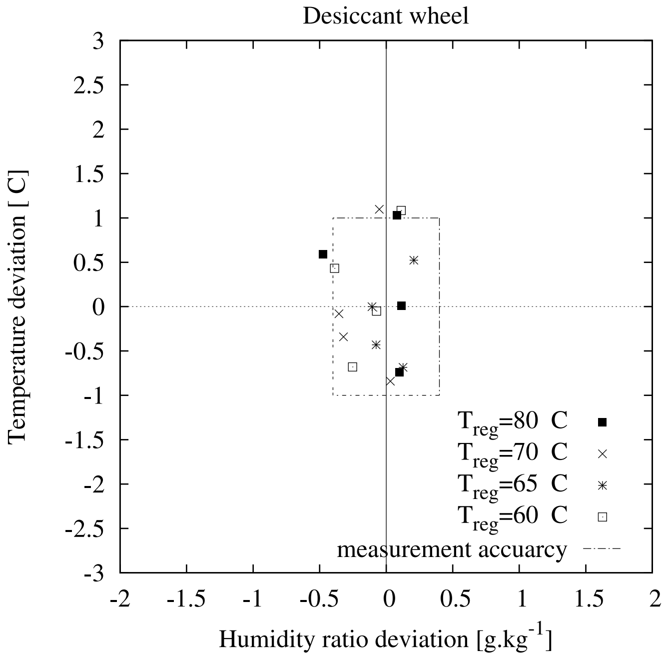


Figure 8: Deviation of the outlet temperature and outlet humidity ratio in comparison with measured results for different operating conditions

The maximum error in predicting the temperature at the outlet of the desiccant wheel is 1°C (a relative error of 2.5%). This error occurs for different regeneration temperatures and for the same inlet temperature of the wheel, 38°C, which is not in the application domain of desiccant cooling. For the humidity ratio, the maximum error is 0.5 g/kg but it should be noted that the accuracy of humidity ratio measurement in the air handling unit is 0.4 g/kg (this accuracy can be tested by comparing the humidity ratio at points 2, 3 and 4 of Figure 1 with the supply humidifier switched off; in this case the points must have the same humidity ratio), so the error is in the validity domain of the measurements.

This accuracy of the model is obtained when the model is dynamically parameterized in function of the operating regeneration temperature; if this is not the case, the deviation will be more significant when the operating regeneration temperature is much higher than that of rating conditions used to parameterize the model [1].

<u>The evaporative cooler</u>

For the evaporative cooler model, Figure 9 demonstrates that the enthalpy is constant across the evaporative cooler and that the constant enthalpy model can thus accurately predict the outlet temperature of the evaporative cooler.

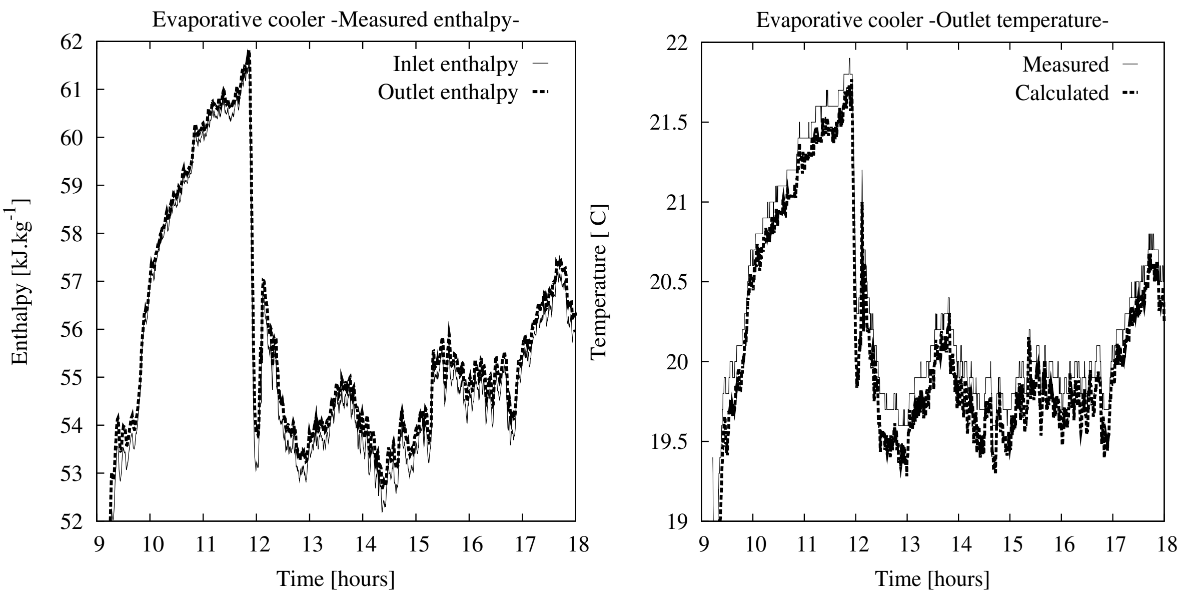


Figure 9: Measured inlet and outlet enthalpy of the evaporative cooler (left) and comparison between the measured and calculated outlet temperature of the evaporative cooler

Solar Collectors

Three different recorded days were used to validate the collector model. The first day is representative of typical summer conditions and the collectors are under storage load only, the second day is of atypical solar radiation with the same storage load and the third day represents a typical desiccant cooling load conditions e.g. storage in the morning and regeneration in the afternoon.

Once the solar radiation in the collectors' plane, the outside temperature, and the inlet temperature were recorded, the computed and measured collector outlet temperatures were compared.

Day 1:

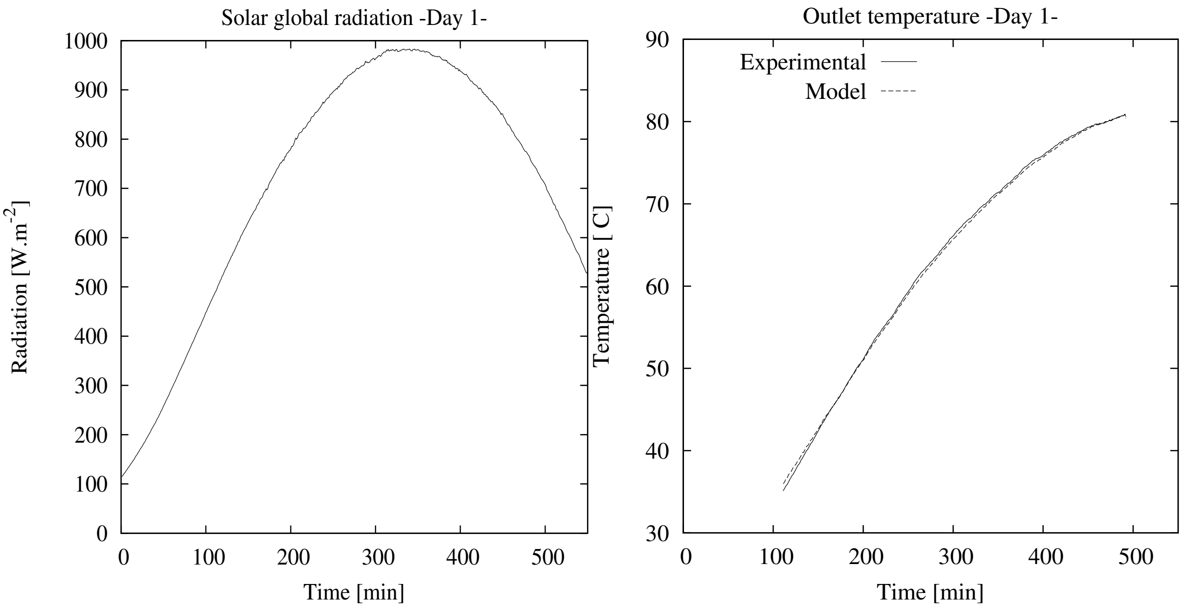


Figure 10: comparison of the predicted outlet temperature of the collectors with the measured one for perfect radiation conditions and a storage load

Comparison between the computed and the measured temperatures for the typical summer day conditions shows the model's high performance in predicting collector outlet temperature with a negligible error. This accuracy is due to the fact that each component is taken into consideration by the model and the calculations are performed in each vacuum tube simultaneously (400 tubes).

While collector outlet temperature can thus be predicted accurately in normal radiation conditions, it is very important to study the performance of the model for atypical conditions.

Day 2:

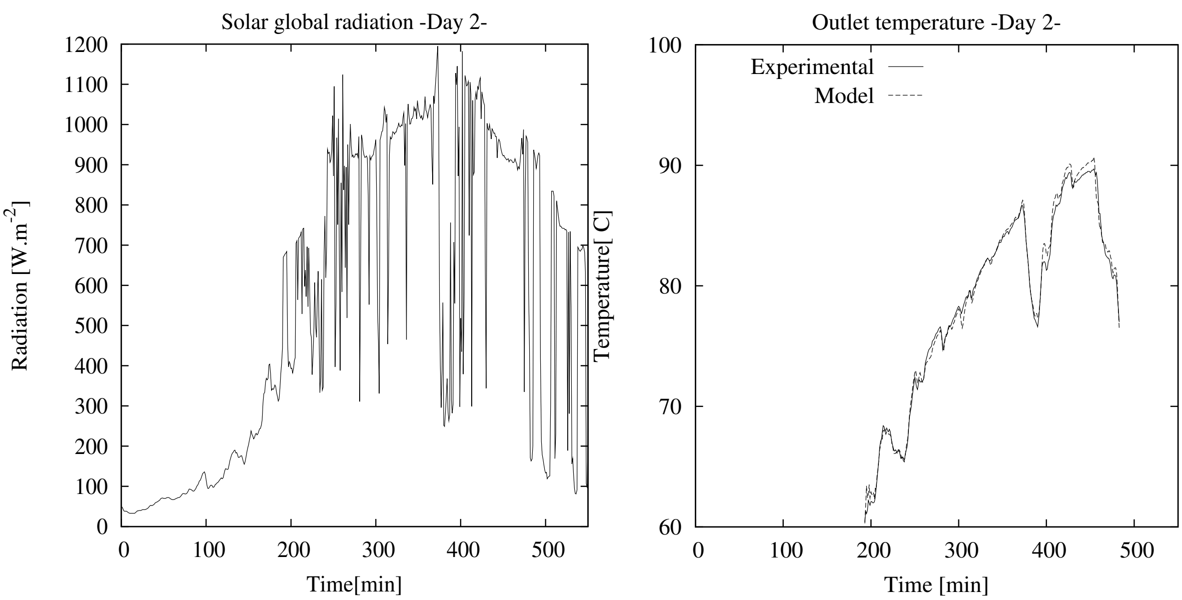


Figure 11: comparison of the predicted outlet temperature of the collectors with the measured one for fluctuating radiation conditions and a storage load

By comparing the calculated and the measured outlet temperatures for the atypical conditions day, when the overall solar radiation fluctuated constantly, it can be seen that the model very accurately predicts the outlet temperature for the whole day, whatever the amplitude of the solar radiation fluctuations and the maximum error is always below 2℃ but the model does not show any time delay with respect to the measurments.

In these first two cases, the model correctly predicts the collector outlet temperature under different radiation conditions but for storage load. The next step in the validation procedure is to study the model's response to a desiccant cooling load, i.e. storage in the morning, with regeneration in the afternoon.

<u>Day 3:</u>

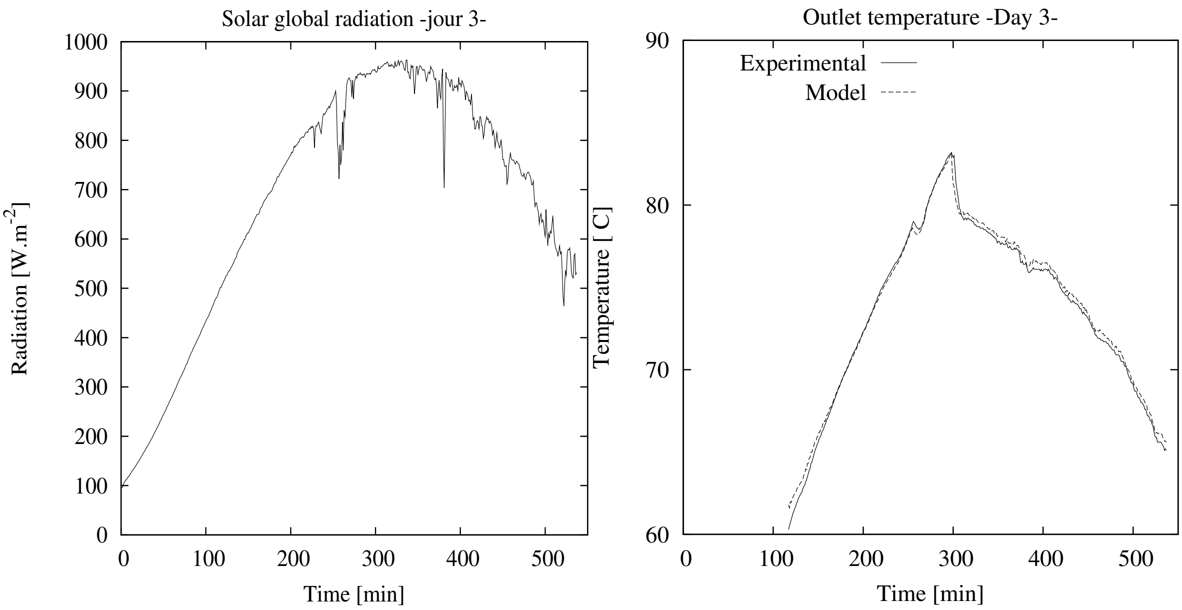


Figure 12: comparison of the predicted outlet temperature of the collectors with the measured one for good radiation conditions and a desiccant load (storage in the morning and regeneration starts in the middle of the day)

Under typical desiccant cooling load there is remarkable agreement between the predicted and the measured outlet temperatures, for both storage and regeneration periods even for a sever transition period at t=300 min.

These three typical days show the capacity of the model to predict the outlet temperature of the collectors with a negligible error.

Storage Tank

Two different scenarios are considered for the validation of the storage tank model. In the first only storage is considered (m_1>0 and m_2=0 in reference to Figure 5) while for the second it concerns desiccant cooling application with storage in the morning and regeneration in the afternoon (m_1>0 and m_2>0). The flow rates m_1 and m_2 are measured as well as the inlet temperature of the storage tank. The predicted and measured temperatures at the top and the bottom of the buffer are compared in the Figure 13 below.

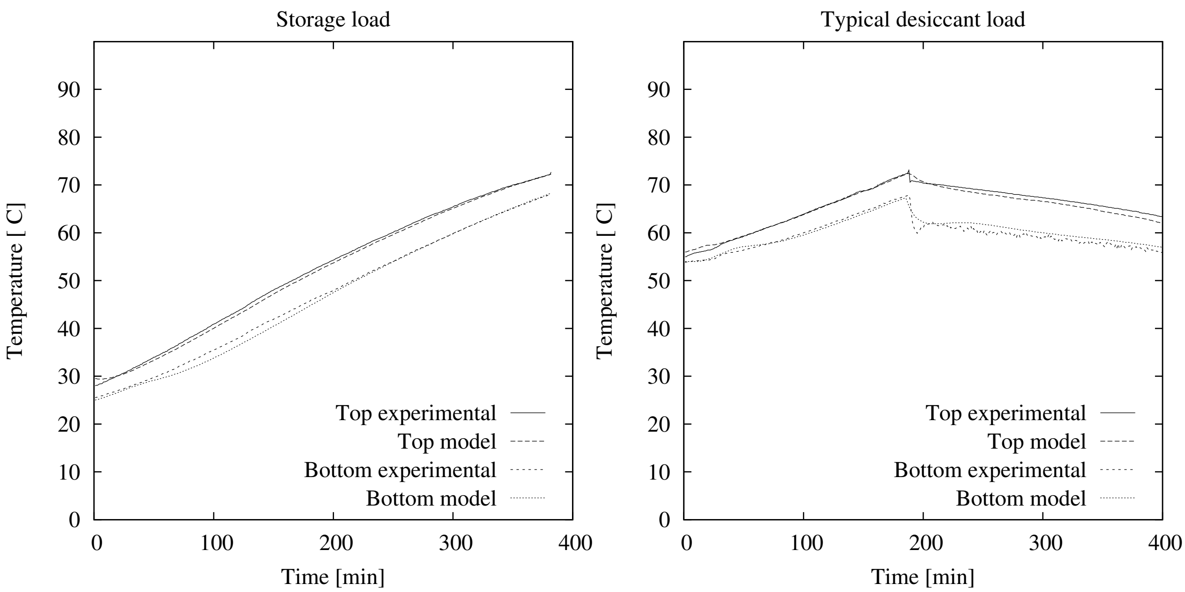


Figure 13: Comparison of the predicted temperature with measured one for the top and the bottom of the storage tank under a storage load (left) and a desiccant load (right)

In both scenarios the model can accurately predict the temperature at the top and the bottom of the buffer. In the first case the stratification remains fairly constant but increases when the storage and the regeneration are combined. This may at first, appear incoherent as more mixing occurs in the latter case. The fact that stratification is greater with the regeneration is that the regeneration load was applied by water loss and the return water was cold and significantly below the tank temperature which amplified the stratification. Even with this extreme case the model was capable to predict accurately the temperature at the top and the bottom of the buffer.

Now if the model has shown an acceptable accuracy when considered solely, we have to be sure that this accuracy will not be lost when coupling the components making the numerical simulation useless. In the next section the error propagation will be analysed and the impact on the supply temperature is examined.

Error propagation

For the solar installation the maximum error in the temperature prediction was that of the tank with a maximum error of 1℃ which yields an er ror in the prediction of the regeneration temperature. Or the experimental results showed that 1℃ difference in the regeneration temperature does not have any practical impact on the performance of the desiccant wheel and thus on the supply conditions.

For the air handling unit the evaporative cooler model and the sensible regenerator models has shown negligible errors. In reversal the desiccant wheel model has shown simultaneous error in temperature and humidity. Let us consider an error *α* in the prediction of the outlet temperature T_2 of the desiccant wheel.

The temperature T_3 at the outlet of the sensible regenerator in function the inlet temperature T_2 and T_6 of the regenerator and its efficiency *η*:

$$T_3 = T_2(1-\eta) + \eta T_6 \tag{24}$$

With the error *α* in the prediction of the temperature T_2 the temperature T_3 is now:

$$T_3^{'} = T_3 + \alpha(1-\eta) \tag{25}$$

This means that an error *α* in the prediction of the T_2 will yield an error *(1-α)* in the prediction of T_3. Now considering an error β in the prediction of the humidity ratio at the outlet of the wheel, this will yield an error at the outlet temperature of the evaporator cooler in the order of:

$$\Delta T = \frac{h_{fg}\beta}{c_{pa}} \tag{26}$$

h_{fg} is the latent heat of vaporization of water.

With a maximum deviation of 2℃ and 0.4 g/kg in the temperature and humidity prediction at the outlet of the desiccant wheel and appropriately combining the impact of the errors we have a maximum error of 1.3℃ in the prediction of the supply temperature. This is the maximum error in the supply temperature that can be committed by the model.

Finlay we will compare the overall model prediction for the air handling unit coupled with the solar installation for day under typical desiccant operations.

Overall performance of the model

A day under desiccant operation is selected to validate the overall model. The response of the model will be compared with the measured performance for the most important components and especially for the supply conditions. Figure 14 below shows the outside conditions.

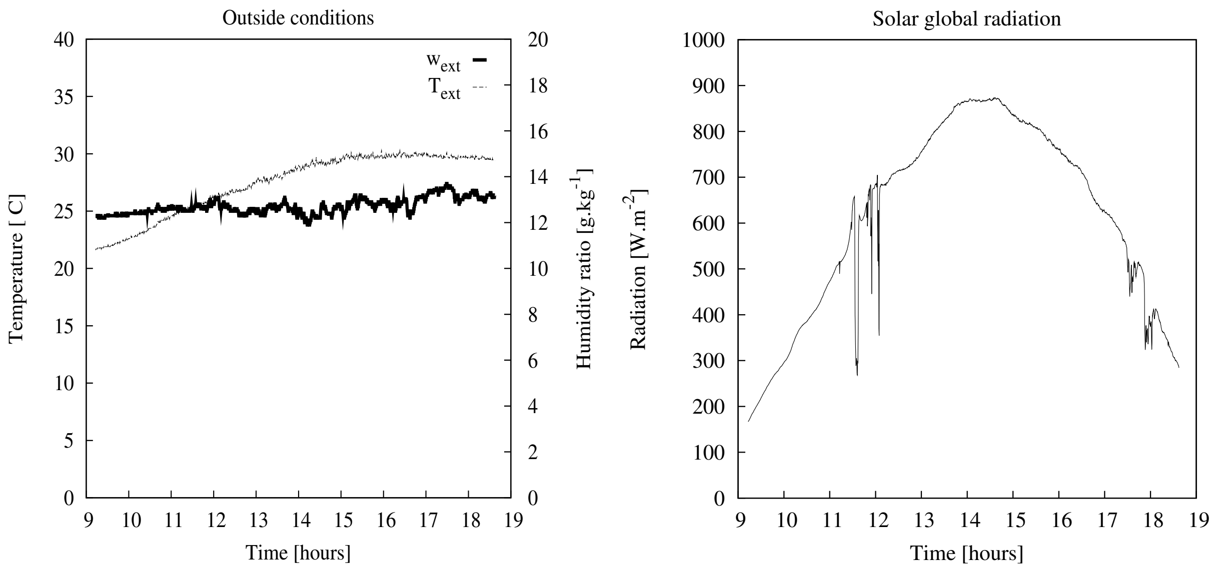


Figure 14: Outside temperature and humidity ratio (left) and solar global radiation for the considered day

The outlet temperature of the collectors, the temperature at the bottom and the top the storage tank as well as the regeneration temperature are shown on the figures below. We notice that the error is below 1℃ in all the cases .

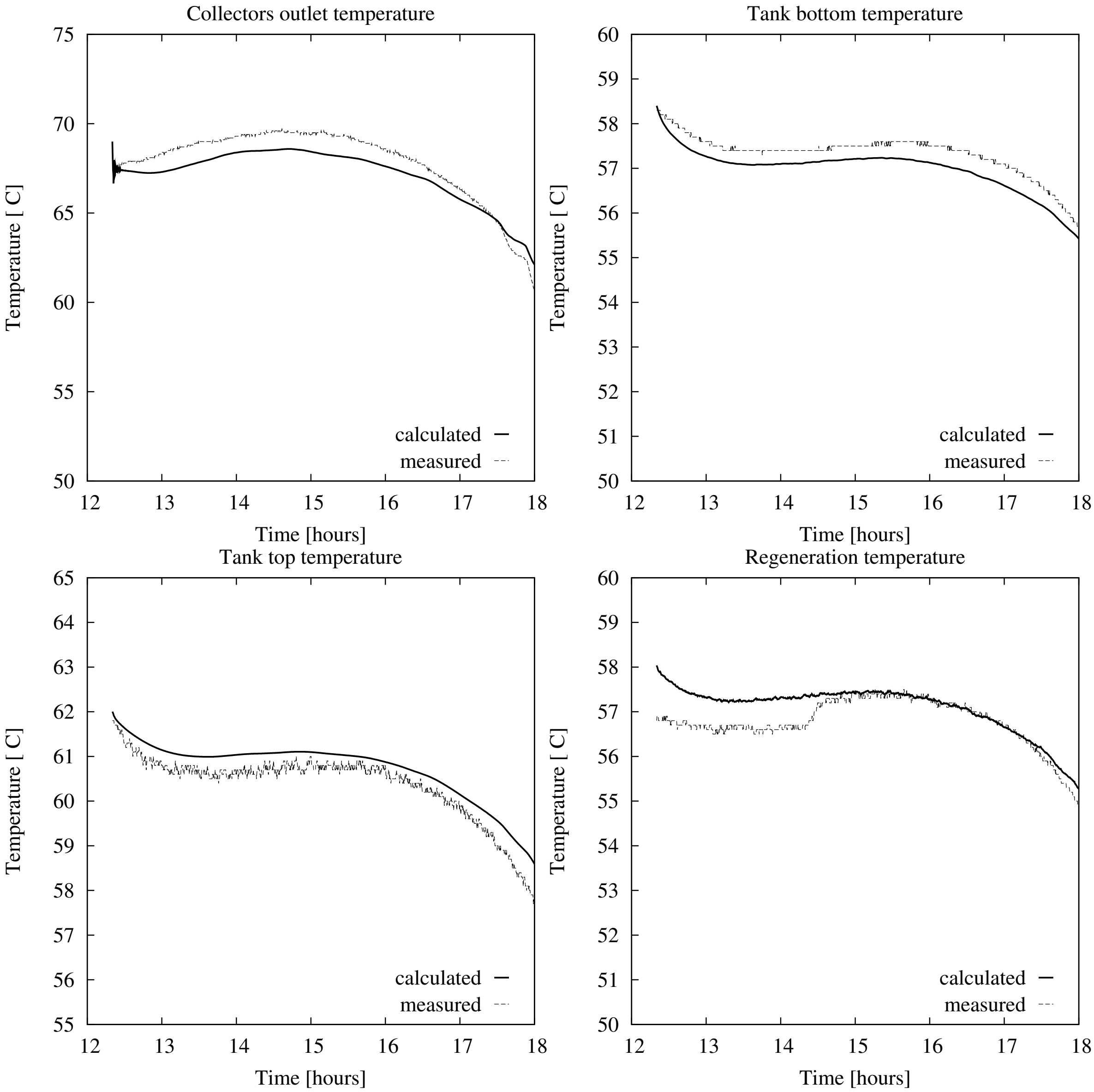


Figure 15: Comparison of the predicted temperature with the measured one for the collectors outlet, storage top, storage bottom, and regeneration air

Figure 16 compares the measured outlet conditions of the desiccant wheel to those predicted by the model. The profiles of the temperature and humidity ratio are very well predicted by the model during the whole day. For the measured temperature we noticed some peaks et we can count 7 peaks per hour and for each peak in the temperature corresponds a minimum in the outlet humidity of the wheel. Or the angular speed of the wheel is 7 rounds per hour this mean for each revolution of the wheel corresponds a peak. The reason behind this behaviour is a probable anomaly in the desiccant wheel with a sector having more desiccant and yields more dehumidification this mean an increase in the outlet temperature of the wheel. It is evident that the desiccant does not take into account this anomaly.

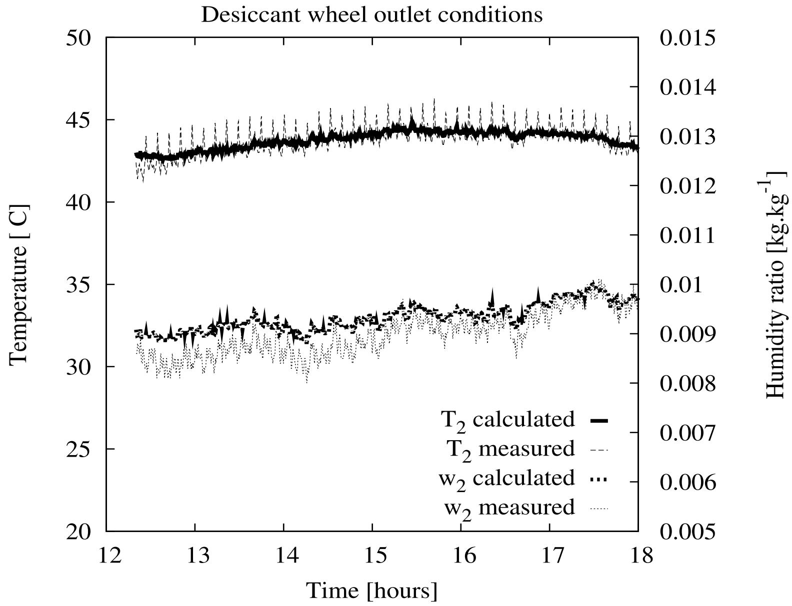


Figure 16: Comparison of the outlet temperature and outlet humidity ratio of the desiccant wheel for the considered day

For the supply temperature, figure 17 shows that the maximum committed error is below 0.5 ℃ and appears during the increase of the humidific ation rate.

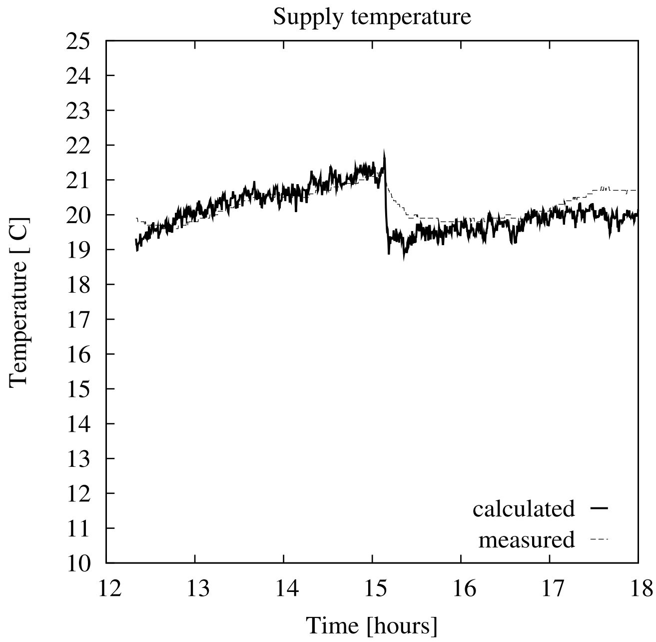


Figure 17: Comparison of the predicted and measured supply temperature of the installation (position 4 on the figure 1) for the considered day

All these results proofs that the model of the installation can predict accurately the supply conditions of the solar desiccant installation.

In the next section a transient model of the desiccant wheel is presented.

3. Transient model of the desiccant wheel and experimental validation

Conventional models of the desiccant wheel predict a mean temperature and a mean humidity ratio at the outlet of the desiccant wheel. In reality and due to the low rotation speed of the desiccant wheel, the temperature and humidity distribution at the outlet is not uniform. The model presented in this section is a two dimensional model that gives the temperature and humidity evolution across the wheel and at its outlets.

Model description

The desiccant wheel scheme is given in the figure below

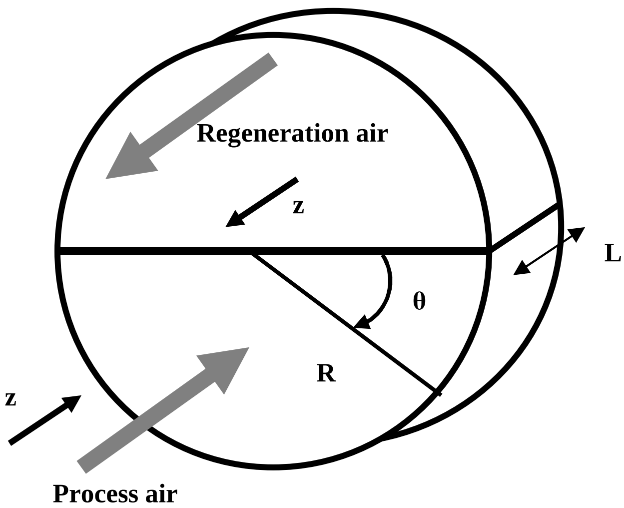


Figure 18: Schematic of the desiccant wheel

In the model development the following assumption are taken:

- The state properties of the air streams are spatially uniform at the desiccant wheel inlet
- The interstices of the porous medium are straight and parallel
- There is no leakage or carry-over of streams
- The interstitial air velocity and pressure are constant
- Heat and mass transfer between the air and the porous desiccant matrix is considered using lumped transfer coefficients
- Diffusion and dispersion in the fluid flow direction are neglected
- No radial variation of the fluid or matrix states

A basic element in the angular and width direction of the desiccant wheel is considered and can be presented as follows:

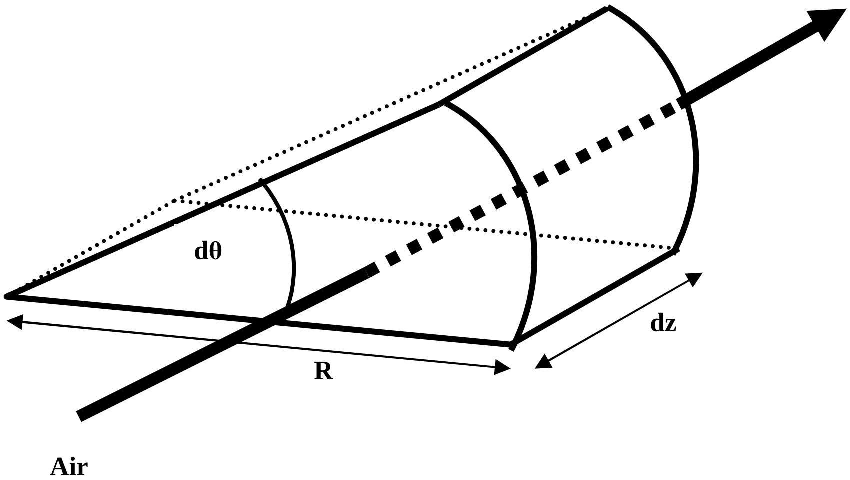


Figure 19: Basic element of the desiccant wheel

Fundamental equations of heat and mass transfer for the basic element

Mass conservation equation

$$M_d \frac{\partial W}{\partial t} + m_a \left(\frac{\partial w_a}{\partial t} + u \frac{\partial w_a}{\partial z} \right) = 0 \tag{27}$$

Mass transfer equation

$$M_d \frac{\partial W}{\partial t} = h_m S \left(w_a - w_{eq} \right) \tag{28}$$

Heat conservation equation

$$M_d \frac{\partial H}{\partial t} + m_a \left(\frac{\partial h_a}{\partial t} + u \frac{\partial h_a}{\partial z} \right) = 0 \tag{29}$$

Heat transfer equation

$$M_d \frac{\partial H}{\partial t} = h_m S \left(w_a - w_{eq} \right)\left(h_{fg} + c_{pv} T_a \right) + h_t S \left(T_a - T_d \right) \tag{30}$$

Variable change

The time t is related to the angular position θ with the following equation

$$\frac{t}{\tau_{ro}} = \frac{\theta}{\pi} \text{ so } t = \frac{\tau_{ro}}{\pi} \theta ,$$

Where τ_{ro} is the rotation period of the process or regeneration and *N* is the angular speed of the wheel.

The partial derivative of a variable *x* with respect to time *t* can be thus written with respect to the angular position θ following: $\frac{\partial x}{\partial t} = \frac{\pi}{\tau_{ro}} \frac{\partial x}{\partial \theta}$

This yields the following system:

<u>Mass conservation equation</u>

$$M_d \frac{\pi}{\tau_{ro}} \frac{\partial W}{\partial \theta} + m_a \left(\frac{\pi}{\tau_{ro}} \frac{\partial w_a}{\partial \theta} + u \frac{\partial w_a}{\partial z} \right) = 0 \tag{31}$$

<u>Mass transfer equation:</u>

$$M_d \frac{\pi}{\tau_{ro}} \frac{\partial W}{\partial \theta} = h_m S \left(w_a - w_{eq} \right) \tag{32}$$

<u>Heat conservation equation</u>

$$M_d \frac{\pi}{\tau_{ro}} \frac{\partial H}{\partial \theta} + m_a \left(\frac{\pi}{\tau_{ro}} \frac{\partial h_a}{\partial \theta} + u \frac{\partial h_a}{\partial z} \right) = 0 \tag{33}$$

<u>Heat transfer equation</u>

$$M_d \frac{\pi}{\tau_{ro}} \frac{\partial H}{\partial \theta} = h_m S \left(w_a - w_{eq} \right) \left(h_{fg} + c_{pv} T_a \right) + h_t S \left(T_a - T_d \right) \tag{34}$$

The obtained equation system is coupled non linear hyperbolic, since the enthalpy of the moist air and of the desiccant depend on the water content and on the temperature.
In the bibliography the equations were solved by Macalaine-cross, [17] and Stabat, [11] by neglecting the terms $\frac{\partial w_a}{\partial \theta}$ and $\frac{\partial h_a}{\partial \theta}$ in front of transport term which yields a simplification of the equations and of the matrix to be solved. In this work we conserved these terms.

Transformation to a discrete problem and restriction to 2 dimensions

The wheel is divided into elementary domains following the figure below

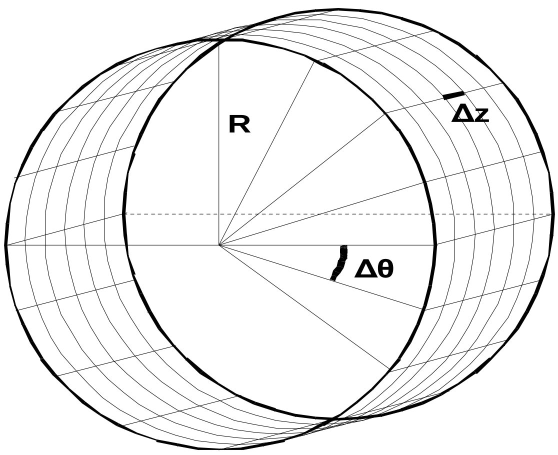


Figure 20 : discretisation of the desiccant wheel

Since in equations the radius R of the wheel does not appear since the variables are assumed to be constant in the radial direction the problem can thus be treated as two dimensional problem following z and θ.

The considered steps are then:

$\Delta\theta = \dfrac{2\pi}{Na}$ and $\Delta z = \dfrac{L}{Np}$

Where *Na* is the number of steps in the angular position θ direction and *Np* is the number of steps in the width z direction. The domain can be represented on Cartesian scale:

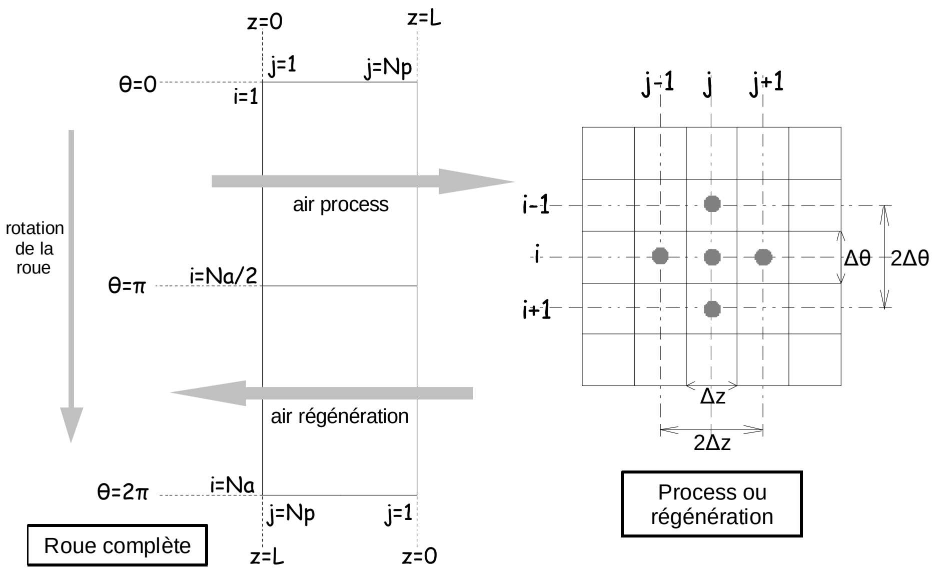


Figure 21: Cartesian representation of the grid

For each node in the center of the elementary domain is identified by the coordinate (i,j) where i corresponds to the angular position and j to the width position:

<u>For the process:</u>

$$0 < \theta < \pi \Leftrightarrow i = 1 \; to \; \frac{Na}{2} \quad \begin{cases} j = 1 & process \;\; inlet \quad (z = 0) \\ j = Np & process \;\; outlet \quad (z = L) \end{cases}$$

<u>For the regeneration:</u>

$$\pi < \theta < 2\pi \Leftrightarrow i = \frac{Na}{2} + 1 \; to \; Na \quad \begin{cases} j = 1 & regeneration \;\; inlet \quad (z = 0) \\ j = Np & regeneration \;\; outlet \quad (z = L) \end{cases}$$

Estimation of the partial derivatives

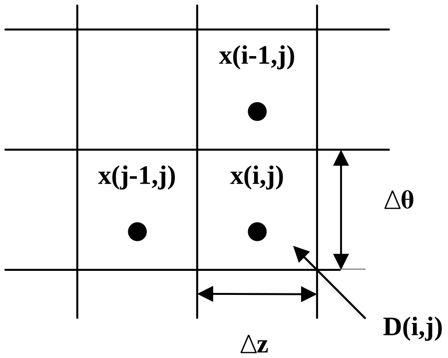


Figure 22: Current and neighbor nodes

The estimation of the partial derivatives is done using an upwind interpolation that permits expressing the frontiers values in function of the considered node and in function of the previous node.

The derivative with respect to θ:

$$\frac{\partial x}{\partial \theta} = \frac{\left(x_{i,j} - x_{i-1,j}\right)}{\Delta \theta}$$

The derivative with respect to z:

$$\frac{\partial x}{\partial z} = \frac{\left(x_{i,j} - x_{i,j-1}\right)}{\Delta z}$$

The moist air enthalpy is expressed by:

$$h_a = c_{pa}.T_a + w_a\left(h_{fg} + c_{pv}.T_a\right) \tag{35}$$

The desiccant enthalpy is expressed following Brandemuehl and Banks [18] and Stabat [11]

$$H = c_{pd}\, T_d + c_{pv}\, W\, T_d + \frac{h_{fg}\,\Delta\lambda}{k} e^{-kW} - 1 \tag{36}$$

Replacing the enthalpies and derivative by the above equations, for each node (i,j) we have finally:

<u>Mass conservation equation</u>

$$M_d{}'\left(W_{i,j} - W_{i-1,j}\right) + m_a{}'\left(w_{a\,i,j} - w_{a\,i-1,j}\right) + m_a * \left(w_{a\,i,j} - w_{a\,i,j-1}\right) = 0 \tag{37}$$

<u>Mass transfer equation</u>

$$M_d{}'\left(W_{i,j} - W_{i-1,j}\right) - h_m S' w_{a\,i,j} = -h_m S' w_{eq\,i,j} \tag{38}$$

<u>Heat conservation equation</u>

$$\begin{aligned}&\left(c_{m\,i,j}\,T_{d\,i,j}-c_{m\,i-1,j}\,T_{d\,i-1,j}\right)+\left(c_{a\,i,j}\,T_{a\,i,j}-c_{a\,i-1,j}\,T_{a\,i-1,j}\right)+m_a{}'\,h_{fg}\left(w_{a\,i,j}-w_{a\,i-1,j}\right)\\&+\left(c_a*_{i,j}\,T_{a\,i,j}-c_a*_{i,j-1}\,T_{a\,i,j-1}\right)+m_a*h_{fg}\left(w_{a\,i,j}-w_{a\,i,j-1}\right)\\&=-M_d{}'\,h_{fg}\,\frac{\Delta\lambda}{k}\left(\exp(-kW_{i,j})-\exp(-kW_{i-1,j})\right)\end{aligned}\tag{39}$$

<u>Heat transfer equation</u>

$$\begin{aligned}&\left(c_{m\,i,j}\,T_{d\,i,j}-c_{m\,i-1,j}\,T_{di-1,j}\right)-h'\,w_{a\,i,j}-h*_{i,j}\,T_{a\,i,j}+h_cS'\,T_{d\,i,j}\\&=-M_d{}'\,h_{fg}\,\frac{\Delta\lambda}{k}\left(\exp(-kW_{i,j})-\exp(-kW_{i-1,j})\right)-h'\,w_{eq\,i,j}\end{aligned}\tag{40}$$

With the following coefficients [19]

$$M_d{}'=\frac{\pi\,M_d\,\Delta z}{\tau_{ro}}\qquad m_a{}'=\frac{\pi\,m_a\,\Delta z}{\tau_{ro}}\qquad m_a*=m_a\,u\;\;\Delta\theta$$

$$S'=S\,\Delta\theta\,\Delta z\qquad h'=h_m\,S'\,h_{fg}$$

$$c_{m\,i,j}=M_d{}'\left(c_{pd}+c_{pv}\,W_{i,j}\right)\qquad c_{a\,i,j}=m_a{}'\left(c_{pa}+c_{pv}\,w_{a\,i,j}\right)$$

$$c_a*_{i,j}=m_a*\left(c_{pa}+c_{pv}\,w_{a\,i,j}\right)\qquad h*_{i,j}=h_m\,S'\,c_{pv}\left(w_{a\,i,j}-w_{eq\,i,j}\right)+h_c\,S'$$

For the details of the boundary conditions of the model please check the French version of [1].

Model performance

The above presented model of the desiccant was implemented in SPARK. The figure below shows the evolution of the air and the desiccant properties inside the wheel for an inlet temperature of 30℃, inlet humidity ratio of 12 g/k g, regeneration temperature of 60℃ and regeneration humidity ratio of 12 g/kg. The figures show the temperature and the humidity ratio distribution as a function of the angular position (0 to 180°) and as function of the width (0 to 0.2m) of the wheel for the process sector.

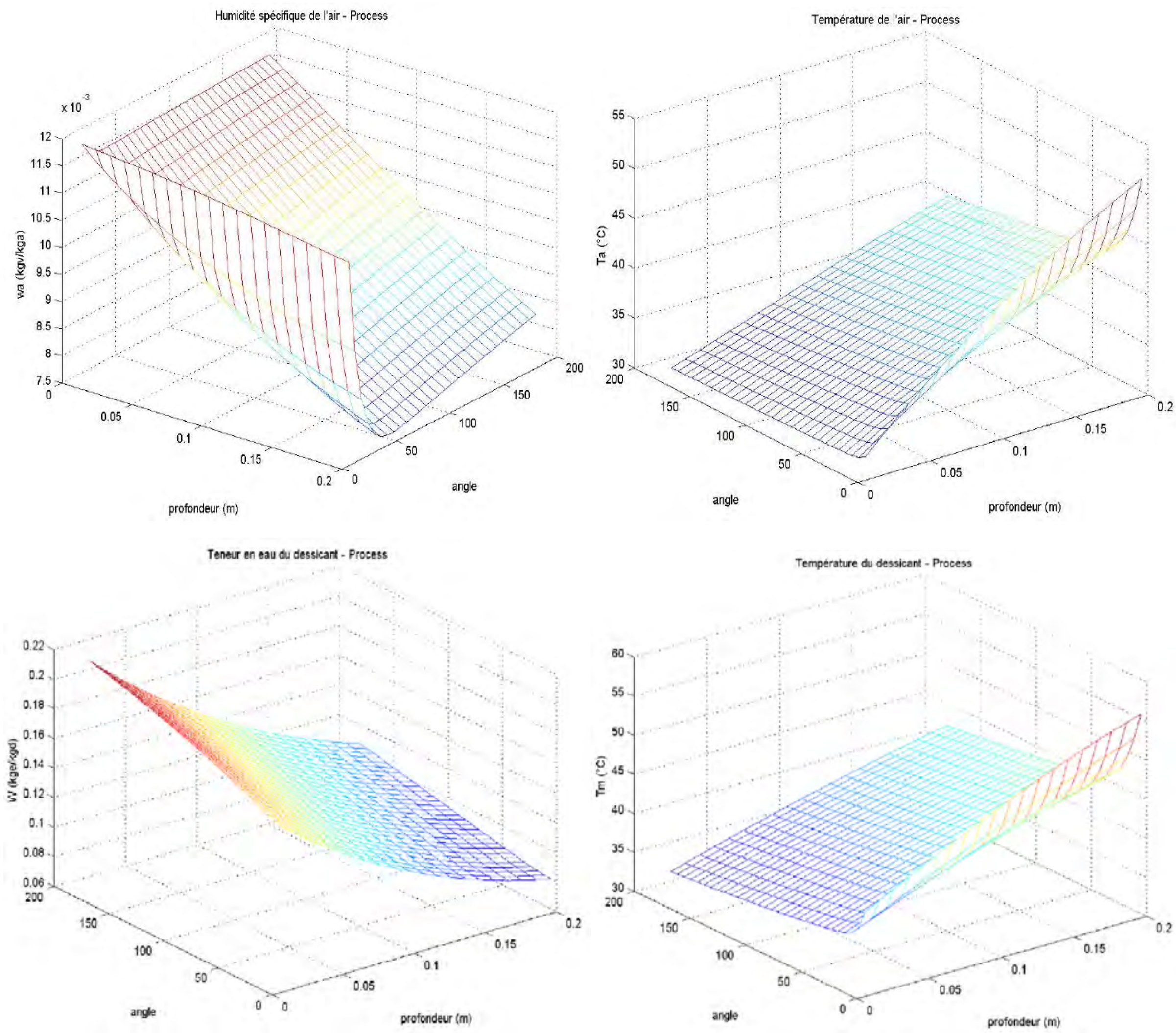
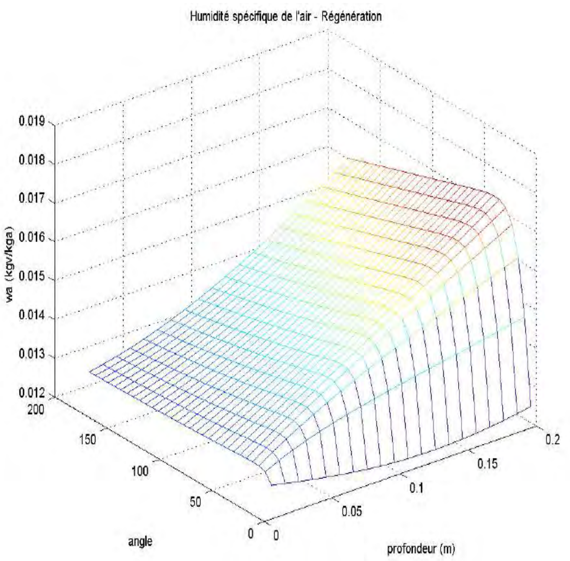


Figure 23: Evolution of air and desiccant properties for the process section in function of the wheel width and the angular position

We notice that for a given angular position the air humidity ratio decreases with the width of the wheel and yields a temperature increase. The dehumidification increase significantly in the first angular sector of the process (0-40°) and reaches its minimum and then increases slightly with the angular position. This behavior of the humidity is due to the fact that after the regeneration; the desiccant is dry and when reaching the process it starts to dehumidify and then starts to saturate. The other reason is the heat of adsorption that heats the desiccant which reduces its adsorption capacity.

While examining the air temperature and humidity ratio profiles we notice a certain angular delay of the dehumidification in comparison with the temperature increase. This can be explained that the desiccant coming from the regeneration is very hot and has a limited adsorption capacity. So in the process sector; outside air must first cool the desiccant and then dehumidification becomes effective.

The figures below show a similar behavior in the regeneration sector.



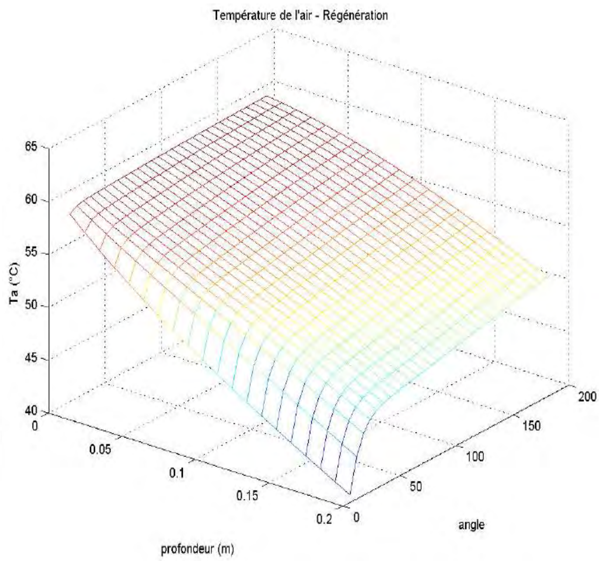


Figure 24: Evolution of the air properties in the regeneration section as a function of the width and angular position

Experimental validation

With the difficulty to measure the temperature and humidity inside the wheel for the detailed model only the temperature and humidity profiles as well as the mean temperature and humidity ratio at the outlet of the wheel will be validated.

In order to validate the temperature and humidity distribution at the outlet of the wheel, 3 psychrometers and 3 Pt100 temperature sensors (3 humidity measurments and 6 temperature measurements) are used in order to validate the profiles as shown in the below.

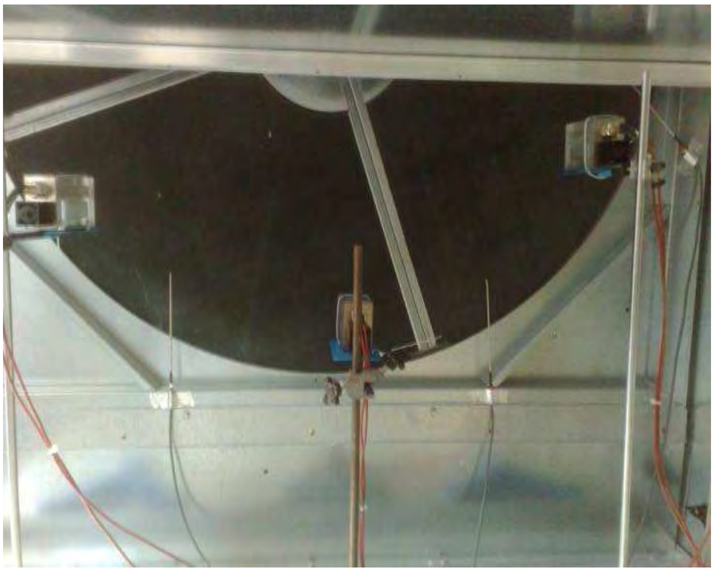


Figure 25: Psychrometers and temperature sensors distribution at the outlet of the desiccant wheel

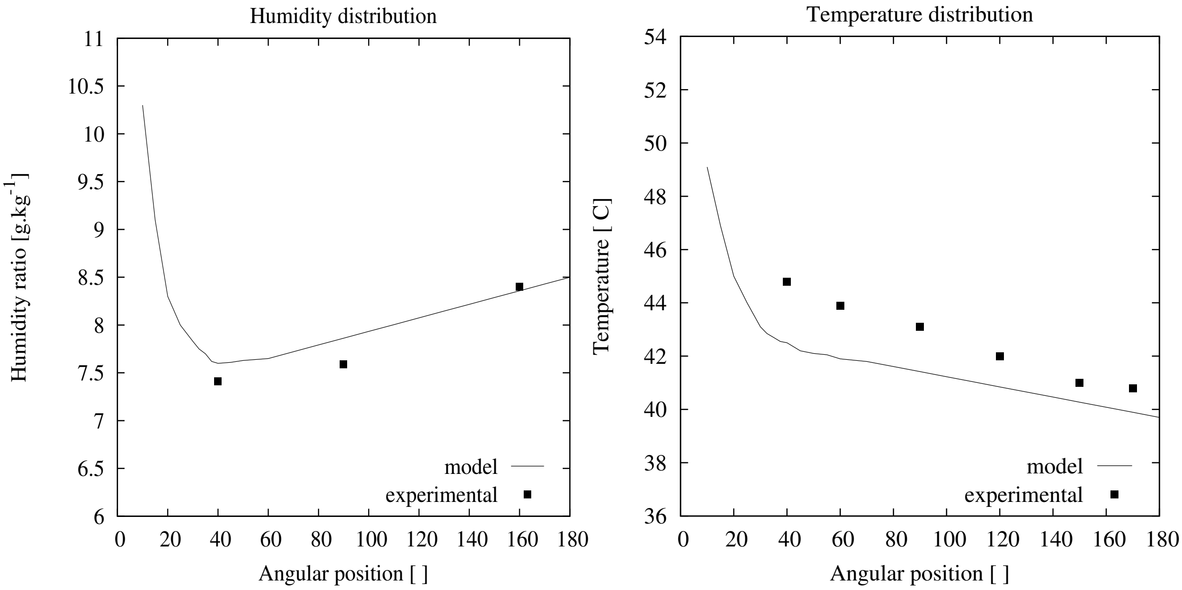


Figure 26: Comparison of the predicted and measured temperature and humidity distribution at the outlet of the desiccant wheel

The profiles obtained numerically are coherent with the measured profiles and we notice 0.3 g/kg error in the humidity ratio while an error of 2℃ in the temperature.

The outlet conditions of the desiccant wheel depend on the temperature and humidity at both inlets of the wheel. In order to validate the detailed and the simplified models of the wheel were considered. Different inlet conditions were considered: the inlet temperature of the process stream varies between 25 and 35℃ while the humidity ratio varies between 10 and 15 g/kg and the regeneration temperature was considered varying from 60 to 80℃.

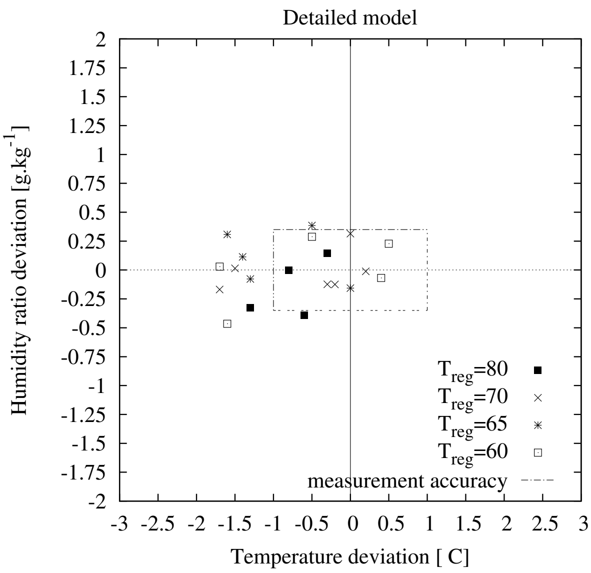


Figure 27: Comparison of the predicted and measured mean outlet temperature and humidity at the outlet of the desiccant wheel.

Figure 27 gives the temperature and humidity ratio deviations at the outlet of the desiccant wheel for the detailed. The maximum humidity ratio deviation of the detailed model is 0.4 g/kg and the maximum temperature deviation is 2℃. The uncertainty of the mean temperature at the outlet of the wheel is in the order of 1 to 1.5℃ (inhomogeneous outlet with 10℃ gradient) and the uncertainty of the humi dity ratio is 0.35 g/kg. The uncertainty of the humidity is evaluated by comparing the mean humidity at the outlet of the wheel (position 2) with the absolute humidity at points 3, 4 and 5 (with reference to figure 1) while turning off the supply humidifier. In these conditions the humidity must be constant. We notice that the humidity ratio deviations are in the domain of the validity of the measurements. The detailed model established of the physical description shows the same accuracy independently of the regeneration temperature.

Conclusion

In this chapter a model of the solar desiccant installation is developed and validated experimentally. First the components model considered solely showed acceptable accuracy and low errors under different operating conditions. Then the overall model of the air handling unit coupled with the solar installation gave satisfactory results for the considered day and predicts accurately the supply air conditions. For the developed solar collector model it showed a good accuracy in predicting the outlet temperature under transient operating conditions and permits a good estimation of the potential of the collectors in transient conditions. This model was used to elaborate control strategies and to optimize the operations of the installation in function of outside and load conditions.

The developed detailed transient model of the desiccant wheel showed a good accuracy in predicting the temperature and humidity distribution at the outlet of the wheel as well as the mean outlet temperature and mean outlet humidity ratio. This detailed model is not suited for seasonal simulation but it is very useful to elaborate control strategy and to study transient behaviour of the desiccant air handling unit. The model can be used to study different desiccant materials.

References

[1]. Bourdoukan, P. "Etude numérique et expérimentale destinée à l'exploitation des techniques de rafraîchissement par dessiccation avec régénération par énergie solaire." *Ph.D Thesis,* Université La Rochelle, (2008) (Available in English)

[2]. Bourdoukan, P., Wurtz, E., Joubert, P., and Sperandio, M.. "Potential of solar heat pipe vacuum collectors in the desiccant cooling process: modelling and experimental results." *Solar Energy* 2008; 82(12): 1209-1219.

[3]. Bourdoukan, P., Wurtz, E., Joubert, P., and Spérandio, M. "Overall cooling efficiency of a solar desiccant plant powered by direct flow vacuum tube collectors: simulation and experimental results" *Journal of Building Performance Simulation* 2008, 1(3):149-162

[4]. Bourdoukan, P., Wurtz, E., Joubert, P." Experimental investigation of a solar desiccant cooling installation Solar Energy 83 (2009) 2059–2073

[5]. Bourdoukan, P., Wurtz, E., Joubert, P "omparison between the conventional and recirculation modes in desiccant ooling cycles and deriving critical efficiencies of components" Energy an International Journal, 2009 1-11 In press Corrected Proof. doi:10.1016/j.energy.2009.06.021

[6]. Kays, W. M., and London, A. L. Compact Heat Exchangers, McGraw-Hill, New York. (1984)

[7]. Maclaine-cross, I. L., and Banks, P. J. "Coupled heat and mass transfer in regenerators- prediction using analogy with heat transfer." *Int. J. Heat Mass Transfer* 1972; 15: 1225-1242.

[8]. Banks, P. J. "Coupled equilibrium heat and single adsorbate transfer in field flow through a porous medium- 1. Characteristics potentials and specific capacity ratios." *Chemical Engineering Science* 1972; 27: 1143-1155.

[9]. Close, D. J., and Banks, P. J. "Coupled equilibrium heat and single adsorbate transfer in field flow through a porous medium-2. Prediction for a silica gel air drier using characteristic charts." *Chemical Engineering Science* 1972; 27: 1157-1168.

[10]. Jurinak, J. J. "Open cycle solid desiccant cooling - component models and system simulation," *Ph.D*, University of Wisconsin, Madison. (1982)

[11]. Stabat, P. "Modélisation de composants de systèmes de climatisation mettant en œuvre l'adsorption et l'évaporation d'eau," *Ph.D Thesis*, Ecole des Mines, Paris. (2003)

[12]. Kamminga, W.. "The approximate temperature within a flate plate solar collector under transient conditions." *Int. J. Heat Mass Transfer*, 28 (1985), 433-440.

[13]. Schnieders, J.. "Comparison of the energy yield predictions of stationary and dynamic solar collector models and the models' accuracy in the description of a vacuum tube collector." *Solar Energy*, 61(1997), 179-190.

[14]. Klein, S. A.. "A Design Procedure for Solar Heating Systems," PhD, University of Wisconsin-Madison (1976)

[15]. SPARK. "Simulation Problem Analysis and Research Kernel." LBNL, Berkeley, California. (2003)

[16]. Sowell, E. F., and Taghavi, K. "Generation of building energy system models." *ASHRAE Transactions* 1984, 106: 465-476.

[17]. Maclaine-cross, I. L. "A theory of combined heat and mass transfer in regenerator," *Ph.D*, Univeristy of Monach, Australie. (1974).

[18]. Brandemuehl, M. J., and Banks, P. J.. "Rotary heat exchangers with time varying or nonuniform inlet temperatures." *ASME Journal of Heat Transfer*, (1984) 106, 750-758.

[19]. Le Garrec, S. "Modélisation des transferts de masse et de chaleur dans une roue dessiccante," *Master*, ENTPE, Lyon. (2007).

Chapter III: New developments in simulation tools and models:

A dynamic simulation model for transient absorption chiller performance.

Paul Kohlenbach
Solem Consulting

Felix Ziegler
Technische Universität Berlin, Institute for Energy Engineering

This chapter presents the development of a dynamic model for single-effect LiBr/water absorption chillers and its experimental validation.
In section (A) the developed model is presented in detail and compared to experimental data. In section (B), general functionality of the model and a more detailed comparison with experimental data are presented. This part also presents a more detailed investigation of the model performance, including performance analysis, sensitivity checks. General model functionality is demonstrated.

1. A dynamic simulation model for transient absorption chiller performance. Part 1: the model

This section presents in detail the development of transient model for an absorption chiller. The model is based on external and internal steady-state enthalpy balances for each main component. Dynamic behaviour is implemented via mass storage terms in the absorber and generator, thermal heat storage terms in all vessels and a delay time in the solution cycle.For verification, the model has been compared to experimental data. The dynamic agreement between experiment and simulation is very good with dynamic deviations around 10s.

Nomenclature

Symbols

A	area, (m^2)
A	Duehring factor (deg C)
B	Duehring factor (-)
c	specific heat capacity ($kJkg^{-1}K^{-1}$)
c	number of simulation steps representing time constants for transport delay (-)
D	dew point temperature (deg C)
g	gravity constant (Nm^2kg^{-2})
h	height difference between generator outlet and absorber inlet (m)
h	enthalpy ($kJkg^{-1)}$
l	specific heat of solution ($kJkg^{-1}$)
$m, \dot{m}$	mass flow rate (kgs^{-1})
M	mass (kg)
p	pressure (Pa)
$Q,\ \dot{Q}$	heat flux (kW)
r	evaporation enthalpy ($kJkg^{-1}$)
R	gas constant for water vapour (Jkg^{-1})
T	temperature (deg C)
t	time (s)
UA	heat transfer coefficient (kWK^{-1})
x	Solution mass fraction ($kg_{Salt}kg_{Sol}^{-1}$)
X	mole ratio (-)
z	solution level in generator sump (m)

Greek letters

η	effectiveness (-)
ρ	density (kgm^{-3})
Δ	difference (-)
ϑ	temperature (deg C)

Subscripts

A	Absorber
C	Condenser
d	Duehring
E	Evaporator
ext	external
G	Generator
i	simulation time interval
in	inlet
int	internal
meas	measured
p	pump
p, pc	at constant pressure
s	strong
sim	simulated
sol	solution
st	storage
SHX	solution heat exchanger
sG	strong solution leaving the generator tube bundle
sA	strong solution leaving the generator sump and entering the absorber
t	tube
tb	tube bundle
out	outlet
v	vapour
w	water, weak
wA	weak solution leaving the absorber tube bundle
wG	weak solution leaving the absorber sump and entering the generator
X	general index for vessels (X=A, C, E, G)
*	time-delay of solution at generator and absorber inlet

Introduction

The dynamic model of an absorption chiller allows the simulation of its transient behaviour for changing input conditions or design parameters. This is important because absorption chillers usually have a high thermal mass, consisting of their internal heat exchangers, the absorbing solution and the externally supplied heat transfer media. The dynamics of an absorption chiller are therefore rather slow compared to similar capacity compression chillers. The time to achieve a new steady-state with all parameters after a change of input conditions is about 15 minutes for the chiller presented in this paper.

If the chiller is implemented in a complex heat supply/cooling demand system, e.g. a solar thermal or waste-heat driven system, the simulation of the chiller is usually being done using steady-state models. They simulate the chiller assuming constant operating conditions and allow the determination of internal and external cycle parameters, such as heat exchanger sizes, pump flow rates, temperatures and heat flows. However, steady-state models do not provide time-dependent information on the thermal behaviour of absorption chillers and are therefore not suitable for transient system simulations. In contrast, the model presented in this work allows the simulation of the dynamic chiller behaviour. It extends the range of applicable models for transient system simulations where the time constants of the chiller significantly influence the system performance.

Research on dynamic system behaviour was carried out for both LiBr/water and water/NH_3 heat pumps, chillers and components of such. Most complete are the recent papers by Bian et al. [1], and Jeong et al. [2]. Bian et. al. [1] have performed a transient simulation of an absorption chiller. They present a chiller model that can be run using variable time steps for the simulation. It includes a temperature change term of each heat exchanger per time step as well as a mass storage term in the generator, i.e. a part of the strong solution is being stored in the generator in each time step. The model has been verified with experimental data and shows good agreement in the transiency of the thermal behaviour, even if absolute values do not exactly match.

Jeong et. al. [2] present the dynamic simulation of a steam-driven LiBr/water absorption heat pump for the use of low-grade waste heat. The model assumes storage terms with thermal capacities and solution mass storage in the vessels. Solution and vapour mass flow rates are calculated in proportion to pressure differences between vessels. The heat transfer coefficients as well as the simulation time step are assumed to be constant. The model has been verified with good agreement using operational data for an absorption chiller.

The dynamic model presented here simulates the reaction of the absorption chiller on a change of external conditions. In contrast to the approach in the cited references above, a simpler model structure was chosen in order to learn more about the most important interdependencies in a direct manner. For instance, emphasis is on the fact that the thermal storage terms respond partly to the temperatures of the external fluids and partly to the internal process streams and these effects are separated. In contrast to this, the steady state was incorporated very coarsely only, because time consuming iterations were to be avoided. The model has been developed for and verified on the 10kW absorption chiller manufactured by Phoenix SonnenWaerme AG [3].

Basic model

Figure 1 shows the single-effect LiBr/water absorption cycle with state points as assumed for the model.

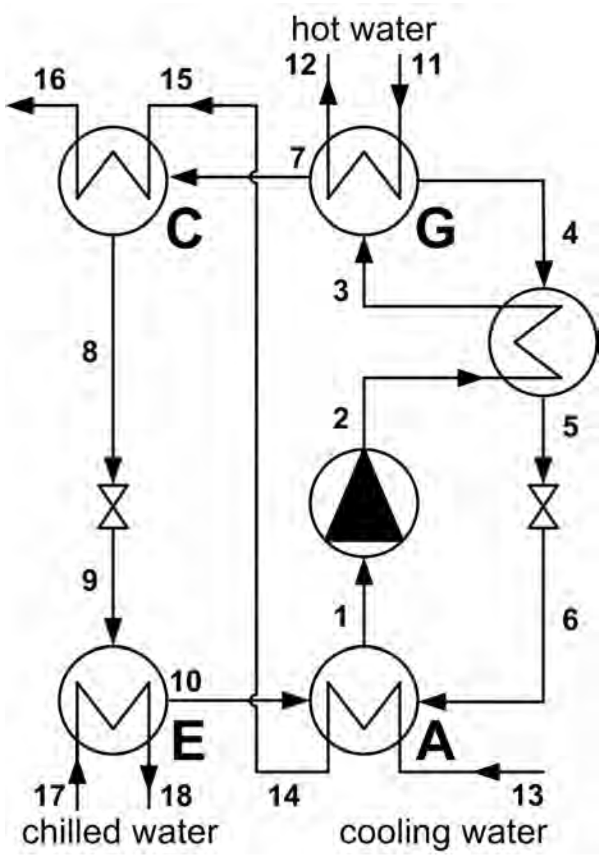


Figure 1. LiBr/water absorption cycle.

Shown are the four main components of an absorption chiller: generator (G), condenser (C), evaporator (E) and absorber (A). Also shown are solution heat exchanger, solution pump and the throttling valves between absorber/solution heat exchanger and condenser/evaporator. The internal state points characterize the thermodynamic properties of LiBr/water solution (points 1-6), water vapour (7, 10) and liquid water (8, 9). The external state points characterize the thermodynamic properties of hot water (11, 12), cooling water (13, 14, 15, 16) and chilled water (17, 18).

The input and output parameter of the model are given in the Appendix.

From Figure 1 the steady-state internal heat flows of the vessels are as follows.

Evaporator:
$$\dot{Q}_{E,\text{int}} = \dot{m}_v \cdot (h_{10} - h_8) \tag{1}$$

Condenser:
$$\dot{Q}_{C,\text{int}} = \dot{m}_v \cdot (h_7 - h_8) \tag{2}$$

Absorber:
$$\dot{Q}_{A,\text{int}} = \dot{m}_v \cdot h_{10} - \dot{m}_{sol,w} \cdot h_1 + (\dot{m}_{sol,w} - \dot{m}_v) \cdot h_5 \tag{3}$$

Generator:
$$\dot{Q}_{G,\text{int}} = \dot{m}_v \cdot h_7 + (\dot{m}_{sol,w} - \dot{m}_v) \cdot h_4 - \dot{m}_{sol,w} \cdot h_3 \tag{4}$$

In equations (1) to (4), $\dot{m}_{sol,w}$ is the mass flow of the diluted solution and $\dot{m}_v$ is the mass flow of the refrigerant water vapour which is of equal value at state points 7 and 10 in Figure 1. Equations (1) to (4) are based on the calculation of internal enthalpies for LiBr/water solution and refrigerant vapour which requires sound knowledge of these properties. For computational purposes this means that a property database of both LiBr/water and water vapour has to be implemented into the simulation model and usually requires time-consuming iterations.

The main aim of the model presented is to account for the dynamics. Therefore, no detailed steady-state simulation with all state points was performed but a simpler approach was taken which considers the most important physical properties only. This may be refined later on but is not intended to be the focus of this work. This kind of refinement will not change the basic findings of this report. So, to keep the model fast, equations (1) to (4) can be simplified using the latent heat of evaporation and sorption, *r* and *l*, as shown in equations (6) to (9). This approach has been proposed by Ziegler [4] to avoid a detailed enthalpy calculation for each state point. Please note that the latent heat is taken at condenser pressure. The dependency of latent heat on temperature is treated according to Plank's rule [4].

Moreover, not all internal temperatures in Figure 1 are being calculated in the model. Instead, each vessel is being modelled using external and internal mean temperatures of the respective fluids (heat carrier and working fluid) as well as a mean vessel temperature. External mean heat carrier temperatures $\overline{\vartheta}_X$ are being calculated using the arithmetic mean temperature of inlet and outlet temperature. Internal mean temperatures $\overline{T}_X$ are assumed to be the mean of the equilibrium temperatures of strong and weak solution and of the condensing or evaporating refrigerant. Mean vessel temperatures T_X are the arithmetic mean temperature of the external and internal mean temperatures.

The boiling temperature of saturated LiBr/water solution can be calculated using the Duehring relation, as described by Feuerecker [5]. In the model, this temperature is by definition the mean of the equilibrium temperatures of strong (or weak) solution, $\overline{T}_x$.

$$\overline{T}_x = A_d(X) + B_d(X) \cdot D \qquad (5)$$

with *X* being the mole fraction and D being the dew point temperature of the water vapour. A_d and B_d are the Duehring coefficients which depend on the solution concentration but are constant with regard to the dew point [5]. Some simplifying assumptions have been made for the model. These are:

(a) There is no heat loss to or gain from the ambient.

(b) The pump work is neglected.

(c) The LiBr/water solution leaving generator and absorber tube bundle is saturated.

(d) The external inlet temperatures t_{11}, t_{13}, t_{17} are used as control parameters.

(e) The absorber cooling water outlet temperature equals the condenser inlet temperature (t_{14}=t_{15}).

(f) The solution heat exchanger has a constant effectiveness η_{SHX}.

(g) Water and LiBr/water solution have constant property data.

(h) The external and internal heat transfer coefficients (UA-values) are constant.

(i) Evaporation and solution enthalpy are constant.

(j) Constant pumping rate: The mass flow of weak solution from absorber to generator is constant (state points 1, 2 and 3).

(k) The generator pressure equals the condenser pressure; the absorber pressure equals the evaporator pressure.

Equations (1) to (4) do not account for dynamic effects. One way of accounting for a more precise dynamic calculation is the introduction of different vapour mass flows for state points 7 and 10. The vapour mass flow from generator to condenser, $\dot{m}_{v,G}$ (state point 7), and from evaporator to absorber, $\dot{m}_{v,A}$ (state point 10), are therefore used in equations (6) to (9) which describe the simplified internal enthalpy calculations.

$$\dot{Q}_{E,\text{int}} = \dot{m}_{v,A} \cdot \left(r_{(pc)} - c_{p,v} \cdot \left(\overline{T}_C - \overline{T}_E \right) \right) \qquad (6)$$

$$\dot{Q}_{C,\text{int}} = \dot{m}_{v,G} \cdot \left(r_{(pc)} + c_{p,v} \cdot \left(\overline{T}_G - \overline{T}_C\right)\right) \quad (7)$$

$$\dot{Q}_{G,\text{int}} = \dot{m}_{v,G} \cdot \left(r_{(pc)} + l_{(pc)} + c_{p,w} \cdot \left(\overline{T}_G - \overline{T}_A\right)\right) + \dot{Q}_{SHX} \quad (8)$$

$$\dot{Q}_{A,\text{int}} = \dot{m}_{v,A} \cdot \left(r_{(pc)} + l_{(pc)} - c_{p,v} \cdot \left(\overline{T}_G - \overline{T}_E\right) + c_{p,w} \cdot \left(\overline{T}_G - \overline{T}_A\right)\right) + \dot{Q}_{SHX} \quad (9)$$

The solution heat exchanger does not exchange the maximum possible heat between its two flows. The difference between actual and ideal heat transfer can be assumed as a parasitic heat flow which has to be added to the generator and has to be removed at the absorber. For a constant heat exchanger effectiveness η_{SHX} it can be calculated as

$$\dot{Q}_{SHX} = \left(1 - \eta_{SHX}\right) \cdot \dot{m}_{sol,sG} \cdot c_{p,sol,s} \cdot \left(\overline{T}_G - \overline{T}_A\right). \quad (10)$$

In equation (10), $\dot{m}_{sol,sG}$ is the mass flow rate of the strong solution from generator to absorber [4].

The dynamic performance of the absorption chiller is influenced by various time-dependent effects, caused by complex heat transfer phenomena in the internal and external heat exchangers. In order to keep the model simple, not all of the dynamic effects have been taken into account. Only the three effects with the estimated biggest influence on chiller performance have been chosen. These include a time delay in the solution transport between generator and absorber, mass storage in the vessel sumps, and thermal storage in the external and internal heat exchangers. These dynamic terms are the backbone of the model and will be discussed in detail.

Dynamic Modelling

Both generator and absorber vessel have been modelled as a serial connection of a tube bundle heat exchanger and a solution sump. The tube bundle is the active part; the sump is a storage and mixing device. Solution can accumulate in absorber and generator sump according to the actual load, however there is also some solution which is always stored on the tube bundle. This hold-up, usually, is small as compared to the amount of liquid in the sump because the film is less than half a millimetre thick. Moreover, the bundle should be wetted all the time with the consequence that the amount of liquid on the bundle will not change significantly. Therefore the amount of liquid on the bundle is neglected.

Due to the storage effect, we have to distinguish between the solution flow entering the vessel, the one leaving the bundle and entering the sump, and the one leaving the sump. Moreover, we have to consider different concentrations at the inlet of the bundle, at the exit of the bundle dripping into the sump, and at the exit of the sump. Figure 2 shows the concentration and mass flow definitions, introducing time-discrete parameters (index *i*).

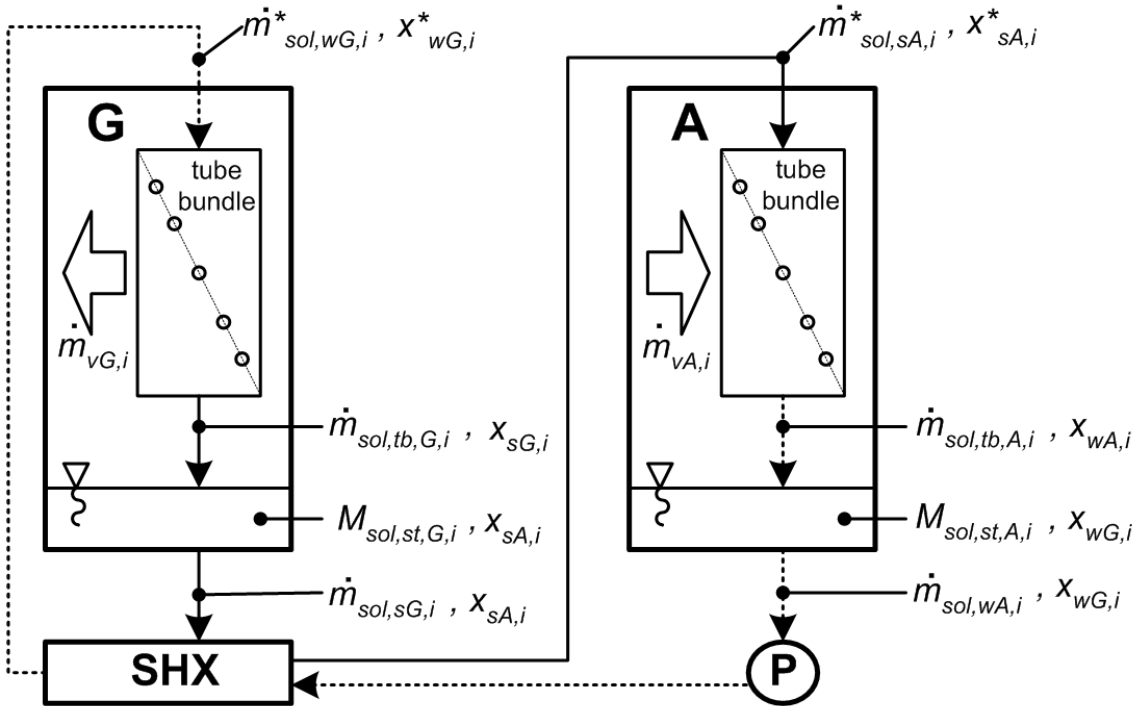


Figure 2. Definition of concentrations and mass flows in generator/absorber. Solid lines: strong solution, dotted lines: weak solution, white arrows: vapour.

Solution transport delay

A transport delay (c) is assumed to occur in both solution circuit legs. Referring to Figure 2, the inlet values of solution mass flow $\dot{m}^*_{sol,sA,i}$ and concentration $x^*_{sA,i}$ at the absorber at time interval *i* are assumed to equal the generator outlet values of time interval (*i*-c_1). The superscript * symbolizes the time-delayed arrival of the solution at absorber and generator inlet. By analogy, the inlet value of concentration $x^*_{wG,i}$ at the generator at time interval *i* is assumed to equal the outlet value of the absorber at time step (*i*-c_2). Constants c_1 and c_2 account for the time which the solution needs to flow from generator to absorber and vice versa. They are integers representing a number of simulation steps.

Absorber:

$$x^*_{sA,i} = x_{sA,i-c1} \tag{11}$$

$$\dot{m}^*_{sol,sA,i} = \dot{m}_{sol,sG,i-c1} \tag{12}$$

Generator:

$$x^*_{wG,i} = x_{wG,i-c2} \tag{13}$$

$$\dot{m}^*_{sol,wG,i} = \dot{m}_{sol,wA,i-c2} = const. \tag{14}$$

As stated in equation (14) above, the mass flow of weak solution is set constant. In equation (12) it is assumed that the mass flow entering the absorber tube bundle at time interval *i* equals the mass flow which has left the generator c_1 time intervals ago. Doing so, the physical incompressibility of the solution in the piping between generator and absorber is not modelled correctly. In reality, the mass flow entering the tube at time interval *i* should equal the mass flow leaving the tube at time interval *i*. The assumption, however, had to be made in order to achieve a correct salt balance in equation (22). The result is a hidden mass storage term in the solution tube. However, the amount of stored solution in the tube is only 0.6 % of the total amount of stored solution in the sumps [3]. The error introduced by this assumption is therefore negligible.

Mass balances

The vessel sumps (see Figure 2) are assumed to be fully mixed at each time interval. Thus vessel sump and solution leaving the vessel are assumed to have the same salt concentration, $x_{sA,i}$ and $x_{wG,i}$, respectively. The outlet of each tube bundle is assumed to exhibit the equilibrium concentration $x_{sG,i}$ and $x_{wA,i}$. The total solution mass stored in generator and absorber sump at time *i* is expressed by $M_{sol,st,G,i}$ and $M_{sol,st,A,i}$, respectively.

We know that the total mass (or salt mass, respectively) in the sump at time interval *i* equals the total mass (or salt mass) at time interval (*i-1)* plus the difference of ingoing and outgoing solution (or salt) flows at time interval *i*. The balances for the total solution contents in the sumps are given in equations (15) and (16). Δt is the time between two simulation intervals.

Generator:

$$\dot{m}_{sol,tb,G,i} - \dot{m}_{sol,sG,i} - \frac{M_{sol,st,G,i} - M_{sol,st,G,i-1}}{\Delta t} = 0 \qquad (15)$$

Absorber:

$$\dot{m}_{sol,tb,A,i} - \dot{m}_{sol,wA,i} - \frac{M_{sol,st,A,i} - M_{sol,st,A,i-1}}{\Delta t} = 0 \qquad (16)$$

The fraction term in equations (15 and 16) is the amount of solution which is added to the solution stored in either generator or absorber sump during the time Δt between consecutive simulation intervals *i* and (*i-1)*.

The salt flow balance, analogously, in both sumps can be written as:

Generator:

$$\dot{m}_{sol,tb,G,i} \cdot x_{sG,i} - \dot{m}_{sol,sG,i} \cdot x_{sA,i} - \frac{M_{sol,st,G,i} \cdot x_{sA,i} - M_{sol,st,G,i-1} \cdot x_{sA,i-1}}{\Delta t} = 0 \qquad (17)$$

Absorber:

$$\dot{m}_{sol,tb,A,i} \cdot x_{wA,i} - \dot{m}_{sol,wA,i} \cdot x_{wG,i} - \frac{M_{sol,st,A,i} \cdot x_{wG,i} - M_{sol,st,A,i-1} \cdot x_{wG,i-1}}{\Delta t} = 0 \qquad (18)$$

Proceeding from the sumps to the tube bundles, the mass flow balance yields

Generator: $$\dot{m}_{sol,wA,i-c2} - \dot{m}_{vG,i} - \dot{m}_{sol,tb,G,i} = 0 \quad (19)$$

Absorber: $$\dot{m}_{sol,sG,i-c1} + \dot{m}_{vA,i} - \dot{m}_{sol,tb,A,i} = 0 \quad (20)$$

The salt flow balances of the tube bundles read

Generator: $$\dot{m}_{sol,wA,i-c2} \cdot x_{wG,i-c2} - \dot{m}_{sol,tb,G,i} \cdot x_{sG,i} = 0 \quad (21)$$

Absorber: $$\dot{m}_{sol,sG,i-c1} \cdot x_{sA,i-c1} - \dot{m}_{sol,tb,A,i} \cdot x_{wA,i} = 0 \quad (22)$$

Pressure drop

The strong solution flow from generator to absorber is driven by gravity and a pressure difference. The pressure loss of the solution heat exchanger and its adjacent piping is assumed constant which is a good approximation as long as the flow variations are not too large; hence the mass flow of the strong solution depends only on the pressure difference between generator/absorber and the liquid solution column at heat exchanger inlet. This assumption, of course, can be refined in future works. During operation, the static pressure of the liquid column has to equal the dynamic pressure loss caused by flow restrictions through piping and solution heat exchanger. The mass flow of the strong solution can therefore be calculated as

$$\dot{m}_{sol,s,i} = A_t \cdot \sqrt{\frac{2 \cdot \rho_{sol,s} \cdot \left(p_{G,i} - p_{A,i} + \rho_{sol,s} \cdot g \cdot (h + z_i)\right)}{\varsigma}}\,, \quad (23)$$

where generator/condenser and evaporator/absorber pressure are calculated using the equation of Clausius-Clapeyron as shown in equation (24). There, the pressure and temperature values of solution interval *(i-1)* are being used to calculate the values of simulation interval *i*. This is a valid approximation as long as the time between two simulation intervals, Δt, is not too long.

$$\ln\left(\frac{p_{X,i}}{p_{X,i-1}}\right) = \frac{r_0}{R} \cdot \left(\frac{1}{\overline{T}_{X,i-1}} - \frac{1}{\overline{T}_{X,i}}\right) \quad (24)$$

Thermal storage

After a temperature change of one of the external flows entering a vessel, the temperatures of all components which are influenced either by the external heat carrier (external components such as the water headers) or by the refrigerant and sorbent (internal components such as the solution heat exchanger) change consecutively. This is, of course, connected with some heat being stored in all external and internal components which are involved in heat transfer mechanisms. The temperature change of these components is a complex three-dimensional heat transfer problem. One simplified approach to modelling it is the assumption that some components or parts of components follow internal temperatures and others follow external temperatures. To account for the accordingly different heat transfer rates the total thermal mass has been divided into an external and an internal part. The external part follows the mean external temperature $\overline{\vartheta}_X$, the internal part follows the mean internal temperature $\overline{T}_X$. Also, different heat transfer coefficients and exchange areas have been assumed for the internal and external parts. The external and internal components that have been incorporated for thermal storage are listed in Table 1.

Table 1. External and internal components involved in heat transfer.

Location	Component	Heat capacity [kJ/kgK]	Mass in E [kg]	Mass in C [kg]	Mass in G [kg]	Mass in A [kg]
external	Vessel wall (10% of total weight)	0.48	2.0	2.0	1.9	1.9
	External water header	0.48	7.6	7.3	9.8	10.6
	Water in external parts	4.19	8.5	7.4	12.6	11.9
	Heat exchanger tube bundles (50%)	0.38	5.7	5.1	8.9	8.0
internal	Heat exchanger tube bundles (50%)	0.38	5.7	5.1	8.9	8.0
	LiBr/water solution (50% each)	3.70	-	-	19.0	19.0
	Refrigerant water (50% each)	4.19	7.5	7.5	-	-
	Solution heat exchanger (50% each)	0.38	-	-	4.1	4.1
	Solution pump (50% each)	0.48	-	-	5.0	5.0
	Vessel walls (50% of total weight)	0.48	10.0	10.0	9.5	9.5

Figures 3 and 4 illustrate the assumptions for the heat transfer from the external heat carrier to the heat exchanger and further on to solution or refrigerant. For generator and absorber the tube bundle is divided into two virtual parts, one being at mean external temperature, $\overline{\vartheta}_X$, and one being at mean internal temperature, $\overline{T}_X$. The mean vessel temperature T_X is shown as a virtual crossover temperature between internal and external temperature levels.

Figure 3 shows the partition of the generator vessel into the external (right side) and internal (left side) part, Figure 4 shows the same partition for the absorber.

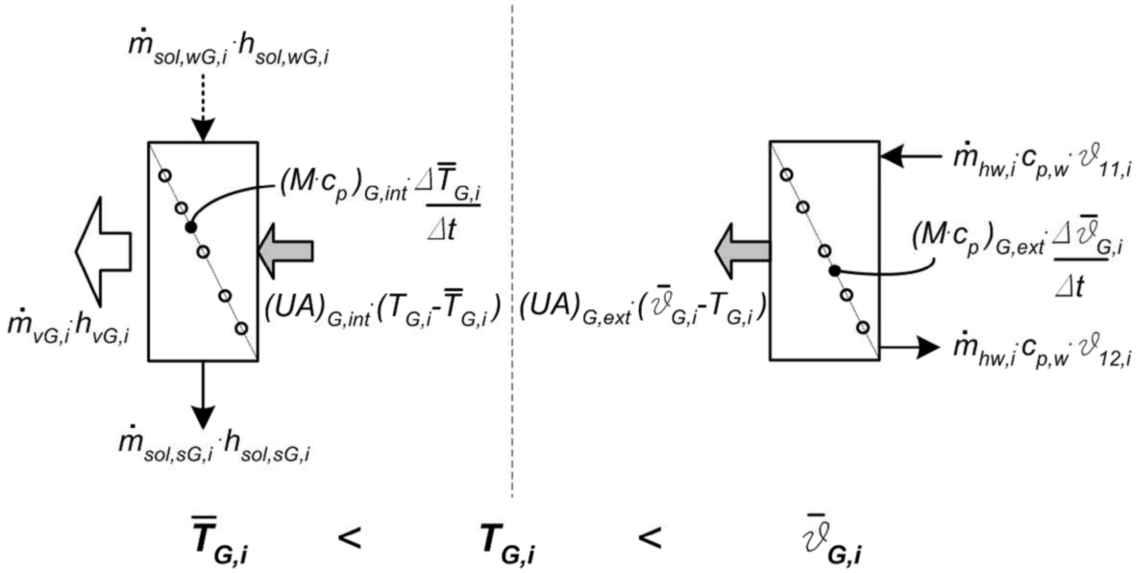


Figure 3. Enthalpy balance for generator vessel. Grey arrows: heat flow, white arrow: vapour flow, black arrows: fluid flow. Left side: internal, right side: external

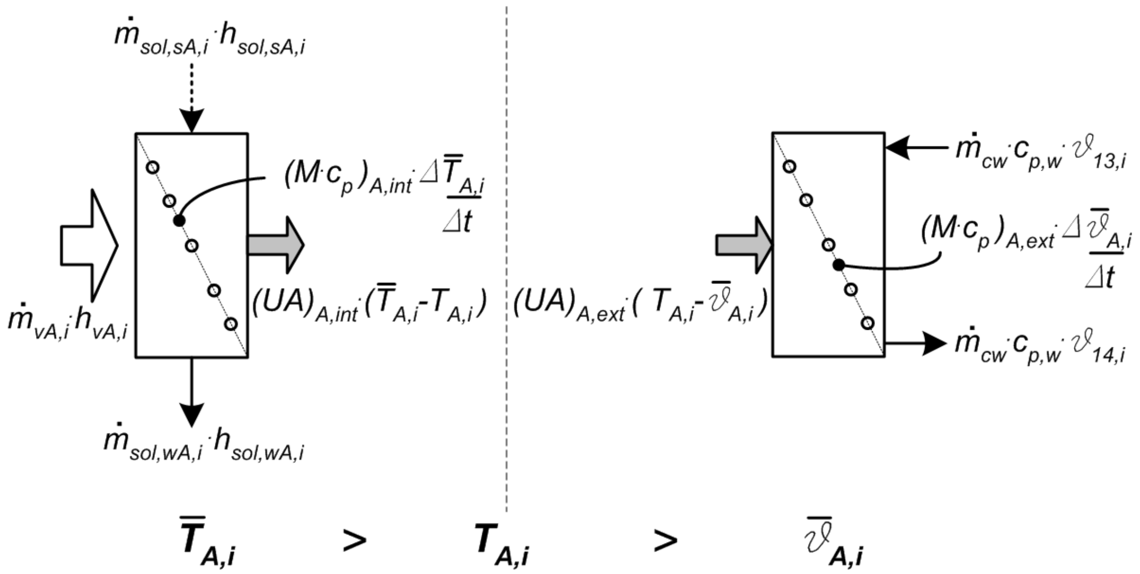


Figure 4. Enthalpy balance for absorber vessel. Grey arrows: heat flow, white arrow: vapour flow, black arrows: fluid flow. Left side: internal, right side: external

Externally, all four chiller vessels incorporate water headers for the distribution of hot, chilled and cooling water through the tube bundles. It is assumed that the headers and the water mass enclosed in those contribute completely to the external heat storage. The tube bundles are assumed to contribute by 50% of their mass to the external heat storage and by 50% to the internal heat storage. This is reasonable because the falling film heat exchanger tubes have flow on both sides, internally by LiBr/water solution and externally by water. The vessel wall is assumed to contribute by 10% of its total weight to the external thermal mass due to the conduction heat transfer from the headers.

Internally, in addition to the tube bundles, the solution contributes by 50% of its mass in generator and absorber. In analogy, the refrigerant is assumed to have a 50% internal contribution in both evaporator and condenser. The mass of solution heat exchanger and solution pump has an internal effect on both generator and absorber heat transfer and is assumed to contribute by 50% of its weight on both. The vessel wall now contributes internally by 50% of its weight due to the larger surface area involved in heat transfer mainly in the sumps.

Enthalpy balances

Taking internal and external thermal storage as in Figures 3 and 4 into account, equations (6) to (9) can be refined and the external and internal enthalpy balances for the vessels can now be expressed. The external enthalpy balance of the vessels describes that the heat which is given off (generator) or taken on (absorber) by the heat carrier (left-hand side of equation (25) or (26)) partly crosses the heat exchanger (first term on the right-hand side, grey arrows in Figures 3 and 4) and reaches the mean vessel temperature, partly is used to heat up parts of the vessel which are more or less at external temperature (second term on right-hand side). For generator and evaporator, the external enthalpy balance reads

$$\dot{m}_{X,w} \cdot c_{p,w} \cdot \left(\vartheta_{X,in} - \vartheta_{X,out}\right) = (UA)_{ext,X} \cdot \left(\overline{\vartheta}_{X,i} - T_{X,i}\right) + \left(M \cdot c_p\right)_{ext,X} \cdot \frac{\overline{\vartheta}_{X,i} - \overline{\vartheta}_{X,i-1}}{\Delta t} \qquad (25)$$

In equation (25), index *X* denotes the different vessels of generator and evaporator (*X* = *G, E*). For absorber and condenser the external enthalpy balance reads

$$\dot{m}_{X,w} \cdot c_{p,w} \cdot \left(\vartheta_{X,in} - \vartheta_{X,out}\right) = -(UA)_{ext,X} \cdot \left(\overline{\vartheta}_{X,i} - T_{X,i}\right) - \left(M \cdot c_p\right)_{ext,X} \cdot \frac{\overline{\vartheta}_{X,i} - \overline{\vartheta}_{X,i-1}}{\Delta t} \qquad (26)$$

Index *X* in equation (26) denotes absorber and condenser (*X* = *A, C*).

The internal enthalpy balances for all vessels read as follows.

$$\dot{m}_{v,A,i} \cdot \left(r_{(pc)} - c_{p,v} \cdot \left(\overline{T}_{C,i} - \overline{T}_{E,i}\right)\right) + \left(M \cdot c_p\right)_{E,\mathrm{int}} \cdot \frac{\overline{T}_{E,i} - \overline{T}_{E,i-1}}{\Delta t} = (UA)_{E,\mathrm{int}} \cdot \left(T_{E,i} - \overline{T}_{E,i}\right) \qquad (27)$$

$$\begin{aligned} &\dot{m}_{v,G,i} \cdot \left(r_{(pc)} + l_{(pc)} + c_{p,w,i} \cdot \left(\overline{T}_{G,i} - \overline{T}_{A,i}\right)\right) + \dot{Q}_{SHX,i} + \left(M \cdot c_p\right)_{G,\mathrm{int}} \cdot \frac{\overline{T}_{G,i} - \overline{T}_{G,i-1}}{\Delta t} \\ &= (UA)_{G,\mathrm{int}} \cdot \left(T_{G,i} - \overline{T}_{G,i}\right) \end{aligned} \qquad (28)$$

$$\dot{m}_{v,G,i} \cdot \left(r_{(pc)} + c_{p,v} \cdot \left(\overline{T}_{G,i} - \overline{T}_{C,i}\right)\right) + \left(M \cdot c_p\right)_{C,\mathrm{int}} \cdot \frac{\overline{T}_{C,i} - \overline{T}_{C,i-1}}{\Delta t} = (UA)_{C,\mathrm{int}} \cdot \left(T_{C,i} - \overline{T}_{C,i}\right) \qquad (29)$$

$$\begin{aligned} &\dot{m}_{v,A,i} \cdot \left(r_{(pc)} + l_{(pc)} - c_{p,v} \cdot \left(\overline{T}_{G,i} - \overline{T}_{E,i}\right) + c_{p,w} \cdot \left(\overline{T}_{G,i} - \overline{T}_{A,i}\right)\right) + \dot{Q}_{SHX,i} + \\ &\left(M \cdot c_p\right)_{A,\mathrm{int}} \cdot \frac{\overline{T}_{A,i} - \overline{T}_{A,i-1}}{\Delta t} = (UA)_{A,\mathrm{int}} \cdot \left(T_{A,i} - \overline{T}_{A,i}\right) \end{aligned} \qquad (30)$$

Equations (27) and (28) describe the internal heat transfer in evaporator and generator, respectively. There, the heat which is given off by the heat exchanger (right-hand side) partly heats up the internal parts of the vessel (second term on left-hand side) and partly is used for the heating of solution and refrigerant (first term on left-hand side). In analogy, equations (29) and (30) describe condenser and absorber, respectively. There, the heat which is given off by the solution and refrigerant (first term on left-hand side) is used partly to heat the internal parts of the heat exchanger (second term on left-hand side) and partly to heat the cooling water (right-hand side).

The equation system is being solved in MATLAB using a Newton-Raphson procedure with finite-difference Jacobian approximation [6].

Experimental Verification

The agreement between simulated and experimental data has been tested using an experimentally measured hot water input step from 75 to 85 °C. The results of the simulation using these input data are shown in Figure 5. The hot water inlet and outlet temperatures in Figure 5 are taken from experimental measurements of the Phoenix absorption chiller. They show the transient behaviour of the chiller for a 10K step in hot water inlet temperature. Cooling and chilled water inlet temperatures as well as all external mass flow rates were kept constant during the step. It can be seen that it takes approx. 600s or 10 minutes to achieve a new steady-state in hot water inlet temperature after the step. This is shorter than the 15 minutes it takes for all parameters to achieve steady-state again. The simulation was performed using the measured data as inputs in the model and comparing simulated and measured hot water outlet temperatures.

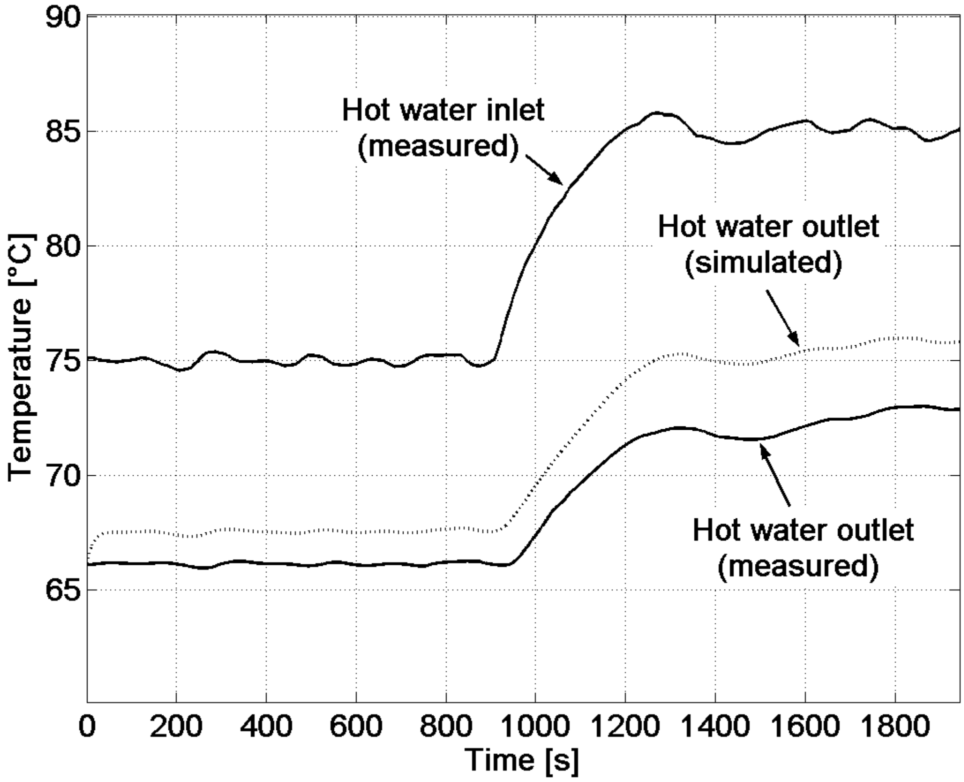


Figure 5. Comparison of simulated and measured generator outlet temperatures.

Figure 5 shows that the steady-state agreement between measured and simulated values is reasonably well despite the fact that steady-state accuracy has been incorporated coarsely only. At simulation start, the simulated temperatures diverge slightly from the initial measured state towards a different steady-state. In general, the temperature difference between experiment and simulation is larger after the step which shows that there is a dependency of the model output on the hot water temperature. This can be related to temperature-dependent parameters which have been assumed constant for the simulation, such as evaporation and solution enthalpy, density and specific heat capacity. As steady-state accuracy is not the focus in this paper this deviation shall not be discussed further.

Much more important is the fact that the dynamic agreement between measured and simulated output data is very good. Figure 5 shows that the fluctuations of simulated and measured hot water outlet temperature are well synchronized, although there is a slight delay of approximately 10 seconds of measured to simulated values. This has to be put into relation to the 600 seconds or 10 minutes that it takes to reach a steady state in the generator temperatures again.

The delay between simulation and experiment can – at least to some extent - be explained with the assumptions of thermal mass for the individual vessels in Table 1 which have not been verified yet. Another reason for the disagreement is the fact that some time delays existing in reality have not been incorporated into the model. These include the forced circulation of refrigerant in the evaporator by means of a second internal pump. Also, the condensed refrigerant flowing from condenser to evaporator has a time delay which has been neglected. There is no vapour storage in the vessels assumed. The transport delay c_1 has been assumed constant although it changes in proportion to the strong solution mass flow. The transport delay in the cooling water between absorber outlet and condenser inlet due to piping length has also been neglected as well as the change in solution storage on the tube bundles. These time delays have not been included in order to keep the model as simple as possible.

The results in Figure 5 are shown here just in order to demonstrate the basic functionality of the model. A sensitivity analysis along with more experimental data is presented in the second part of this paper [7].

Conclusions

In this paper a dynamic model for absorption chillers has been presented. It is based on internal energy balances as well as mass balances. Dynamic behaviour is implemented via eight thermal and two mass storage terms as well as by two delay times. The heat transfer has been divided into two parts: one which transfers heat from the external heat carrier to the exchanger material and the second from there on to the refrigerant or solution side, or vice versa. In this way the thermal mass of the vessel could be introduced easily. The agreement between experiment and simulation is very good with deviations of 10s for the generator. The total time to achieve a new steady-state after an input temperature step amounts to approximately 15 minutes. Compared to this, the present dynamic deviations are in the magnitude of approximately 1%.

More information about the simulation results, comparison with experiments and sensitivity checks will be presented in the second part of this paper [7].

Appendix

Inputs and outputs of the dynamic model

Table 2. Input parameter of the dynamic model.

No.	***Inputs***	***Unit***	***Description***
1	$\vartheta_{11,i}$	℃	*Generator hot water inlet temperature at time*
2	$\vartheta_{17,i}$	℃	*Evaporator chilled water inlet temperature at*
3	$\vartheta_{13,i}$	℃	*Absorber cooling water inlet temperature at time*
4	$\vartheta_{18,i-1}$	℃	*Evaporator chilled water outlet temperature at*
5	$\vartheta_{16,i-1}$	℃	*Condenser cooling water outlet temperature at*
6	$\vartheta_{12,i-1}$	℃	*Generator hot water outlet temperature at time*
7	$\vartheta_{14,i-1}$	℃	*Absorber cooling water outlet temperature at*
8	$\overline{T}_{E,i-1}$	℃	*Internal mean temperature of evaporator at time*
9	$\overline{T}_{C,i-1}$	℃	*Internal mean temperature of condenser at time*
10	$\overline{T}_{G,i-1}$	℃	*Internal mean temperature of generator at time*
11	$\overline{T}_{A,i-1}$	℃	*Internal mean temperature of absorber at time*
12	$x_{sol,w,G,i-c2}$	$kg_{Salt}kg_{Sol}^{-}$	*Solution concentration of absorber sump at time*
13	$x_{sol,s,A,i-c1}$	$kg_{Salt}kg_{Sol}^{-}$	*Solution concentration of generator sump at time*
14	$\dot{m}_{sol,s,i-c1}$	kgs^{-1}	*Strong solution mass flow at time interval i-c1*
15	$x_{s,G,i-1}$	$kg_{Salt}kg_{Sol}^{-}$	*Equilibrium concentration in generator at time*
16	$x_{s,A,i-1}$	$kg_{Salt}kg_{Sol}^{-}$	*Solution concentration of generator sump at time*
17	$x_{w,G,i-1}$	$kg_{Salt}kg_{Sol}^{-}$	*Solution concentration of absorber sump at time*
18	$x_{w,A,i-1}$	$kg_{Salt}kg_{Sol}^{-}$	*Equilibrium concentration in absorber at time*
19	$M_{st,sol,G,i-1}$	*kg*	*Total mass in generator sump at time interval i-1*
20	$M_{st,sol,A,i-1}$	*kg*	*Total mass in absorber sump at time interval i-1*
21	Δt	*s*	*Time period between time intervals i and i-1*

Table 3. Output parameter of the dynamic model.

No.	***Outputs***	***Unit***	***Description***
1	$\vartheta_{18,i}$	°C	*Evaporator chilled water outlet temperature at time*
2	$\vartheta_{16,i}$	°C	*Condenser cooling water outlet temperature at time*
3	$\vartheta_{12,i}$	°C	*Generator hot water outlet temperature at time*
4	$\vartheta_{14,i}$	°C	*Absorber cooling water outlet temperature at time*
5	$\overline{T}_{E,i}$	°C	*Internal mean temperature of evaporator at time*
6	$\overline{T}_{C,i}$	°C	*Internal mean temperature of condenser at time*
7	$\overline{T}_{G,i}$	°C	*Internal mean temperature of generator at time*
8	$\overline{T}_{A,i}$	°C	*Internal mean temperature of absorber at time*
9	$\dot{m}_{v,G,i}$	kgs^{-1}	*Vapour mass flow from generator to condenser at*
10	$\dot{m}_{v,A,i}$	kgs^{-1}	*Vapour mass flow from evaporator to absorber at*
11	$x_{s,G,i}$	$kg_{Salt}kg_{Sol}^{-}$	*Equilibrium concentration in generator at time*
12	$x_{s,A,i}$	$kg_{Salt}kg_{Sol}^{-}$	*Solution concentration of generator sump at time*
13	$x_{w,A,i}$	$kg_{Salt}kg_{Sol}^{-}$	*Equilibrium concentration in absorber at time interval*
14	$x_{w,G,i}$	$kg_{Salt}kg_{Sol}^{-}$	*Solution concentration of absorber sump at time*
15	$\dot{m}_{sol,tb,G,i}$	kgs^{-1}	*Solution mass flow after tube bundle in generator*
16	$\dot{m}_{sol,tb,A,i}$	kgs^{-1}	*Solution mass flow after tube bundle in absorber*
17	$m_{st,sol,G,i}$	*kg*	*Stored solution in generator sump at time interval i*
18	$m_{st,sol,A,i}$	*kg*	*Stored solution in absorber sump at time interval i*
19	z_i	*m*	*Height of solution level in generator sump at time*
20	$\dot{m}_{sol,s,i}$	kgs^{-1}	*Strong solution mass flow at time interval i*
21	$p_{G,i}$	*Pa*	*Generator/condenser pressure at time interval i*
22	$p_{A,i}$	*Pa*	*Evaporator/absorber pressure at time interval i*
23	$M_{st,sol,G,i}$	*kg*	*Total solution mass in generator at time interval i*
24	$M_{st,sol,A,i}$	*kg*	*Total solution mass in absorber at time interval i*

References

[1] Bian J, Radermacher R, Moran D. Transient simulation of an absorption chiller in a CHP system. In Proceedings of the International Sorption Heat Pump Conference, June 22-24, *2005*, Denver, CO, USA.

[2] Jeong S, Kang BH, Karng SW. Dynamic simulation of an absorption heat pump for recovering low grade waste heat. *Applied Thermal Engineering 1998*; 18 (1-2); 1-12

[3] Kohlenbach P. Solar cooling systems with absorption chillers: Control strategies and transient chiller performance. 2006; PhD Thesis; Technical University of Berlin, Forschungsberichte des Deutschen Kälte- und Klimatechnischen Vereins; Nr. 74, Erding, Germany

[4] Ziegler F. Sorptionswärmepumpen. 1997; Habilitation Thesis; Forschungsberichte des Deutschen Kälte- und Klimatechnischen Vereins; Nr. 57, Erding, Germany.

[5] Feuerecker G. Entropieanalyse für Wärmepumpensysteme: Methoden und Stoffdaten. 1994; PhD thesis; Faculty of Physics; Technical University of Munich; Germany.

[6] Urroz GE. Solution of non-linear equations. 2004; Lecture handout; Utah State University; U.S.A.

[7] Kohlenbach P, Ziegler F. A dynamic simulation model for transient absorption chiller performance. Part II: Numerical results and experimental verification. 2007; *Int J Refrigeration*, this issue.

2. A dynamic simulation model for transient absorption chiller performance: Numerical results and experimental verification

This section describes the performance and experimental verification of a dynamic absorption chiller model. In [1] the model itself was described with regard to dynamic effects, such as transport delays in the solution circuit, thermal storage and mass storage. In detail, the size of the solution sumps in absorber and generator, the time for the solution to flow from absorber to generator and vice-versa and the thermal mass of the main components has been accounted for. As a special feature, the thermal mass of the components has been split into two parts, one which responds to the temperature of the external fluids, and the other which responds to the temperature of the solution and the refrigerant (internal fluids). These are the main parameters which determine the dynamic behaviour of the chiller.
This second section is looking at internal consistency, sensitivity and accuracy of the model. Results of a performance analysis using ideal conditions to prove correct model behaviour are shown. A sensitivity analysis on thermal storage and solution transport delay has been performed to investigate the influence of the dynamic parameters on the chiller performance. Finally, a model verification using experimental results is also given in this paper.

Nomenclature

Symbols

A	area, (m^2)
A	Duehring factor (deg C)
B	Duehring factor (-)
c	specific heat capacity ($kJkg^{-1}K^{-1}$)
c	number of simulation steps representing time constants for transport delay (-)
D	dew point temperature (deg C)
g	gravity constant (Nm^2kg^{-2})
h	height difference between generator outlet and absorber inlet (m)
h	enthalpy ($kJkg^{-1)}$
l	specific heat of solution ($kJkg^{-1}$)
$m, \dot{m}$	mass flow rate (kgs^{-1})
M	mass (kg)
p	pressure (Pa)
$\dot{Q}, Q$	heat flux (kW)
r	evaporation enthalpy ($kJkg^{-1}$)
R	gas constant for water vapour (Jkg^{-1})
T	temperature (deg C)
t	time (s)
UA	heat transfer coefficient (kWK^{-1})
x	Solution mass fraction ($kg_{Salt}kg_{Sol}^{-1}$)
X	mole ratio (-)
z	solution level in generator sump (m)

Greek letters

η	effectiveness (-)
ρ	density (kgm^{-3})
Δ	difference (-)
ϑ	temperature (deg C)

Subscripts

A	Absorber
Acc	Cooling water inlet
Ach	Cooling water outlet
C	Condenser
d	Duehring
E	Evaporator
Eh	Chilled water inlet
Ec	Chilled water outlet
ext	external
G	Generator
Gc	Hot water outlet
i	simulation time interval
in	inlet
int	internal
meas	measured
p	pump
p, pc	at constant pressure
s	strong
sim	simulated
sol	solution
st	storage
SHX	solution heat exchanger
sG	strong solution leaving the generator tube bundle
sA	strong solution leaving the generator sump and entering the absorber
t	tube
tb	tube bundle
out	outlet
v	vapour
w	water, weak
wA	weak solution leaving the absorber tube bundle
wG	weak solution leaving the absorber sump and entering the generator
X	general index for vessels (X=A, C, E, G)

Performance analysis

The internal consistency of the model can be analysed by applying a step change to one of the external parameters in the model. A step from 75°C to 85°C in the hot water inlet temperature has been used for this purpose. Cooling and chilled water inlet temperatures have been kept constant at 27°C and 18°C, respectiv ely. The simulation interval was 1s. The temperature step was set at 200s after simulation start. This time period is necessary because the preset steady-state is not exactly met by the model: after starting the simulation first the steady state with the given input values has to be reached. 200s are enough in order to equalize the differences between initial and steady-state values.

For the consistency analysis no thermal storage terms were assumed, just solution mass storage and a transport delay of 2 steps or seconds between generator and absorber and 1 step or second the other way. These very short transport delays were chosen to speed up the simulation. Figure 1 shows some internal variables for a short time period before and after the temperature step. Figure 2 shows the external heat flows before and after the temperature step. Each marker symbol in the graph denotes one simulation step.

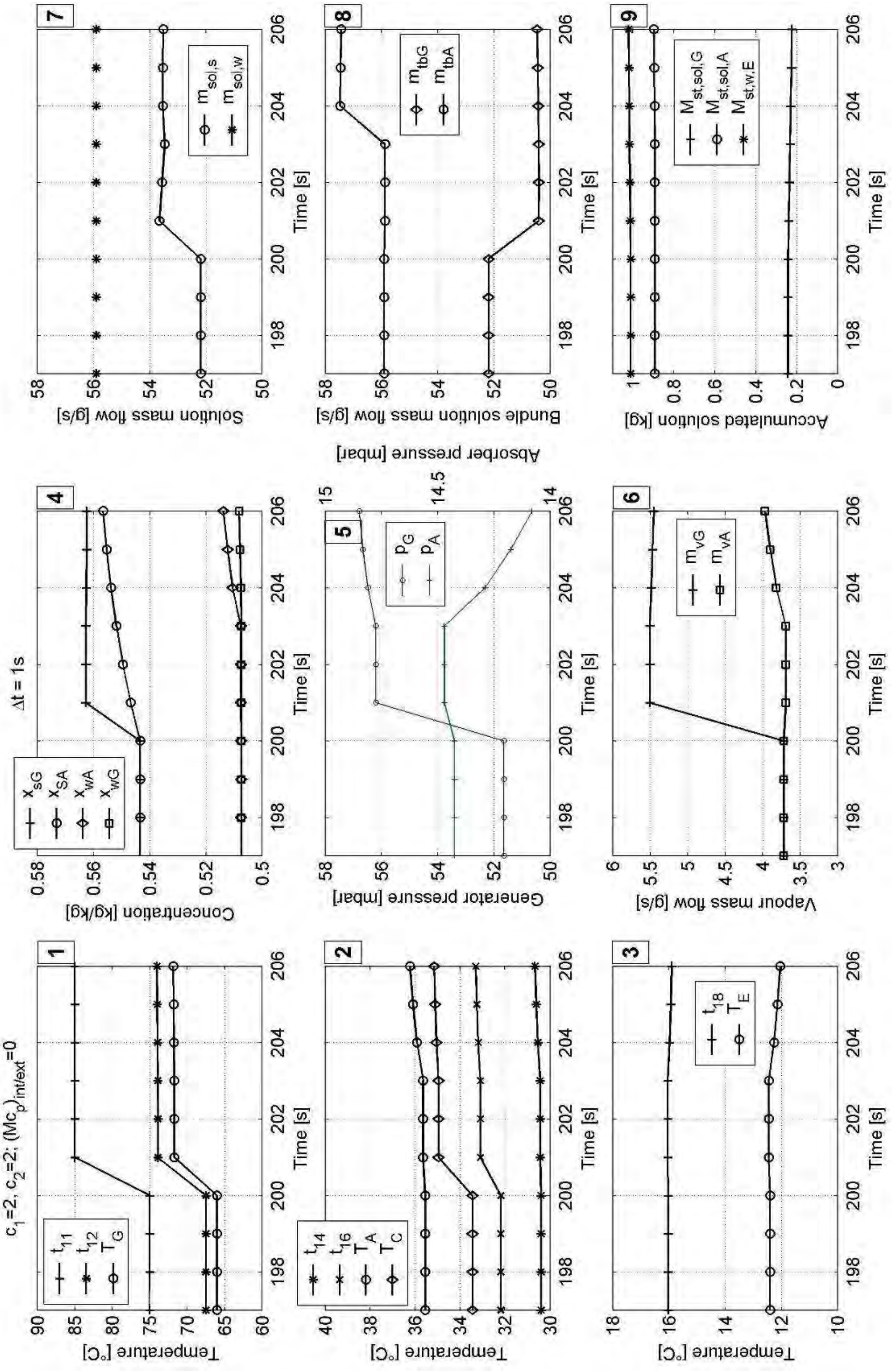


Figure 1. Simulated chiller response on 10K step in generator inlet temperature without thermal mass.

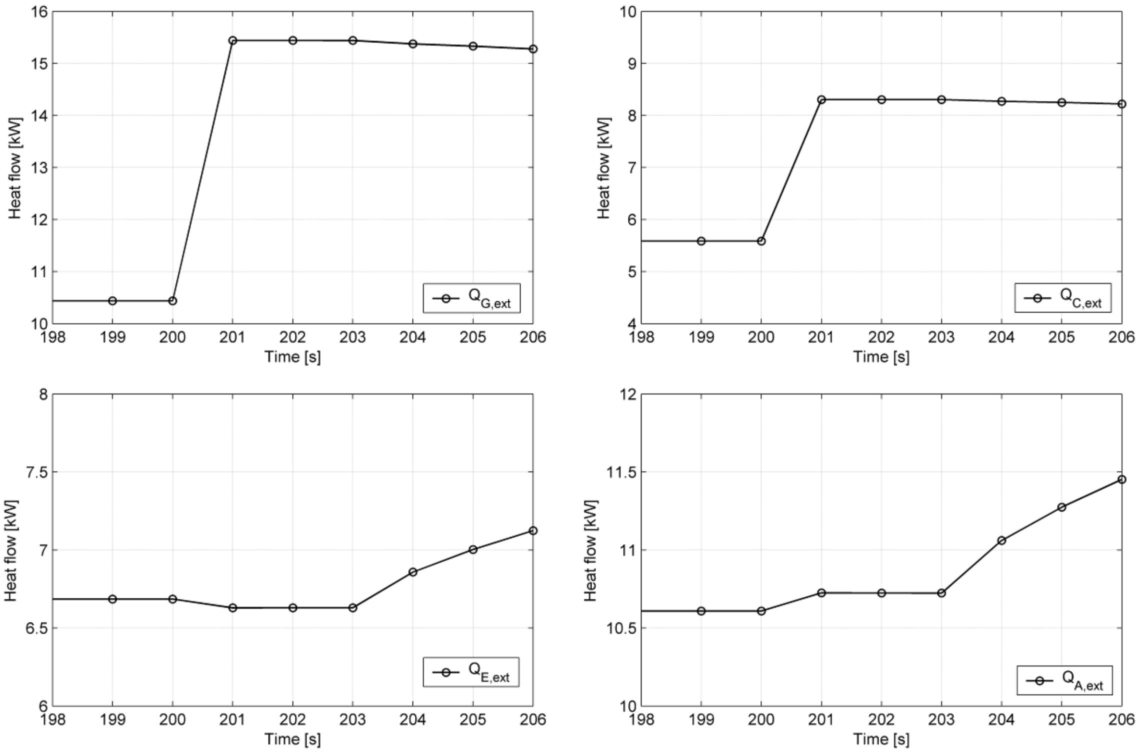


Figure 2. External heat flows of simulated chiller response on 10K step in generator inlet temperature without thermal mass

In Figure 1, graph 1 shows the temperatures in the generator which follow immediately the step of the external inlet temperature, t_{11}, because no thermal storage has been assumed. Accordingly, the heat flow to the generator increases (Figure 2, top left graph). The strong solution leaving the generator bundle follows immediately (x_{sG} in graph 4 of Figure 1). The strong solution leaving the generator sump follows in a damped fashion (x_{sA} in graph 4) because of the mass storage in the sump. The first change in mass fraction of the strong solution reaches the absorber inlet two steps later. Three steps after the initial change the mass fraction of the weak solution leaving the absorber bundle begins to change as well (x_{wA} in graph 4); the solution in the sump follows as well (x_{wG} in graph 4) but this is hardly visible due to the damping effect of the mass which is stored in the sump.
The flow of weak solution is kept constant ($m_{sol,w}$ in graph 7). The flow of strong solution increases immediately ($m_{sol,s}$ in graph 7). This due to the rise in generator pressure (p_G in graph 5), which drives more solution through the expansion valve. The rise in pressure, in turn is a consequence of the rise in the condenser heat flow (Figure 2, top right graph) due to the rise of vapour mass flow to the condenser (m_{vG} in graph 6). The flow of strong solution from the generator bundle to the generator sump (m_{tbG} in graph 8) decreases because the inlet flow is constant and more vapour is being produced. Of course, also the mean generator and condenser temperature (T_G in graph 1 and T_C in graph 2), and the condenser cooling water outlet temperature (t_{16} in graph 2) increases.
When the more concentrated solution reaches the absorber in step 204, the pressure immediately falls (p_A in graph 5). The absorber absorbs more vapour, the vapour flow from evaporator to absorber increases (m_{vA} in graph 6 of Figure 1), the flow from the absorber bundle to the sump increases (m_{tbA} in graph 8), and the absorber heat increases (Figure 2, bottom right graph). Accordingly, the temperature of the cooling water leaving the absorber increases (t_{14} in graph 2). At the same time, the evaporator heat increases (Figure 2, bottom

left graph) and the evaporator temperature as well as the chilled water outlet temperature decreases (T_E in graph 3 of Figure 1 and t_{18} in graph 3).
Summarising Figures 1 and 2, the model predicts the performance of the absorption chiller after an input temperature change with very good consistency and logic. However, there is a small inaccuracy in the model due to the way the solution heat exchanger is being modelled [1]. With the assumed transport delay in the solution circuit, no changes of absorber and evaporator parameters should occur during the first 2s after the temperature step. There is, however, a very small change in the equilibrium mass fraction of the absorber at time interval 201, barely visible in Figure 1 (x_{wA} in graph 4). This also results in small changes of the vapour mass flow from evaporator to absorber (m_{vA} in graph 6), the evaporator equilibrium temperature (T_E in graph 3) and the absorber pressure (p_A in graph 5).
The reason is as follows. In time interval 201, the increase of strong solution mass flow and equilibrium temperature in the generator results in an immediate increase of the solution heat exchanger heat flow as well which, in reality, would be dampened due to the thermal mass of the solution heat exchanger. Therefore, more heat has to be dissipated to the cooling water in the absorber, as visible in Figure 2 (bottom right graph). In consequence, the absorber pressure rises and less vapour can be absorbed, resulting in an equilibrium concentration change in the absorber and a temperature rise in the evaporator. The order of magnitude of these deviations is however very small and can be neglected for the overall model performance. The concentration changes by a value of 0.0002 kg/kg, the equilibrium temperature in the evaporator changes by 0.017K. This results in a vapour mass flow change of 2 10^{-5} kg/s.

Sensitivity analysis

The sensitivity of the model on its dynamic terms is important information for the model user as well as for developers of chillers. In the following, the influence of thermal storage and solution transport delay on the overall model performance will be investigated. Characterising parameters for the thermal storage are the values of internal and external heat capacity, $\left(Mc_p\right)_{X,\text{int}}$ and $\left(Mc_p\right)_{X,ext}$. The transport delay is characterised by the transport delay constants c_1 and c_2.

Thermal storage variations

To investigate the influence of thermal storage variations, three different cases have been simulated. In the first case, the actual thermal capacity of the absorption chiller on which the model is based is used. See Table 1 for details. The second case assumes 50% of the actual thermal capacity and in the third case no thermal capacity is assumed. In all cases, the external capacity equals the internal capacity and the actual solution transport delays are utilized, i.e. 67s and 61s for c_1 and c_2, respectively. Again a hot water temperature step from 75℃ to 85℃ at 200 s after simulation start has be en used as the model input for all cases. Figure 3 shows the results, expressed by the heat flow from the external heat transfer fluid (the water to be chilled) into the evaporator.

Table 1. Assumed heat capacities for the model.

	Heat capacities, $(Mc_p)_{X,\text{int}}$ and $(Mc_p)_{X,ext}$							
	E,ext	**A,ext**	**G,ext**	**C,ext**	**E,int**	**A,int**	**G,int**	**C,int**
	kJ/K	kJ/K	kJ/K	kJ/K	kJ/K	kJ/K	kJ/K	kJ/K
100%	42	59	62	38	42	59	62	42
50%	21	30	31	19	21	30	31	21
0%	0	0	0	0	0	0	0	0

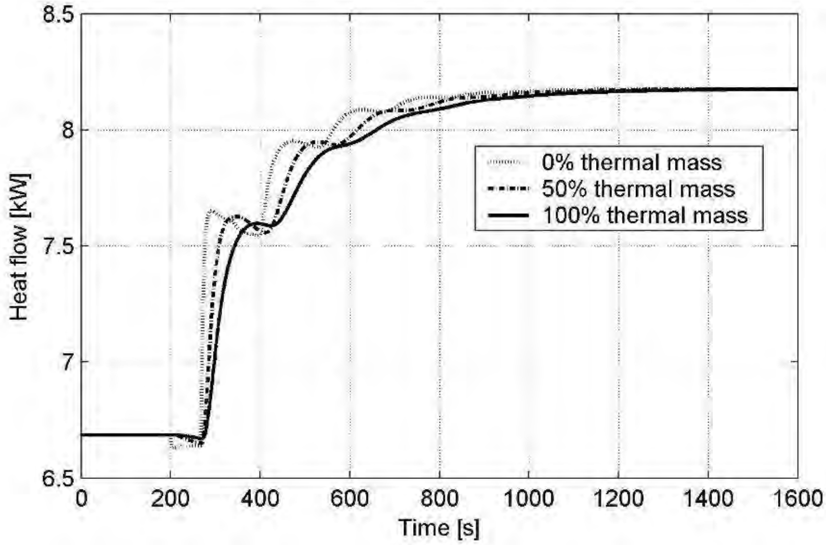


Figure 3. Evolution of evaporator heat flow for different thermal masses.

It can be seen in Figure 3 that the influence of the heat capacity is twofold. A lower capacity value results in a faster time until steady-state but also in bigger variations during the transient period. The first effect, the faster response, is rather obvious, as less thermal mass has to be heated or cooled in order to reach the new steady-state. This can be seen in more detail in Figure 4, where the time period around the step of Figure 3 is shown in enlargement.

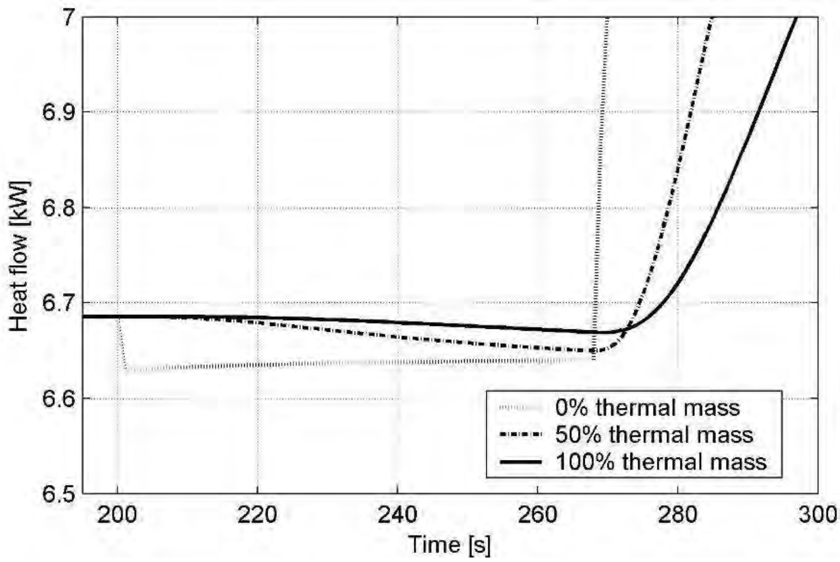


Figure 4. Detail of Figure 3: Time period around step only.

It is clearly visible in Figure 4 that an increase in thermal mass results in a slower response to the step change. Without thermal mass, the heat flow decreases abruptly after the step. This is equivalent to Figure 2 (bottom left graph) where the evaporator heat flow decreases for the time period between the step and the elapsing of time constant c_1. The lower evaporator heat flow is due to the increased heat flow in the solution heat exchanger just after the step which in turn results in increased absorber pressure, an equilibrium concentration change in the absorber, less absorbed vapour and hence a temperature rise in the evaporator. The evaporator heat flow increases sharply again after time constant c_1 has elapsed at time interval 267, i.e. after the change in solution mass fraction reaches the absorber. For the other two cases with higher thermal mass the changes are much less abrupt and the transients are not as steep.
The second effect visible in Figure 3 is bigger variations during the transient period for less thermal mass. This is related to the transport delay in the solution circuit. There, less mass results in more abrupt variations of the heat flow, induced by sharp changes of the solution parameters after either c_1 or c_2 have elapsed. An increase in both internal and external thermal mass provides a damping effect on the heat flow oscillations in the chiller. Regardless of the assumed thermal mass, the model still converges into the same steady-state value after the step.

For Figures 3 and 4, it has been assumed that the external capacity equals the internal capacity. This does not give any information on the influence of either of those capacities by itself and whether the assumption of equal capacities is reasonable. Therefore, another simulation has been performed assuming 100% capacity either only on the external or on the internal side. Figures 5 and 6 show the results for external and internal heat capacity only. As before, a hot water temperature step from 75°C to 85°C at 200s after simulation start has been used as the model input.

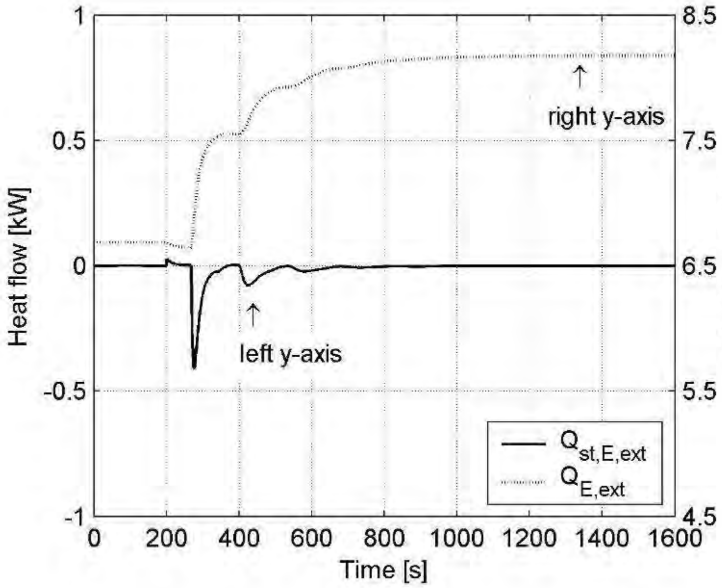


Figure 5. Evolution of evaporator heat flow for externally coupled thermal mass only.

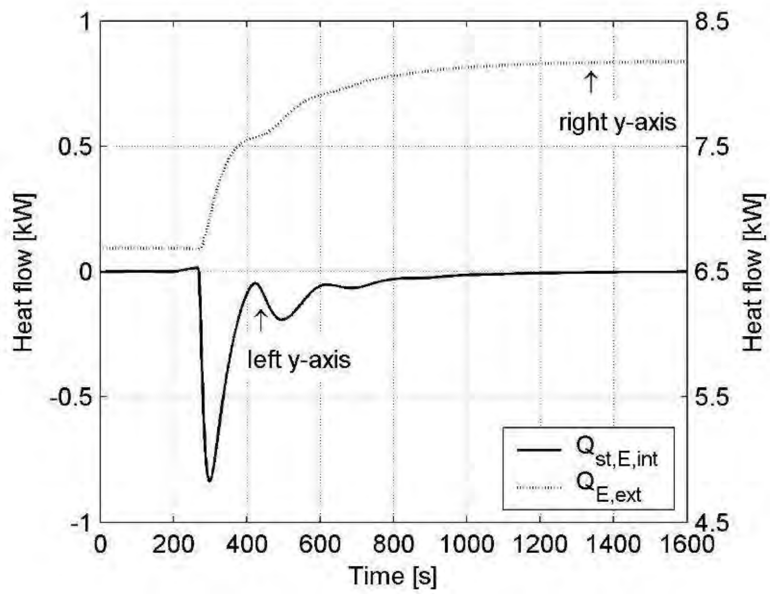


Figure 6. Evolution of evaporator heat flow for internally coupled thermal mass only.

Figure 5 shows the external heat flow from the chilled water, $Q_{E,ext}$, and the heat flow into the external thermal mass, $Q_{st,E,ext}$. The difference between both is the heat flow into the refrigerant. In the first seconds after the step, the temperature of the evaporator rises somewhat due to the increased absorber pressure as previously explained. The external heat flow drops, some heat is stored in the external mass. At 67s after the step (elapsing of c_1) a sharp negative peak can be seen in the heat flow which is stored in the external mass: the evaporator cools down because more absorption takes place as the change in solution mass fraction reaches the absorber. Consequently the external heat flow into the evaporator increases with a steep gradient. In parallel to this process, solution with the initial weak concentration from the absorber reaches the generator at 61s after the step (elapsing of c_2). Therefore the equilibrium concentration in the generator sump decreases due to the mixing. This slightly weaker solution leaves the generator sump and reaches the absorber with a time difference of 6s (Δc_{1-2}). It increases the absorber pressure and decreases the evaporator heat flow, visible in Figure 5 at 128s after the step where the evaporator heat flow plateaus at constant level. At approx. 200s after the step new stronger solution reaches the absorber from the generator, the absorber pressure decreases and the evaporator heat flow increases again. The gradient is, however, less steep than the first time due to the increased mixing in both generator and absorber sump and the corresponding equalisation of solution mass fraction. After this second increase another plateau can be observed in the evaporator heat flow at approx. 260s after the step. This process of alternating solution concentration changes is repeated a few more times with less and less intensity until at 500 seconds after the initial step the heat flow out of the thermal mass has stopped almost completely. A new steady-state is reached at 1200s after the initial step.

Figure 6 shows a similar evolution but with allocation of the thermal mass to the internal evaporator temperature. The behaviour is very similar, but there is an influence on the time to reach a new steady-state and also on the course of the heat flow after the step. Figure 6 shows that allocating the thermal mass to the internal parts results in a slower response with approx. 1400s to reach the new steady-state. Even the cooling down of the evaporator takes about 1000 seconds. The influence of the solution transport delay is much stronger, as the internal mass now includes the solution, the solution heat exchanger, and the solution pump

[1]. Thus the heat flow transient is less steep than in Figure 5 and the plateaus are less distinct.

Comparing Figures 5 and 6 it can be observed that allocating the thermal mass to the external parts as in Figure 5 yields a faster response with regard to achieving a new steady-state. The dynamic course of Figure 5 clearly shows obvious plateaus created by the transport delay in the solution circuit. As no internal thermal mass has been allocated, the solution changes occur quite rapidly after the step and then become weaker and are dampened with time. The course in Figure 6 is much less pronounced and the gradients of heat flows are less steep. The model behaviour shown in Figures 5 and 6 demonstrates that in order to attain a short response time the absorption chiller should be designed with possibly small internal thermal mass, however this will result in stronger variations during the transient period. Internally coupled thermal mass is more important than externally coupled thermal mass.

Transport delay variations

A second analysis has been performed to determine the sensitivity of the model results on time constants c_1 and c_2 which represent the transport time of the solution from generator to absorber and vice versa. The time constants have been set to 0, 50 and 100% of their original values which have been estimated from the design of the experimental plant. Table 2 shows the time constants used. For these simulations, the same hot water inlet step from 75℃ to 85℃ as before has been applied. Also, no t hermal mass has been assumed. Figure 7 shows the results, again expressed using the external heat flow to the evaporator.

Table 2. Assumed transport delays for the model.

	Transport delay	
	c_1	**c_2**
100%	67	61
50%	33	30
0%	2	1

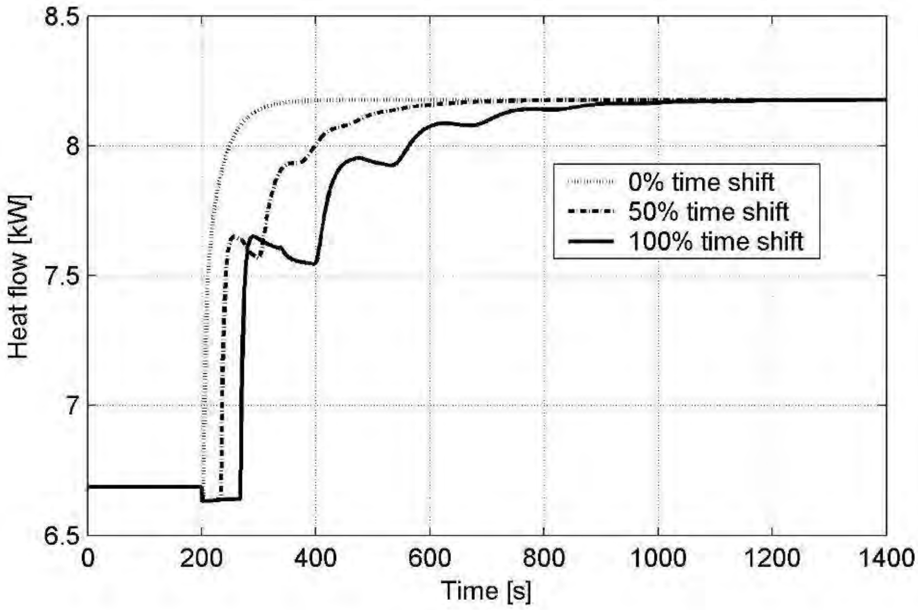


Figure 7. Evolution of evaporator heat flow for different transport delay times.

It is visible that the transport delay has a significant influence on the model performance. Without a transport delay, the new steady-state is reached at approx. 200s after the step. With a transport delay of 50% (which corresponds to about half a minute) it takes approx. 500s, e.g. more than double the time to reach it. For the assumed 100% of transport delay (around 1 minute) it takes approx. 900s or more than four times as long.

Also, with 100% transport delay, the transient period is not only longer but also quite different in its course. After the elapsing of each transport delay constant, a change in heat flow can be observed in a repeated pattern. The pattern consists of a small change and a larger change following each other which is due to the repetitive change in solution mass fraction as previously explained. The changes become less intensive with time and after approx. 12 delay times the new steady-state is reached. In a real absorption chiller the transport delay is determined by the flow rate of the solution. The model results in Figure 7 show that a significantly faster return to steady-state can be achieved if the flow rate of the solution is high and the transport delays are small.

Experimental verification

In Part 1 of this paper, we showed a first comparison of the simulated generator outlet temperature using experimentally measured inlet and outlet temperatures to measured data [1]. In this second part the comparison is also done similarly for the evaporator and absorber. Flow rates and inlet temperatures of the external flows are taken as input data to the simulation model. All internal parameters, the heat flows, and the outlet temperatures of the external flows are simulated. Figure 8 shows measured inlet and outlet temperatures of the chilled water together with the simulated outlet temperature; Figure 9 shows the same for the cooling water through absorber/condenser. The constant simulation parameters for the experimental verification can be found in the Appendix.

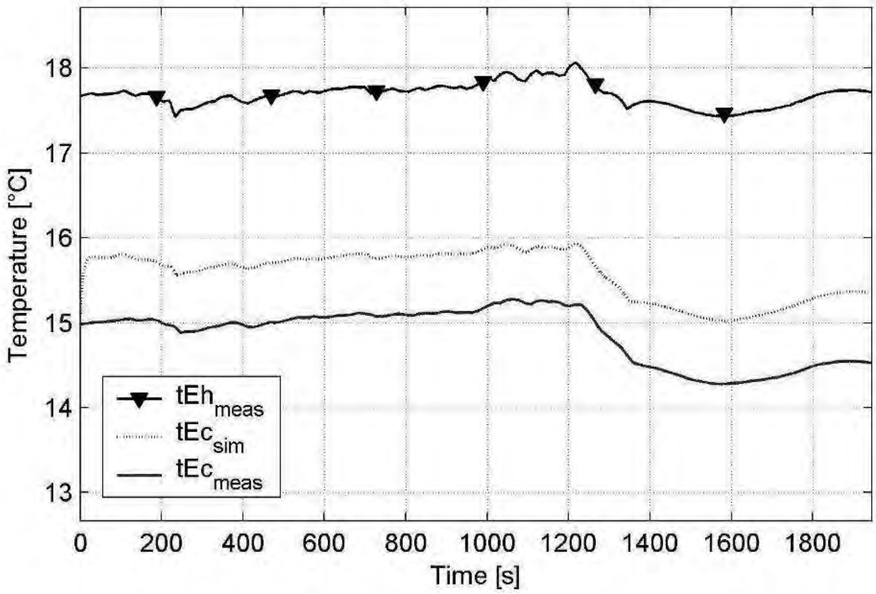


Figure 8. Simulated external evaporator temperatures using experimentally measured input data.

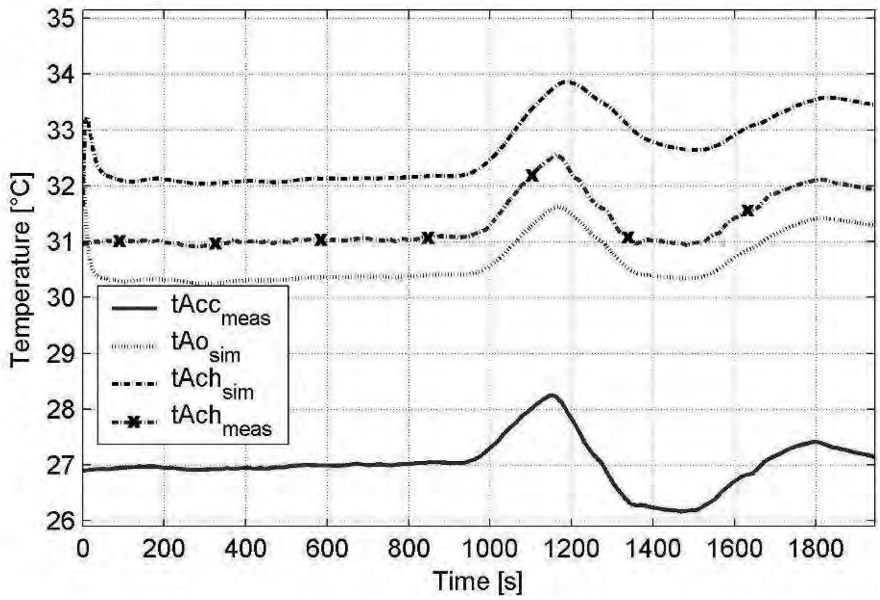


Figure 9. Simulated external absorber temperatures using experimentally measured input data.

Both figures show that the dynamic behaviour including small oscillations is in very good agreement. As discussed in part I, the temperature levels are not reproduced satisfactorily because the thermodynamic steady state model is a very coarse one only. This is tolerated because the focus is laid on the dynamics. In Figure 8, the dynamics are being reproduced very well with only a slight delay of 10s between course of the two temperatures. Figure 9 shows a slightly worse agreement for the cooling water temperatures, with simulated values being approx. 25s ahead of measured ones. Also shown in Figure 9 is the simulated outlet temperature of the absorber, $t_{Ao,sim}$, for which no measured data was available. We want to emphasize that no tuning of the dynamic parameters has been performed up to now: the dynamic response seems to be very robust according to the sensitivity analysis which has been shown before.

Conclusions

In this paper, the performance of a dynamic model for absorption chillers has been investigated. General model functionality is demonstrated and the thermodynamics have been found to be consistent with reality. A sensitivity analysis has been performed on external and internal heat exchange related thermal mass. The analysis shows that an increase in both external and internal thermal mass results in a slower response to the step change but also in smaller heat flow oscillations during the transient period. Also, the thermal mass has been found to influence the heat flow transients more significantly if allocated internally. A time difference of 200s for reaching the steady state (response time) was observed between a complete internal and external allocation of thermal mass.

The transport delay in the solution cycle has been found to influence both the response time and the transients of the heat flow. A smaller transport delay leads to significantly faster response. Assuming half the value of the real transport delay in the absorption chiller leads to a 33% reduction of response time.

The comparison with experimental data shows that the dynamic agreement between experiment and simulation is very good with dynamic temperature deviations between 10 and 25s. The total time to achieve a new steady-state after a 10K input temperature step amounts to approximately 15 minutes for the experimental chiller. Compared to this, the present dynamic deviations of the model are in the magnitude of approximately 1 to 3%. Steady-state results are being reproduced with temperature deviations between 0.7 and 3.5K in the model. Accuracy in this respect, however, was not the aim of the present study.

The dynamic simulation model presented in this paper is a useful tool in the overall design process of absorption chillers. Technical changes in the construction of an existing absorption chiller model can be tested quickly and easily by incorporating the design changes in the model. Also, new chiller designs can be tested on their performance without the need to build a prototype. The model also allows the identification of transfer functions and control parameters of absorption chillers without the need to perform experiments. The model has been designed for the Phoenix 10kW chiller but can easily be adapted to other LiBr/water absorption chillers if the required design data of these chillers are available.

Appendix

Table 3. Constant parameters of the dynamic model.

No.	Constants	Unit	Value	Description
1	$r_{(pc)}$	$kJkg^{-1}$	2450	Evaporation enthalpy of water
2	$l_{(pc)}$	$kJkg^{-1}$	272	Solution enthalpy of LiBr/water solution
3	$(UA)_{E,ext}$	kWK^{-1}	2.9	External heat transfer coefficient value of evaporator
4	$(UA)_{C,ext}$	kWK^{-1}	5.2	External heat transfer coefficient value of condenser
5	$(UA)_{G,ext}$	kWK^{-1}	4.0	External heat transfer coefficient value of generator
6	$(UA)_{A,ext}$	kWK^{-1}	3.2	External heat transfer coefficient value of absorber
7	$(UA)_{E,\mathrm{int}}$	kWK^{-1}	3.9	Internal heat transfer coefficient value of evaporator
8	$(UA)_{C,\mathrm{int}}$	kWK^{-1}	8.7	Internal heat transfer coefficient value of condenser
9	$(UA)_{G,\mathrm{int}}$	kWK^{-1}	4.3	Internal heat transfer coefficient value of generator
10	$(UA)_{A,\mathrm{int}}$	kWK^{-1}	3.2	Internal heat transfer coefficient value of absorber
11	$\dot{m}_{sol,w}$	kgs^{-1}	0.0559	Mass flow of weak solution
12	$\dot{m}_{hw}$	kgs^{-1}	0.3322	Mass flow of hot water
13	$\dot{m}_{cw}$	kgs^{-1}	0.7462	Mass flow of cooling water
14	$\dot{m}_{chw}$	kgs^{-1}	0.808	Mass flow of chilled water
15	A_t	m^2	$4.9*10^{-4}$	Cross area of solution tube
16	h	m	0.1	height between solution outlet at generator and inlet at absorber
17	$M_{st,sol,G,start}$	kg	0.24	Initial condition of accumulated solution in generator sump
18	$M_{st,sol,A,start}$	kg	0.19	Initial condition of accumulated solution in absorber sump
19	ς	-	2100	Resistance coefficient of solution heat exchanger and piping
20	c_1	-	67, 2	time constant for solution flow from generator to absorber
21	c_2	-	61, 1	time constant for solution flow from absorber to generator
22	$(Mc_p)_{E,ext}$	kJK^{-1}	6.8	Cumulated heat capacity of evaporator, external
23	$(Mc_p)_{C,ext}$	kJK^{-1}	6.4	Cumulated heat capacity of condenser, external
24	$(Mc_p)_{G,ext}$	kJK^{-1}	9.0	Cumulated heat capacity of generator, external
25	$(Mc_p)_{A,ext}$	kJK^{-1}	9.0	Cumulated heat capacity of absorber, external
26	$(Mc_p)_{E,\mathrm{int}}$	kJK^{-1}	38.4	Cumulated heat capacity of evaporator, internal
27	$(Mc_p)_{C,\mathrm{int}}$	kJK^{-1}	38.2	Cumulated heat capacity of condenser , internal

28	$\left(Mc_p\right)_{G,\text{int}}$	kJK^{-1}	82.2	Cumulated heat capacity of generator , internal
29	$\left(Mc_p\right)_{A,\text{int}}$	kJK^{-1}	81.9	Cumulated heat capacity of absorber , internal
30	η_{SHX}	-	0.9	Efficiency of solution heat exchanger
31	cp_w	$kJkg^{-1}K^{-1}$	4.19	Specific heat capacity of liquid water
32	cp_v	$kJkg^{-1}K^{-1}$	1.86	Specific heat capacity of water vapour
33	$cp_{sol,s}$	$kJkg^{-1}K^{-1}$	3.8	Specific heat capacity of strong solution
34	$\rho_{sol,s}$	kgm^{3-1}	1600	Density of strong solution
35	Δt	s	1	Time increment of simulation

References

[1] Kohlenbach P, Ziegler F. A dynamic simulation model for transient absorption chiller performance. Part I: The model. *Int J Refrigeration 2007*,

Conclusion

In this report the new development concerning simulation tools for solar cooling are presented. First are presented the most commonly used simulation tools in the domain of solar air conditioning. It was shown that most of simulation tools requires high knowledge of the simulated process as well as internal development for each project, except some simple simulation tools with predefined configurations and fixed boundaries. Most of simulation environment used in each research group in the solar cooling domain are internally developed and used. In the commercially available detailed softwares; the models of the solar installation are widely used as in found in the libraries, however the model of the cooling process itself are usually re-developed by the users.

New development of solar cooling models and their validation are then presented. A detailed desiccant air handling unit powered by heat pipe vacuum tube collectors is developed and implemented in SPARK. The model of the unit showed good accuracy on a component level and on a system level under different operating conditions. The supply conditions of the air handling unit are estimated with good accuracy as well as the potential of solar energy in the desiccant cooling process. The transient behaviour of the collector is very well predicted by the model. A transient model of the desiccant wheel is also developed and experimentally validated. The model calculates the evolution of the temperature and humidity ratio of the air across the wheel and at its outlet. It gives also a good estimation of the mean air outlet conditions of the desiccant wheel.

A transient detailed model of a Li/Br absorption chiller is developed. The model is experimentally validated and is capable to predict the transient behaviour of the chiller. The comparison with experimental data shows that the dynamic agreement between experiment and simulation is very good with dynamic temperature deviations between 10 and 25 s The dynamic simulation model presented is a useful tool in the overall design process of absorption chillers. Technical changes in the construction of an existing absorption chiller model can be tested quickly and easily by incorporating the design changes in the model.

Task 38
Solar Air-Conditioning
and Refrigeration

Hygienic Aspect of Small Wet Cooling Towers

A technical report of subtask C

Date: 17.12.2009

By Harald Moser[1] and Erich Podesser[2]

[1] **Graz University of Technology**
Institute of Thermal Engineering
Inffeldgasse 25 B
A-8010 Graz

Phone	**+43 (0)316 873 7304**
Fax	**+43 316 873 7305**
e-mail	**harald.moser@tugraz.at**

[2] **on behalf of AEE - Institut für Nachhaltige Technologien**
Feldgasse 19
A-8200 Gleisdorf

Phone	**+43 (0)316 672605**
Fax	**+43 (0)316 672605**
e-mail	**erich.podesser@utanet.at**

Contents

1 Introduction

This chapter shall give a very short description of thermally driven heat pumps especially with respect to the heat rejection sub-system and temperature level. Furthermore different technologies for rejecting the heat to the air are briefly discussed and compared to each other for different climatic conditions.

1.1 General

Heat rejection for thermally driven heat pumps is a crucial subsystem especially in solar assisted air conditioning, because:

- The necessary temperature level of the driving heat and the efficiency of the system depends on the temperature level of the heat rejection system significantly
- The amount of heat to be rejected is about twice to triple bigger than the cooling load
- The electrical energy consumption as well as the initial and operating costs of the heat rejection system are significantly high

In order to minimize the temperature level of the heat rejection wet cooling towers can be used. As these systems bring the air and cooling water into direct contact hygienic problems can occur. This leads to a high maintenance effort and operational costs and to legislative restrictions.

This report focuses on the heat rejection system for small thermally driven heat pumps. It describes in a comprehensive way the potential, operation and design criteria as well as hygienic aspects of wet cooling towers. Furthermore possible solutions to overcome the drawback of the poor hygienic conditions of wet cooling towers are discussed.

This report is structured in 6 chapters. Chapter 1 gives a very shot overview on available heat rejection technologies for thermally driven heat pumps. In Chapter 2 the special needs of small scale wet cooling towers are discussed and Chapter 3 describes a calculation procedure for a wet cooling tower which can be used for commissioning optimization purpose. Chapter 4 is focused on Legionella in small scale wet cooling towers and Chapter 5 describes measures to avoid uncontrolled Legionella multiplication especially using UV-light and silver-copper ionisation. Chapter 6 summarizes the report content and gives short conclusions.

1.2 Thermally driven Heat Pumps

Neglecting the electrical (mechanical) energy input a thermally driven heat pump (THP) for cooling purpose is characterized only by heat flows at three temperature levels:

- at high temperature level the driving heat (Q_{DRV}) is taken up,
- at medium temperature level the waste heat (Q_{HRJ}) of the process is rejected, and
- at low temperature level the cooling load (Q_{COL}) is taken up.

Due to energy conservation the amount of heat which has to be rejected at medium temperature level has to be the driving heat plus the cooling load (compare Figure 1-1).

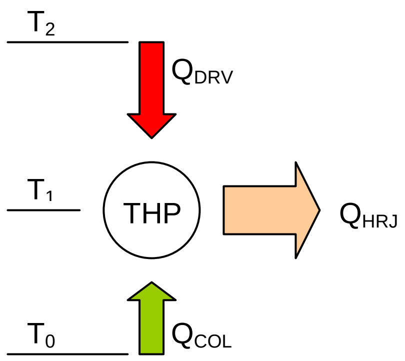


Figure 1-1: Sketch of heat flows and temperature levels for a thermally driven heat pump

The "Coefficient of Performance" for cooling (COP_C) is defined as shown in Eq. 1 (if the mechanical energy can be neglected). It can be easily shown, that the amount of heat which has to be rejected (Q_{HRJ}) per cooling capacity (Q_{COL}) is directly dependent on the COP_C (compare Eq. 2)

$$\mathrm{COP_C} = \frac{Q_{COL}}{Q_{DRV}} \qquad \text{Eq. 1}$$

$$\frac{Q_{HRJ}}{Q_{COL}} = 1 + \frac{1}{\mathrm{COP_C}} \qquad \text{Eq. 2}$$

For different technologies (ad- or absorption), process configurations and working pairs (e.g. $H_2O/LiBr$ or NH_3/H_2O) of thermally driven heat pumps the COP of a real application varies depending on the three temperature levels. Furthermore the temperature level of the driving heat has to be at a certain minimum level, depending on the temperature level of the cooling load and heat rejection.

However, in order to estimated the relevance of the temperature level of the heat rejection system a theoretical thermodynamic approach can be used. In terms of energy conversion a thermally driven heat pump combines two cycles, a power generation and a heat pump cycle. Using the Carnot efficiency of these cycles the theoretical possible efficiency of the thermally driven heat pump can be calculated using the three temperature levels mentioned above.

In Figure 1-2 the two Carnot cycles for a thermally driven heat pump are shown. The cycle on the left hand side generates the work to drive the heat pump cycle on the right hand side.

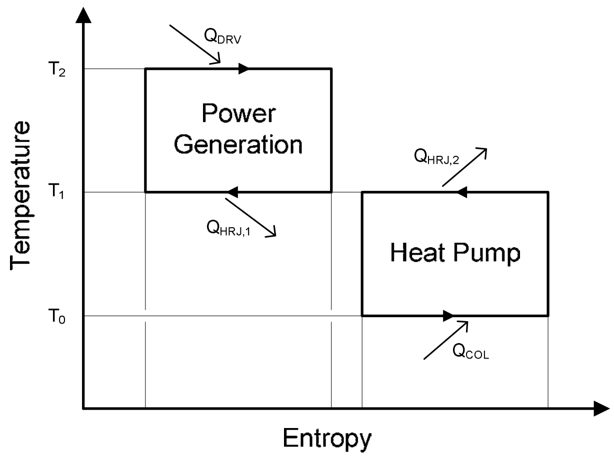


Figure 1-2: *Carnot cycles of a thermally driven heat pump*

Acc. to Eq. 3 the theoretical COP of the Carnot process for cooling of a thermally driven heat pump can be calculated.

$$COP_{th,C} = \eta_{PG} \cdot \varepsilon_{HP} = \frac{(T_2 - T_1)}{T_2} \cdot \frac{T_0}{(T_1 - T_0)} \qquad \textit{Eq. 3}$$

Evaluating Eq. 3 using different temperature levels for heat rejection and constant temperature levels of the driving heat T_2 = 80℃ and of the cooling load T_0 = 5℃ leads to a characteristic shown in Figure 1-3.

The Carnot-efficiency shows, what is thermodynamically possible. A real technical solution will be far below this value. Thus a further line has been drawn in Figure 1-3 which represents 40% of the Carnot efficiency ($COP_{th,C,40\%}$). The 40%-value has been chosen arbitrarily but the results represent approximately the performance of real NH3/H2O AHP-applications for a temperature level of the heat rejection of approx. 30℃. The COP_{th} and $COP_{th,C,40\%}$ show a strong dependency on the temperature level of the heat rejection system, e.g. the $COP_{th,C,40\%}$ is above 0.6 for a heat rejection temperature of 30℃ and decreases below 0.4 for 40℃.

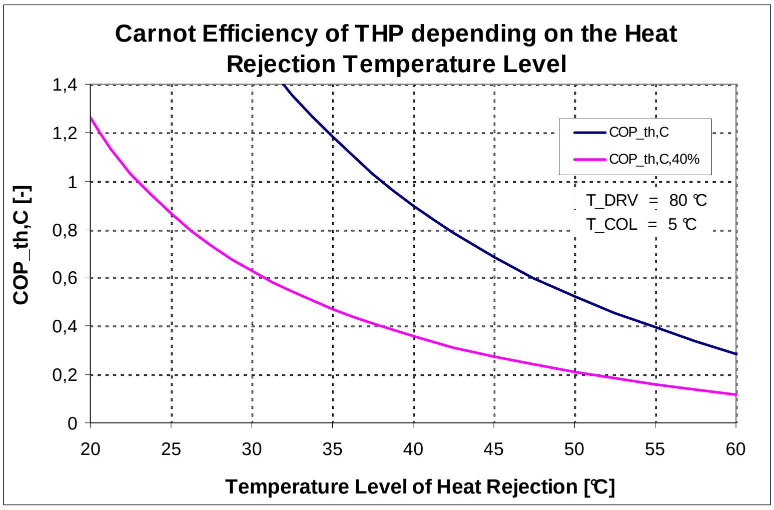


Figure 1-3: Carnot efficiency for different temperature levels of the heat rejection system (T_{DRV}=80℃, T_{COL}=5℃)

For the calculation of the Carnot-efficiency represented in Figure 1-3 the temperature level of the driving heat was constant. Operating a real thermally driven heat pump a minimum temperature level for the generator is required. E.g. in an Absorption Heat Pump (AHP) a certain temperature level is needed in order to be able to evaporate the rich solution in the generator. This temperature level is generally dependant on the medium and low temperature level.

In order to discuss this dependence thermodynamic calculation of a AHP with the working pair NH3/H2O has been set up. In Figure 1-4 left the result for the theoretical minimum temperature level in the generator, were the evaporation of the rich solution starts is shown. In order to be able to evaporate a certain portion of the rich solution the temperature level of the driving heat has to be higher than this theoretical minimum. In Figure 1-4 right typical temperature levels for NH3/H2O-AHP are shown.

The shown figures and values should not be treated as assured absolute values for real applications but as theoretical approach in order to show the general tendency of the heat rejection temperature level to the driving heat temperature level.

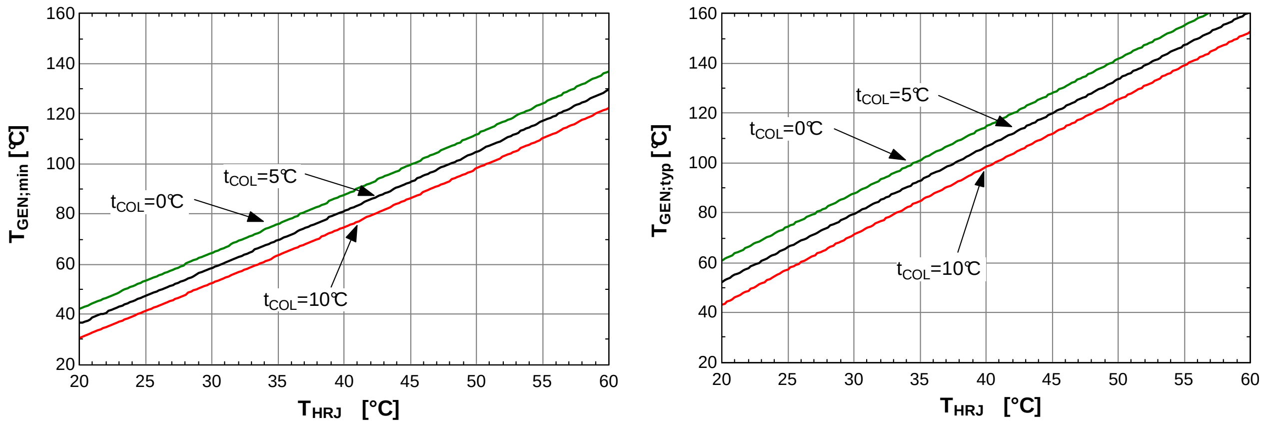


Figure 1-4: Thermodynamic calculation of the minimum (left) and a typical (right) temperature level of driving heat depending on heat rejection temperature level.

In conclusion it can be seen from Figure 1-4, that the temperature level of the driving heat has to be increased with increasing heat rejection temperature level significantly. For a low temperature heat source of 5℃ and a heat rejection temperature of 30℃ the minimum temperature level of the driving heat is at approx. 60℃. If the heat rejection temperature increases to 50℃ the minimum driving temperature level has to be increased to approx. 105℃ as well.

In Figure 1-5 the efficiency of different solar collectors at varying working temperature levels are shown (Heß, 2007). The efficiency of standard collectors decreases dramatically with increasing working temperature. This effect underlines the need of low heat rejection temperature levels to enable as low driving temperature levels as possible, standard flat-plate collectors and increase system efficiency.

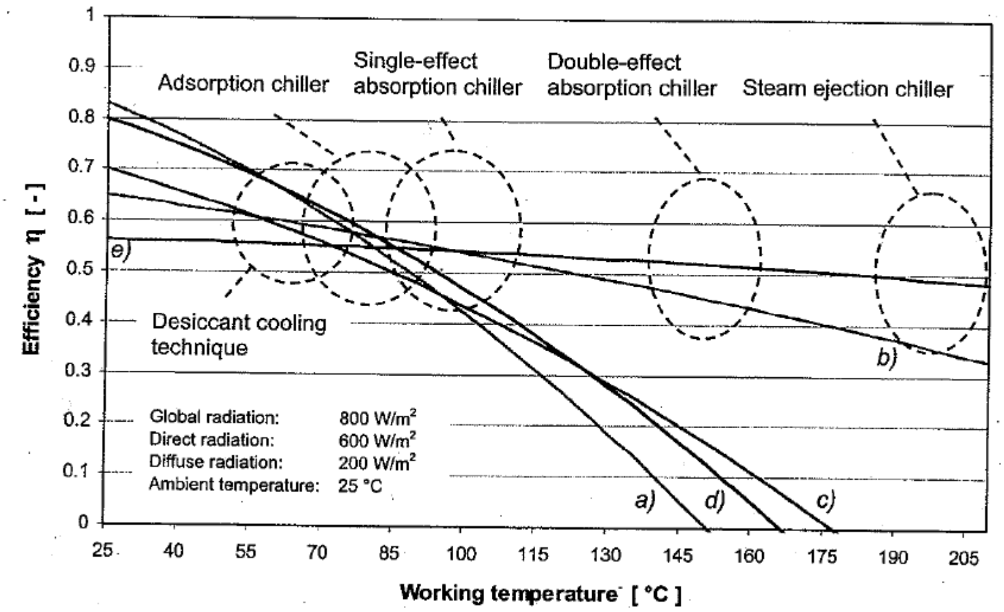


Figure 3: Comparison of different collectors at 800 W/m². a) Single glazed flat-plate with AR, b) Evacuated tube collector of the Sydney type, c) CPC flat-plate with Teflon foil, d) Flat-plate with double AR-glazing and inert gas filling, e) Small parabolic trough (under development; only the fraction of direct radiation = 600 W/m² can be used). The values are for normal irradiance and refer to the aperture area.

Figure 1-5: Comparison of the efficiency of different solar collectors designs at varying working temperature levels. (Heß, 2007)

Note: Dealing with solar cooling applications one important factor is the part load operation. In other words: what cooling capacity can be achieved at certain temperature levels of the heat sink and the heat sources. By the use of different assumptions (e.g. that the mass flow through the solutions pump is constant) a "characteristic equation" can be derived. This equation brings together in a linear correlation the cooling load (Q_{COL}) and a function of the temperature levels ($\Delta\Delta t$) in the four main components (Genarator, Absorber, Condenser and Evaporator) of an AHP. A detailed description of this correlation can be found in Ziegler (1997). Evaluating the temperature level of the heat rejection system it can be concluded, that if the heat rejection temperature increases the cooling capacity of an AHP will decrease significantly.

1.3 Comparison of Heat Rejection Technologies

As discussed the temperature level of the heat rejection system should be as low as possible in order to reduce the necessary temperature level of the driving heat and increase the system efficiency and capacity.

Generally different heat sinks are possible to reject the heat, e.g. air, ground or water. While the use of ground and water depends strongly on the local conditions air is available for almost all applications.

For rejection of heat to the ambient air in principle two types of systems are considered, open cooling towers (or wet cooling towers) and closed cooling towers or (dry coolers). As a combination of these adiabatic pre-cooling of the air in e dry cooler and hybrid cooling towers should be mentioned.

The main difference between these technologies is that in the dry cooler the cooling water rejects the heats to the air via a heat exchanger und in wet cooling towers the cooling water is sprayed into the air and a direct heat and mass transfer takes place. Thus in dry coolers only sensible heat and in wet cooling towers mainly latent heat is exchanged.

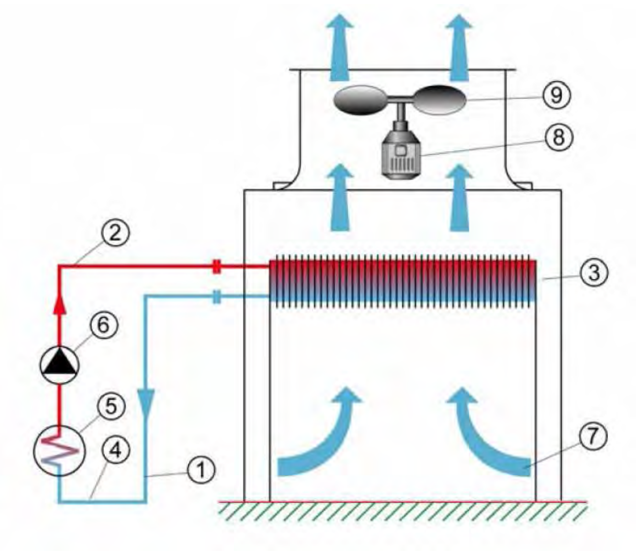
1.3.1 Dry Cooler

Dry coolers consist generally of finned heat exchangers (air to water), fans and a casing. The water circulates in a closed circuit and by passing ambient air over the finned surfaces the heat is rejected to the air (compare Figure 1-6).

With air-cooled heat exchangers, it is not possible to cool the medium below the ambient dry bulb temperature. In this case the approach temperature between the medium outlet temperature and the inlet temperature of the dry air depends mainly on the size and capacity of the dry cooler - typical values of the approach temperatures are 5 to 9 K (SWKI, 2003).

Dry coolers are often used for cooling refrigerants, oils or water/glycol mixtures. Compared to wet cooling towers they have lower operational and maintenance cost and because the cooling water comes not in direct contact to the air they have no hygienic problems or legionnaires risk. Further advantages are low noise, easy installation and a low profile.

The main disadvantages compared to wet cooling towers are higher re-cooling temperatures much higher investment costs, energy consumption and space requirement.



1. cooling circuit
2. inlet flow
3. cooling element (heat exchanger)
4. return flow
5. heat source
6. circulating pump
7. cooling air
8. fan drive
9. fan

Figure 1-6: Sketch of a dry cooler (SWKI, 2003)

Note: One alternative of dry-cooler is the adiabatic pre-cooling of the air upstream the heat exchanger in case of high ambient air temperatures. In that case water is sprayed into the air inlet stream and should evaporate before it arrives at the heat exchanger surface. Thus the air is cooled down near the wet bulb temperature and the dry-cooler can be operated at lower operating temperatures. The spraying should only take place at limited operating hours when it is needed due to the operating conditions, because the excessive use can lead to increased corrosion and the formation limestone at the heat exchanger.

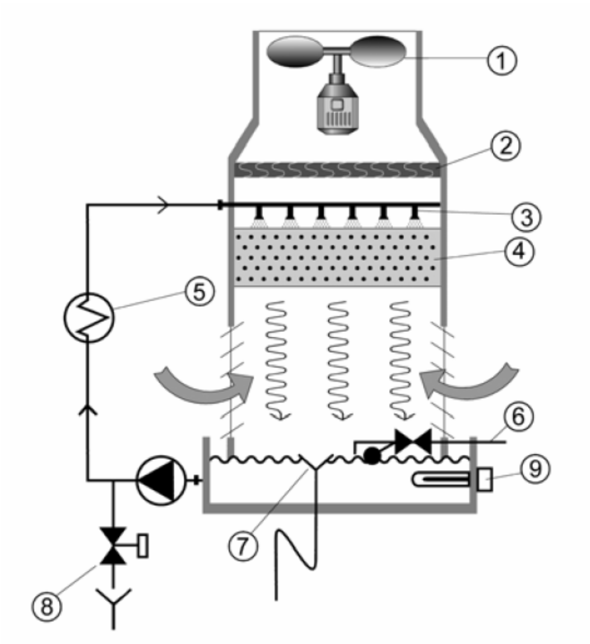
1.3.2 Wet Cooling Towers

Wet cooling towers (open loop evaporative cooling tower) consists of a shell containing packing/fill material with a large surface area. Nozzles arranged above the packing, spray and distribute the cooling water onto the packing. The water trickles through the packing into a basin from which it is pumped back to the condenser. The water is cooled by air, drawn or blown through the packing by means of a fan. The air flow, which is either in counter flow or cross flow to the water flow, causes some of the water to evaporate, thus latent heat, is exchanged from the water to the air.

The evaporated water is continuously replenished by make-up water. Due to the fact, that evaporation also increases the concentration of the dissolved solids in the cooling water, blow down of the cooling water is necessary.

In wet cooling towers the wet-bulb temperature determines the degree of cooling and thus cooling below the ambient dry bulb temperature is possible. The characteristic approach temperature, which is the difference between the water outlet temperature and the ambient wet-bulb temperature, of wet cooling towers lies between 4 to 8ºK (SWKI, 2003).

Compared to dry coolers wet cooling towers, are able to cool the cooling water to lower temperature level, requires less space and have much lower investment costs. The main disadvantages of wet cooling towers are hygienic problems, water consumption and high maintenance effort.



1. fan with drive
2. drift eliminator
3. spray nozzels
4. trickle packing
5. heat source
6. float valve and fresh water inlet
7. overflow
8. bleed off
9. frost protection heating

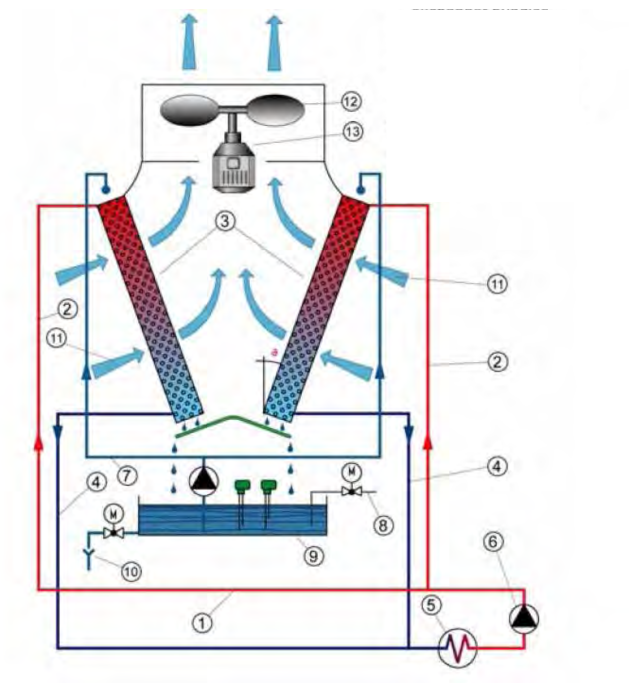
Figure 1-7: Sketch of a wet cooling tower (Jaeggi, http://www.guentner.ch/pdfs/Evaluation of Aircooled Cooling Systems.pdf, 26.03.2009)

1.3.3 Hybrid

Hybrid dry cooler combines the two methods of dry cooling and evaporative cooling. The cooling water is circulated by a pump in a closed primary cooling circuit from the heat source to cross current air to water heat exchanger.

In cool weather conditions this process cools down the cooling water sufficiently and the hybrid cooler operates like a dry cooler. At high air temperatures the hybrid cooler uses the principle of evaporative cooling in order to achieve lower cooling temperatures. Therefore a pump circulates water from a basin to the cooling element where the water flows back via the finned surface of the air to water heat exchanger. The air flowing past the heat exchanger causes the water to evaporate on the fin surface, and takes the heat from the fins.

Comparing a hybrid dry cooler to common dry cooler it has the advantage to use evaporative cooling at hot whether conditions and therefore cools down the cooling water below the dry bulb temperature, it has a higher capacity and lower energy consumption. On the other side the hybrid dry cooler has higher investment costs, maintenance effort and water consumption. Furthermore, hygienic measures have to be taken as for the wet cooling tower.



1. primary cooling circuit
2. inlet flow
3. finned tube heat exchanger
4. return flow
5. heat source
6. circulating pump
7. deluging water circuit
8. make up water inlet
9. water collector
10. bleed off
11. cooling air
12. fan
13. fan drive

Figure 1-8: Sketch of a hybrid dry cooling system (Jaeggi, http://www.guentner.ch/pdfs/Evaluation of Aircooled Cooling Systems.pdf, 26.03.2009)

1.4 Climatic Conditions

Wet cooling towers are able to cool down the cooling water near the wet bulb temperature and dry cooler refer to the dry air temperature. The difference between these temperatures depends on the geographic location of the plant or the climatic conditions respectively.

For comparison of the climatic conditions four different locations have been chosen:

- Frankfurt, Germany: represents a moderate, Central European climate.
- Stockholm, Sweden: represents a moderate, Northern European climate.
- Madird, Spain: represents a Mediterranean, continental climate with high temperatures during summer but moderate air humidity.
- Palermo, Italy: represents Mediterranean coastal climates with high humidity and temperature during summer

Using the commercial database Meteonorm® (Meteotest, 2003) it is possible to calculate on hourly bases climate date for one year for a certain site, e.g. for the dry and wet bulb temperature. This is shown in Figure 1-9 (left) for Frankfurt. By sorting these data according the dry bulb temperature the annual temperature distribution curves can be drawn as shown in Figure 1-9 (right). The dots represent the wet bulb temperatures dedicated to the dry bulb temperatures.

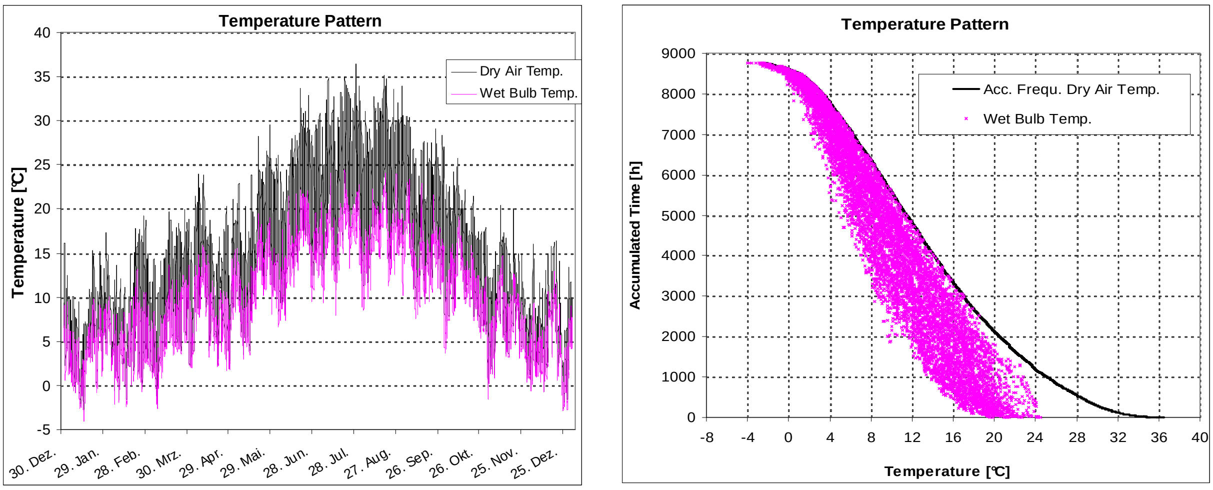


Figure 1-9: Dry and wet bulb temperature on hourly basis for one year of Frankfurt (left) and annual temperature distribution curve for dry bulb temperature with dedicated wet bulb temperatures (right).

In Figure 1-10 and Figure 1-11 the lower section of this diagram is shown for four different sites and a further line (blue) is drawn, which represents the annual temperature distribution curve of the wet bulb temperature. Comparing the temperature distribution of the dry air and wet bulb temperature for Frankfurt Figure 1-10, left) to each other it can be seen, that the dry air temperature is approx. 940 hours above 20°C and the wet bulb temperature only approx. 170 hours. The mean temperature difference between dry and wet bulb temperature is ca. 7.4 K when the cooler operates only at dry air temperature above 20°C.

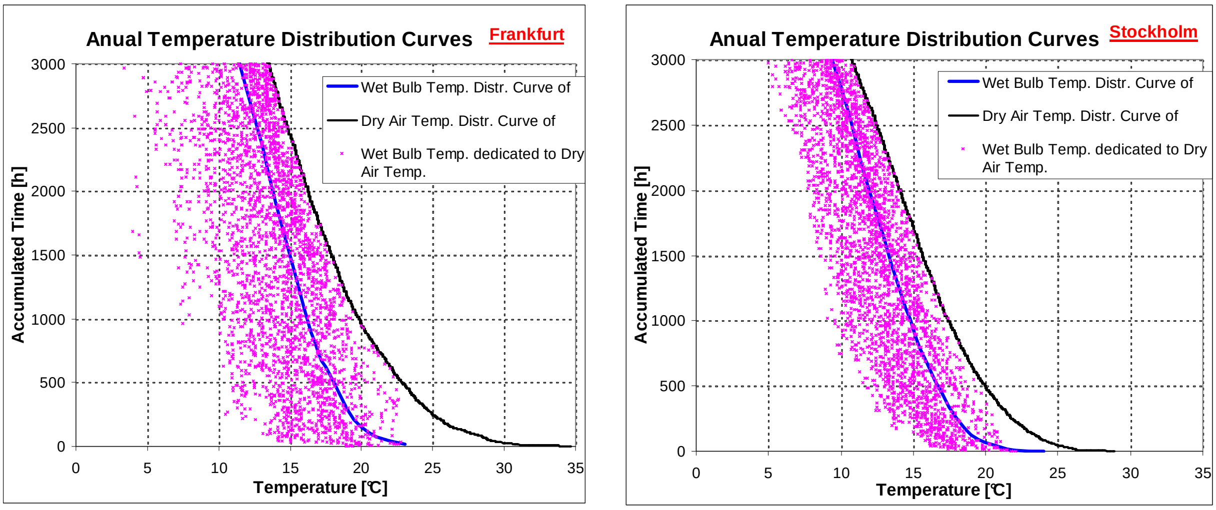


Figure 1-10: Annual dry air temperature distribution curve with dedicated wet bulb temperatures and annual wet bulb temperature distribution curve for Frankfurt (left) and Stockholm (right)

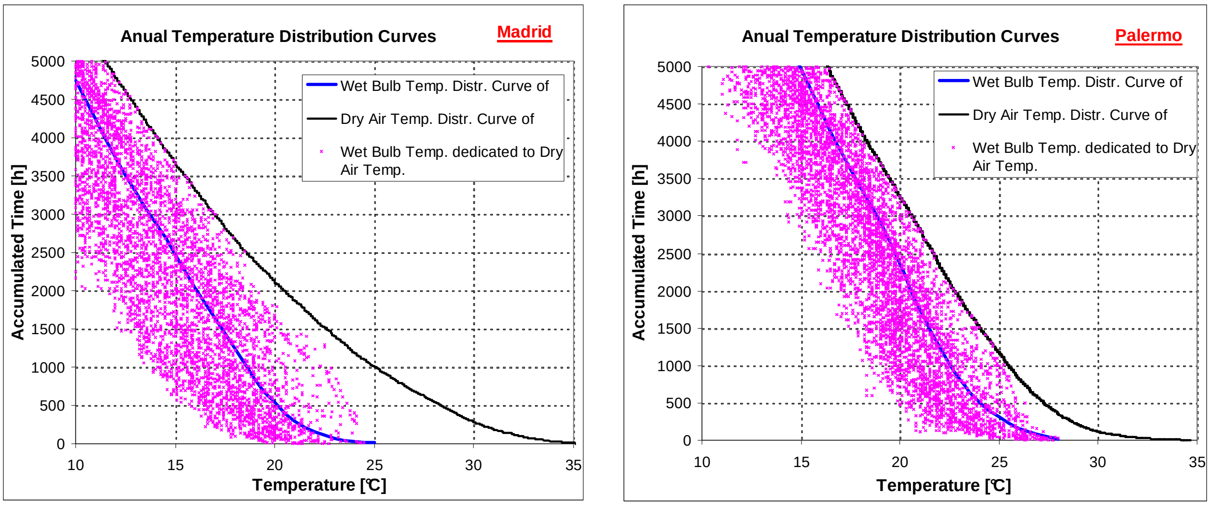


Figure 1-11: Annual dry air temperature distribution curve with dedicated wet bulb temperatures and annual wet bulb temperature distribution curve for Madrid (left) and Palermo (right)

Table 1-1 shows an overview of the operating hours and the maximum and mean temperature levels for different sites for dry cooler and wet cooling tower assuming that the cooler is in operation when the dry air temperature exceeds 20°C.

The locations with moderate climatic conditions Frankfurt and Stockholm show only few operating hours with a low mean wet bulb temperature of about 16.5°C. The mean temperature difference between dry cooler and wet cooling towers is 5.5 K for Stockholm and 7.4 K for Frankfurt. At maximum air temperature the temperature difference is at ca. 12 K in Frankfurt.

At Mediterranean climate conditions (e.g. Palermo or Madrid) the period of time with dry air temperatures of more than 20°C is far longer, e.g. 3237 h for Palermo. While the maximum dry air temperature in Madrid (36.4°C) is higher than in Palermo (34.7°C) the maximum wet bulb temperature is much lower (Madrid: 24.5°C and Palermo 28°C).

As expected, the wet cooling tower benefits most in dry hot climates but also in humid coastal climates the resulting cooling water temperatures are significantly lower, especially at very hot weather conditions.

When the maximum cooling load is required the dry cooler delivers cooling water with a temperature level of 42.4°C and the wet cooling tower of 30.5°C in Madrid which represents a difference of 11.9 K. In Palermo the dry cooler delivers a temperature of 40.7°C and the wet cooling tower of 34°C thus the temperature difference is still 6.7 K.

Table 1-1: Overview heat rejection temperature levels for dry coolers and wet cooling towers at different locations.

		Frankfurt		Stockholm		Palermo		Madrid	
Operating hours for $T_{dry,air} > 20°C$	[h]	937		482		3237		2102	
		max	mean	max	mean	max	mean	max	mean
Dry air temperature	[°C]	34,6	23,7	28,9	22,3	34,7	24,2	36,4	25,4
Wet bulb temperature	[°C]	22,7	16,3	22,1	16,8	28	20,5	24,5	17,4
Temperature difference dry - wet	[K]	11,9	7,4	6,8	5,5	6,7	3,7	11,9	8
Cooling water temperature- dry cooler ($\Delta T_{approach,dry}$ = 6 K)	[°C]	40,6	29,7	34,9	28,3	40,7	30,2	42,4	31,4
Cooling water temperature for wet cooling tower ($\Delta T_{approach,wet}$ = 5 K)	[°C]	28,7	22,3	28,1	22,8	34	26,5	30,5	23,4
Temperature difference dry - wet	**[K]**	**11,9**	**7,4**	**6,8**	**5,5**	**6,7**	**3,7**	**11,9**	**8**

Wet cooling tower are able to cool down the cooling water to a significantly lower temperature than dry coolers. Furthermore the electricity demand and investment costs are much lower than for dry coolers. The main drawbacks of wet cooling towers are water consumption, hygienic problems and linked to that high maintenance costs.

2 Wet Cooling Tower - Operation

A short description of wet cooling towers is already given in section 1.3.2. Within this chapter the special needs of small scale wet cooling towers should be discussed.

In wet cooling towers an intensive heat and mass transfer takes place by direct contact of the cooling water and the air.

2.1 Nutrients and Biofilm

Because wet cooling towers bring large quantities of air in contact with the cooling water they are highly efficient air scrubber and organic material and other debris can be accumulated in the cooling water. This material may serve as nutrient source for micro-organism like Legionellae and may cause the formation of biofilm on any wetted surface in the cooling tower. Thus it is necessary to remove organic material from the cooling tower frequently during maintenance work.

2.2 Water consumption

The water consumption in a wet cooling tower consists of three different losses, because of:

1. evaporation,
2. blow off or drift and
3. blow down

Cooling occurs in a cooling tower mainly by the mechanisms of evaporative cooling (latent heat) and minor by the exchange of sensible heat. Approx. 2-3% of the cooling water flow rate evaporates in the cooling tower and leaves its dissolved salts behind in the bulk of the water which has not been evaporated. Thus the salt concentration in the circulating cooling water is rising. To prevent the salt concentration of the water from becoming too high, a portion of the water is drawn off (blow down) for disposal. Furthermore a small portion of the cooling water is lost in form of mist carried out of the tower with the waste air. In order to limit these losses drift baffles or drift eliminators are installed in the cooling tower.

In small cooling towers the blow down mechanism is often controlled manually but a better approach is to use a conductivity controller to continuously bleed and refill water in the system. Continuous systems maintain water quality at a more consistent level without wide fluctuations in the dissolved salts. Thermal efficiency, proper operation, and life of the cooling tower are related directly to the quality of the recirculating water in the tower.

To compensate the losses fresh water (makeup water) has to be supplied to the tower basin. In praxis higher water consumption is often needed due to e.g. back flush of filters, cleaning operations and exchange of water due to contaminations of the cooling water.

Water supply to cooling towers is a limiting source to operation of cooling towers. To reduce fouling in cooling towers and the remaining cooling system make-up water needs to contain relative low concentrations of nutrients that may support growth of biofilm and *Legionella* (i.e. biofouling). Hence, when ground water or drinking water is scarce and expensive resource pre-treatment of make-up water from other sources, e.g. rivers and lakes, is required. Furthermore, the efficiency of chemicals for prevention of corrosion and biofouling is dependent on the water quality of the circulating cooling water.

2.3 Water treatment

The makeup water is fresh water added to the cooling towers to replace evaporation, blow down, and drift losses. The amount and chemistry of makeup water added directly affects the quality of water in the systems.

The relationship between blow down water quality and make-up water quality can be expressed as a concentration ratio. This concentration ratio is typically between 2 and 5 (VDMA 24649), which means that the salt concentration in the circulating water is 2 to 5 times higher than in the makeup water. Limits for the concentration of different minerals are usually given in the operating manuals of the cooling tower manufacturer.

As the concentration of salts increase the water may not be able to hold the minerals in solution and they can precipitate out as mineral solids and cause fouling and heat exchange problems in the cooling tower.

Beyond the necessary limitation of the mineral content water treatment might be required to avoid scale, corrosion and growing micro organism.

2.4 Aerosols in the Waste Air

As discussed a small portion of the cooling water is lost in form of mist carried out of the tower with the waste air. In Order to limit these aerosols in the waste air drift eliminator are installed in the air outlet. The effectiveness of these drift eliminator varies in a far range between 0.0005 and 0.1% of the circulation cooling water flow rate is released to the waste air (Aquaprox, 2007). This means that the drift eliminator affects the aerosol discharge of the cooling tower and with that also the release of micro-organism very much.

However, it must be assumed that some droplets are within the critical size for human intake of 5 micrometer or smaller. Larger droplets leaving the cooling tower may be reduced to 5 micrometer or less by evaporation (ASHRAE Guideline 12-2000).

Even the best drift eliminators do not eliminate aerosols entirely. Thus it cannot be the “stand alone“ preventive measure but high efficiency drift eliminator are able to reduce the release of micro-organism significantly. Furthermore the evaporative cooling equipment should be positioned such that it is away from occupied areas or where drift can enter directly into the windows or air intakes of buildings in the vicinity of the installation. The prevailing wind direction should be taken into account wherever possible EUROVENT 9/5 (2002)

2.5 Electricity Consumption

The electricity consumption of a cooling tower is mainly determined by the electricity demand of the pumps and fans. As discussed, compared to the cooling capacity of a thermally driven heat pump the cooling capacity of the heat rejection system is approx. 2 to 3 times bigger. This leads to a significant electricity consumption of the heat rejection system in solar cooling applications. In order to reduce this demand variable speed controlled pumps and fans can be considered.

2.6 Anti freeze

Solar cooling applications are not in operation during winter season and cooling towers will be drained during this period of time thus no freezing problems occur. However, if there is a risk of freezing appropriate counter measures are needed e.g. electric immersion heater in the collection basin.

2.7 Part Load Operation

Typically the speed of the fans can be controlled in two steps. Alternatively it is possible to equip the fan with frequency control in order to control the required cooling capacity.

During part load operation the reduction ratio of the water flow rate must not exceed 1:5 to avoid clogging, whereas it is even possible to switch off the fan when running at about 10% cooling load. (Henning, 2004)

2.8 Maintenance

The key requirements for maintaining system efficiency are adequate control of the recalculating water quality and a programme of maintenance to keep the equipment clean and in good condition.

Maintenance requirements are given in manufacturer's instructions or in Guidelines e.g. EUROVENT 9/5 (2002) or VDMA 24649 (2005).

A typical mechanical maintenance schedule is shown in Table 2-1

Table 2-1 Typical mechanical maintenance schedule (EUROVENT 9/5,2002)

Description of Service	**Start–Up (see Note 1)**	**Monthly**	**Every six months**	**Shut- Down**	**Annually**
Inspect general condition of the system	X			X	X
Inspect heat transfer section(s) for fouling	X		X		
Inspect water distribution	X		X		
Inspect drift eliminators for cleanliness and proper installation	X		X		
Inspect sump	X		X		
Check and adjust sump water level and make-up	X		X		
Check chemical feed equipment	X	X			
Check proper functioning of blow-down	X	X			
Check operation of sump heaters (if applicable)	X		X		
Clean sump strainer	X		X		
Drain sump & piping				X	

Note 1: Initial start-up and after seasonal shut-down period.

3 Wet Cooling Tower - Process Calculation

The need of special information of wet cooling tower technology begins with the purchase of a wet cooling tower. The owner and later on the user are interested in the technical behavior of the cooling tower. Therefore this chapter contains information according the process calculation of wet cooling towers. With the help of these knowledge the commissioning, adaptations and also changes of the technical features of wet cooling towers can be executed.

3.1 General aspects

The heat transferred from the heat sources (solar collectors or other heat sources and the air conditioned rooms) to the sorption process have to be rejected to the ambient due to the requirements of the continuous thermodynamic process.

If a dry heat rejection system is used, the return temperature of the cooling water will be about 7 to 10 °C higher than the dry temperature of the ambient air. This causes cooling water temperatures in the Mediterranean countries in the range of 40 to 55 °C. This requires for the operation of sorption cooling machines temperatures of the heat source of 110 °C and more, which can not be managed by relatively cheap flat plate collectors.

If a wet heat rejection system is used, the return temperature of the cooling water will be about 3 to 8 °C lower than the dry temperature of the ambient air. In these cases of application flat plate collectors can be used.

This calculation procedure for a wet cooling tower is used above all for commissioning, adaptations and changes of the technical features. In addition to this the formulas and graphs can be computerized and used for several kinds of simulations of changing the mass flows of water and air.

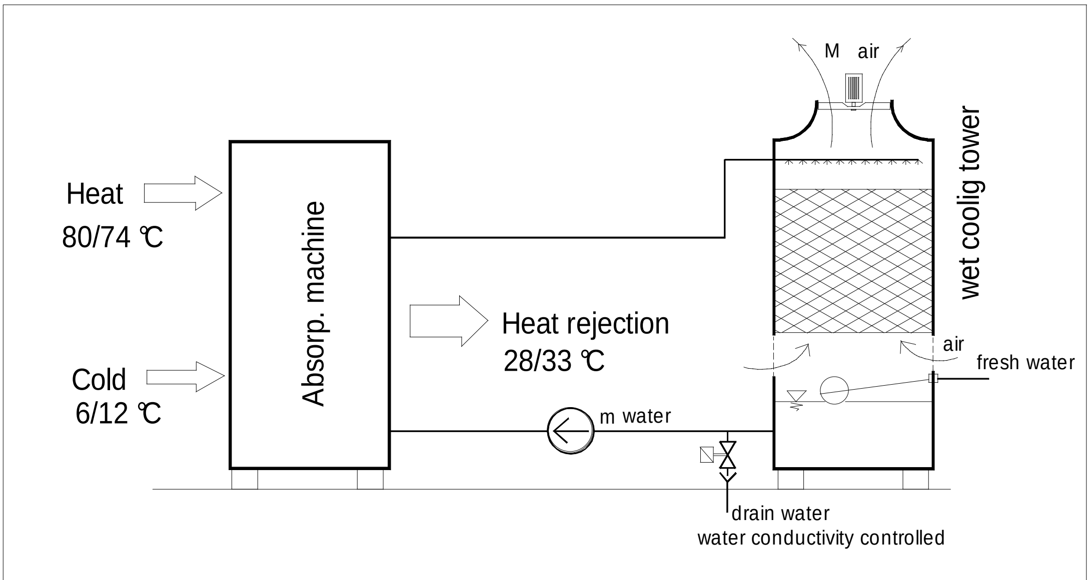


Figure 3-1: Principle of a thermal driven cooling with wet heat rejection

3.2 State of air and temperatures inside the cooling tower

The following equation describes the heat balance at the wet cooling tower.

$$m * c * (\vartheta w1 - \vartheta w2) = M * (h1 - h2) \qquad (3\text{-}1)$$

Legend

m………………… mass flow of the cooling water ………… …kg/s

M…………………mass flow of the air…………………………......kg/s

c …………………..specific isobar heat capacity kJ/kg K

$\vartheta\ w1, \vartheta w2$ ……….cooling water temperature (inlet/outlet)………°C

$h2 - h1$ …………..enthalpy difference of the air (inlet/outlet)…..kJ/kg

Figure 3-2 shows the temperatures of the cooling water between inlet and outlet in the cooling tower and also the state of the air on the way from the entrance to the outlet.

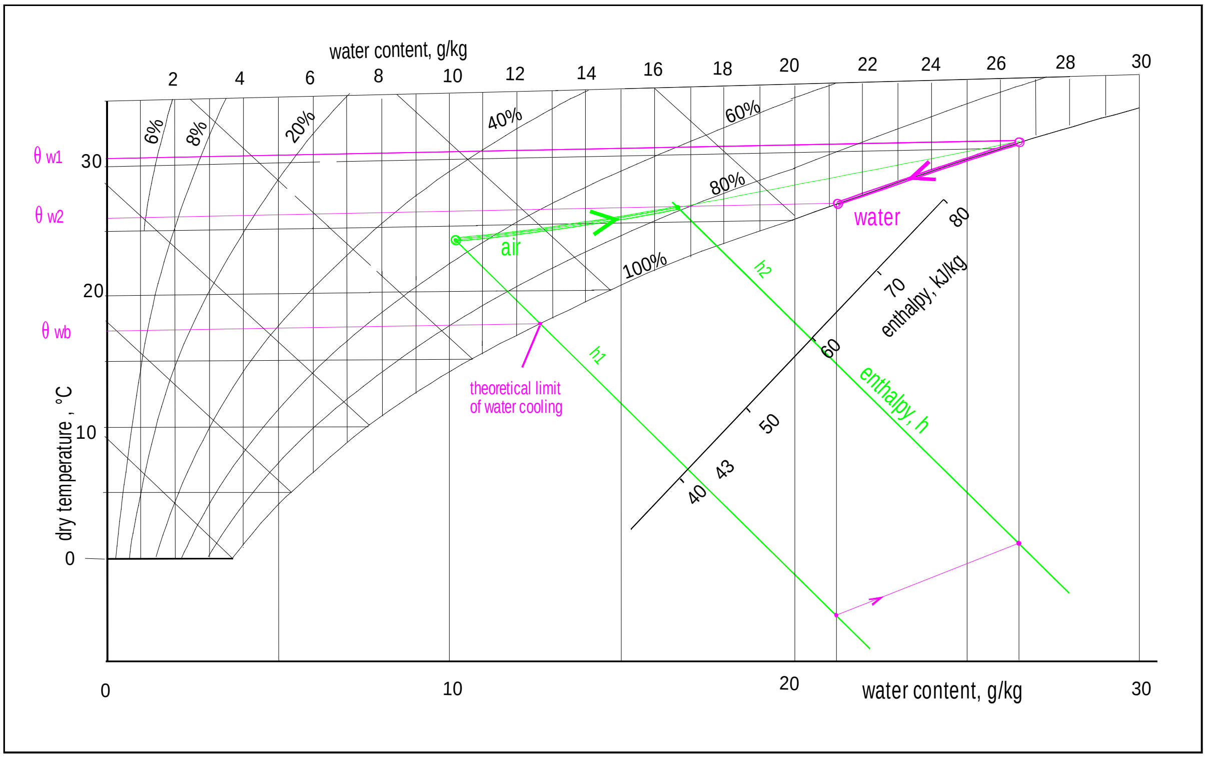


Figure 3-2: States of the air and temperatures of the cooling water inside of the cooling tower

The theoretical limit of water cooling is given by the wet bulb temperature of the air. In this case the contact area between air and water has to be infinite. In practice only a part of the theoretical water cooling gradation is realized due to economical reasons of the dripping body and air flow. The "water cooling gradation" η describes this part of the theoretical possible water cooling.

$$\eta = \frac{\vartheta w1 - \vartheta w2}{\vartheta w1 - \vartheta wb} \qquad (3.2)$$

η ……….. water cooling gradation …………………… -

ϑwb …….wet bulb temperature ………………………℃

Following the information in Recknagel Sprenger (1997/98) the water cooling gradation (η) can be calculated with the aid of the cooling tower operational data (M, m), the cooling tower constant (C_K) and the state of the ambient air at the location of the cooling tower. The important result is the return temperature of the cooling tower, which can be determined with the cooling ratio η.

In a first step the *relative minimum quantity of air (l_{min})* is determined with the aid of the diagram in Figure 3-3 and the state of the ambient air at the site. A typical state of the ambient air is assumed with $\vartheta\ w1$ = 33 ℃, and φ = 54% at noon on a sunny summer day. Out of the h,x-diagram the wet bulb temperature can be determined with 16,5 ℃. With the aid of the wet bulb temperature and Figure 3-3 now the relative minimum quantity of air can be found.

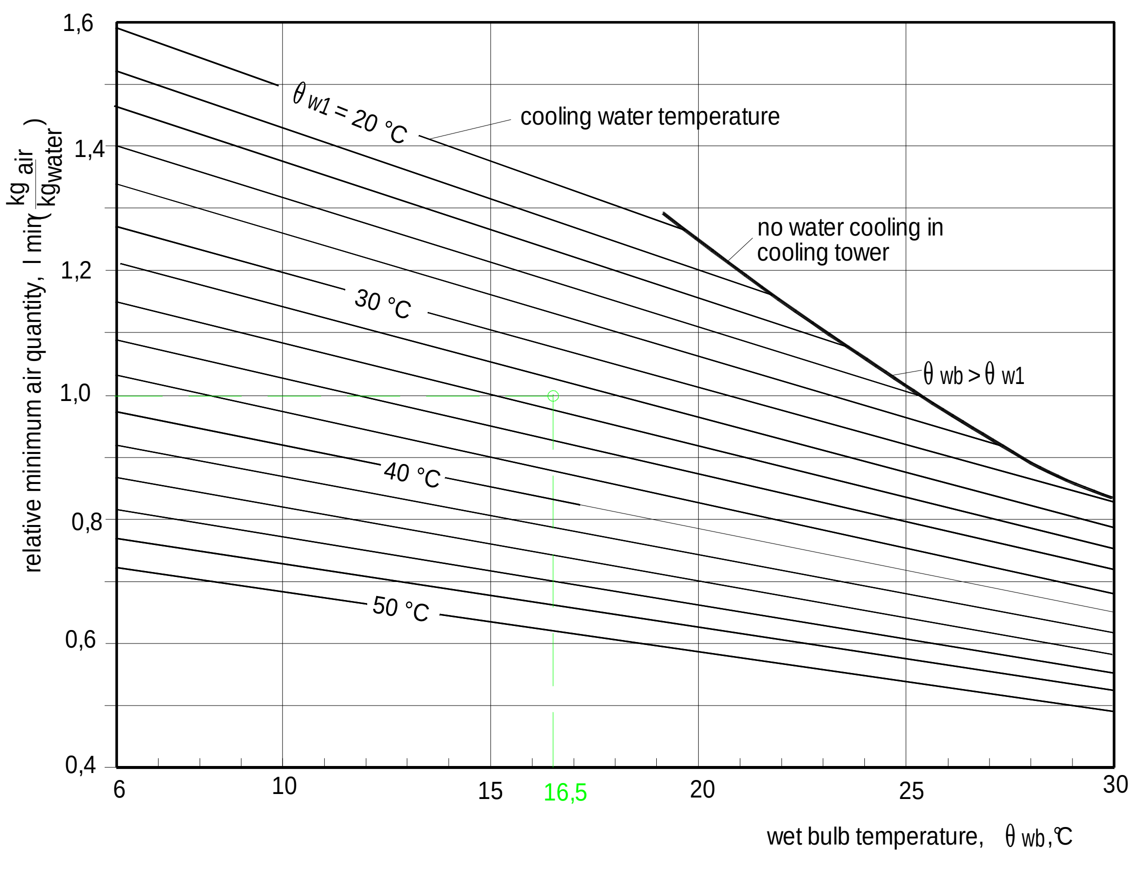


Figure 3-3: Relative minimum quantity of air (Recknagel Sprenger, 1997/98)

The relative minimum air quantity is determined with l_{min} = 1,0

For the further calculations the equations below are used:

$$l_{\min} = \frac{M_{\min}}{m} \tag{3.3}$$

$$l_0 = \frac{M}{m} \qquad (3.4)$$

$$\lambda = \frac{l_0}{l_{min}} \qquad (3.5)$$

l_{min} ……………relative minimum air quantity……… ……………………..-
M_{min} …………..minimum air quantity, corresponding to (h2 – h1) ………kg/s
(M_{min} is the minimum air quantity for an <u>ideal cooling tower</u>
to cool down the water from ϑ_{w1} to ϑ_{wb}

l_0 …………….. relative real air quantity ……….…………………………..-

M ……………..real air quantity …………………………………………kg/s
λ ……………. air ration of a wet cooling tower ………… ……………. –

The next step is the calculation of the cooling tower constant. This can be managed by a relation between the water cooling gradation and the air ratio. Equation 3.6 shows this.

$\eta = C_k * (1 - e^{-\lambda})$ …………………………………………………………………………..3.6

C_k ……………...cooling tower constant ……………………………..………-

C_k can be determined by a diagram which is provided by the cooling tower producers and is shown in Figure 3-4

The water cooling gradation (η) can now be read out of the diagram in Figure 3-5

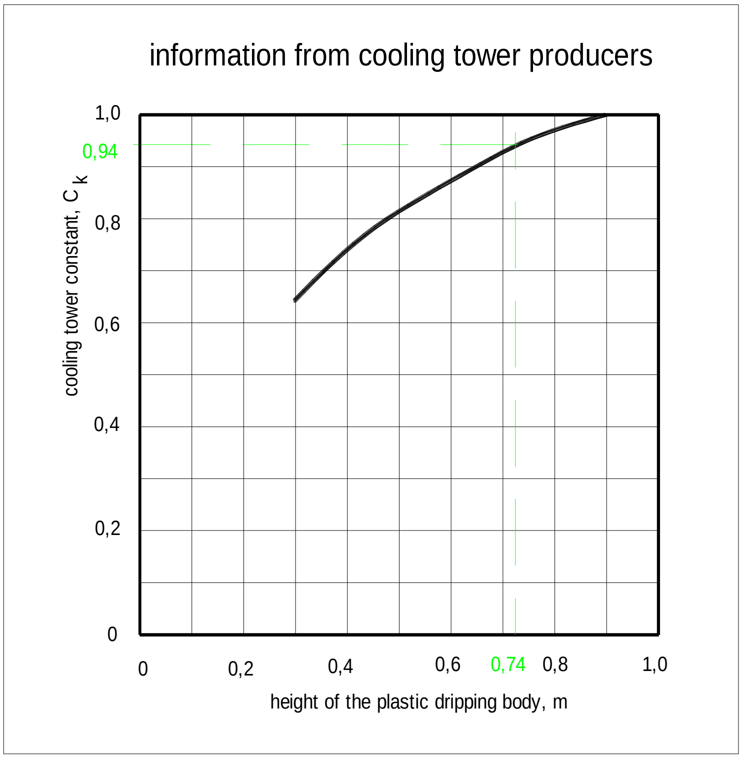


Figure 3-4: Cooling tower constant as a function of the height of the dripping body (on the basis of Recknagel Sprenger, 1997/98)

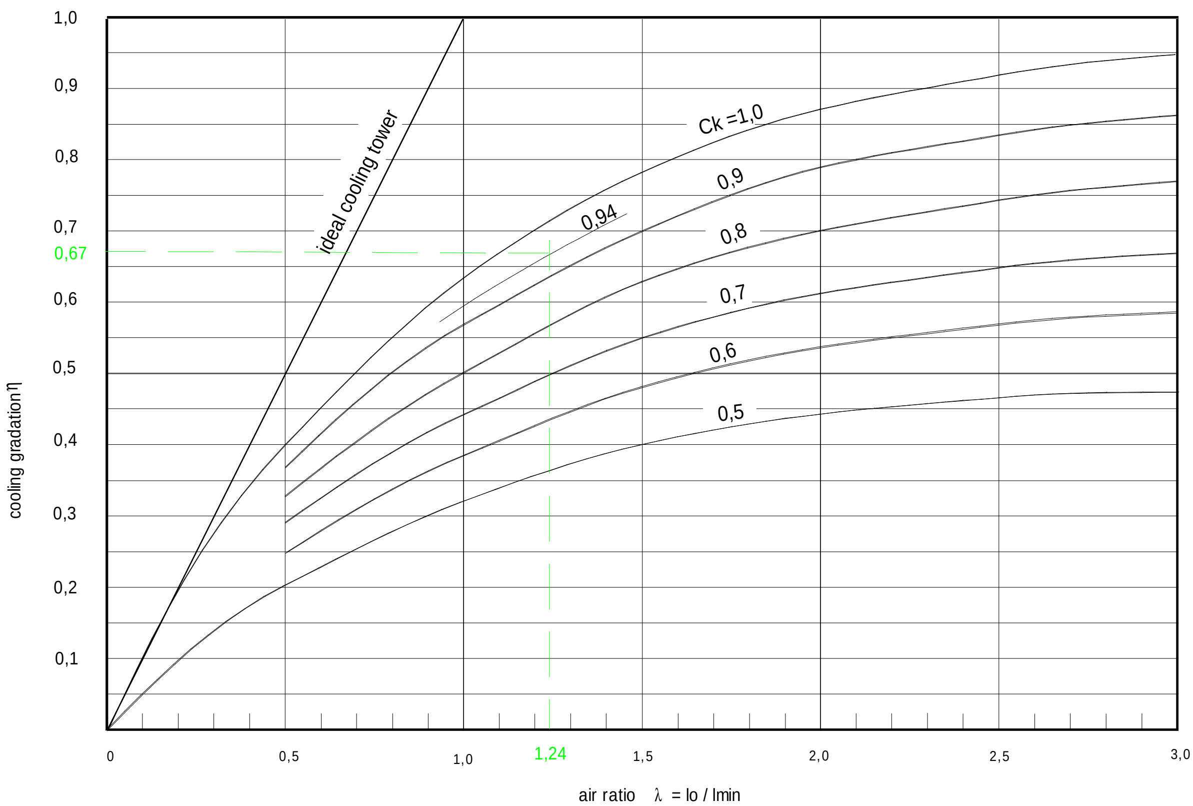


Figure 3-5: Cooling gradation η as a function of the air ratio with the parameter of the cooling tower constant (Recknagel Sprenger, 1997/98)

3.3 Example

Cooling water:	ϑ_{w1} = 33 ℃
Cooling water mass flow	m = 1,11 kg/s = 4.000 kg/h
Air flow rate	M = 1,38 kg/s (information of the producer)
Wet bulb temperature	ϑ_{wb} = 16,5 ℃
Ideal minimum air ratio	$l_{\min}$ = 1,0
Real air ratio	l_0 = 1,243
Air ratio of a wet cooling tower	$\lambda = \frac{l_0}{l_{\min}} = 1{,}24$
Cooling gradation	$\eta =$ 0,67 (out of figure 3.5)
Result: Cooling water ϑ_{w2}	$\vartheta_{w2} = \frac{\vartheta_{w1} - \vartheta_{w2}}{\vartheta_{w1} - \vartheta wb}$ = 0,64 $\rightarrow \vartheta_{w2} = 22{,}09°C$

4 Legionella

Due to the fact, that limitation of micro-organism and especially Legionella is an important issue in many applications like water supply, domestic hot water systems, spas, cooling technologies etc. a lot of corresponding literature can be found.

According the risk of contracting to Legionnaires' disease associated with cooling towers the following guidelines have been found and reviewed:

- EUROVENT 9/5 (2002)
- VDMA 24649 (2005)
- ASHRAE Guideline 12-2000
- CTI Guideline WTB 148 (2008)
- IIR 18th Informatory Note on Refrigeration Technologies (2005)

In Austria the following national standards dealing with hygienic aspects in "Heating Ventilation and Air Conditioning" equipment were found:

- ÖNORM H6021 (2003) Lüftungstechnische Anlagen – Reinhaltung und Hygiene
- ÖNORM H6020-2 (2007) Lüftungstechnische Anlagen in Krankenanstalten – Betrieb, Instandhaltung, technische und hygienische Kontrollen,
- ÖNORM B5019 (2007) Hygienerelevante Planung, Ausführung, Betrieb, Wartung, Überwachung und Sanierung von zentralen Trinkwassererwärmungsanlagen

The ÖNORM H 6021 states that wet cooling tower need to have regular blow down and at least twice a year mechanical cleaning and water quality monitoring. If the water quality is not sufficient the period of time between cleaning and monitoring measures has to be reduced.

The description in the subsequent sections is manly derived from these guidelines and shall give a short overview how it is generally intended to avoid a too high contamination of Legionella bacteria in the cooling water system. The values given below should only be used for orientation; some local or national regulations may differ from these values.

4.1 Legionellosis

Legionella is a family of bacteria, commonly present in low concentrations, in natural and man-made aquatic environments. Most of them are not virulent. However, pneumophila causes legionellosis, which have two distinct clinical forms:

- Legionaire´s disease is a form of pneumonia. The fatality rate is estimated by 10 to 20%.
- Pontiac fever is an easily treated flu-like illness.

Legionnaires' disease is an uncommon but serious form of pneumonia. Although healthy people can develop Legionnaires' disease, but mainly people who are susceptible to an infection of this kind like smokers, patients with cancer etc. are affected. People are contracted by inhaling contaminated aerosols deeply into there lungs and not by drinking contaminated water.

Over 40 species of Legionella are known. The Legionella Pneumophila (LP) appears to be the most virulent and is associated with approx. 90% of cases of Legionnaires' disease. The bacterium is commonly found in surface water and is likely to exist in low concentration in most water systems (ASHRAE Guideline, 12-2000).

4.2 Conditions of Legionella Growth

Legionella and similar bacteria develop in ground, surface water and mud. They grow in slime and biofilms, which are layered groups of microbial populations. The biofilms protect the bacteria from inactivation agents and provide nutrients. If the conditions for the growth are well, the proliferation of the Legionella and also of other bacteria may increase significantly.

Legionella growth is sensitive to the prevailing temperature level of the water. The following temperature ranges can be distinguished:

- 70 - 80℃ Disinfection range
- 60 ℃ 90% of Legionella die within 2 minutes
- 20 - 50℃ Legionella growth range
- 35 - 46℃ Ideal growth range
- < 20 ℃ Legionella can survive but do not grow

In solar cooling systems the cooling water temperatures will be at a critical level for Legionella growth most period of operation time.

Furthermore a ph-value between 5.5 and 8.5 and the presence of nutrients like sediments, sludge, corrosion debris, untreated wood or natural rubber support microbiological growth. Biofilms, algae, slimes and fungi also provide nutrients and protection for Legionella multiplication especially in stagnant water e.g. in hydraulically dead ends which can provide a haven for the growth of Legionella. Legionella can invade and replicate within other organisms like amoeba which protect them e.g. from disinfectants – compare Figure 4-1.

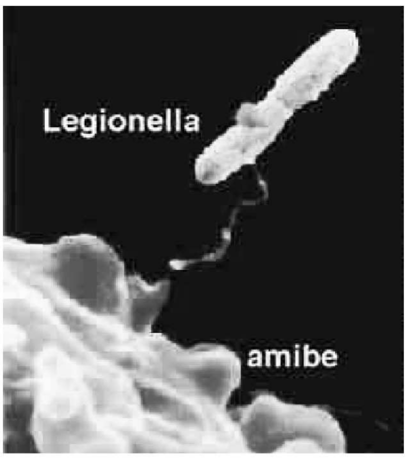


Figure 4-1 Picture of a Legionella growing on a amoeba (Aquaprox, 2007)

Water is essential for the survival of the bacteria; if the water evaporates and the Legionella dry out it will die.

4.3 Monitoring of Legionella and Micro-Organisms

The simplest method to monitor the bacteriological levels in water is to measure the total aerobic bacteria (TAB) concentration by the use of dip slides. Therefore dip slides are immersed in the cooling water and stored in an incubator for 24 - 48 hours. Depending on the concentration of total aerobic bacteria (TAB) in the water the colour of the dip slide changes and by comparison of the dip slide with a colour chart supplied by the producer the amount of total aerobic bacteria can be determined. The results are indicated in colony forming units of TAB per millilitre (cfu/ml)

Weekly monitoring of the total aerobic bacteria levels in cooling water is recommended by many legislators and professional authorities (e.g. EUROVENT 9/5, 2002; VDMA 24649, 2005) as a visual performance indicator to both system and treatment regime.

Beside dip slides ATP-based technologies can be used for real-time assessment of microbial populations. ATP is the abbreviation for “Adenosine-5'-triphosphate” which is a multifunctional nucleotide, that plays an important role in cell biology for intracellular energy transfer. The ATP-based measurement device measures the light produced when enzyme reagents react with ATP and the amount of light production correlates with the amount of ATP in the sample, which in turn is a relative measure for the microbial activity. The advantage of this system is the short determination time of several minutes.

It should be noted that dip slides and ATP-based measurements alone do not detect Legionella as a select micro-organism. However it is generally accepted that overall bacteria levels (TAB) are considered able to support Legionella and therefore are an indicator of serious risk for Legionnaires’ Disease.

Unless otherwise specified by local regulations, a concentration of total aerobic bacteria up to 10^4 cfu/ml mean the system is under control. Between 10^4 and 10^5 cfu/ml the test should be repeated and if the concentration is confirmed the biocide treatment should be increased. Concentrations of more than 10^5 cfu/ml require immediate corrective actions to reduce the bacterial level.

Most of professional and governmental agencies that have issued Legionella position statement and guidelines do not recommend testing of Legionella bacteria on a routine basis. According to CTI Guideline WTB 148 (2008) the reasons derive from difficulties in interpreting Legionella test results and the following aspects are mentioned:

- Not all Legionella serogroups are associated with Legionnaires’ Disease
- Culture-based testing methods to quantify Legionella have a 10 to 14 day turnaround for results which is too long for effective treatment control
- Legionella can repopulate within a few days and can be released from biofilms etc.
- An infectious dose level for Legionella has not been established

However, testing of Legionella is needed if:

- Legionella contamination is suspected
- The TAB concentration remains above 10^4 cfu/ml after corrective measures have been taken

EUROVENT 9/5 (2002) specifies that, if the level of Legionella bacteria has been separately tested and the result are below 10^3 cfu/l, no action is required, otherwise the test should be repeated and corrective measures are necessary. If the concentration is above 10^5 cfu/l immediate cleaning and disinfection is required.

Two international standards should be mentioned which describe cultural methods for isolation of Legionella organisms and estimation of their numbers in environmental samples. The ISO 11731 (1998) method is applicable to all kinds of environmental samples including potable, industrial and natural waters and associated materials such as sediments, deposits and slime. The ISO 11731-2 (2004) is intended for water for human use (e.g. hot and cold water, water used for washing). It is especially suitable for waters with prospected low numbers of Legionella.

The EUROVENT 9/5; 2002 guideline suggests a Typical Water Quality Monitoring Schedule shown in Table 4-1. For a specific application the local regulations has to be reviewed.

Table 4-1: Typical Water Quality Monitoring Schedule (EUROVENT 9/5; 2002)

Control Activity	**Time of Execution**
Check operation of water treatment system	Initial start-up & after seasonal shut-down period. Thereafter monthly.
Check stock of chemicals	Initial start-up & after seasonal shut-down period. Thereafter weekly.
Monitor TAB concentration	Weekly
Monitor recirculating water quality against Control Parameters	Monthly
Visual inspection for algae, biofilm formation	Every 6 months (see text)
Check LP concentration	If TAB remains high (see Table 5) after corrective action (see text). If LP contamination is suspected.
System cleaning & disinfection	Prior to start-up, annually, after a shut-down longer than one month. If TAB is above 10^5 cfu/ml. If LP concentration is above 10^4 cfu/l. If excessive growth of organic material is noticed.

Note: Microbiological molecular methods for enumeration of pathogenic and other troublesome microbes are now commercially available. These methods are based on oligonucleotide probes that target the specific microorganisms of interest. For specific analyses of *Legionella* sp. a new method based on quantitative polymerase chain reaction (real time PCR or qPCR) is now commercially available and normally analyses can be conducted within a day. This tool can be used on a routine basis and is suitable for monitoring increased growth of Legionella in cooling systems. However, guidelines are still using the culture based method for detection of *Legionella* to determine the risk of *Legionella* in a cooling system. This includes also decisions for weather cooling systems should be closed down and disinfected.

5 Avoidance of Legionella and Micro Organisms

This chapter describes measures to avoid uncontrolled Legionella multiplication in order to operate the system safe. The conventional biocides for water treatment like Chlorine or Ozon are state of the art technologies and therefore described hereafter very briefly only.

Based on a survey of water treatment technologies it seems to be worth to look more in detail to two possibilities of water treatment, ultra violet light and silver copper ions. For small scale wet cooling towers these technologies seem to be promising, thus the focus of this chapter is on these two technologies. However these technologies can not be seen as state of the art and further investigations are strongly needed.

5.1 Chain of Events

An outbreak of Legionnaires' Disease associated with a cooling tower requires a 'Chain of Events' with all events in the chain linked together and occurring in sequence (comp. Figure 5-1)

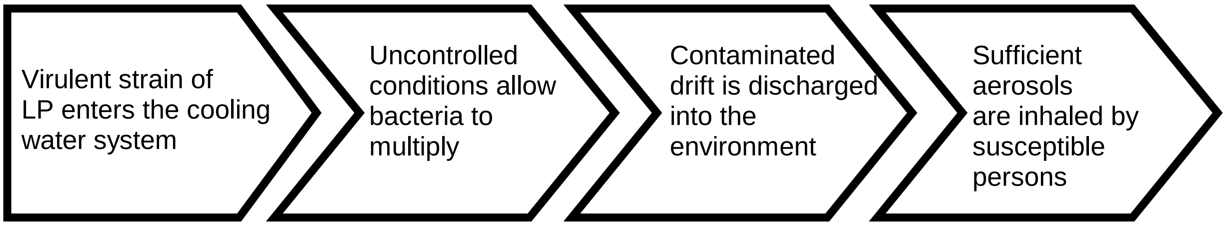


Figure 5-1: Chain of Events for the outbreak of Legionnaires' Disease associated with a cooling tower (EUROVENT 9/5, 2002))

To effectively prevent the risk of Legionnaires' Disease, it is necessary to break this chain of events at any link. There are three chain links, which can be broken by good design and correct operation of the cooling system:

1. prevent conditions that encourage multiplication of bacteria
2. minimise drift or aerosol effect in the discharge air stream
3. reduce chances of inhalation by people through equipment location and/ or personal protection.

The measures mentioned above are not equally effective in terms of prevention. By far the most important measure is to prevent uncontrolled conditions that allow the bacteria to multiply.

5.2 Design and Operational Measures to Avoid Legionella

Proper system design, regular inspection and, if required, cleaning and disinfection are needed to minimise Legionella bacteria within the cooling system. With respect of the design and installation of evaporative cooling systems a large number of items should be considered:

- Easy and safe access to the cooling tower should be included for inspection and to take samples.

- Regular maintenance and cleaning should be scheduled.
- The open Cooling device should be located as far as possible from the fresh air intakes of a building the areas with organic intake, like kitchen exhaust fan.
- As far as possible remote from out door public areas.
- Reduction of the water droplet rate by high performed separator without a hindrance for inspection and maintenance

Additional to the design active measure are necessary like injection of biocide. A possible alternative could be the installation of UV-C disinfection facility including the necessary control measures and the installation of suitable metal ion electrolysis, e.g. silver-copper ion method which is discussed hereafter.

The above mentioned measures together with a regular maintenance will avoid the growth of virulent bacteria like Legionella pneumophila.

5.3 Water Treatment with Disinfectants

In wet cooling towers different water treatment chemicals are used for scale and corrosion inhibition, anti-foaming, cleaning, biofilm control and disinfection. Many different commercial products are on the market which uses different chemical substances. Within this section only a very short overview on the used biocides for Legionella control should be given. Good information on commercially available products can e.g. be found on the web pages of the companies

- Lenntech (http://www.lenntech.com/water-treatment-chemicals.htm) and
- Accepta (http://www.accepta.com/water_treatment_chemicals/biocides.asp)

In principle the biocides for Legionella control can be divided up into oxidising biocides and non-oxidising biocides.

The commonly used oxidising agents are:

- Chlorine
- Chlorine dioxide
- Ozone
- Hypochlorite

Every disinfection technique has its specific advantages and its own application area. The company Lenntech assigns the following advantages and disadvantages to different technologies (compare: Table 5-1)

Table 5-1: Pro and Cons of different water treatment technologies (http://www.lenntech.com/water-treatment-chemicals.htm, 07.04.2009; 14:53)

Technology	Environm. friendly	Byproducts	Effectivity	Investment	Operational costs	Fluids	Surfaces
Ozone	+	+	++	-	+	++	++
UV	++	++	+	+/-	++	+	++
Chlorine dioxide	+/-	+/-	++	++	+	++	--
Chlorine gas	--	--	-	+	++	+/-	--
Hypochlorite	--	--	-	+	++	+/-	--

From the table above it can be concluded, that ultra violet light (UV) and Chlorine Dioxide are the most promising technologies for the application in small scale wet cooling towers because the investment and operational cost of the used technology are moderate or low and the effectivity against Legionella is high. Chlorine Dioxide offers further the advantage to be effective against biofilms. For large installations Chlorine Dioxide is usually manufactured on site because it is an unstable gas that dissociates into chlorine gas (Cl_2) and oxygen gas (O_2). For small applications Chlorine Dioxide is also commercial available in stabilized form as powder, tablet or solution.

For the water treatment with Ozone an expansive reactor for production of the Ozone on site is necessary, thus this technology is applicable for high quantities of water only. The treatment with Chlorine or Hyperchloride is less effective compared to other options and does not act against biofilm formation.

5.4 Disinfection by UV-Radiation

This chapter describes the principle structure of bacteria and the mechanism how they can be destroyed by ultra violet light (UV). Furthermore the design of market available lamps and there possible application are discussed.

5.4.1 General

For a better understanding of the destruction mechanism in case of disinfection by UV-radiation, of water or air, the Figure 5-2 shows the principle structure of bacteria. It consists of a cell nucleus, the cell fluid and the cell wall. The cell wall consists of a Murein membrane and an additional Lipid membrane. Murein is a biochemical strong net of Polysacharine chains and Peptide chains and build up a strong protection around the cell fluid. The chemist Gram discovered, that there are two groups of bacteria. The group 1, he calls it “negative”, do have a thin Murein membrane around the nucleus. The Murein membrane is lower than 10 % of the wall dry mass. The group 2, he calls it “positive” do have a Murein membrane with 30 to 70 % of the dry wall mass.

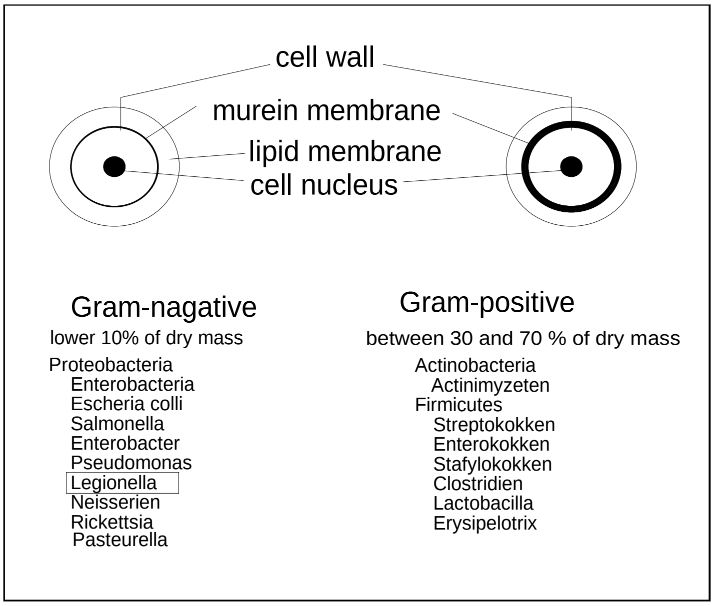


Figure 5-2: Principle construction of Gram positive and Gram negative bacteria.

Figure 4.1 characterizes a lot of well known bacteria which are assigned to the Gram-positive and Gram-negative groups. The Legionella bacteria belong to the Gram-negative group and have therefore a thin protecting Murein membrane which opens the possibility to kill the dangerous bacteria by UV-disinfection.

The ultraviolet (UV) radiation is an electromagnetic wave with a wave length in the range of 100 to 380 nm, or a frequency of more than 789 THz. Generally it is known that ultraviolet radiation is able to destroy bacteria.

Especially the Gram-negative bacteria, which have only a relatively thin Murein membrane, can be destroyed by UV-radiation with a high lethal rate. Figure 5-3 shows the range of the ultraviolet radiation and the curve of effectiveness of destroying the Gram-negative bacteria. The highest effective destruction can be detected at 254 nm in the range of UV-C radiation (200 to 280 nm)

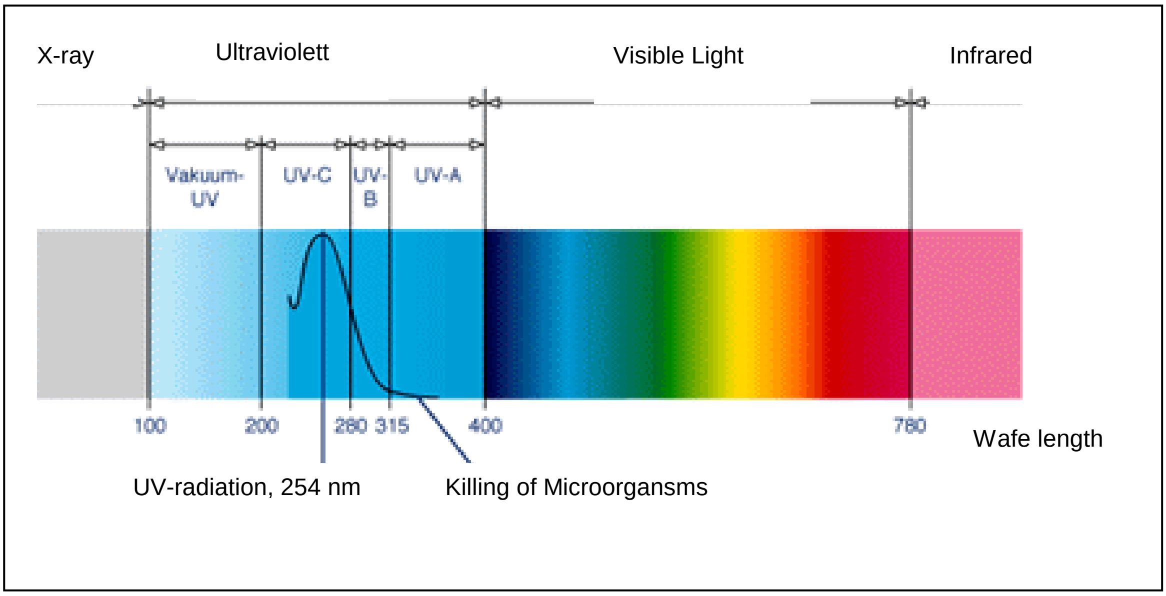


Figure 5-3: Position of the UV-C radiation in the frequency band from 100 to 750 nm

5.4.2 Mechanism of destruction

All microorganisms contain the Nuclein Acids (DNA), which carries their genetic information. The general shape of the Nuclein Acids is displayed in Figure 5-4. It is similar to a double helix with connecting rods. The Nuclein Acids absorbs the energy of the UV-radiation at 254 nm, due to their resonance frequency. The energy absorbed destroys the molecule structure and the proliferation of the Gram-negative bacteria, like Legionella, is stopped suddenly.

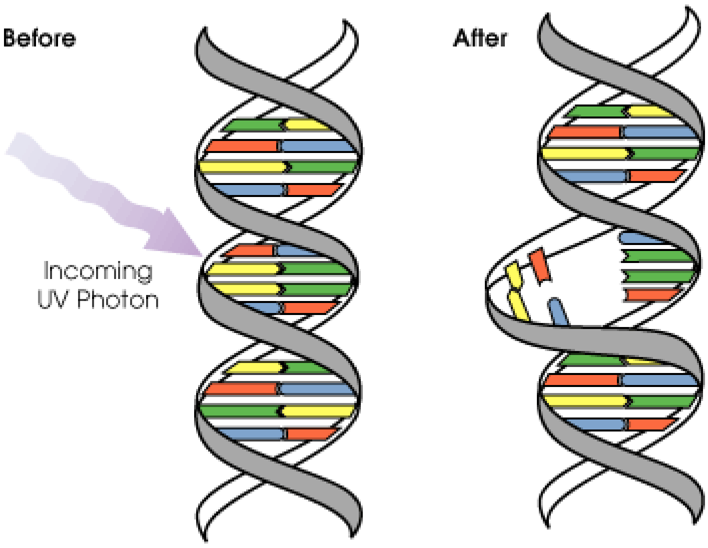


Figure 5-4: Schematic shape of the Nuclein Acid (DNA) before and after radiation treatment (http://waterecotechnology.com/?page_id=414, 09.12.2009)

5.4.3 UV-C light production

The construction of UV-C lamps needs a special technology: the "mercury vapor low pressure technology". An important component, a quartz glass tube, is necessary to envelop the mercury vapor low pressure process, which generates the UV-C radiation. Quartz glass has the important property that UV-C can pass the quartz glass wall without significant losses.

Some companies offer UV-C lamps with the above mentioned technology. Normally lamps with 18, 20, 30 and 36 Watt are available. For the production of UV-C radiation the lamp itself has to be equipped with additional electrical components, which consumes additional electric energy too. The UV-C radiation drops after an operation time of about 10.000 to 12.000 hours down to 75%. After 12.000 operation hours the change of the lamp is recommended.

5.4.4 Market available products

Figure 5-5 illustrates a product of company sterilAqua, which is besides of other similar products available on the market. The UV-C lamp including the additional components is waterproof and ready for a use in a water basin, like the basin at the bottom of a wet cooling tower. The effective penetration of the UV-C light is for relatively clean water about 30 cm. The cost of a UV-C unit regarding figure 4.5 lies in the range of 250 to 350 Euro excl. VAT.

Figure 5-5: Waterproof UV-C lamp (http://www.albkoi.de/upload/files/1-AQT-018-PDE.pdf, 09.12.2009)

A lot of other similar UV-C products are available on the market produced by several companies. Figure 5-6 shows an UV-C product for integration in a tube system, which is on a higher cost level.

The maintenance for the UV-C unit belongs to the change of the lamps after 10.000 to 12.000 operational hours and the cleaning of the quartz glass depending on the purity of the water.

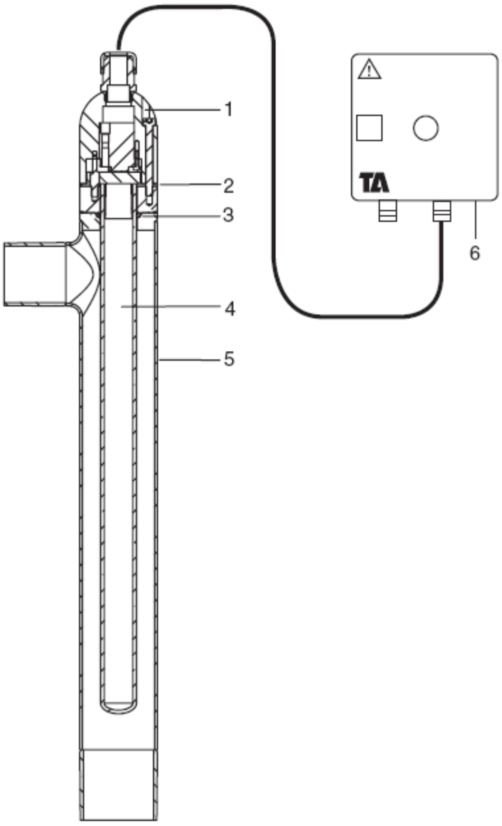


Figure 5-6: UV-C disinfection facility for integration in the installation (http://www.hydroteam.gr/pdf/TA-Aqua.pdf, 09.12.2009)

5.5 Water disinfection by metal ions

The technology of metal ion disinfection is well known since some thousand years. Already the old Egyptians used silver vessels for the drinking water, in order to keep it sterile. The old Greeks and Romans know the disinfecting effects of silver and copper too. Also the settler of North America put into the drinking water barrels silver and copper coins, which produce by the motion of the water in the barrels silver and copper ions, and the water remains potable.

In order to verify, if the disinfection effects of silver and copper ions can also be used in open wet cooling towers this old knowledge and the experience of water treatment in swimming pools has been reviewed and further investigated. The aim of this work is to make a step forward in finding a simple and cheap method to kill Gram-negative bacteria and especially Legionella bacteria in the open cooling water circuits.

This sub-chapter describes the results of a pre-test which has been carried out in order to verify the feasibility of the disinfection method with silver and cooper ions. Furthermore a method for production and control of silver and cooper ions in a water system is discussed and a test device is presented in order to show a possible technical realisation.

5.5.1 Pre-Test of the method

From the river Mur in Graz, Austria, contaminated water was taken out without a cleaning up for the test of the disinfection method with silver and cooper ions. The contamination of the Mur water was tested with the method of Petri shells. With the aid of the Petri shells the number of germs of 1 cm^3 water can be determined. The Petri shell consists of a plastic housing and a culture medium at the bottom. One cm^3 of Mur water was distributed continuously across the culture medium of the Petri shell and taken into a warm atmosphere for about 24 hours with temperatures of 33 °C, whic h is near the optimum for the growth of bacteria like Legionella.

Figure 5-7 illustrates the effectiveness of killing the bacteria by the silver/copper ion disinfection method.

Petri shell a) shows the bacteria, which are grown up without a treatment of the contaminated Mur water. 370 germs (bacteria) could be determined in 1 cm^3 of contaminated water by this Petri shell method.

Petri shell b) shows that after a treatment of only 6 hours with the silver/copper ions method the water seems to be dead and all bacteria killed. The same can be said for the Petri shells c) and d).

This simple test indicates that the silver/copper ions could be used obvious successfully in the open water cycles of wet cooling towers.

a) b) c) d)

Figure 5-7: Test of the content of bacteria in the Petri shell

a … contaminated water with out silver/copper ion treatment
b … contaminated water after silver/copper ion treatment of 6 hours
c … contaminated water after silver/copper ion treatment of 12 hours
d … contaminated water after silver/copper ion treatment of 18 hours

5.5.2 Production of silver ions and copper ions

The Faraday equation and the measurement of the electric current allow calculating, how many metal ions are produced in a certain period of time. With the help of the Petri shell tests series can be detected that a concentration of about 10^{-9} (mass of ions per mass of water) results in a satisfying bacteria dead rate. The chemical analysis of the water used in the cooling tower cycle is furthermore very important for a successful implementation of the metal ion disinfection.

The control of the silver/copper ions disinfection works in two stages. After filling or refilling the cooling water cycle with fresh water the main task of the control is the production of those number of silver ions, which enables a concentration of around 10^{-9}. After this stage one the electric current has to be reduced, so that only that quantity of the silver/copper ions are produced, which correspond to the replaced water in the cooling water cycle.

Important is also the presence of calcium, chlorine, magnesium or other kind of anions in the water. With the help of a suitable computer program like *phreeqc*, which can be cost free downloaded from the internet, and the above mentioned chemical water analysis it is possible to calculate the saturation concentration of silver ions and copper ions in the ambient of several anions The chemical analysis for the water from the river Mur in Graz was used as an example to calculate the limits of concentration. The results show that the limits of concentration for the active copper ions lies at 0.58 µg/liter, which is very low. In contrast to this the limit of the concentration of active silver ions is much higher and reaches 87 µg/liter. These results show that the silver ions can meet very easily the necessary concentration of 20 to 30 µg/liter, which lie in the range of a satisfying bacteria dead rate.

The limits of concentration of silver and copper ions depend on the kind of used water and on its chemical balance.

5.5.3 Realization of the method

A suitable concentration of silver/copper ions can be produced with the help of electric current. Only some milliamperes of electric current are necessary to produce the required quantity of silver/copper ions for the disinfection of the open water circuit of wet cooling towers. An electronic device produces this necessary electric current on a low voltage level. The electronic device is designed in such a manner that also the personal safety of the people making the maintenance is secured. Figure 5-8 and Figure 5-9 displays the realized device for the test described.

Figure 5-8: Test device for silver/copper ions production (courtesy of ECONICsystems)

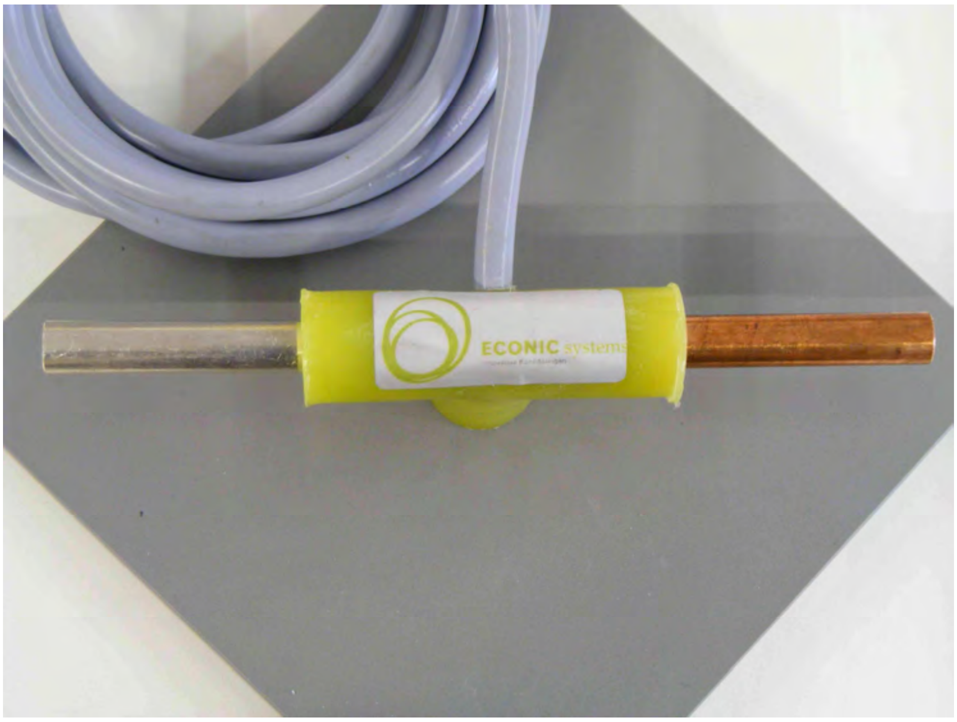


Figure 5-9: Silver/copper electrodes for cooling tower basin (courtesy of ECONICsystems)

The silver/copper electrodes are also available for integration in the cooling water pipe system.

In summary it can be said, that the first results are promising to the development of a suitable disinfection method for small scale wet cooling towers. However, further research on the disinfection method and tests in real heat rejection applications are necessary to verify the effectiveness of the disinfection method in long term tests and for different chemical composition of the cooling water. This shall be carried out in near future.

5.6 Cost comparison

After analysis and description of the two physical water treatments for a hygienic operation of small wet cooling towers, it is also worth to compare the estimated cost of the proposed measures. In the table below the investment and operation cost are listed over an operation period of 15 years.

Table 5-2 Annual cost of hygienic measures for a small cooling tower (30 kW) over a operation time of 15 years

	Investment	Operational cost	Maintenance & replacement	Remarks
	Euro	Euro/a	Euro/a	
UV-C disinfection	450[1)] to 1000[2)]	90	65	Cleaning, lamp
Silver/copper ions disinfection	100 to 300	0,5	27,5	Inspection

Note[1)]: Equipment according to Figure 5-5

Note[2)]: Equipment according to Figure 5-6

The numbers in the Table 5-2 are calculated for a small wet cooling tower with cooling capacity of 30 kW under the following conditions:

UV-C disinfection

➢ Annual operation time:	1.500 h/a
➢ Electricity demand of a 30 W UV-C lamp: (40W consumption)	60 kWh/a
➢ Electricity cost: (0,16 €/kWh)	9,6 €/a
➢ Maintenance: Change of UV-C lamp every 6,6 years (cost: 200 €/lamp)	40 €/a
➢ Maintenance time/a: 30 minutes (50 €/h)	25 €/a

Silver/copper ion disinfection

➢ 0,5 W in sum with 1500 h/a	0,12 €
➢ Maintenance silver/copper electrodes 15 €/piece, once in 6 years is	2,5 €/a
➢ Maintenance time/a: 30 Minutes	25 €/a

6 Conclusion

Heat rejection for thermally driven heat pumps is a crucial subsystem especially in solar assisted air conditioning, because:

- The necessary temperature level of the driving heat and the efficiency of the system depends on the temperature level of the heat rejection system significantly
- The amount of heat to be rejected is about twice to triple bigger than the cooling load
- The electrical energy consumption as well as the initial and operating costs of the heat rejection system are significantly high

In order to minimize the temperature level of the heat rejection wet cooling towers can be used. Wet cooling tower are able to cool down the cooling water to a significantly lower temperature than dry coolers. The typical cooling water temperature of a dry cooler and a wet cooling tower has been calculated for different climates. E.g. when the maximum cooling load is required the dry cooler delivers cooling water with a temperature level of 42.4°C and the wet cooling tower of 30.5°C in Madrid which rep resents a difference of 11.9 K. In Palermo the dry cooler delivers a temperature of 40.7°C and the wet cooling tower of 34°C thus the temperature difference is still 6.7 K.

Furthermore the electricity demand and investment costs are much lower for wet cooling towers than for dry coolers. The main drawbacks of wet cooling towers are water consumption, hygienic problems and linked to that high maintenance costs.

Large wet cooling towers are normally integrated in power stations or industrial production lines and well educated people are available for the necessary service work at the site. In contrast to this is the necessary service at small wet cooling towers below 100 kW not every time secured, due to the lack of suitable workers and owner´s technical information. Therefore a more or less automatic maintenance system for small cooling towers is highly desirable.

The need of special information of wet cooling tower technology begins with the purchase of a wet cooling tower. The owner and later on the user are interested in the technical behavior of the cooling tower. Therefore this report contains a small but easy written chapter of "Wet Cooling Tower – Process Calculation". With the help of these knowledge the commissioning, adaptations and also changes of the technical features of wet cooling towers can be executed.

The main task of an automatically working maintenance system of a wet cooling tower is the secure hygienic operation. It seems that a physical disinfection might be better than biocide injection or other chemical water treatments. Therefore especially the UV-disinfection and also the not so well known silver/cooper ion disinfection were analyzed and even tested. Due to the simple installation and low cost operation, the silver/copper disinfection method seems to be a recommendable disinfection technique for small open water cycles, like there are in small wet cooling towers. However, further research on the disinfection method and tests in real heat rejection applications are necessary to verify the effectiveness of the disinfection method in long term tests and for different chemical composition of the cooling water. This shall be carried out in near future.

Other important tasks of an automatically working maintenance system are the minimization of the water consumption and also the electric energy consumption. The measurement and the limitation of the conductivity of the water by an automatic drain valve can guarantee low water consumption. In addition to this also the speed of the cooling tower fan should be controlled and leaded by a temperature signal of the cooling water. The water temperature generates a control signal, which can be used by a broad variety of market available frequency converters for the fan.

7 Bibliography

Aquaprox (2007), „Kühlwasserbehandlung“, Springer Verlag, 2007

ASHRAE Guideline 12-2000, (2000): „Minimizing the Risk of Legionellosis Associated wit Building Water Systems“ American Society of Heating Refrigerating and Air-Conditioning Engineers, Inc.

CTI Guideline WTB 148 (2008): „Legionellosis; Guildline: Best Practices for Control of Legionella”

EUROVENT 9/5 (2002): „Recommended code of practice to keep your cooling system efficient and safe“; http://www.eurovent-association.eu/web/eurovent/web/documents.asp; (downloaded 07.04.2009, 11:30)

Henning Hans-Martin (2004) “Solar-Assisted Air-Conditioning in Buildings”, Springer Wien NewYork

Heß Stefan (2007) “Application of Medium Temperature Collectors for Solar Air-Conditioning” 2nd International Conference Solar Air-Conditioning, Tarragona, Spain

IIR 18th Informatory Note on Refrigeration Technologies (2005): International Institute of Refrigeration: “Evaporating Cooling and Legionella. “A Risk which can be prevented by Using good practices”. February 2005, www.iifiir.org.

ISO 11731 (1998) “Water quality- Detection and enumeration of Legionella”

ISO 11731-2 (2004) “Water quality- Detection and enumeration of Legionella - Part 2: Direct membrane filtration method for waters with low bacterial counts”

Meteotest, 2003: Software “Meteonorm Version 6.0.2.5”, Meteotest

ÖNORM H6021 (2003) „Lüftungstechnische Anlagen – Reinhaltung und Hygiene“

ÖNORM H6020-2 (2007) „Lüftungstechnische Anlagen in Krankenanstalten – Betrieb, Instandhaltung, technische und hygienische Kontrollen“

ÖNORM B5019 (2007) „Hygienerelevante Planung, Ausführung, Betrieb, Wartung, Überwachung und Sanierung von zentralen Trinkwassererwärmungsanlagen“

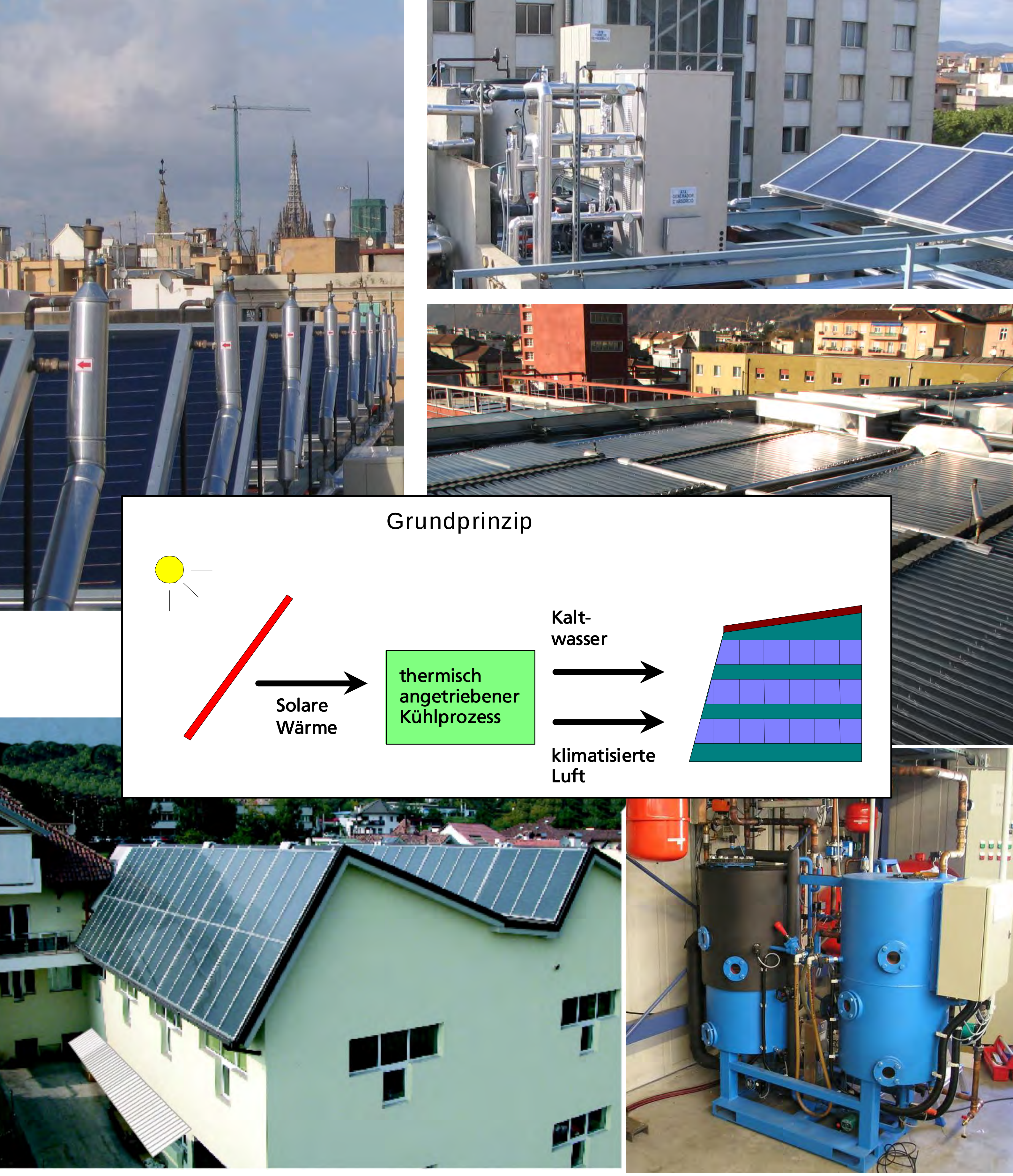
Peischl S., (2004): „Zum Vorkommen von Legionellen in wasserführenden technischen Systemen und der Wirksamkeit von Sanierungsmaßnahmen unter Praxisbedingungen“, Dissertation an der Rheinischen Friedrich-Wilhelms-Universität Bonn

Recknagel Sprenger (1997/98): „Taschenbuch für Heizung- +Klimatechnik“, Oldenbourg Industrieverlag

SWKI (2005), Richtlinie 2003-3 „Rückkühlung“, Schweizer Verein für Wärme und Klimaingenieure

VDMA 24649 (2005): „Hinweise und Empfehlungen zum wirksamen und sicheren Betrieb von Verdunstungskühlanlagen“, Verband Deutscher Maschinen- und Anlagenbau e.V

Ziegler F. (1997): „Sorptionswärmepumpen“, Habilitationsschrift, Forschungsbericht des Deutschen Kälte- und Klimatechnischen Vereins Nr. 57, Stuttgart, Deutschland



SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning and Refrigeration

Industrie Newsletter

Erste Ausgabe, 01-2009

IEA – SHC Task 38

Solar Air-Conditioning and Refrigeration

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning and Refrigeration

Hintergrund

Operating Agent:
Hans-Martin Henning
Fraunhofer-Institut für Solare Energiesysteme ISE
Heidenhofstraße 2, 79110 Freiburg
GERMANY
email: hans-martin.henning@ise.fraunhofer.de

In vielen Ländern entfällt mittlerweile ein Großteil des Energieverbrauches in Gebäuden auf die Bereitstellung komfortabler Raumklimabedingungen. Trotz ihres teilweise hohen Standards bezüglich des Energieverbrauches führen insbesondere elektrisch betriebene Raumklimageräte hierbei zu Spitzenbelastungen in den Elektroenergienetzen. Gerade die wachsende Anzahl dieser Geräte entwickelt sich zu einem zunehmenden Problem in Gebieten mit hohem Kühlbedarf durch die daraus resultierenden Energieenpässe in stark belasteten Netzen. In den vergangenen Jahren kam es deshalb in zunehmendem Maße zu sommerlichen Elektroenergieengpässen bis hin zu Blackouts, welche auf den Betrieb von Klimageräten zurückzuführen sind. Aus diesem Grund wurden in einigen Gemeinden und Regionen Gebäuderichtlinien eingeführt, um damit die Nutzung von Klimatisierungssystemen einzuschränken, falls diese nicht mit erneuerbaren Energien betrieben werden. Dadurch wird die Notwendigkeit neuer Lösungsansätze mit geringerem Energieverbrauch betont, insbesondere ein geringerer Verbrauch in Spitzenlastzeiten. Die Nutzung von thermischer Solarenergie in Kombination mit thermisch angetriebenen Kühlprozessen (Kältemaschinen, offene Sorptionssysteme) können hierbei ein möglicher Lösungsweg sein.
Das Hauptziel der internationalen Zusammenarbeit innerhalb der Task 38 “Solar Air-Conditioning and Refrigeration” im Rahmen des Solar Heating & Cooling Programmes der Internationalen Energieagentur (IEA) ist die Förderung von Maßnahmen zur beschleunigten Markteinführung von Systemen zur Solaren Klimatisierung und Kühlung mit dem Hauptaugenmerk auf verbesserte Komponenten und Systemkonzepte. Die Arbeiten in dieser Task sollen einen Beitrag zur Zunahme der Technologieakzeptanz leisten und gleichzeitig zur Überwindung technologischer und informationsbedingter Hürden beitragen.
Die Nutzung von Solarenergie für Kühlanwendungen, wie z. B. Klimatisierung, erscheint sinnvoll, weil der Kühlbedarf und die Solarerträge annähernd gleichzeitig auftreten. Bei Kälteanwendungen, wie beispielsweise im Lebensmittelbereich, ist diese zeitliche Übereinstimmung nicht notwendigerweise gegeben. Jedoch ist auch in diesem Bereich ein saisonaler Zusammenhang zwischen Kältebedarf und Solarerträgen festzustellen.

Allgemein kann unter solar unterstützter Kühlung sowohl die Elektroenergieerzeugung durch Photovoltaik und damit der Betrieb von elektrischen Kältemaschinen als auch die Erzeugung von Solarwärme mittels thermischer Solarkollektoren und deren Nutzung für einen thermisch getriebenen Kälteprozess verstanden werden. Das Interesse an thermisch angetriebenen Technologien besteht besonders bei Anwendungsfällen mit Kühl- und Heizbedarf, weil hierbei der solarthermische Teil der Installation ganzjährig genutzt werden kann.

Task 38 "Solar Air-Conditioning and Refrigeration"
Der hauptsächliche Aufgabenbereich der Task 38 sind Technologien zur Erzeugung von Kaltwasser oder klimatisierter Luft unter Nutzung von Solarwärme. Die Hauptanwendung innerhalb des Projektes ist die Klimatisierung von Gebäuden, aber auch industrielle Kühlung, z. B. im Lebensmittelbereich, wird betrachtet. Heutzutage hat die solar unterstützte Kühlung die besten Chancen für den Markteinstieg bei Anwendungen in großen Gebäuden mit zentralen Klimatisierungssystemen. Es wird aber auch ein wachsender Markt für Kühlung im kleineren Wohnbereich und kleinen kommerziellen Bereich gesehen. Hier sind insbesondere neue Lösungen notwendig, bei denen der Solarkollektor ganzjährig die Wärmeversorgung übernimmt; für die Heizung im Winter, für die Kühlung im Sommer und für die Warmwassererzeugung während des ganzen Jahres. Sogenannte vorgefertigte Systeme (pre-engineered systems)werden als Lösung für diesen Anwendungsbereich gesehen. Deshalb werden innerhalb der Task 38 sowohl kundenspezifische Systeme mit großer Leistung als auch vorgefertigte Systeme mit kleiner Leistung betrachtet.

Ziele der Task
Das Hauptziel der Task 38 ist die Förderung von Maßnahmen zur beschleunigten Markteinführung von Systemen zur Solaren Klimatisierung und Kühlung mit dem Hauptaugenmerk auf verbesserte Komponenten und Systemkonzepte.
Das wird durch folgende Aktivitäten unterstützt:

- Entwicklung und Test von Geräten und Ausstattung zu Kühlung für den Bereich von Wohngebäuden und kleinerem kommerziellen Bereich;
- Entwicklung von vorgefertigten Systemkonzepten für kleine und mittelgroße Systeme und die Entwicklung von optimierten und standardiserten Systemen für kundenspezifische Systeme;
- Berichte über Erfahrungen mit neuen Pilot- und Demonstrationsanlagen sowie die Auswertung und Leistungsbewertungprozedur;
- Bereitstellung von Begleitdokumenten zur Unterstützung von Planung, Installation und Inbetriebnahme von Anlagen zur Solaren Kühlung;
- Analyse neuer Konzepte und Technologien mit dem Schwerpunkt der Betrachtung thermodynamischer Prinzipien und eines bibliografischen Reviews

- Leistungsvergleich von verfügbaren Simulationswerkzeugen und Anwendbarkeit für Planung und Systemanalyse;
- Aktivitäten zum Markttransfer und Marktanreiz, welche Informationsschriften, Workshops und Schulungsmaterial beinhalten, außerdem die zweite Auflage des Handbuches „Solar Cooling for Planners“

Laufzeit der Task 38: September 2006 bis Dezember 2010.

Struktur der Task 38

Die Arbeiten in der Task 38 sind in 4 Subtasks aufgeteilt und jede Subtask beinhaltet verschiedene Arbeitspakete zu spezifischen Themen

Subtask A	Subtask B
Pre-engineered systems for residential and small commercial applications	**Custom-made systems for large non-residential buildings and industrial applications**

Subtask C

Modeling and fundamental analysis

Subtask D

Market transfer activities

Subtask A:
Pre-engineered systems for residential and small commercial applications

Subtask Leitung:
Dagmar Jähnig
AEE INTEC, Feldgasse 19, A-8200 Gleisdorf
AUSTRIA
Email: d.jaehnig@aee.at

Das Ziel von Subtask A besteht in der Unterstützung von Maßnahmen zur Entwicklung kleiner vorgefertigter Systeme (pre-engineered systems), die sich durch folgende Punkte auszeichnen:

- Kühlleistung < 20 kW.
- Hoher Vorfertigungsgrad des Gesamtsystems
- Kein zusätzlicher Planungsaufwand notwendig
- Vorgefertigte Systeme bestehen im allgemeinen aus folgenden Hauptkomponenten: Solarkollektor, Speicher, Back-up System, Kältemaschine, Rückkühleinheit und Steuerung. Diese Systeme können vom Installateur direkt an die raumseitigen Verteilsysteme angeschlossen werden.

Die Arbeiten in Subtask A umfassen folgende Bereiche:

- Erstellung einer Übersicht der marktverfügbaren Komponenten sowie über aktuell laufende Entwicklungen im Bereich von Kombisystemen für Heizungs- und Kühlungsanwendungen im entsprechenden Leistungsbereich.
- Basierend auf den bereits marktverfügbaren kleinen Systemen für solares Heizen und Kühlen werden generische Systemschemata erarbeitet
- Den Hauptteil von Subtask A bildet das Monitoring sowohl von Versuchsanlagen als auch von kommerziell installierten solaren Heiz- und Kühlsystemen. Bisher umfasst das Monitoring 11 Systeme, 8 weitere sollen im Lauf des Jahres 2009 hinzukommen.
- Die an den installierten Systemen gesammelten Erfahrungen werden in Richtlinien/Hilfen für Installation, Inbetriebnahme und Wartung zusammengefasst. Zusätzlich wird eine Übersicht von Erwartungen der Endanwender in Bezug auf ihre Erwartungen an Installation, Betrieb und Wartung von vorgefertigten System erstellt.

Beispiele realisierter kleiner Systeme:

Abb. 1:
Installation am Bürogebäude des Unternehmens SOLID in Graz, Österreich (17.5 kW Absorptions-Kältemaschine, Yazaki WFC-SC5). Foto: SOLID

Abb. 2:
Zwei chillii® Cooling Kit's PSC10 mit einer Kälteleistung von 20 kW sind in einer Bankfiliale in Miesbach, Deutschland, installiert. 99.8 m² Flachkollektoren erzeugen die benötigte Antriebswärme, welche in einem Warmwasserspeicher mit 7.500 l Volumen gespeichert wird. Weiterhin verfügt das System über einen 1.000 l Kaltwasserspeicher sowie über einen Nasskühlturm für die Rückkühlung. Fotos: SolarNext

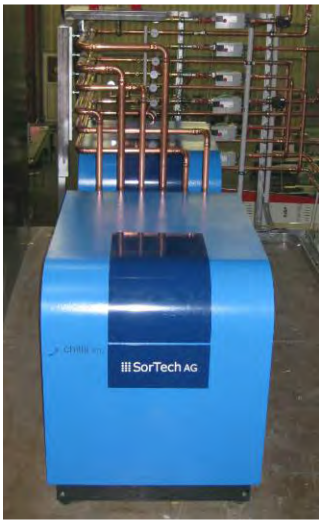


Abb. 3:
Das Bürogebäude in St. Schörfling, Österreich, verfügt über 162 m² Fassadenkollektoren als zweite Gebäudehülle. Zwei chillii® Cooling Kits STC8 werden für die Erzeugung von 15 kW Kälte genutzt. Das System besteht aus zwei Wasser-Silikagel Adsorptionskältemaschinen chillii® SCT8 und einem Wärme- sowie einem Kältespeicher mit Volumina von 15.000 l bzw. 1.500 l. Für die Rückkühlung wird ein Trockenkühlturm mit Befeuchtungseinheit genutzt. Fotos: SolarNext

Abb. 4:
In einer Seniorenresidenz in Maclas, in der französischen Rhône-Alp Region, wird der Freizeitraum sowie das Restaurant solar klimatisiert. Das System besteht aus einer Absorptionskältemaschine (Sonnenklima Suninverse, Deutschland) mit einer Kälteleistung von 10 kW und einem Kollektorfeld mit 24 m² Vakuumröhrenkollektoren. Fotos: TECSOL

Subtask B:
Custom-made systems for large non-residential buildings and industrial applications

Subtask Leitung:

Wolfram Sparber
EURAC Research, Viale Druso 1, I-39100 Bolzano
ITALY
Email: wolfram.sparber@eurac.edu

Das Ziel von Subtask B ist die Überwindung hauptsächlich technologiebezogener Hürden, um eine breitere Einführung von mittleren und großen Systemen zur solar unterstützten Kühlung zu erreichen. Diese Systeme charakterisieren sich durch:

- Kälteleistung > 20 kW.
- Individuell geplante Systeme für spezielle Anwendungen unter Einbeziehung des Planungsingenieurs
- Ausschreibungen üblicherweise für die Einzelkomponenten und nicht für das Gesamtsystem

Der Zielmarkt sind Endnutzer großer Klimatisierungs- und Kälteanwendungen (große Büro- und andere Nichtwohngebäude, Hotels, Industrie, etc.).

Die Arbeiten in Subtask B umfassen folgende Bereiche:

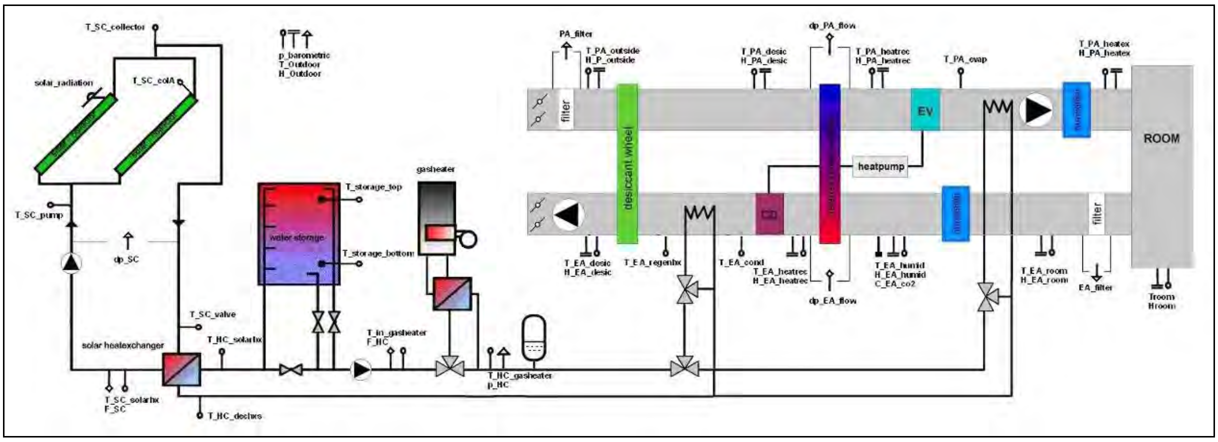
- Erstellung eines Berichtes mit einem Überblick über große Systeme zur solaren Kühlung, wobei sowohl Absorptions- und Adsorptionstechnologien als auch DEC-Systeme (Desiccant Evaporative Cooling) betrachtet werden.
- Als Unterstützung für zukünftige Installationen erfolgt eine Zusammenstellung der bisherigen Erfahrungen für Entwurf und Design der Systeme sowie für Regelstrategien.
- Einen Hauptanteil in Subtask B bildet das Monitoring von insgesamt 12 Demonstrationsprojekten großer Anlagen zur solaren Klimatisierung. Die Erarbeitung und Anwendung von allgemein akzeptierten Bewertungsverfahren garantiert die Vergleichbarkeit der Monitoringergebnisse.
- Mit dem verfügbaren Expertenwissen wurde eine Methode für einen schnellen Vorentwurf entwickelt. Damit ist eine Vorauswahl des technischen und hydraulischen Schemas in Bezug auf das Gebäude und die meteorologischen Randbedingungen, ebenso wie eine Größenauslegung der hauptsächlichen Systemkomponenten möglich.
- Ein weiteres Ergebnis der Arbeiten in Subtask B werden Richtlinien für die Ausschreibung sowie für die Installation und Inbetriebnahme sein.

Beispiele realisierter großer Systeme:


Abb. 5:
Solar unterstützte Klimatisierung des FESTO Technologiezentrum in Esslingen, Deutschland, mit 1218 m² Vakuumröhrenkollektoren und 3 Adsorptionskältemaschinen (Mayekawa ADR-100). Fotos: FESTO


Abb. 6:
Solar unterstützte Klimatisierung des Gebäudes der EURAC research Bozen, Italien, mit 615 m² Vakuumröhrenkollektoren und einer 300 kW Absorptionskältemaschine (THERMAX - THW LT 14). Fotos: EURAC

a) b)

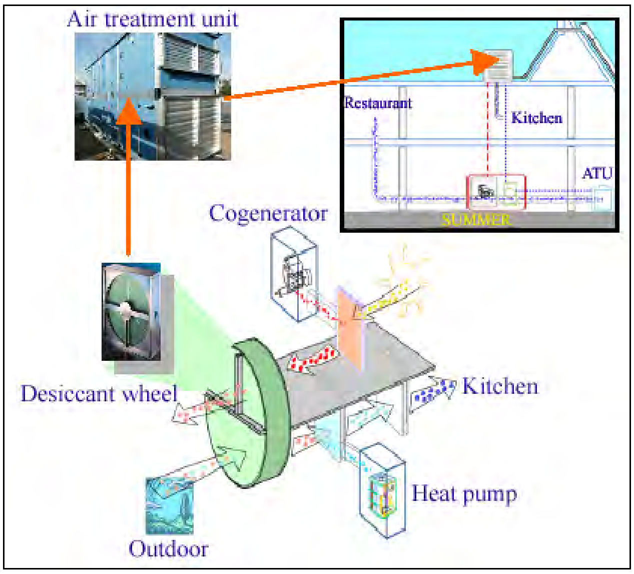


c)

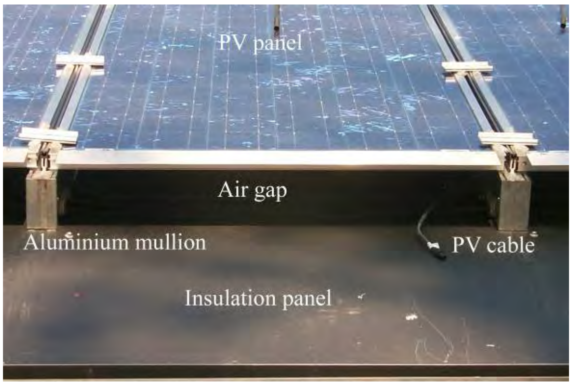
Abb. 8:

Solar unterstützte Klimatisierung des Gebäudes der Abteilung Erneuerbare Energien von INETI in Lissabon, Portugal. Die Antriebswärme für den Systembetrieb wird über eine Kombination aus Wärmepumpe und 24 CPC Solarkollektoren (44 m² Aperturfläche) bereitgestellt. Das DEC-System hat eine maximale Leistungsfähigkeit von 5.000 m³/h und versorgt 12 Büroräume mit konditionierter Luft. Fotos: INETI

a)



b)



c)

Abb. 9:

Installation eines Hybrid Photovoltaisch/Solarthermischen DEC-Systems am Fiat Forschungszentrum in Turin, Italien. Das System besteht aus einer Kombination von 163 m² Hybrid Photovoltaik-Thermie Kollektoren und 32 m² solarthermischen Kollektoren (Booster Funktion) die die Antriebswärme für das DEC-System mit einem Nennvolumenstrom von 15.000 m³/h bereitstellen. Neben der Lufterwärmung liefert das PV-System mit einer Nennleistung von 19.5 kWp einen jährlichen Ertrag von 100 MWh. Fotos: POLIMI-BEST

Abb. 10:

Solare Klimatisierungsanlage an der Technischen Universität in Saint Pierre (La Réunion), Frankreich. Die Installation wird für die Kühlung von Unterrichtsräumen (Fläche: 180 m²) genutzt und besteht aus einem Kollektorfeld mit 90 m² doppelt verglasten SCHÜCO Kollektoren, einem 30 kW SCHÜCO LiBr Absorptionskältemaschine und einem Nasskühlturm. Randbedingungen: tropisches Klima, kein Back-up. Foto: TECSOL / LPBS

Subtask C:
Modelling and fundamental analysis

Subtask Leitung:

Etienne Wurtz
Institut National d'Energie Solaire, 50, avenue du lac Léman BP 332
73375 Le Bourget du Lac Cedex
FRANCE
Email: etienne.wurtz@univ-savoie.fr

In Subtask C werden hauptsächlich folgende Ziele verfolgt:

- Weitere Entwicklung und Bewertung bereits existierender sowie neuer Komponentenmodelle und Simulationswerkzeuge in Hinblick auf deren Anwendbarkeit für Planung und Systemanalyse
- Analyse neuer und weiterentwickelter Konzepte für solare Kühlung, die noch nicht marktreif sind sondern sich noch im Forschungsstadium befinden

Die Arbeiten in Subtask C umfassen folgende Bereiche:

- Erstellung einer Übersicht über neue Entwicklungen im Bereich solarer Kühlung:
    - Geschlossene Flüssigsorptionsverfahren
    - Geschlossene Feststoffsorptionsverfahren
    - Offene Flüssigsorptionsverfahren
    - Offene Feststoffsorptionsverfahren
    - Dampfstrahlkälteverfahren
- Modellierung von offenen Flüssigkeits- und Feststoffsorptionsverfahren unter Nutzung verschiedener Software und Erstellung einer Vergleichsstudie der Simulationsergebnisse mit verfügbaren, experimentell ermittelten Daten
- Thermodynamisch/exergetische Analyse der solaren Kühlung und vergleichende Bewertung der Effizienz installierter Systeme an Hand von entsprechend erstellten Leistungskriterien
- Erarbeitung einer Übersicht und Analyse unterschiedlicher Konzepte zur Rückkühlung

Subtask D:
Market transfer activities

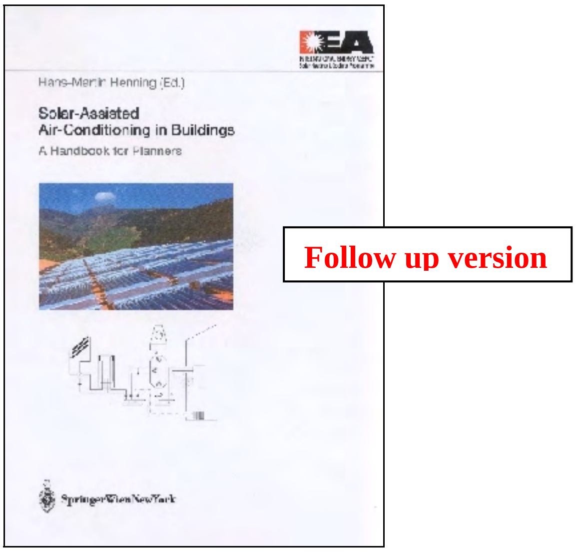
Subtask Leitung:

Mario Motta
Politecnico di Milano, Dip. Energetica, Piazza Leonardo da Vinci 32, 20133, Milano
ITALY
Email: mario.motta@polimi.it

In Subtask C werden hauptsächlich folgende Ziele verfolgt:

- Identifizierung von Zielmärkten für Technologien zur solaren Klimatisierung und Kühlung
- Informationstransfer der Ergebnisse der Task 38 zu den entsprechenden Zielgruppen

Eines der Hauptergebnisse wird die Neuauflage des Handbuches "Solar Cooling for Planners" sein, in welches die Ergebnisse aus der gesamten Task einfließen.



- Basierend auf den Ergebnisse aus Subtask A und B werden sowohl Berichte zur Lebensdaueranalyse von konventionellen Systemen und Systemen zur solaren Kühlung als auch zur Kosten- und Leistungsbewertung erarbeitet.
- Die Verbreitungsaktivitäten in Subtask D umfassen die Bereitstellung von Trainingsmaterial für Installateure und Planer von Anlagen zur solaren kühlung und außerdem die Organisation nationaler Industrieworkshops, bei denen die Ergebnisse der Task 38 präsentiert werden.

Liste der Systeme im Monitoring von Subtask A

bei denen das Monitoring bereits durchgeführt wird

Land	Stadt	Bezeichung der Installation / Projekt	Nenn-kälte-leistung	Art der Kühlung / Kältemaschine
Austria	Sattledt	Office building; Head-quarter of SOLution	15 kW	Absorption - H2O/LiBr; EAW WEGRACAL SE15
Austria	Vienna	Municipal building MA 34	7.5 kW	Adsorption – H20/LiBr; Sortech ACS08
France	Maclas	Résidence du Lac / SIEL	10 kW	Absorption - H2O/LiBr; Suninverse Sonnenklima
France	Perpignan	SOLACLIM	7.5 kW	Adsorption – H20/LiBr; Sortech ACS08
Germany	Berlin	Radiological Practice	10 kW	Absorption – H20/LiBr; Sonnenkli-ma
Germany	Freiburg	Canteen; Fraunhofer Institute FhG-ISE	5.5 kW	Adsorption - Sila-cagel/Water; SorTech (prototype)
Germany	Garching	ZAE Bayern	10 kW	Absorption - H2O/LiBr; Sonnen-klima
Germany	Moosburg	Office building; Head-quarter of CitrinSolar	5.5 kW	Adsorption - H2O/Silicagel; chillii® STC6
Germany	Rimsting	Office building; Compa-ny SolarNext	15 kW	Absorption - H2O/LiBr; chillii® ESC15
Italy	Milan	ISSA	4.5 kW	Absorption - H2O/LiBr; Rotartica Solar v45
Italy	Milan	Kindergarten Politecnico di Milano	7.5 kW	Adsorption – H20/LiBr; Sortech ACS08

Liste der Systeme im Monitoring von Subtask A

bei denen das Monitoring für 2009 geplant ist

Land	Stadt	Bezeichnung der Installation / Projekt	Nenn-kälte-leistung	Art der Kühlung / Kältemaschine
Austria	Graz	Office building; Company SOLID	17.5 kW	Absorption – H_2O/LiBr; Yazaki WFC-SC5
Austria	Gröbming	Training centre and office building Bachler	10 kW	Absorption - NH_3/H20; chillii® PSC10
Denmark		AC-Sun ApS	10 kW	Ideal Rankin / Carnot cycle with overheating; AC-Sun
Germany	Miesbach	Raiffeisenbank Miesbach	2x 10 kW	Absorption - NH_3/H20; chillii® PSC10
Germany	Stuttgart	ITW, University Stuttgart	10 kW	Absorption - NH_3/H20; Prototype
Malta	Kordin	Headquarter of Eco Group	10 kW	Absorption - NH_3/H20; chillii® PSC10
Malta	Kalkhara	Retirement home	10 kW	Absorption - NH_3/H20; chillii® PSC10
Portugal	Lisbon	AoSol	8 kW	Absorption – NH_3/H_2O; AoSol

Liste der Systeme im Monitoring von Subtask B

Land	Stadt	Bezeichnung der Installation / Projekt	Nenn-kälteleistung (Volumenstrom bei offenen Systemen)	Art der Kühlung / Kältemaschine
Australia	Ipswich	Ipswich Hospital	300 kW	Absorption – BROAD BZH 25 (double effect)
Austria	Rohrbach	BH Rohrbach	30 kW	Absorption – EAW
Austria	Gleisdorf	Town hall	6250 m³/h 35 kW	DEC system Absorption – Yazaki WFC 10
Belgium	Brussels	Renewable Energy House	35 kW	Absorption – Yazaki WFC 10
Denmark	Skive	Municipal administration building of Skive	70 kW	Absorption
France	La Réunion Island	RAFSOL	30 kW	LiBr - Absorption chiller – SCHÜCO
Germany	Ingolstadt	AUDI logistic center	8000 m³/h	DEC system
Germany	Esslingen	FESTO technology center	1.05 MW	Adsorption – 3x MAYEKAWA ADR-100
Italy	Bolzano	EURAC	300 kW	Absorption – THERMAX – THW LT 14
Italy	Turin	ECOMENSA, Fiat Research Center	15.000 m³/h	Adsorption – DEC system
Italy	Palermo	DREAM	1250 m³/h	DEC system
Portugal	Lisbon	Renewable Energy Department INETI	5.000 m³/h	DEC system
Spain	Valladolid	CARTIF, Boecillo Technology Park	35 kw	Absorption – Yazaki WFC 10
Spain	Barcelona	PERACAMPS	35 kW	Absorption – Yazaki WFC 10

Allgemeine Task 38-bezogene Veröffentlichungen:

Sparber, W., Napolitano, A. and Schmitt, Y.: Solares Kühlen & Heizen - aktueller Stand installierter Systeme großer Leistung und ein Ausblick auf neue Gebäude. Energy Forum - Solararchitektur & Solares Bauen. Brixen (I): December 2007.

Sparber, W., Napolitano, A.: Solar cooling, il condizionamento alternativo e pulito. Indagine sugli impianti di raffrescamento solare installati in Europa, in Ilsoleatrecentosessantagradi / Ilsolea360gradi, Newsletter mensile di ISES, Anno XV – n°8 settembre 2008

Sparber, W., Napolitano, A. and Melograno, P.: Overview in world wide installed solar cooling systems. 2nd International Conference Solar Air Conditioning, Tarragona – Spain, October 2007

Presentations auf der EUROSUN 2008:

(Published in the proceedings of the EUROSUN 2008, the 1st International Conference on Solar Heating, Cooling and Buildings, Lisbon, Portugal, October 7 to 10)

Aprile, M., Ayadi, O. and Motta, M.: The application of a novel solar refrigeration concept in the Tunisian food and agro-industry: Simulations and first experimental results.

Ayadi, O. Doell, J.Aprile, M. Motta, M. and Núñez, T.: Solar energy cools milk.

Beccali, M., Finocchiaro, P., Luna, M. and Nocke, B.: Monitoring of a solar desiccant cooling system in Palermo, (Italy). First results and test planning.

Besana, F., Franchini, G., Perdichizzi, A., Rodriguez, J., Sparber, W. and Witte, K.: Heat rejection as a control strategy for Solar Combi+ systems.

Bongs, C., Morgenstern, A. and Henning, H.-M.: Modelling and first experimental characterization of a sorptive heat exchanger prototype for application in a novel desiccant evaporative cooling cycle.

Bourdoukan, P., Wurtz, E., Joubert, P. and Spérandio M.: A sensitivity analysis of a desiccant wheel.

Jakob, U. and Saulich, S.: Development and Investigation of solar cooling systems based on small-scale sorption heat pumps.

Jones, B. M. and Harrison, S. J.: First results of a solar-thermal liquid desiccant air conditioning concept.

Kohlenbach, P., Rossington, D. and Weigand, A.: A novel material for desiccant wheels: Performance testing results

Koller, T., Zetzsche, M., Brendel, T. and Müller-Steinhagen, H.: Operation of a small scale ice store.

Kühn, A., Corrales Ciganda, J. L. and Ziegler, F.: Comparison of control strategies of solar absorption chillers.

Marletta, L., Evola, G. and Sicurella, F.: Energy and exergy analysis of advanced cycles for solar cooling.

Mehling, H., Hiebler, S., Schweigler, C., Keil, C. and Helm, M.: Test results from a latent heat storage developed for a solar heating and cooling system.

Minds, S. and Ellehauge, K.: The AC-Sun, a new concept for air conditioning.

Mugnier, D. and Le Denn, A.: Fast pre-design method for the selection and the pre-design of solar cooling systems in buildings.

Napolitano, A.: Coupling solar collectors and co-generation units in solar assisted heating and cooling systems.

Núñez, T., Nienborg, B. and Tiedtke, Y.: Heating and cooling with a small scale solar driven adsorption chiller combined with a borehole system.

Pietruschka, D., Jakob, U., Hanby, V. and Eicker, U.: Simulation Based Optimisation of a Newly Developed System Controller for Solar Cooling and Heating Systems

Sparber, W., Thuer, A., Besana, F., Streicher, W. and Henning, H.-M.: Unified monitoring procedure and performance assessment for solar assisted heating and cooling systems

Wiemken, E.: Solar cooling in the German funding program SOLARTHERMIE 2000plus.

White, S.D., Kohlenbach, P. and Bongs, C.: Desiccant Cooling System Modelling and Optimisation

Witte, K. T., Albers, J., Krause, M., Safarik, M., Besana, F. and Sparber, W.: Absorption chiller modelling with TRNSYS - requirements and adaptation to the machine EAW Wegracal SE 15.

Zetzsche, M., Koller, T. and Müller-Steinhagen, H.: Solar cooling with an ammonia/water absorption chiller

Task 38 - Teilnehmer

AC-Sun **Rudolfgardsvej 19** **DK-8260 Viby J.** **DENMARK** **www.ac-sun.com**	**AEE INTEC** **AEE - Institute for Sustainable Technologies** **Feldgasse 19** **A-8200 Gleisdorf** **AUSTRIA** **www.aee-intec.at**	**AIGUASOL ENGINYERIA** **C/ Roger de Llúria, 29 3r 2a** **08009 Barcelona** **SPAIN** **www.aiguasol.com**
Ambiente Italia S.r.l. - Research Institute **Via Orbassano 16** **10090 BRUINO TO** **ITALY** **www.ambienteitalia.it**	**AMG Energia S.p.A.** **via Ammiraglio Gravina, 2/e** **90139 Palermo** **ITALY** **www.amg.pa.it, www.ambienteitalia.it**	**arsenal research** **Giefinggasse 2** **A-1210 Wien** **AUSTRIA** **www.arsenal.ac.at**
ASIC-Austria Solar Innovation Center **Durisolstraße 7/Top 50** **4600 Wels** **AUSTRIA** **www.asic.at**	**ZAE Bayern** **Division 1: Technology for Energy Systems and Renewable Energy** **Walther-Meissner-Strasse 6** **D-85748 Garching** **GERMANY** **www.zae-bayern.de**	**Ciemat – Centro de Investigaciones Energéticas, Medíoambientales y Tecnológicas** **Unidad de Eficiencia Energética en la Edificatión** **Av. Complutense 22** **28040 Madrid** **SPAIN** **www.ciemat.es**
CIE-UNAM (Centro de Investigacion en Energia, **Universidad Nacional Autonoma de Mexico)** **Privada Xochicalco S/N** **Temixco, Morelos** **62580, México**	**Center for Renewable Energy Sources** **19th km Marathonos Ave** **19009, Pikermi Attiki** **GREECE** **www.cres.gr**	**CSIRO Division of Energy Technology** **PO Box 330** **Newcastle, NSW, 2300** **AUSTRALIA** **www.csiro.au**
Acciona Infraestructuras **Departamento de Investigación, Desarrollo e Innovación** **Polígono Industrial de Alcobendas** **C/Valportillo II, 8** **28108 Alcobendas (MADRID)** **SPAIN** **www.acciona-infraestructuras.com**	**DER/INETI - Instituto Nacional de Engenharia, Tecnologia e Inovacao** **Estrada do Paço do Lumiar** **1649-038 Lisboa** **PORTUGAL** **www.ineti.pt**	**EDF R&D - Département Enerbat** **Centre des Renardières** **Avenue des Renardières Ecuelles** **77818 Moret-sur-Loing** **FRANCE** **www.edf.fr**

Ellehauge & Kildemoes **Vestergade 48H, 2.tv.** **8000 Århus C** **DENMARK** **www.elle-kilde.dk**	**EURAC research** **Viale Druso/Drususallee 1** **39100 Bolzano/Bozen** **ITALY** **www.eurac.edu**	**Fraunhofer-Institute for Solar Energy Systems ISE** **Heidenhofstraße 2** **79110 Freiburg** **GERMANY** **www.ise.fraunhofer.de**
Fraunhofer-Institute for Environ-mental, Safety and Energy Technol-ogy UMSICHT **Osterfelder Str. 3** **46047 Oberhausen** **GERMANY** **www.umsicht.fraunhofer.de**	**Fundación CARTIF** **Energy Division, Renewable Energies Area** **Parque Tecnológico de Boecillo, parcela 205** **47151 Boecillo, Valladolid** **SPAIN** **www.cartif.es**	**HEIG-VD - School of Business and Engineering,** **Laboratory of Solar Energetics and Building Physics (LESBAT),** **Route de Cheseaux 1** **CH - 1400 Yverdon-les-Bains** **SWITZERLAND** **www.heig-vd.ch**
IKERLAN - Centro de investigation tecnològicas **Parque Tecnológico de Álava** **Juan de la Cierva 1** **01510 Miñao** **SPAIN** **www.ikerlan.es**	**Institut für Luft- und Kältetechnik Gemeinnützige Gesellschaft mbH** **Bertolt-Brecht-Allee 20** **01309 Dresden** **GERMANY** **www.ilkdresden.de**	**Joanneum Research** **Institut für Energieforschung IEF** **Elisabethstraße 5/I** **A-8010 Graz** **AUSTRIA** **www.joanneum.at**
Institut für Solartechnik SPF **Hochschule für Technik Rapperswil HSR** **Oberseestrasse 10** **CH-8640 Rapperswil** **SWITZERLAND** **www.solarenergy.ch**	**Institut National de l'Energie Solaire** **Parc Technologique de Savoie Tech-nolac** **50 av. du Lac Léman, Bâtiment Lynx- BP 258** **F- 73375 LE BOURGET DU LAC - CEDEX** **FRANCE** **www.ines-solaire.com**	**INTA** **Ctra. San Juan del Puerto-Matalascañas, km. 34** **21130 Mazagón,Huelva** **SPAIN** **www.inta.es**

University of Kassel Institute of Solar Engineering Kurt-Wolters-Str. 3 / Raum 3127 34125 Kassel GERMANY http://www.solar.uni-kassel.de/	LEPTAB - University of La Rochelle POLE SCIENCES ET TECHNOLOGIE Laboratoire d'Etudes des Phé-nomènes de Transfert Appliqués au Bâtiment Avenue Michel CREPEAU 17042 La Rochelle Cedex 1 www.univ-lr.fr/labo/leptab/	LOCIE - INES Université de Savoie Campus scientifique - Savoie Tech-nolac 73376 Le Bourget du Lac CEDEX FRANCE www.polytech.univ-savoie.fr
LPBS - Université de La Réunion http://lpbs.univ-reunion.fr/	Olymp Italia S.r.l. Via Orbassano 16 10090 Bruino (TO) ITALY www.olympitalia.it	PlanEnergi Jyllandsgade 1 DK 9520 Skørping DENMARK www.planenergi.dk
Politecnico di Milano Dip. di Energetica Piazza Leonardo da Vinci 32 20133 Milano ITALY www.polimi.it	Politecnico di Milano Dept. Building Environment Sciences & Technology (BEST) Via Garofalo 39 20133 Milano ITALY www.polimi.it	Queen´s University Dept. of Mechanical Engineering McLaughlin Hall Kingston, ON CANADA www.queensu.ca
Institute for Environmental Research & Sustainable Development (IERSD) National Observatory of Athens (NOA) Group Energy Conservation (GREC) I. Metaxa & Vas. Pavlou GR-15236 Palea Penteli GREECE www.meteo.noa.gr	S.O.L.I.D. GmbH Puchstraße 85 A-8020Graz AUSTRIA www.solid.at	Department of Mechanical Engineering, University of Malta, Msida MSD06, Malta
Solarnext AG Nordstrasse 10 D-83253 Rimsting GERMANY www.solarnext.de	Teknologisk Institut DENMARK www.dti.dk	TECSOL SA. 105 av Alfred Kastler - BP 90434 66 004 PERPIGNAN Cedex FRANCE www.tecsol.fr

Tekniker
Centro Tecnológico Tekniker
Apdo. 44
Avda. Otaola 20 · 20600 Eibar · Guipúzcoa
SPAIN
www.tekniker.es

Tecnical University of Berlin
Institut für Energietechnik, KT 2
FG Maschinen- und Energieanlagen-technik
Marchstraße 18
10587 Berlin
GERMANY
www.iet.tu-berlin.de

TU Graz
Institut für Wärmetechnik (IWT)
Infeldgasse 25/B
A-8010 Graz
AUSTRIA
www.tugraz.at

Universidad Carlos III de Madrid, Departamento de Ingeniería Térmica y de Fluidos
Escuela Politécnica Superior
Avda. de la Universidad 30
28911 Leganés, Madrid
SPAIN
www.uc3m.es

Universidad Miguel Hernández
Avenida de la Universidad
03202 Elche-Alicante
SPAIN
www.umh.es

Universidad Nacional Autónoma de México
Centro de Investigación den Energía
Departamento de Sistemas Energéti-cos
Privada Xochicalco S/N
Temixco, Morelos
62580, México
MEXICO
www.unam.mx

Universidad Politécnica de Cataluña
C. Jordi Girona, 31
08034 Barcelona
SPAIN
www.upc.es

UNIVERSITA' DI CATANIA
Dipartimento di Ingegneria Industria-le e Meccanica (DIIM)
Viale A. Doria 6 ; 95125 CATANIA
ITALY
www.unict.it

Università di Palermo
Dipt. DREAM
Viale delle Scienze 9
90128 Palermo
ITALY
www.dream.unipa.it

Solar and Systems Engineering
Institute of Thermal Engineering
University of Kassel
Kurt-Wolters-Str. 3
34125 Kassel
http://www.solar.uni-kassel.de/

Stuttgart University
Institut für Thermodynamik und Wärmetechnik (ITW)
Pfaffenwaldring 6
D-70550 Stuttgart
www.itw.uni-stuttgart.de

University of Applied Sciences Stutt-gart
Schellingstr.24
D-70174 Stuttgart
www.hft-stuttgart.de

Università degli Studi di Firenze (CREAR)
P.zza S.Marco, 4
50121 Firenze - Centralino
ITALY
www.unifi.it

Solem Consulting
Postfach 2127
71370 Weinstadt, Germany
www.solem-consulting.com
info@solem-consulting.com