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IEA TASK 38 „Solar Air Conditioning and Refrigeration" des Implementing Agreements on Solar Heating and Cooling

T. Selke

Berichte aus Energie- und Umweltforschung

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IEA TASK 38 „Solar Air Conditioning and Refrigeration" des Implementing Agreements on Solar Heating and Cooling

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Wien, Juli 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 Publikationsreihe und die entsprechende Homepage **www.nachhaltigwirtschaften.at** gewährleistet wird.

Dipl. Ing. Michael Paula
Leiter der Abt. Energie- und Umwelttechnologien
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INHALTSVERZEICHNIS

KURZFASSUNG (Deutsch)

AUSGANGSLAGE

Als die Vorgänger Task 25 »Solar Assisted Air Conditioning of Buildings« des Implementing Agreements on Solar Heating and Cooling der Internationalen Energieagentur begann, konnte man auf nur sehr wenig Erfahrung mit Anlagen zur solaren Kühlung bzw. Klimatisierung blicken. Frühere Aktivitäten, die im Wesentlichen in den Ländern USA und Japan in den 1980er unternommen wurden, fanden keine Fortsetzung in der folgenden Dekade. So war ein Neubeginn der Aktivitäten zu dieser solaren Technologie notwendig und dazu stimulierten einige neue Projekte der Task 25 die Wiederaufnahme recht erfolgreich.

Heute, findet die Solartechnologie eine wachsende Beachtung aufgrund des kontinuierlichen Wachstums der globalen Energienachfrage und ambitionierte Zielsetzungen werden weltweit vereinbart, die Beteiligungen von erneuerbaren Energiequellen im allgemeinen und Solarenergie im speziellen an der zukünftigen Energieversorgung beinhalten. Diese Ziele werden nur dann erreichbar sein, wenn alle Bereiche der Energienachfrage, d.h. auch Kühlung und Klimatisierung, berücksichtig werden.

SHC TASK 38

Die IEA SHC Task 38 setzte auf den Erkenntnisse der abgeschlossenen IEA SHC Task 25 »Solar Assisted Air Conditioning of Buildings« auf. Es zeigte sich einerseits das große Marktpotential dieser Technologie zur Gebäudeklimatisierung – insbesondere in sonnigen Regionen – und anderseits die Notwendigkeit zur Fortsetzung der Forschungsarbeit, um die Entwicklung von Systemen hoher Qualitätsstandards und zuverlässigen Anlagenbetrieb über viele Jahre zu forcieren.

In der SHC Task 38 wurden im Wesentlichen geeignete Maßnahmen zur verbesserten Markteinführung der solaren Klimatisierung und Kälteerzeugung aufgesetzt, wobei der Hauptfokus auf verbesserten Komponenten und Systemkonzepten lag.

Die SHC Task 38 wurde in vier folgender Struktur organisiert:

Subtask A: Kompaktsysteme für den Wohngebäude und kommerzielle Anwendungen im kleinen Leistungsbereich

Subtask B:	Kundenspezifische Systeme für Anwendungen im großen Nicht-Wohnbau und in Industrie
Subtask C:	Modellbildung und Grundlagenuntersuchungen
Subtask:	Markttransfer

Die SHC Task 38 »Solar Air-Conditioning & Refrigeration« war hinsichtlich Teilnehmerzahl eine der größten Einzelaktivitäten im Rahmen der SHC Programms. 49 Organisationen aus 12 Ländern haben an dieser Task teilgenommen. 16 Forschungsinstitute, 20 Universitäten und 13 private Unternehmen (Planungsbüros, Hersteller und Installationsbetriebe) haben sich regelmäßig auf den 9 Expertentreffen zur gemeinsamen Projektarbeit und Ergebnisdiskussion zusammengesetzt und ausgetauscht.

Die Ergebnisse sind in Berichten mit hoher Qualität dokumentiert und auf zahlreichen Konferenzen wurden diese verschiedensten Zielgruppen präsentiert. Die öffentlich verfügbaren Berichte sind unter http://iea-shc-task38.org/reports im pdf-Format herunter zu laden. Die SHC Task 38 hat einen essentiellen Beitrag zur Technologieentwicklung und der Markteinführung der solaren Kühlung geleistet.

Vertiefende Informationen zur SHC Task 38 finden sich im Web unter http://www.iea-shc.org/publications/task.aspx?Task=38

ÖSTERREICHISCHE BETEILIGUNG

Die Aktivitäten der Österreichischen Forschungspartner wurden im Auftrag des Bundesministeriums für Verkehr, Innovation und Technologie durchgeführt.

Österreich war mit folgenden Institutionen und Unternehmen an der SHC Task 38 beteiligt:

- AIT - Austrian Institute of Technology (former arsenal research)
- AEE INTEC, AEE - Institute for Sustainable Technologies
- ASIC - Austria Solar Innovation Center
- IWT - Institute of Thermal Engineering Graz University of Technology
- S.O.L.I.D. Gesellschaft für Solarinstallation und Design m.b.H.

Der vorliegende Bericht enthält einerseits eine Dokumentation der erreichten Ergebnisse der gesamten SHC Task38 und anderseits wird eine Auswahl der Projektarbeit der Österreich-ischen Forschungspartner Austrian Institute of Technology, Austria Solar Innovation Center und dem Institute of Thermal Engineering Graz University of Technology vorgestellt. Weiters enthält dieser Bericht eine Auflistung des Knowhow-Transfers der SHC Task38 Ergebnisse, der durch Veröffentlichungen und öffentliche Veranstaltungen erfolgte. Im Anhang befinden sich ausgewählte Ergebnisdokumentationen der einzeln österreichischen Forschungspartner

Für den Herbst 2011 ist die Herausgabe eines neuen Englisch sprachigen Handbuches zur Technologie der solaren Kühlung und Kälteerzeugung geplant. Dieses Buch wird die wesent-lichen Arbeitsergebnisse der SHC Task 38 enthalten und bietet einen umfassenden und vertiefenden Überblick zu dieser Technologie.

AUSBLICK

Internationale Forschungsarbeiten

Zusammenfassend zeigen die Resultate der SHC Task 38, dass die Qualität in allen Phasen der Projektierung von Anlagen zum solaren Kühlen eine Herausforderung ist. Das bezieht sich sowohl auf deren energetische Leistungsfähigkeit (Energieeinsparung gegenüber einem konventionellen Referenzsystem) als auch auf die Planungs- und Installationsphase. Ebenso wirken sich die eingesetzten Komponenten (z.B. Rückkühlung, Pumpen, Kältemaschinen, Kollektoren), besonders aber auch das Gesamtsystem inklusive Hydraulikkonzept und Regelungsstrategien auf die erreichbare Qualität und Zuverlässigkeit der Anlage aus.

Vor diesem Hintergrund wird auf internationale Ebene derzeit an der Definition einer Nachfolge Task im Rahmen des *Implementing Agreements on Solar Heating and Cooling* gearbeitet. Im März 2011 erfolgte dazu ein Definition Work in Paris und der derzeitige Arbeitstitel dieses neuen Tasks lautet »Quality assurance measures for solar thermally driven heating and cooling systems«.

Nationale Forschungsarbeiten

Auf nationaler Ebene arbeiten Forschungspartner unter der Leitung des AIT an der „Entwicklung einer Roadmap zur solarthermische Kühlung in Österreich“ im Rahmen des österreichischen Forschungsprogramms ENERGIE DER ZUKUNFT - gefördert aus Mitteln des Klima- und Energiefonds (Projektnummer 819031). D.h. angeregt über die internationale

Kooperation der SHC TASK 38 wurde und wird die nationale Zusammenarbeit aller relevanten Österreichischen Akteure intensiviert, um eine Technologie-Roadmap mit zukünftigen Milestones zu definieren.

Im Förderprojekt „HdZ-GLF: SolarCoolingMonitoring“ wird an der Evaluierung der Energieeffizienz und des Betriebsverhaltens von solarthermisch Kühlanlagen zur Gebäudekühlung gearbeitet. Dieses Projekt wird im Rahmen des österreichischen Forschungsprogramms Haus der Zukunft plus (FFG Projektnummer 822265) gefördert.

Weiters beschäftigen sich österreichische Forschungspartner und Unternehmen im Rahmen des Projektes „Neue Energien 2020 – SolarCoolingOpt“ mit der primärenergetische Optimierung von Anlagen zur solaren Kühlung mit effizienter Anlagentechnik und innovativen Regelstrategien. Dieses Projekt wird im Rahmen des österreichischen Forschungsprogramms Neue Energien (FFG Projektnummer 825544) gefördert.

ABSTRACT (English)

Task Objectives

The main objective of the Task is the implementation of measures for an accelerated market introduction of solar air conditioning and refrigeration with focus on improved components and system concepts. The market introduction will be supported through

- activities in development and testing of cooling equipment for the residential and small commercial sector;
- development of pre-engineered system concepts for small and medium size systems and development of optimised and standardised schemes for custom made systems;
- reports on the experiences with new pilot and demonstration plants and on the evaluation and performance assessment procedure;
- provision of accompanying documents supporting the planning, installation and commissioning of solar cooling plants;
- analysis of novel concepts and technologies with special emphasis on thermodynamic principles and a bibliographic review;
- performance comparison of available simulation tools and applicability for planning and system analysis;
- market transfer and market stimulation activities, which include information letters, workshops and training material as well as the 2nd edition of the Handbook for Solar Cooling for Planners.

Scope

The scope of the Task are the technologies for production of cold water or conditioned air by means of solar heat, i.e., the subject which is covered by the Task starts with the solar radiation reaching the collector and ends with the chilled water and/or conditioned air transferred to the application. However, although the distribution system, the building and the interaction of both with the technical equipment are not the main topic of the Task this interaction will be considered where necessary. In particular, for small scale systems which may use the solar collector as the only heat source the overall system including the building and its thermal mass is focused inn order to optimize the overall performance. The Task also covers solar refrigeration for other than comfort air-conditioning applications such as industrial processes and other applications (e.g. food conservation).

Means

The work in this Task is organised in four Subtasks:

Subtask A: Pre-engineered systems for residential and small commercial applications

Subtask B: Custom-made systems for large non-residential buildings and industrial applications

Subtask C: Modelling and fundamental analysis

Subtask D: Market transfer activities

Each Subtask consists of several work packages with specific focus and results.

Duration

Start date: September, 1st , 2006

Completion date: August, 31st , 2009

ExCo-Extention: December, 31st, 2010

EINLEITUNG

Markt der Klimatisierung weltweit

Ein beachtliches Wachstum des Marktes zur Klimatisierung ist weltweit beobachtbar. Abbildung 1 zeigt die Verkaufszahlen vom Raumklimatisierungsgeräten (RAC-Geräte) unterschiedlicher Regionen der Welt. Die Zahl verkaufter Geräte stieg weltweit von etwa 26 Mio. Stück im Jahre 1998 auf über 40 Mio. Stück im Jahr 2006 (Prognose) an. Im gleichen Zeitraum blieben die Verkaufszahlen großer zentraler Klimaanlagen beinahe unverändert. Welche Technologie zum Einsatz kommt – kleine RAC Splitgeräte, Multi-Splitsysteme, Systeme mit zentrale Kaltwassererzeugung oder Zentralklimaanlagen - hängt sowohl stark von der jeweiligen Region als auch vom Kühllastprofil ab. Die Dominanz kleiner Splitsysteme im Bereich von Privatwohnungen stellt einen weltweit gültigen Trend dar. Aus diesem Grunde sind insbesondere neue umweltfreundliche Konzepte für Klimaanlagen im kleineren Leistungsbereich von hoher Wichtigkeit und Bedeutung.

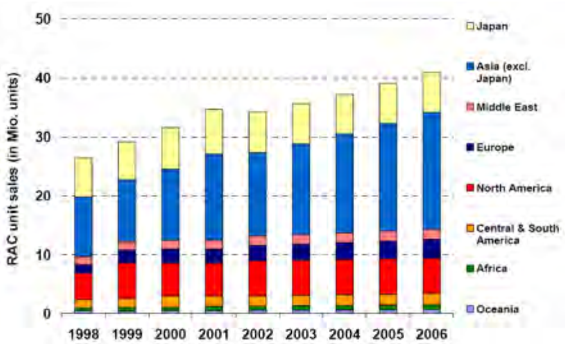

Abbildung 1 Verkaufszahlen vom Raumklima Geräten (RAC-Geräte) unterschiedlicher Regionen der Welt.

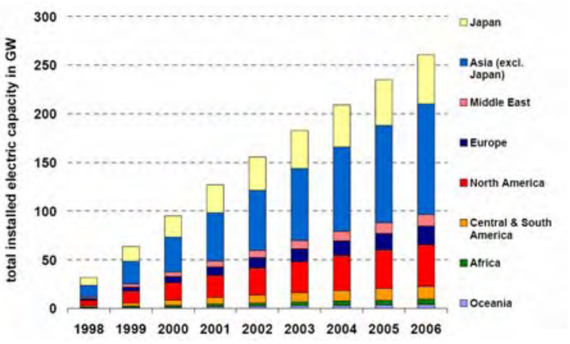

Abbildung 2 - Neu installierte elektrische Leistung aufgrund Verwendung von RAC-Geräten

Energie und Umwelt

Elektrisch betriebene Kälte- Klimageräte zeichnen sich durch relativ hohen Standard hinsichtlich ihres Energieverbrauches aus. Diese Geräte weisen hohe Stromverbräuche auf – und was noch mehr wiegt – sie verursachen signifikante Spitzenlasten im öffentlichen Stromnetz. Zunehmend stellt dies in Regionen mit hohem Kühlbedarf ein Problem dar. Abbildung 2 illustriert die neu installierte elektrische Leistung, die sich aufgrund der Verwendung von RAC-Geräten im Jahr ergibt (Annahme 10% Altgeräte werden durch neue ersetzt und haben durchschnittlich eine elektrische nominelle Leistung von 1.2 kW). In jüngster Vergangenheit stieg die Zahl der auftretenden Engpässe der öffentlichen Stromversorgung aufgrund der sommerlichen Nutzung von elektrisch betriebenen Klimageräten. In verschiedenen Teilen der Welt sind unterschiedliche Richtlinien initiiert worden, um den Einsatz von aktiver Klimatisierung ohne Nutzung Erneuerbarer Energien im Gebäude zu limitieren oder sogar zu vermeiden. Dies unterstreicht deutlich die Notwendigkeit neue technische Lösungen bei geringerem Stromverbrauch auf den Weg zu bringen. Insbesondere sind effektive Konzepte zur Verminderung der elektrischen Spitzenlast im öffentlichen Netz von hohem Interesse.

Ein zusätzlicher umweltrelevanter Aspekt betrifft das Potenzial globaler Erwärmung aufgrund der eingesetzten Kältemittel. Leckagen von Kältemittel zur Klimatisierung – insbesondere im Automotive Sektor – führten zu einzelnen Gesetzesinitiativen, die eine Limitierung oder sogar ein Verbot klassischer Fluorierter Kältemittel zum Ziele haben. Beinahe alle thermisch angetriebenen Technologien verwenden Kältemittel gänzlich ohne Potenzial zur Globalerwärmung

In der SHC Task 38 wurden im Wesentlichen geeignete Maßnahmen zur verbesserten Markteinführung der solaren Klimatisierung und Kälteerzeugung aufgesetzt, wobei der Hauptfokus auf verbesserten Komponenten und Systemkonzepten lag. Im Einzelnen wurde die Markteinführung durch folgende Aktivitäten unterstützt:

- Entwicklung und Prüfung von Komponenten der Kühl- bzw. Klimatechnologie für Wohngebäude und Gewerbe im kleineren Leistungsbereich;
- Entwicklung von Kompaktlösungen für Systeme kleinerer und mittlerer Leistung und Erarbeitung von optimierten und standardisierten Schemata kundenspezifischer Systeme;
- Berichterstellung zu Erfahrungen mit neuen Demonstrationsanlagen und zur Methode der Evaluierung und Bewertung von Betriebsverhalten;
- Erstellung von Begleitdokumenten zur Unterstützung der Planung, der Installation und der Inbetriebnahmen von solaren Klimatisierungs- bzw. Kühlanlagen;
- Analyse von neuen Konzepten und Technologien mit speziellem Schwerpunkt auf thermodynamischen Prinzipien und einem bibliografischen »Review«;
- Wissenschaftlicher Vergleich zwischen verfügbaren Simulationstools und Eignung zur Planung und Systemanalyse
- Marktorientierter Wissenstransfer und Maßnahmen zur Marktstimulation, es wird neben der Bereitstellung von Informationsmaterialien und Ausbildungsunterlagen sowie der Durchführung von Workshops auch eine zweite Ausgabe des Handbuches

zur solaren Klimatisierung und Kühlung für Planer herausgegeben. Geplante Erscheinung ist Ende 2011.

Methoden und Inhalte

Zur Erreichung der oben angeführten Ziele ist die Task 38 in vier Subtasks unterteilt. Im Weiteren wird auf die Inhalte und Methoden eingegangen.

Subtask A: Kompaktsysteme für den Wohngebäude und kommerzielle Anwendungen im kleinen Leistungsbereich

Diese Subtask konzentriert sich auf unterstützende Maßnahmen zur Entwicklung von Kompaktsystemen im kleinen Leistungsbereich. Die betreffenden Systeme haben eine nominelle Leistung kleiner 20 kW Kälte und zeichnen sich durch einen höheren Grad der Vorfertigung im Vergleich zum Gesamtsystem aus. Kein zusätzlicher Planungsaufwand ist erforderlich. Die Kompaktsysteme – im Wesentlichen bestehend aus Solarkollektor, Speicher, Backup-System, Kälteerzeuger, Rückkühlung und Regelungseinheit – können direkt vom Installateur mit den raumseitigen Haustechnik Komponenten verbunden werden. Erfahrungen durch Entwicklungen neuer Kälteerzeuger werden in dieser Subtask einfließen. Die Subtask setzt sich aus folgenden Arbeitspaketen zusammen:

- *A1: Marktüberblick* - Ein umfangreicher und detaillierter Marktüberblick bestehender Komponenten wird erarbeitet und Marktspezifische Bedürfnisse und Erfordernisse im technischen und ökonomischen Sinne werden skizziert. Neue Komponenten und *Methoden* - derzeit in der Entwicklungsphase – werden in die Bestandaufnahme inkludiert und ihre Aussichten für den Markt der solaren Klimatisierung und Kälteerzeugung bewertet.
- *A2: Sammlung ausgewählter Auswahl der Systemauslegung und Regelungsstrategien* - Durch vergangene und heutige Erfahrungen mit Anlagen zur solaren Klimatisierung (kleinerer/mittlerer Leistung) werden Auslegung und Regelungsstrategien abgeleitet, die einen einwandfreien Betrieb der Anlage mit optimierter Performance und hoher Zuverlässigkeit sicherzustellen. Ein besonderes Augenmerk wird auf die Standardisierung der Systemauslegung gelegt.
- *A3: Feldtest-Monitoring und Ergebnisse* - Es werden Anlagen zur solaren Klimatisierung (kleiner Leistungsbereich) im Feldtestbetrieb dokumentiert und bewertet. Die Monitoring-Daten werden evaluiert unter Berücksichtigung der verschiedenen Rahmenbedingungen und Anwendungsbereichen. Die dokumentierten

Erfahrungen vom realen Anlagenbetrieb bilden die Wissensbasis für zukünftige Installationen.

- *A4: Vorschläge für den Bewertungsprozess* - In diesem Arbeitspaket wird eine Zusammenfassung von Ideen zur Bewertungsmethode ausgearbeitet. Diese bildet die Grundlage, um installierte Anlagen vergleichend zu bewerten unabhängig vom Standort und von anderen Rahmenbedingungen.
- *A5: Leitfäden zur Installationen und Inbetriebnahme* - Die vorangegangenen Erfahrungen und Erkenntnisse werden in Leitfäden zur Installation und Inbetriebnahme gebündelt.

Subtask B: Kundenspezifische Systeme für Anwendungen im großen Nicht-Wohnbau und in Industrie

Diese Subtask konzentriert sich auf unterstützende Maßnahmen zur Entwicklung von Systemen der solar unterstützten Kühlung mittlerer und größeren Leistung (> 20 KW Kälte). Ziel der Aktivitäten ist eine Überwindung der technischen Barrieren zur vermehrten Einführung dieser Systeme. Marktzielgruppe bilden hier Anwender großer Klima- und Kälteerzeugungsanlagen (Großbüro, und Nicht-Wohnbau, Hotels, Industrie etc.). Einige unterschiedliche Demoanlagen werden dazu aufgesetzt und durch ein kontinuierliches Monitoring und eine Bewertungsmethode untersucht und analysiert.

- *B1: Marktüberblick* - Dieses Arbeitspaket beinhaltet eine Erhebung von neuesten Fakten marktverfügbarer thermisch angetriebener Kälte-/ Klimatechnologie und geeigneter solarer Systemkomponenten. Basis bildet der derzeitige Stand der Technik in umgesetzten Anlagen. Eine Marktanalyse zur Darstellung des Potenzials solarer Kühlung und Klimatisierung insbesondere für das Hotelwesen und die Nahrungsmittelbranche wird durchgeführt.
- *B2: Systemauswahl und Anlagenregelung* - Ziel ist die Unterstützung der Marktakteure im Planungsprozess. In Form einer Auswahl erprobter Anlagendesigns mit hydraulischen Schemata, werden Auslegungshilfen zur Kühlung/ Klimatisierung und Lüftung von kommerziellen Gebäuden zusammengestellt. Außerdem wird ein umfassender Überblick zur Regelung von Anlagen zur solaren Kühlung/ Klimatisierung erarbeitet. Erfahrungen der unterschiedlichen Demonstrationsanlagen werden mit

einander verglichen und ausgewertet. Die wichtigsten Aspekte der Regelung werden als Ergebnis dieser Arbeit zusammengefasst.

- *B3: Demonstrationsanlagen/ Monitoring/ Ergebnisse und Entwicklung eines Bewertungsprozess* - Hauptaugenmerk dieses Arbeitspaketes liegt auf der Beschreibung und der Bewertung von ausgewählten existierenden Demonstrationsanlagen mit großen Anlagen zur Klimatisierung oder Kälteerzeugung (Große Bürogebäude; Nicht-Wohnbau; Hotels; Industrie; etc.). Die Untersuchungen betrachten einerseits unterschiedliche Anwendungsfälle und Rahmenbedingungen zu Experimenten und Monitoring der ausgewählten Anlagen und anderseits die Ergebnisanalyse und die Bewertung der technischen Einrichtungen.
- *B4: Methoden zur Auslegung erfolgreicher Projekte* - Basierend auf den Ergebnissen erfolgreicher Projekte wird ein Softtool-Paket entwickelt, um eine rasche Bewertung in der Auslegungsphase für den Planer zu ermöglichen. Dies inkludiert technische und nicht-technische Aspekte. Dies unterstützt Marktakteure im Bewertungsprozess und in der Machbarkeitsanalyse bei ähnlichen Projekten zur solaren Klimatisierung und Kälteerzeugung.
- *B5: Leitfäden für Installation und Inbetriebnahme* - Erste Erfahrungen werden in Leitfäden für die Installation und Inbetriebnahme zusammengefasst. Diese beinhalten Auslegungskennwerte und technische Hinweise. Außerdem werden in diesem Arbeitspaket Leitfäden für die Ausschreibung und Checklisten für die technische Anlagenauslegung erstellt. Diese unterstützen Architekten und Planer in der ersten Projektentwicklungsphase.

Subtask C: Modellbildung und Grundlagenuntersuchungen

Die Arbeiten in dieser Subtask sind in zwei Teile gruppiert. Der erste Teil widmet sich der Weiterentwicklung und der eingehenden Prüfung von bereits verfügbaren Modellen von Systemkomponenten und Simulationstools. Bei den Untersuchung zu Simulationstools wird besonderer Wert auf Ihre Anwendbarkeit in den unterschiedlichen Schritten des Planungsprozesses gelegt. Im zweiten Teil konzentriert sich die Arbeit auf neue und viel versprechende Konzepte der solaren Klimatisierung und Kälteerzeugung, die bisher noch wenig bzw. gar nicht am Markt sind. Diese Subtask hat folgende Arbeitspakete:

- *C1: Technischer Review und Bericht von neuen Entwicklungen* - Die Arbeit beinhaltet einen technischen Review zu neuen Entwicklungen von Komponenten, Prozessen und geeigneten Systemen der solaren Klimatisierung und Kälteerzeugung. Weiters wird versucht Informationen über wichtige nicht veröffentlichte Forschungsaktivitäten zu sammeln und zu beschreiben.
- *C2: Simulationstools – Analyse und Entwicklung* - Ziel dieser Untersuchung hat zwei Ausrichtungen: Einerseits werden Neuentwicklungen im bereich der Simulationstools präsentiert und vorgestellt und andererseits werden bekannte Simulationstools analysiert. Die Analyse dazu prüft die Anwendbarkeit der Software hinsichtlich auf unterschiedlichen Ebenen der Auslegung und Konfiguration. Außerdem wird die Vergleichbarkeit der Modellbasierten Ergebnisse untersucht.
- *C3: Thermodynamische Untersuchungen (Exergie)* – Ziel ist es thermodynamische Analysen existierenden Anlagenkonfigurationen und neuer Konzepte der solaren Kühlung und Kälteerzeugung durchzuführen. Die Methode soll interessante Lösungen hinsichtlich Exergie und wirtschaftlicher Betrachtung aufzeigen und mittels Sensitivitätsanalyse werden signifikante Systemparameter identifiziert.
- *C4: Performance Kriterien* – Diese Aktivität zielt darauf ab, Exergie-Analysen durchzuführen, um Systemkonfigurationen zu identifizieren mit einem Minimum an Irreversibilität.
- *C5: Konzepte der Rückkühlung* - Dieses Arbeitspaket widmet sich den Untersuchungen zu bestehenden und neuen Konzepten zur Rückkühlung. Dazu wird ein Überblick erstellt. Ein Empfehlungskatalog wird ausgearbeitet, um aufzuzeigen welche Maßnahmen zur Rückkühlung unter Berücksichtigung verschiedenen Randbedingungen (Klima, Anlagenkonzept, etc.) sinnvoll umsetzbar sind.

Subtask D: Markttransfer

Die Arbeiten dieser Subtask erstrecken sich horizontal über alle drei zuvor genannten Subtasks. Das Ziele dieser Subtask sind die Aufdeckung der maßgeblichen Barrieren (hauptsächlich die nicht technischen) für eine weitere Verbreitung der solarer Klimatisierung/ Kühlung, die Promotion von Ergebnissen und die Verbreitung von Infomaterial als Diskussionsgrundlage, die Umsetzung von Wissenstransfermaßnahmen für die technische Interessengruppe, die Entwicklung von Instrumenten und Ihre Entsendung zu politisch

Verantwortlichen und die Bewerbung von Zertifizierung und Standardisierung. Die Subtask hat folgende Arbeitspakete:

- *D1: Vorschläge für eine Methode zur Performance-Bewertung* - Hier wird eine Methode zur Bewertung der Performance von Anlagen zur solaren Klimatisierung und Kälteerzeugung entwickelt. Diese erlaubt in gewissen Grenzen eine beinahe standardisierte Bewertung der gesamten Performance sowohl energetisch als auch ökonomisch.
- *D2: Zertifizierungs- und Standardisierungspläne* - Der Inhalt dieses Arbeitspaketes zielt einerseits auf die Erstellung von Zertifizierungs- und Standardisierungsplänen und anderseits auf ihre Unterstützung von kleine bis mittleren Kompaktanlagen sowie große Anlagen zur solaren Klimatisierung und Kälteerzeugung.
- *D3: LCA (Lebenszyklusanalyse)* - Hier wird eine Methode zur Bewertung der Lebenszyklen von Anlagen zur solaren Klimatisierung und Kälteerzeugung entwickelt.
- *D4: Handbuch 2. Ausgabe* - Dieses Arbeitspaket beinhaltet die Zusammenfassung der Task-Ergebnisse. In einer 2. Auflage des Handbuches zur solaren Klimatisierung und Kälteerzeugung für Planer wird veröffentlicht. Die Vorgehensweise im zweiten Handbuch ist mit dem ersten vergleichbar. Die Beiträge werden von unterschiedlichen Tast-Teilnehmern erbracht und zahlreiche technische Beispiele aus den teilnehmenden Ländern werden eingearbeitet. Hauptfokus der zweiten Ausgabe liegt auf den praktischen Erfahrungen, Detailbeschreibungen zu hydraulischen Verschaltungen, Anlagenregelung, Inbetriebnahme und Leitfäden, technisch ökonomisch Optimierung und Konzepte für sehr spezielle Anwendungen.
- *D5.1 Politik* - Zielgruppe der zuarbeitenden Materialien sind lokale und nationale politisch Verantwortliche. Hier werden Vorschläge für mögliche politische Maßnahmen zur Förderung der solaren Kühlung erarbeitet.
- *D5.2: Ausbildungsmaterialien* - Das Ergebnis dieses Arbeitspakets ist die Sammlung und Erstellung von Ausbildungsmaterial für Installateure und Planer. Referenzen zu den wichtigen nationalen oder internationalen Standards werden hergestellt, so weit dies möglich ist.
- *D5.3: Nationale Industrie Workshops* - Teilnehmer der Task werden nationale Workshops organisieren, die speziell für Vertreter der Zielgruppe aus Industrie

(Solarkollektorhersteller, Installateure, Industrie der thermischen Kältetechnik, Planer etc.) ausgerichtet sind.

- *D5.4: Industrie Newsletter* - Für weitere Verbreitung der erzielten Ergebnisse aus den F&E Aktivitäten der Task wird ein halbjährlicher Newsletter für die Zielgruppe aus der Industrie veröffentlicht.
- *D6: Marktanalyse* - Durch die Marktanalyse sollen die maßgeblichen Barrieren (hauptsächlich die nicht technischen) für eine weitere Verbreitung der solarer Klimatisierung/ Kühlung aufgedeckt werden. Insbesondere werden hier Nischenmärkte und Zukunftsstrategien beleuchtet.

ÜBERSICHT über das Implementing Agreement

Der Energiekonsum für Heizung, Kühlung, Beleuchtung und Warmwasser in Gebäuden beträgt beinahe 30 % des Gesamtenergieverbrauchs in den IEA-Mitgliedsstaaten. Zwar trägt die Nutzung der Solarenergie bereits signifikant zur Reduktion des Bedarfs an konventionellen Energieformen in Gebäuden bei, jedoch existiert immer noch ein großes Potential für zusätzliche Beiträge. Das Implementing Agreement zu Solar Heating and Cooling zielt auf diese Bereiche ab.

Beschreibung

Im Programm "Solares Heizen und Kühlen" der IEA werden seit den 70er Jahren zahlreiche Aktivitäten im Bereich der aktiven und passiven Solarenergienutzung mit dem Schwerpunkt Gebäude durchgeführt. Durch die Teilnahme von 21 Ländern und der Europäischen Kommission ist in diesem Forschungsprogramm ein breiter internationaler Erfahrungsaustausch möglich.

Ziele des Implementing Agreements für SHC

- Hilfe zur Erreichung einer signifikanten Steigerung der Leistung von solaren Heizungs- und Kühl-Technologien und -Designs.
- Hilfestellung für Industrien und Regierungen bei der Erhöhung des Marktanteils von solaren Heiz- und Kühltechnologien.
- Primäre Quelle für technische Informationen und Analysen der Technologien, Designs und Anwendungen im Bereich solares Heizen und Kühlen zu sein.
- Hilfe für Bildung und Aufklärung von Entscheidungsträgern und der Öffentlichkeit über den Status und Wert von solarem Heizen und Kühlen zu bieten.

IEA-SHC TASK 38 „Solar Air Conditioning and Refrigeration" des Implementing Agreements on Solar Heating and Cooling

Die Ergebnisse der abgeschlossenen IEA SHC Task 25 »Solar Assisted Air Conditioning of Buildings« zeigten einerseits das große Marktpotential dieser Technologie zur Gebäudeklimatisierung – insbesondere in sonnigen Regionen – und anderseits die Notwendigkeit zur Fortsetzung der Forschungsarbeit, um die Entwicklung von Systemen hoher Qualitätsstandards und zuverlässigen Anlagenbetrieb über viele Jahre zu forcieren. Die Arbeiten und Erkenntnisse der TASK 25 haben weltweit zahlreiche Entwicklungen im Bereich der solaren Klimatisierung initiiert. Trotz dieser guten Erfolge existieren viele Hemmnisse zur

Markteinführung von Systemen der solaren Klimatisierung, diesen wird mit der Entwicklung von standardisierten Abläufen im Konzeptionisierungs- und Planungsprozess und mit verstärktem Wissenstransfer entgegengetreten.

Der Technologiefokus dieser SHC Task 38 liegt auf thermisch angetriebenen Kühl- und Klimatisierungsverfahren, die entweder Kaltwasser produzieren oder direkt Luft konditionieren wobei solare Wärme als Antriebsenergie zum Einsatz gelangt. D.h. die gesamte Technologiekette vom solarthermischen Kollektor angefangen über die thermische Kälteerzeugung oder Klimatisierung von Luft bis hin zu den technischen Kälte und Luftverteilsysteme im Gebäude wird im Rahmen der Aktivitäten der SHC TASK 38 betrachtet.

Obwohl die technischen Verteilsysteme der Kälte und Klimatechnik, das Gebäude und die thermo-dynamisch Interaction beider keinen zentralen Schwerpunkt der TASK bildet, werden diese jedoch in essentiellen Fällen in die Betrachtungen mit einbezogen. Insbesondere gilt dies für die Kompaktsysteme für Wohngebäude und andere Anwendungen im kleinen Leistungsbereich, welche überwiegend auf einen hohen Anteil des solarthermischen Antriebs setzen, ist die Betrachtung des gesamten Systems inklusive Gebäude zur Optimierung der Anlagenperformance essentiell. Die SHC TASK 38 beschäftigte sich neben der Gebäudekühlung und –Klimatisierung auch um die solarthermische Erzeugung von Gefrierkälte in anderen kommerziellen und industriellen Anwendungen.

ERGEBNISSE DES PROJEKTES

Internationale Inhalte und Ergebnisse

Der nachfolgende Text basiert im Wesentlichen auf dem Newsletter of the International Energy Agency Solar Heating and Cooling Programme solarupdate Vol.53 | January 2011.

Die SHC Task 38, Solar Air-Conditioning & Refrigeration war hinsichtlich Teilnehmerzahl eine der größten Einzelaktivitäten im Rahmen des SHC Programms. 49 Organisationen aus 12 Ländern haben an dieser Task teilgenommen. 16 Forschungsinstitute, 20 Universitäten und 13 private Unternehmen (Planungsbüros, Hersteller und Installationsbetriebe) haben sich regelmäßig auf den neun Expert Meetings zur gemeinsamen Projektarbeit und Ergebnisdiskussion getroffen und ausgetauscht. Die Ergebnisse sind in Berichten in hoher Qualität dokumentiert und wurden auf zahlreichen Konferenzen verschiedensten Zielgruppen präsentiert. Tabelle 1 beinhaltet eine Übersicht zu den Berichten und Ergebnissen der SHC Task 38. Die öffentlich verfügbaren Berichte sind unter http://iea-shc-task38.org/reports im pdf-Format herunter zu laden. Die SHC Task 38 hat einen essentiellen Beitrag zur Technologieentwicklung und der Markteinführung der solaren Kühlung geleistet.

Ein Forschungsschwerpunkt der SHC Task 38 war das detaillierte Monitoring von 24 Anlagen zur Bewertung des energetischen Anlagenbetriebs unter realen Bedingungen. Davon wurden 13 Anlagen im kleinen Leistungsbereich vermessen und untersucht, wobei die SHC Task 38 Ergebnisse die Entwicklung der ‚pre-engineered systems‘ für den Wohnbau und Gebäude kommerzieller Nutzung im kleinen Leistungsbereich unterstützt hat. Im größeren Leistungsbereich wurden 11 Anlagen zur Kühlung bzw. Klimatisierung von Nichtwohn-Gebäude untersucht. Diese Anlagen sind speziell für eine gewünschte Nutzung und den Kunden ausgelegt und konzipiert. Auf Basis dieser Monitoring¬-Arbeit lässt sich belegen, dass Anlagen zur solaren Kühlung/ Klimatisierung signifikante Primärenergieeinsparungen gegenüber konventionellen Standardlösungen mit elektrisch angetriebener Kompressionskälteanlage erreichen können. Um diese Einsparungsziele in der Umsetzung zu erreichen, ist auf jeden Fall eine hohe Qualität in allen Phasen der Projektierung einer solar thermisch angetriebenen Variante zur Gebäudeklimatisierung zu gewährleisten. Eine sorgfältige Planung, eine hohe Qualität hinsichtlich Installation, eine professionelle Inbetriebnahme und ebenso ein permanentes Anlagenmonitoring über die gesamte Betriebszeit sind erforderliche Maßnahmen, um einen langjährigen, stabilen und zuver¬lässigen Anlagenbetrieb mit geringerem (Primär-)Energieaufwand und geringeren Betriebskosten gegenüber Standardlösungen zu gewährleisten.

Insbesondere hat die SHC Task 38 zahlreiche Werkzeuge/ Leitfäden und Konzepte entwickelt und dokumentiert, damit solare Kühl- bzw. Klimatisierungsprojekte erfolgreich umgesetzt werden können. Außerdem wurden übergreifende Forschungsarbeiten, wie einerseits Exergie-Analysen von solaren Kühl- bzw. Klimatisierungsverfahren und anderseits Vergleichsuntersuchungen von Simulationswerkzeugen hinsichtlich Zuverlässigkeit und Nutzerfreundlichkeit, durchgeführt. Dies sind wichtige Beiträge um die Systeme energetisch zu optimieren. Erstmals wurde eine extensive Lebenszyklusanalyse zu Anlagen der solaren Kühlung bzw. Klimatisierung durchgerechnet. Die drei nachfolgenden Beispiele sind eine Auswahl der SHC Task 38 Ergebnisse mit praktischem Nutzen in kommerziellen Umsetzungsprojekten und weiteren Forschungsarbeiten.

ONLINE TOOL

Die “Check-list method for the selection and the success in the integration of a solar cooling system in buildings” ist auf der Website der Firma TECSOL SA zu finden – siehe http://www.tecsol.fr/checklist/. Diese Checkliste resultiert aus Erfahrungen und inhaltlichen Beiträgen Europäischer Experten der solaren Kühlung aus der SHC Task 38. Ziel war es eine Methode und ein Tool zu entwickeln, welches in der frühen Projektphase Hinweise zum technischen Potenzial gibt. Dieses Werkzeug berücksichtigt alle wesentlichen Aspekte wie technischen Rahmenbedingungen, Finanzierungsfragen, erforderliche Gebäudeflächen und Interessen der involvierten Stakeholder. Letztlich generiert dieses Werkzeug wichtige Hinweise über die Machbarkeit eines bestimmten Projekts der solaren Kühlung.

MONITORING PROZEDUR

Eine einheitliche Vorgehensweise zum Monitoring sowohl für solar thermisch angetriebene Kältemaschinen als auch für die ‘Desiccant Evaporative Cooling Systems‘ wurde definiert, entwickelt und dokumentiert. Hiermit wird eine strukturierte Datenakquise eines Monitorings ermöglicht und es steht eine allgemein gültige Methode zur Bewertung des energetischen Anlagenverhaltens zur Verfügung. Für 13 vermessene Anlagen aus dem kleinen Leistungsbereich und für 11 Großanlagen wurde diese Vorgehensweise angewendet und überprüft. Diese Methode beinhaltet die Evaluierung des Anlagenverhaltens, einen Vergleich mit konventionellen Lösungen und mit anderen Systemkonfigurationen der solaren Kühlung. Eine gut geeignete technische Lösung lässt sich mittels dieser einheitlichen Prozedur identifizieren und definiert gleichzeitig minimal notwendige messtechnische Feldgeräte zur Durchführung eines Monitoring und bietet eine erprobten Evaluierung des Anlagenverhaltens.

POLICY POSITION PAPER

Im Rahmen der SHC Task 38 wurde ein Positionspapier zur Solaren Kühlung verfasst. Das Papier besteht hauptsächlich aus drei Teilen 1) Aktueller Status (technischer Reifegrad, energetische Performance, Wirtschaftlichkeit und Marktdurchdringung), 2) technisches Potenzial, Kosten und wirtschaftliche Aspekt und Marktchancen und 3) erforderliche Maßnahmen (technischen Weiterentwicklung, Qualitätsverfahren und Belange des Marktes und der Politik). Das Positionspaper ist ein Konvolut aus den Anregungen und Beiträgen wichtiger Interessensgruppen – wie Experten aus F&E, Firmenchefs der Herstellerfirmen.

ZWEITE AUSGABE DES HANDBUCHES

Als Erscheinungstermin wird der Herbst 2011 erwartet. Die neue Ausgabe des Handbuches unterteilt sich in zwei Hauptteile und die Inhaltsangabe gibt die nachfolgende Aufstellung (Stand Dezember 2009) wieder:

Part 1

Chapter 1 - Introduction
Chapter 2 - Meteorological data and heating and cooling loads calculation
Chapter 3 - Components of solar thermal systems
Chapter 4 - Heat driven cooling technologies: closed cycles
Chapter 5 - Heat driven cooling technologies: open cycle systems
Chapter 6 - Solar cooling systems characterization
Chapter 7 - Energy and economic performance figures

Part 2

Chapter 8 - Overall system design, sizing, tools
Chapter 9 - Solar thermal system design issues
Chapter 10 - Experiences & lessons learned including control: Pre-engineered
Chapter 11 - Experiences & lessons learned including control: Custom made
Chapter 12 - Experiences & lessons learned including control: DEC systems
Chapter 13 - Trends and new developments
Chapter 14 - Summary and outlook

Annexes

Annex A - Review of market available heat driven cooling technologies

Annex B - Commissioning guidelines and checklists

Tabelle 1 - Überblick zu den Berichten und Ergebnissen der TASK 38 (Status 11.01.2011)

Report-No.	Topic of report	Publication level (PUblic, REstricted)*	Comment / status
A1	Market overview (small scale)	PU	**Published on SHC website**
A2	Generic Systems	PU	**Published on SHC website**
A3	Monitoring (small scale)	RE	**Final report outstanding**
A5	Installation and maintenance guidelines	PU	**Final report outstanding**
B1	Market overview (large scale)	PU	**Published on SHC website**
B2	System design and control	RE	**Approved by ExCo; available on Task 38 internal website**
B3-A	Monitoring (large scale)	RE	**Final draft sent to related Task participants for review**
B3-B	Technical report on monitoring procedure	PU	**Draft version sent to related Task participants for review**
B4-A	Fast pre-design tool	PU	**Available on TECSOL website**
B4-B	Fast pre-design technical report	RE	**Ready for ExCo approval**
B5	Commissioning guideline	PU	**Ready for ExCo approval**
C1	Survey on new solar cooling developments	PU	**Ready for ExCo approval**
C2-A	Simulation tools	PU	**Published on SHC website**
C2-B	Absorption chiller simulation	PU	**Published on SHC website**
C2-C	Desiccant simulation	PU	**Final report outstanding**
C3	Thermodynamic / Exergy Analysis	PU	**Ready for ExCo approval**
C5	Heat rejection	PU	**Final draft sent to related Task participants for review**
D3	Life cycle analysis	PU	**Ready for ExCo approval**
-	Solar Cooling Position Paper	PU	**Draft presented to ExCo in June 2010 and at expert meeting in September 2010; final version until end 2010**
-	**New Handbook**	**PU**	**Publication expected for mid 2011**

* Public: published via IEA SHC website or by publisher; restricted: available only to Task 38 participants via internal Task 38 website

BERICHTE – STATUS ÖFFENTLICH (STAND 26. JULI 2010)

Quellen

- http://www.iea-shc.org/publications/task.aspx?Task=38
- http://iea-shc-task38.org/reports

Report A1	Market Available Components for Systems for Solar Heating and Cooling with a Cooling Capacity < 20 kW
Report A2	Collection of selected systems schemes “Generic Systems”
Report B1	State of the art on existing solar heating and cooling systems
Report C2-A	Description of simulation tools used in solar cooling - New developments in simulation tools and models and their validation - Solid desiccant cooling - Absorption chiller
Report C2-B	Benchmarks for comparison of system simulation tools – Absorption chiller simulation comparison

Inhalte und Ergebnisse der österreichischen Partner

Das österreichische Projektteam der IEA SHC TASK38 setzt sich aus den folgenden Partnern zusammen

- Austrian Institute of Technology, Wien - AIT
- Austrian Solar Innovation Centre, Wels - ASIC
- Institut für Wärmetechnik, TU Graz, Graz - IWT

Die Beschreibung der Inhalte und Ergebnisse erfolgt pro Forschungspartner. Eine allgemeine Dokumentation der Aktivitäten bildet den Hauptteil und ein Anhang dokumentiert die wichtigsten Publikationen und Veröffentlichungen.

Austrian Institute of Technology

Im Folgenden werden die wesentlichen Ergebnisse und Inhalte des Forschungspartners AIT vorgestellt.

- Das AIT führte wissenschaftliches Monitoring von zwei österreichischen Anlagen zur solaren Gebäudekühlung bzw. Klimatisierung durch und stellte die Ergebnisse der SHC TASK 38 zur Verfügung. Der nachfolgende Text enthält einerseits Ergebnisse und Forschungstätigkeiten zum Monitoring der Anlage zur solaren Gebäudeklimatisierung im Bürogebäude ENERGYbase am Standort Wien Floridsdorf und andererseits Angaben zum Monitoring der solaren Adsorptionskälteanlage der Wiener Magistratsabteilung 34.
- Erfahrungen und Erkenntnisse hinsichtlich Installation und Inbetriebnahme wurden in einem Leitfaden gebündelt.
- Das AIT war mit der Koordination des Arbeitspaketes B2 Design und Anlagenregelung betraut und hat eine Sammlung zu 14 Case Studies zusammengestellt.
- Hinsichtlich Erstellung der zweiten Ausgabe eines englischsprachigen Handbuches verfasst das AIT Textbeiträge zu folgenden Kapiteln:
  - CHAPTER 11 - Experiences from installed custom made systems
  - CHAPTER 12 - DEC Systems: Built Examples and Experiences

MONITORING DER SOLAR-GESTÜTZTEN KLIMATISEIRUNG DES BÜROGEBÄUDES ENERGYBASE, WIEN

Anlagenbeschreibung

Die Kernidee zur haustechnischen Versorgung des nachhaltigen Bürogebäudes ENERGYbase ist eine Aufteilung in ein wasserbasiertes und ein luftgeführtes Verteilsystem. Die Grundtemperierung der Innenräume wird über das wassergeführte Heiz- und Kühlsystem realisiert. Das luftgeführte System übernimmt die technische Aufgabe der Zuluftkonditionierung. D.h. die technischen Anforderungen an die Klimatisierung der Zuluft sind moderater, da die Grundtemperierung mittels der Betonkernaktivierung die Büros sensible gekühlt bzw. erwärmt werden. Das luftgeführte System bewerkstelligt somit in diesem Konzept die Abfuhr der latenten thermischen Lasten, d.h. die überwiegende Kontrolle der Raumluftfeuchtigkeit.

Dieser grundsätzliche haustechnische Ansatz ermöglicht den Einsatz der solaren Klimatisierung. Diese energieeffiziente Verfahren zur Luftbehandlung nutzt im Wesentlichen drei Prozessschritte der Luftbehandlung; 1.) Lufttrocknung - mit einem so genannten Sorptionsrotor, 2.)Wärmerückgewinnung und 3.) adiabate Verdunstungskühlung.

Zwei baugleiche Lüftungsgeräte der Firma robatherm mit einem Nennvolumenstrom von 8 240 m³ pro Stunde arbeiten mit dem Verfahren der so genannte sorptionsgestützten Klimatisierung. Zur Regeneration der eingebauten Lufttrocknungsrotoren wird solare Wärme verwendet. Das solare Kollektorfeld weist eine Fläche von ca. 285 m² auf und besteht aus thermischen Flachkollektoren des Typs SONNENKRAFT SK500N-ECO. Die Solaranlage wird im „Low flow“ Modus betrieben. Als thermischer Speicher wird ein wassergefüllter Schichtenspeicher mit 15 m³ Volumen verwendet. Die Beladung erfolgt über eine Lanze und das solar beheizte Wasser wird im oberen Bereich des Tanks entnommen. Die Sollwerte der Zuluft sind im Sommer T_{Zuluft} = 23°C und RF_{Zuluft} = 50%.

Forschungstätigkeiten

Ziel der Tätigkeiten im Rahmen des SHC Task 38 war es die ‚Desiccant Evaporative Cooling’-Anlagen (DEC) in der ENERGYbase mit dem im SHC Task 38 entwickelten Monitoring-Procedere zu bewerten. Dies gibt die Möglichkeit einerseits die Performance der Anlage zu einer Referenzanlage zu bewerten und andererseits unterschiedliche solar-gestützter DEC-Anlagen miteinander zu vergleichen. Folgende Tätigkeiten wurden im Rahmen des SHC Task 38 dafür durchgeführt:

- Gewährleistung des Datenzugriffs und der Datenübertragung
- Überprüfung der Vollständigkeit der erforderlichen Messsensoren
- Überprüfung der Durchgängigkeit der aufgezeichneten Messserien

- Adaptierung der Basisvariante laut Task 38 Monitoring-Procedere zur Ermittlung der für die DEC-Anlagen relevanten und gemessenen Energieströme (thermisch und elektrisch) - siehe Anhang AIT1
- Berechnung der Enthalpiedifferenz pro Monat für jede der beiden DEC-Anlagen anhand eines im SHC Task 38 entwickelten Excel-Tool - siehe Beispiel für Juli 2010 für die DEC-Anlage West in Anhang AIT2.
- Aufbereitung der Monitoringdaten zur Bewertung der DEC-Anlagen auf Monatsbasis in drei Levels:
    - Level 1: Basisinformationen zu Primärenergieeinsatz und Kosten
    - Level 2: Einfache Analyse des Solarenergieeinsatzes
    - Level 3: Detailliertes Monitoring-Procedere

Für die Level 3 Analyse wurden die Monatssummen aus der Monitoringaufzeichnung verwendet und die wesentlichen Erkenntnisse daraus in einen Monitoringbericht zusammengefasst - siehe Anhang AIT3.

Zusammenfassung der wesentlichen Erkenntnisse:

- Bei der Positionierung der Außenluftansaugung und Fortluftausblasung von DEC-Anlagen ist besondere Sorgfalt geboten. Bei ungünstiger Positionierung der Öffnungen kann es unter Umständen zu einer Beimischung von Fortluft zur Ansaugluft kommen. Dies bewirkt besonders im solaren Kühlbetrieb mit hohen Fortlufttemperaturen eine signifikante Erwärmung der Ansaugluft.
- In der Übergangszeit kann die vorhandene Solarenergie im Solarspeicher kaum genutzt werden, da aufgrund der Passivhausbauweise kein Heizbedarf mehr besteht und die Zuluftkühlung über adiabatische Kühlung gewährleistet werden kann. Das Kollektorfeld geht bei Erreichung der maximalen Solarspeichertemperatur von 105 °C in Stagnation; der theoretisch mögliche Kollektorertrag kann daher bei dieser Anlage nicht genutzt werden.
- Eine Geruchsübertragung von der Abluft in die Zuluft trat in den DEC-Anlagen der ENERGYbase im Gegensatz zu anderen DEC-Anlagen nicht auf, was auf den verwendeten Lithium-Chlorid Rotor zurück zu führen ist.
- Die hier eingesetzte Wasseraufbereitungsanlage bestehend aus Enthärtungsanlage, Umkehrosmose und Vorratstank hat sich im Vergleich zu anderen Wasseraufbereitungsanlagen als technisch und energetisch günstige Ausführung erwiesen.
- Die Feuchterückgewinnung über den Sorptionsrotor im Winter trägt wesentlich zur Energieeffizienz der DEC-Anlagen bei, wodurch ein mittlerer elektrischer Coefficent of

Performance (COP_{el}) von 8,12 erreicht wurde (siehe Anhang AIT3). Im Jänner 2010 wird ein COP_{el} von 18,4 erreicht. Der Einsatz eines Sorptionsrotors zur Feuchterückgewinnung ist daher in unserem Klima aus energetischer Sicht generell für Lüftungsgeräte zu empfehlen.

MONITORING DER SOLAREN ADSORPTIONSKÄLTEANLAGE MAGISTRATSABTEILUNG 34, WIEN

Anlagenbeschreibung

Das Kernstück der Anlage zur solaren Kaltwassererzeugung der Wiener Magistratsabteilung 34 ist eine Adsorptionskältemaschine, Nennkälteleistung 7,5 kW Typ SOL ACS 08 der SorTech AG in Halle an der Saale/ Deutschland. Die Rückkühlung erfolgt durch einen trockenen Rückkühler RCS 08, der über eine drehzahlgeregelte Lüftertechnik verfügt und mit einem zusätzlichen Frischwassersprühsystem ausgestattet ist.

Die Antriebswärme für die Kältemaschine wird über 12 Universalflachkollektoren mit einer Gesamtbruttofläche von 32,40 m² der Firma SOLution Solartechnik GmbH generiert. Die Flachkollektoren sind in etwa 40° Neigung zur Horizontalen mit Südorientierung auf einem bestehenden Hallendach montiert. Im Anlagenkonzept ist keine Nacherwärmung durch andere Wärmeerzeuger vorgesehen. Die solare Wärme wird in einem 2000 l Solarpuffer zwischengespeichert – wobei die Beladung aus dem Sekundärsolarkreislauf über eine Schichtladeeinheit erfolgt. Im solaren Kühlbetrieb wird das über die Adsorptionskältemaschine erzeugte Kaltwasser in einem Kaltwasserpuffer mit dem Fassungsvermögen von 800 l zwischengespeichert. Je nach Kühlanforderung der verschiedenen Räume werden die einzelnen Fan-Coils mit Kaltwasser aus dem Kaltwasserspeicher beschickt.

Forschungstätigkeiten

Im Rahmen des IEA SHC Task 38 wurde für die solare Adsorptionskälteanlage der Wiener Magistratsabteilung 34 (MA34) folgende Untersuchungen durchgeführt:

- COP_{el} und COP_{th} der solaren Adsorptionskälteanlage
- Einsatz des Hybridkühlers im Trocken-/ bzw. Nassmodus
- Auswirkungen einer Drehzahlregelung der Ventilatoren im Rückkühlwerk auf die Energieeffizienz

Zusammenfassung der wesentlichen Erkenntnisse:

- Der Rückkühler verursacht in dieser Anlage ¾ des Strombedarfs, daher ist zur Erreichung eines hohen COP_{el} der Einsatz eines Rückkühlwerks mit hoher Effizienzklasse (vorzugsweise Nasskühlturm oder zumindest Hybridkühler) essentiell.
- Einsatz von drehzahlgeregelten, energieeffizienten Pumpen ist ebenfalls wesentlich um einen hohen COP_{el} zu erreichen.
- Die Monitoringergebnisse des 25. August 2010 haben gezeigt, dass ein COP_{el} von 4,55 derzeit schon mit dieser Anlage möglich ist. Wichtig dafür sind hohe Desorptionstemperaturen (> 70 °C) und ein Nassbetrieb des Rückkühlers. Dieser Wert liegt somit deutlich über den maximalen Tageswert des COP_{el} von August 2009 mit 2,3.
- Aufgrund der Wassertemperaturen in allen drei hydraulischen Kreisen (Kalt-, Kühl- und Heißwasserkreis) der Adsorptionskältemaschine wurden tägliche thermische Arbeitszahlen für Juli 2010 im Bereich von 0,16 bis 0,58 gemessen. Der nominelle $COPt_{th}$ liegt bei 0,56.
- Der Anlagenoptimierungsversuch durch Nutzung einer Drehzahlregelung der Ventilatoren im Rückkühler hat durch die nicht statt gefundene Kopplung von Regelung des Kältemaschinenherstellers mit der Gesamtanlagenregelung im Sommer 2010 nicht zur Energieeffizienz beigetragen, im Gegenteil, der COP_{el} wurde dadurch verschlechtert. Generell wäre eine Regelung der Gesamtanlage welche die Kältemaschinenregelung inkludiert zu empfehlen, wodurch derartige Fehler vermieden werden können.
- Durch eine regelungstechnische Kopplung der Energiebereitstellung über die solarthermische Kühlanlage, mit der Kälteverteilung in den Räumen, könnte ein effizienterer Anlagenbetrieb erreicht werden.

INSTALLATION, OPERATION AND MAINTENANCE GUIDELINES FOR PRE-ENGINEERED SYSTEMS

Im Rahmen dieser Working Group wurden zwei Aktivitäten durchgeführt:

- Endnutzer Erhebung zu solaren Kühlanlagen kleiner Leistungen (< 20 kW)
- Erhebung von Paketlösungen für solare Kühlanlagen kleiner Leistungen

Wesentliche Erkenntnisse durch Endnutzer Befragung:

- Typische Planungsindikatoren (m² Kollektorfläche/kW Kälte, Speichervolumen/m² Kollektorfläche) sind aus den evaluierten 18 Anlagen (alle in Europa) nicht ersichtlich, welche einerseits durch unterschiedliche Anwendungen bzw. klimatischen Bedingungen, als auch auf individuelle solaren Deckungsgrade zurück zu führen sind

- Die Installationszeiten vieler dieser Anlagen als auch die Anzahl der involvierten Personen zeigt, dass hier nicht von *pre-engineered systems* gesprochen werden kann
- Die Kommunikation zwischen den involvierten Unternehmen (Kältemaschinenhersteller, Solaranlagenhersteller, Installationsunternehmen, Planer, Endnutzer) war in manchen Fällen sehr langwierig, was auch für *pre-engineered systems* sprechen würde
- Die Inbetriebnahme ist in manchen Fällen nicht passiert
- Die erforderliche Betreuung der Anlage durch die Endnutzer während des Betriebs reichte von nahezu gar keinen Betreuungsaufwand bis zu täglichen Adaptierungen
- Die erhobenen Investitionskosten der Anlagen sind immer noch deutlich höher als derzeit übliche Marktpreise für solare Kühlanlagen größerer Kälteleistungen (>20 kW)

Wesentliche Erkenntnisse Paketlösungen (5 Anbieter):
Die befragten Anbieter von Paketlösungen haben keine einheitliche Strategie zur Auslegung der Anlagen. Das reicht derzeit von eigens entwickelten Tools, mit denen vordefinierte Paketgrößen bestimmt werden können, bis zu individueller Planung jeder Anlage.

Die Strategien der Anbieter von Paketlösungen hinsichtlich der inkludierten Komponenten eines Pakets sind auch sehr unterschiedlich. Die Kältemachine, die Solaranlage und die hydraulische Verschaltung ist bei jedem Anbieter enthalten, jedoch gilt dies nicht für den Gesamtanlagenregler. Die Energieverteilung im Raum wird bisher bei keinem dieser Pakete mit eingebunden.

Die Inbetriebnahme der Anlage ist bei jedem Anbieter von Paketlösungen enthalten, die Wartung kann ebenfalls von denselben Unternehmen durchgeführt werden oder es werden Unternehmen empfohlen. Nur ein einziger Anbieter von Paketlösungen bietet Anlagenoptimierung als Teil eines Paketes an.

Ein Monitoring der Anlage ist bei den meisten Anbietern von Paketlösungen als Teil des Paketes möglich, aber nicht bei allen. Nur ein Anbieter von Paketlösungen hat standardmäßig ein Monitoring im Gesamtanlagenregler enthalten.

Diese Erhebungen haben gezeigt, dass die derzeit verfügbaren Paketlösungen für solare Kühlanlagen kaum miteinander vergleichbar sind und für den Laien auf den ersten Blick nicht erkennbar ist, wo die großen Vor- und Nachteile der einzelnen Paketlösungen liegen. Folgende Teile sollten aus unserer Sicht in einer hochqualitativen Paketlösung neben den Hauptkomponenten wie Kältemaschine, Solaranlage und dazugehöriger hydraulische Verschaltung enthalten sein:

- Gesamtanlagenregler (vorzugsweise mit Einbezug der Kälteabgabe in den Raum)
- Inbetriebnahme mit Handbuch für Endnutzer
- Wartungsvertrag
- Monitoring der Anlage zur Anlagenoptimierung

DESIGN UND ANLAGENREGELUNG

Das AIT war mit der Koordination des Arbeitspaketes‚ Design und Anlagenregelung' betraut und seit Oktober 2007 hat das AIT auf jedem Expertentreffen eine Arbeitsgruppe zu diesem Thema geleitet.

In Zusammenarbeit mit ausgewählten SHC TASK 38 Teilnehmern wurde eine Basisdokumentation mit positiven und negativen Erfahrungen hinsichtlich Konzept, Planung und Betrieb von kundenspezifischen Anlagen zur solaren Kühlung/ Klimatisierung erstellt. Ausgewählte Erkenntnisse dieser grundlegenden Datensammlung werden in die zweite Ausgabe des geplanten Handbuches einfließen.

Das AIT hat ein weiteres Arbeitsdokument entwickelt, um methodisch und strukturiert eine wesentliche Expertise hinsichtlich Regelung und Steuerung von kundenspezifischen Anlagen zur solaren Kühlung/ Klimatisierung zu erheben. Das AIT hat in Zusammenarbeit mit den SHC TASK 38 Teilnehmern 14 CASE STUDIES dokumentiert und signifikanten Hinweise für Regelungsstrategien für solare Kühl- bzw. Klimatisierungsanlagen abgeleitet und ein umfangreicher Bericht wurde verfasst. Ausgewählte Erkenntnisse und Erfahrungen dieser grundlegenden Datensammlung werden in die zweite Ausgabe des Handbuches eingearbeitet. Aus Gründen vertraulicher Projektinformationen ist dieser Bericht auf Wunsch einiger Partner als „vertraulich eingestuft" und ist nicht öffentlich zugänglich. Anhang AIT4 ist eine Veröffentlichung zum Thema , SURVEY OF CONTROL AND CONFIGURATION OF SOLAR HEAT DRIVEN CHILLER SYSTEMS", die auf der EUROSUN 2010 in Graz im Oktober 2010 vorgestellt wurde. Die Dokumentation enthält Ergebnisse einer Analyse von bestehenden Regelungsstrategien für thermisch angetriebene Kaltwassererzeugern.

HANDBOOK 2ND EDITION

Für die zweite Ausgabe des englischsprachigen Handbuches verfasst das AIT Textbeiträge zu folgenden Kapiteln:

- CHAPTER 11 - Experiences from installed custom made systems
- CHAPTER 12 - DEC Systems: Built Examples and Experiences

Die Herausgabe des Handbuches ist für den Herbst 2011 geplant.

ASIC- Austria Solar Innovation Center

Das ASIC als jüngster Partner im Projektteam, ist häufig mit vielfältigen, eher praktischen Anfragen im Bereich der solaren Kühlung konfrontiert. Das ASIC war am Vorgängertask 25 nicht beteiligt. Die Anfragen der Solartechnikfirmen, und Planungsbüros mit denen das ASIC in Kontakt steht, erlauben es aber speziell praktische Erfahrungen und Erkenntnisse einzubringen.

Ein starker Heimmarkt ist Voraussetzung für einen zukünftigen Exportmarkt: Die grundlegende Frage, ob solare Kühlung in Österreich sinnvoll oder notwendig ist, kann durch Abbildung 3 bzw. Abbildung 4 beantwortet werden.

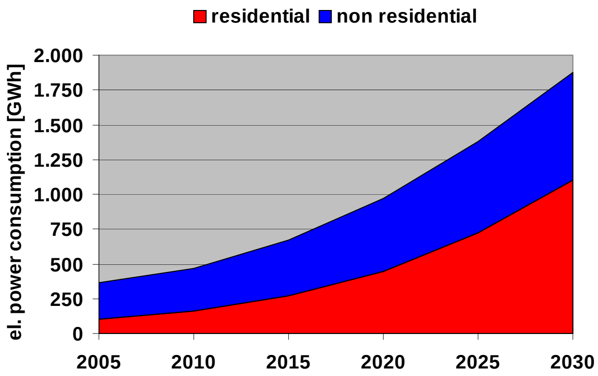

Abbildung 3 – Zunahme des elektrischen Verbrauchs für den Wohn- und Nichtwohnbereich

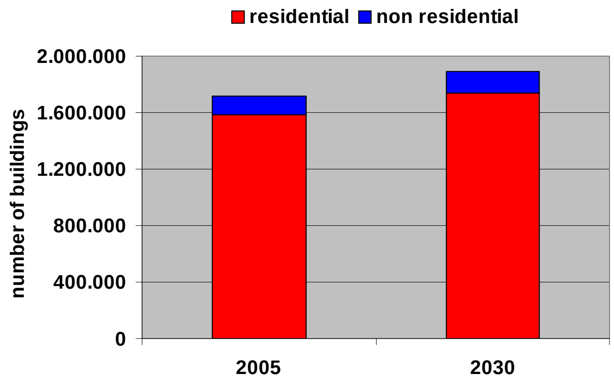

Abbildung 4 - Anzahl an Gebäuden mit Nachrüstung durch Klimageräte

Abbildung 3 zeigt die massive Zunahme des Verbrauchs an elektrischer Endenergie, besonders im Wohnbereich (residential) bis 2030[1]. Abbildung 4 zeigt die Anzahl an Gebäuden, in denen mit Klimageräten nachgerüstet werden, um den steigenden Komfortansprüchen gerecht zu werden - vergleiche dazu Anhang ASIC 1.

Die Notwendigkeit im Bereich Kälte und Klimatisierung Energie zu sparen, wurde durch stark steigende Energiepreise primär aber von Industriebetrieben erkannt. In einem metallverarbeitenden Betrieb in Oberösterreich wurden aus diesem Grund über ein Kalenderjahr der Kältebedarf und die meteorologischen Daten in 10 Minuten Schritten aufgezeichnet. Basierend auf diesen Messwerten wurden dann sechs Anlagenkonfigurationen zur Deckung der Kühllast durchsimuliert und energetisch bilanziert:

- Case 1: Gegenwärtige Situation, elektrische Kompressionskältemaschine

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[1] Haas et al, Wärme und Kälte aus Erneuerbaren 2030, Dachverband Energie-Klima, Wien 10/2007

- Case 2: Elektrische Kompressionskältemaschine +solare Kühlung
- Case 3: Elektrische Kompressionskältemaschine + solare Kühlung
- Case 4: Elektrische Kompressionskältemaschine + freie Kühlung
- Case 5: Elektrische Kompressionskältemaschine + freie Kühlung + solare Kühlung
- Case 6: Elektrische Kompressionskältemaschine + freie Kühlung + solare Kühlung

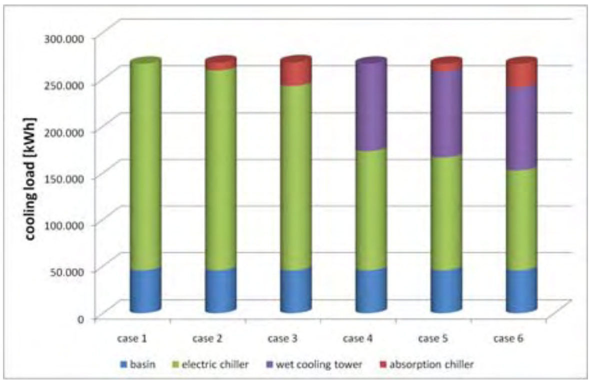


Abbildung 5 – Simulationsergebnisse der Falluntersuchung

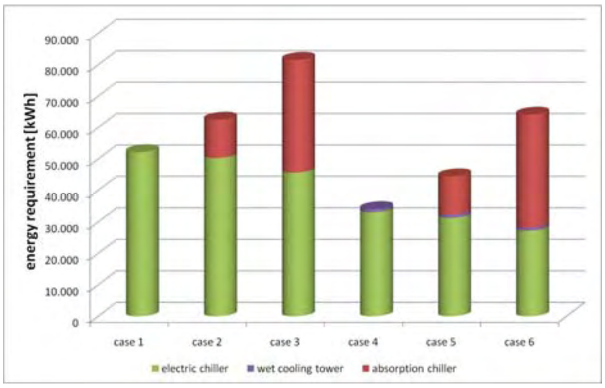


Abbildung 6 - Simulationsergebnisse der Falluntersuchung

Die Ergebnisse zeigen ein eher bescheidenes Potential der solarthermisch betriebenen Kühlung. Das theoretische Potential der freien Kühlung erscheint jedoch beträchtlich. Ein Blick auf die zu erwarteten Kosten unterstreicht das Gebot die solarthermische Anlage auch zur Warmwasserbereitung und Heizungsunterstützung einzusetzen.

Zusammenfassend bestätigt diese Untersuchung die Notwendigkeit Temperaturniveaus, Lastprofile und Einsparpotentiale vor der Planung einer solaren Kühlanlage genauestens zu erheben. Betreffend den Zielen von Subtask B4 muss ein Schwerpunkt auf diese Punkte gelegt werden! Vergleiche dazu Anhang ASIC 2.

Zu Beginn des Jahres 2006 begannen die Planungen am Neubau der Bezirkshauptmannschaft Rohrbach in Oberösterreich. Mit dem am 7.2.1994 von der O.Ö. Landesregierung einstimmig beschlossenen Energiekonzept wurden vorerst konkrete Ziele bis zum Jahr 2000 formuliert, die sowohl die Verbrauchs- als auch die Angebotsseite umfassen. In der zweiten Phase des Konzeptes werden diese bis zum Jahr 2010 erweitert bzw. ergänzt.

Die bei diesem Projekt als Demonstrationsanlage mit Vorbildwirkung geplante 120 m² große thermische Solaranlage wird im Sommer zur Erzeugung der Antriebsenergie der 30 kW Absorptionskältemaschine und im Winter zur Heizungsunterstützung eingesetzt (Maximierung solarer Gewinne!) Der Warmwasserbedarf ist hingegen gering und wird ausschließlich von Untertischspeichern abgedeckt. Die Absorptionskältemaschine unterstützt die wesentlich größere Kompressionskältemaschine und wird in den Rücklauf des Kaltwasserkreises eingebunden. Somit können geringe Kühllasten von der Absorptionsmaschine alleine gedeckt werden. Bei hohem Kühlbedarf tritt zumindest eine Entlastung der Kompressionsmaschine und eine Einsparung von elektrischer Energie ein.

Die Heiz- und Kühlenergieverteilung im Gebäude wird über zentrale Zuluftwärmetauscher und Fan-Coils bewerkstelligt. Zur generellen Kühllastvermeidung kommen ein außen liegender Sonnenschutz und eine Sonnenschutzverglasung zum Einsatz. Die Überwachung der Anlage und die messtechnische Erfassung und Speicherung der Betriebszustände erfolgt durch ein installiertes Monitoring-System. Die Daten können internetbasiert standortunabhängig abgefragt werden. Das ASIC berichtete unter anderem über die genaue Kostenstruktur des Projektes im Rahmen von Subtask B2.

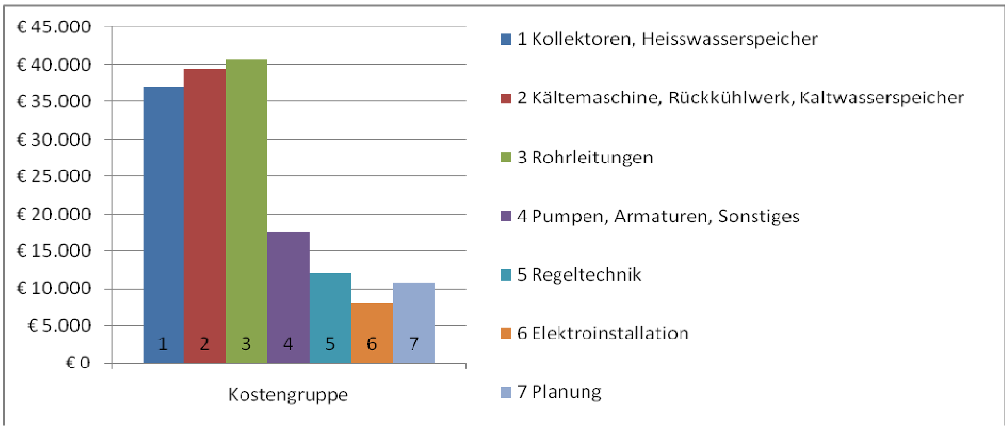


Abbildung 7 Gesamtkosten der solaren Kühlung: 127.000 € (30 kW Kälteleistung)
Spezifische Kosten : 4.233 €/kW
(jeweils ohne solarthermische Anlage und ohne Förderung)

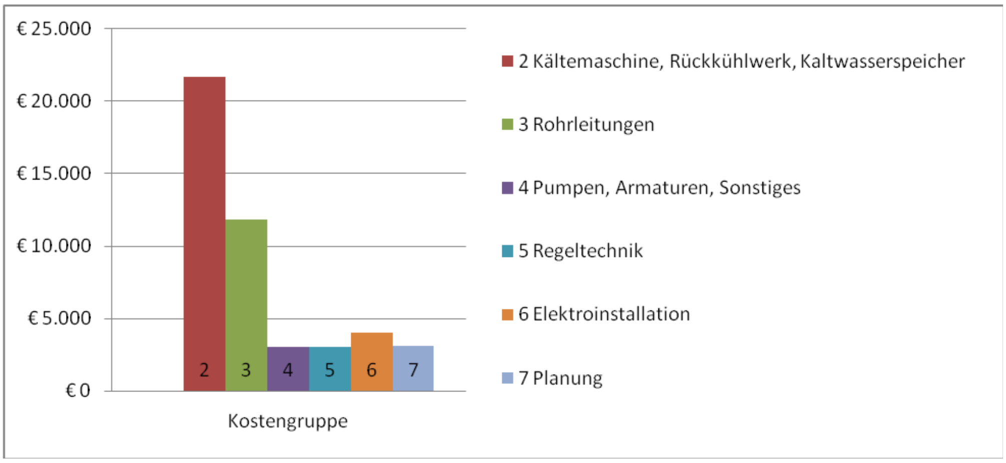


Abbildung 8 Gesamtkosten der Kompressionskühlung : 47.000 € (100 kW Kälteleistung)
Spezifische Kosten : 470 €/kW

Das Monitoringsystem wurde konform der in Subtask A3 entwickelten Richtlinie geplant und errichtet. Die Inbetriebnahme der Anlage erfolgte im Frühling 2008, im Sommer konnten daher bereits erste Erfahrungen im laufenden Betrieb gesammelt werden. So waren umfangreiche Adaptierungen an der Regelungsstrategie nötig, welche im Bericht „Solar Cooling System Design and Control“ zusammengefasst wurden. Die Schwerpunkte in diesem Beitrag liegen auf den Betriebsstrategien für Sommer- bzw. Winterbetrieb, dem Betrieb in der Übergangszeit, der Einbindung der Regelung der Kältemaschine in das Gebäudeleitsystem und auf diversen Änderungen welche einen optimalen Betrieb der Anlage gewährleisten sollen – siehe dazu auch Anhang ASIC 3.

Die enge Zusammenarbeit der österreichischen Partner im Task 38 und die Tatsache, dass jeder Partner in seinem Umfeld mit mehreren Anlagen zur solaren Kühlung beschäftigt ist, führte zu dem Vorhaben, dass im Rahmen der 1. Ausschreibung im Programm Haus der Zukunft PLUS folgendes Gemeinschaftsprojekt eingereicht wurde: SolarCooling Monitor - Evaluierung Energieeffizienz und Betriebsverhalten von solar-thermischen Kühlanlagen zur Gebäudekühlung in Österreich. Das Projekt wurde im Mai 2009 genehmigt und läuft noch bis Oktober 2011. Die in dem Projekt SolarCooling Monitor erarbeiteten Erkenntnisse wurden laufend in den SHC Task 38 eingebracht, und werden nachfolgend zusammengefasst:

Abbildung 9 - Gebäudeansicht

Abbildung 10 - Blick auf die Rückkühlwerke

Aufgrund der oben beschriebenen Adaptierungen an der Regelungsstrategie konnten im Sommer 2010 nur die Monitoringdaten der Monate Juli und August ausgewertet werden. Tabelle 2 zeigt einen Auszug, welcher im Report Subtask B3 „Monitoring Large Scale“ veröffentlicht wurde – siehe Anhang ASIC 4, ASIC 5 und ASIC 6.

Tabelle 2 – Ergebnisse des Monitoring

Größe	Ergebnis mit Einheit	Bemerkung
COP elektrisch	2,65	Gesamtsystem solare Kühlung
COP elektrisch	22,9	nur die solar betriebene Kältemaschine
COP thermisch	0,37	nur die solar betriebene Kältemaschine
Primärenergiekennzahl	1,06	Incl. Kompressionskältemaschine
Kollektorertrag	61,4 kWh/m²	
COP elektrisch	2,8	Gesamtsystem Kompressionskühlung

Als einzige Anlage in Österreich wird bei dieser Installation auch die elektrisch betriebene Kompressionskältemaschine vermessen. Die wesentlichsten Erkenntnisse lassen sich folgendermaßen zusammenfassen:

- Kompressionskälteanlagen arbeiten nicht mit den in den Datenblättern angegebenen Kenndaten
- Die Effizienz einer Kompressionskälteanlage hängt ebenfalls wesentlich von der Planung und Ausführung ab
- Der im Task 38 angenommene COP Wert von 2,8 für ein Referenzsystem mit elektrisch betriebener Kompressionskältemaschine scheint vernünftig gewählt zu sein
- Die Außentemperatur wirkt sich kaum auf die Effizienz der Kältemaschine aus
- Schlechte Daten hinsichtlich Effizienz ergeben sich vor allem im Teillastbetrieb (Start Stop)
- Der Standby - Betrieb in der Übergangszeit ist energetisch ungünstig

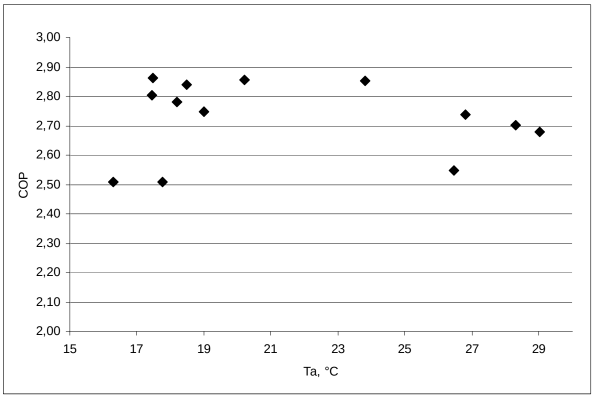


Abbildung 11 - Tages- COPs der Kompressionskältemaschine in der BH Rohrbach in Abhängigkeit der Außentemperatur Juli 2010

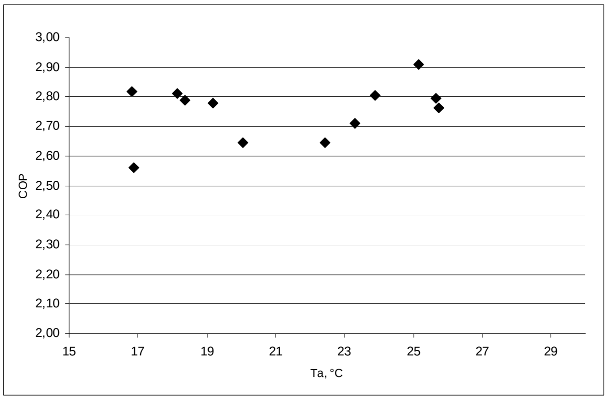


Abbildung 12 - Tages- COPs der Kompressionskältemaschine in der BH Rohrbach in Abhängigkeit der Außentemperatur August 2010

Durch das Projekt SolarCooling Monitor wurde die Zusammenarbeit der österreichischen Partner im SHC Task 38 noch intensiviert, da nach dem Sommer 2010 belastbare Erfahrungen und Messdaten von 11 Anlagen zur Verfügung standen. Daher wurde das Projekt Primärenergetische Optimierung von Anlagen zur solaren Kühlung mit effizienter Anlagentechnik und innovativen Regelstrategien und im Rahmen der 3. Ausschreibung „Neue Energien 2020“ eingereicht.

Ziel des Projektes ist es, den Primärenergieverbrauch von solarthermischen Kühlanlagen zu reduzieren. Dazu werden in einem ersten Schritt verbesserte Simulationsmodelle entwickelt. Anschließend werden diese Modelle verwendet, um mit Hilfe von detaillierten Systemsimulationen für typische Anwendungsfälle im Gebäude- und im Industriebereich sowohl Anlagenkonzepte, den Stromverbrauch von Komponenten als auch die Regelungskonzepte zu optimieren. Die optimierten Konzepte werden im letzen Projektjahr an drei bereits bestehenden Beispielanlagen umgesetzt und die Wirksamkeit der gesetzten Maßnahmen durch Monitoring verifiziert.

Institut für Wärmetechnik, TU Graz

EXERGIEANALYSE

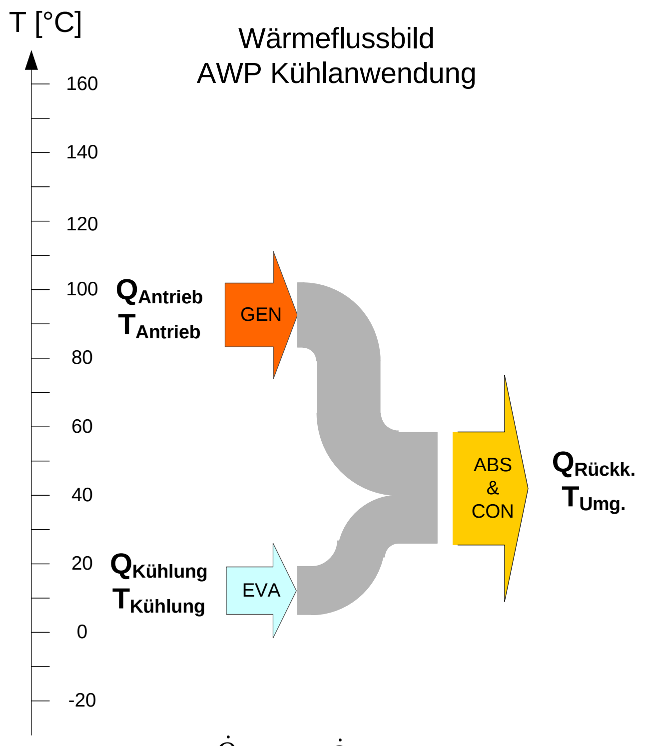
Im Rahmen unterschiedlicher Projekte wurde am Institut für Wärmetechnik eine Prototyp-entwicklung einer indirekt beheizten, stufenlos leistungsgeregelten Absorptionswärmepumpe kleiner Leistung, welche zur Gebäudeheizung und -kühlung (5 kW Kühl- und 15 kW Heiz-leistung) eingesetzt werden kann, und mit Ammoniak & Wasser als Arbeitspaar arbeitet, entwickelt. Dieser Prototyp ist hauptsächlich aus marktgängigen Standardkomponenten aufgebaut, was eine Minimierung der Gestehungskosten verspricht. Für alle Wärmetauscher wurden Plattenwärmetauscher eingesetzt, wobei der Absorber und Generator von der Dampf- und Flüssigphase der Lösung im Gleichstrom durchflossen wird.

Im Rahmen des Subtask C des IEA SHC Task 38 wurden nun die Eignung dieses Konzeptes für die Solare Kühlung untersucht. Es wurde dabei mittels Exergieanalyse die Auswirkung des konstruktiven Aufbaus des Generators und der Rektifikationskolonne auf die Effizienz und die internen exegetischen Verluste untersucht. Die Ergebnisse dieser Untersuchung sind im Anhang IWT1, IWT2 und IWT3 dargestellt:

HYGIENE VON NASSKÜHLTÜRMEN

Thermisch angetriebene Kälteanlagen haben im Vergleich zu elektrisch betriebenen Kompressionskälteanlagen einen Nachteil, sie benötigen bei gleicher Kühlleistung wesentlich höhere Rückkühlleistungen. Abbildung 13 zeigt, dass die gesamte dem System zugeführte Wärme, auf hohem Temperaturniveau als Antriebsenergie und auf niederem Temperaturniveau als Kühlung (Nutzen), wieder an die Umgebung auf mittlerem Temperaturniveau abgeführt wird. Weiters beeinflusst die Rückkühltemperatur wesentlich das notwendige Temperaturniveau der Antriebsenergie (also die benötigte Temperatur im Solarkollektor) und die Effizienz der Kältemaschine.

Aufgrund der hohen Leistungen hängen der Strombedarf und die Wirtschaftlichkeit des gesamten solaren Kühlsystems im großen Maße vom Rückkühlsystem ab.



$$COP_C \cong \frac{\dot{Q}_{Kühlung}}{\dot{Q}_{Antrieb}} = \frac{\dot{Q}_{EVA}}{\dot{Q}_{GEN}} \cong 0{,}6 - 0{,}9$$

Abbildung 13 - Energieflussdiagram für eine thermisch angetriebene Kältemaschine

Um das Temperaturniveau und den Strombedarf der Rückkühlung zu minimieren bietet sich der Einsatz von Nasskühltürme an. Allerdings können diese hygienische Probleme mit sich bringen, insbesondere hinsichtlich der möglichen unkontrollierten Vermehrung von Legionellen im Kühlwasser.

Im Rahmen des IEA SHC Task 38 wurde eine Studie über die Vor- und Nachteile von Nasskühltürmen in solaren Kühlanlagen kleiner Leistung erstellt. Diese Studie liegt ist Anhang IWT4 zu entnehmen.

ENTWICKLUNG EINER BEWERTUNGSMETHODE

Für den Subtask A wurde zu Beginn des Tasks an der Entwicklung der Bewertungsmethode (Erweiterung der FSC-Methode nach IEA SHC Task 26 und Task 32 auf solare Kühlung) und des Monitoringverfahrens mitgearbeitet (Anhang IWT5 & IWT6)

MONITORING

Die Hauptaufgabe im Bereich Anlagenanalyse lag in der Vermessung, Analyse und Optimierung von zwei, in Österreich installierten solarthermischen Kühlsystemen im kleinen Leistungsbereich. Das Monitoring wurde nach dem im Subtask A entwickelten, standardisierten IEA SHC Task 38 Level 3 durchgeführt. Diese Arbeiten wurden im Rahmen des FFG Projekts „Solar Cooling Monitor“, FFG Nr. 822265 und „Solar Cooling OPT“, FFG Nr. 825544 durchgeführt und dem IEA SHC Task 38 teilweise zur Verfügung gestellt. Vor Beginn der Messungen mussten beide Anlagen mit der entsprechenden Messtechnik ausgestattet bzw. die bestehende Messtechnik erweitert werden. Die Messungen laufen seit Sommer 2009 und werden seither ausgewertet und bewertet. Ausgehend von den Daten der ersten gemessenen Kühlsaison wurden Verbesserungsmaßnahmen erarbeitet, teilweise umgesetzt und wiederum neu bewertet.

Der Großteil der Ergebnisse zeigte ein enormes Verbesserungspotential auf. Insbesondere die elektrischen Leistungszahlen (COP) haben die Erwartungen wenig zufriedenstellend erfüllt. Die höchste monatliche elektrische Leistungszahl vor den Verbesserungen der ersten Anlage betrug 1,87 (August 2009). Nach den ersten Optimierungen lag die elektrische Leistungszahl bei durchschnittlich 3,4. Die zweite Anlage erreichte zwischen August und September 2009 durchschnittliche elektrische COPs von 3,09. Trotz der Ausarbeitung von Verbesserungsvorschlägen ging diese Anlage 2010 nicht in Betrieb (Details siehe Anhang IWT7 & IWT8 bzw. Taskreports A3b).

SIMULATION

Um die Potentiale der beiden Anlagen aufzuzeigen, wurden diese in der Simulationsumgebung TRNSYS abgebildet und auf theoretischer Basis weiter optimiert. Dazu wurden die Einzelmodelle an die spezifischen Randbedingungen der Anlagen angepasst und mit den Messdaten validiert. Die Ergebnisse dieser Validierung wurden mit jenen aus Subtask C2 verglichen und mit den österreichischen Partnern diskutiert. Das Potential, das mit Hilfe der Simulation aufgezeigt werden konnte, ist beträchtlich. Z.B. sind elektrische COPs >5 bei korrekter Auslegung und optimierten Regelstrategien durchaus realistisch – siehe Anhang). Einige wenige umgesetzte Beispiele konnten diese Ergebnisse auch im Betrieb erreichen.

Weitere Tätigkeiten zur Unterstützung der Taskaktivitäten wurden u.a. mit der Durchführung von Interviews mit den Betreibern der Anlagen (Subtask A5), sowie durch Testen von Checklisten und Auslegungstools (Subtask C) erbracht. Die 2. Auflage des Handbuchs

(Subtask D5) wurde mit einem kompletten Kapitel (Kapitel 9: Solar thermal system design issues) und einem Beitrag in Kapitel 10.2 (Built Examples) unterstützt (siehe Handbook).

KOSTENVERGLEICH

Weiters wurde ein Kostenvergleich zwischen den bestehenden solarthermisch betriebenen Anlagen und mit Photovoltaik (PV) betriebenen Kompressionskältemaschinen erstellt. Für diesen ersten abschätzenden Vergleich wurden etliche Annahmen und Vereinfachungen getroffen, dennoch zeigt diese Studie eine eindeutige Position auf. Die Ergebnisse zeigen 4- bis 7-mal höhere Kältegestehungskosten (kWh) der vermessenen solarthermischen Kühlsysteme gegenüber der PV-Variante und sogar 11- bis 19-mal höhere Kosten gegenüber konventionellen, elektrisch betriebenen Systemen. (siehe Anhang IWT9 & IWT10)

Schlussfolgerungen zu den Projektergebnissen

Die Ergebnisse sind in Berichten mit hoher Qualität dokumentiert und auf zahlreichen Konferenzen wurden diese verschiedensten Zielgruppen präsentiert. Die öffentlich verfügbaren Berichte sind unter http://iea-shc-task38.org/reports im pdf-Format herunter zu laden. Die SHC Task 38 hat einen essentiellen Beitrag zur Technologieentwicklung und der Markteinführung der solaren Kühlung geleistet.

Repräsentativ werden im Folgenden zwei Beispiele angeführt wie die österreichischen Partner auf Basis der SHC Task 38 Beteiligung Erkenntnisse und Ergebnisse der internationalen Zusammenarbeit in nationalen Forschungsprojekten einfließen lassen:

A) MONITORING

Die entwickelte SHC TASK 38 Prozedere zum energetischen Monitoring von Anlagen der solaren Kühlung wird in nationalen Forschungsprojekten angewendet. D.h. die Methode trägt durch die Definition der erforderlichen Messgeräte und -positionen zur Konzipierung eines Anlagen-Monitorings bei und ermöglicht durch eine einheitliche Bewertungsmethode den Vergleich des realen Anlagenbetriebs eines solaren Kühl- bzw. Klimatisierungssystems mit der konventionellen Ausführung auf Basis elektrischer Kompressionskälte.

B) ANLAGENOPTIMIERUNG und -REGELUNG

Solarthermisch angetriebene Kühl- bzw. Klimatisierungssysteme sind energetisch sinnvoll, wenn sie deutlich weniger elektrische (Hilfs-)Energie benötigen als konventionelle Kompressionskältemaschinen. Messergebnisse der durchgeführten Monitoringaktivitäten zeigen, dass es unter bestimmten Systembedingungen nicht zu den möglichen Stromeinsparungen kommt. Hauptverantwortlich für den Stromverbrauch sind die Hilfsaggregate wie Pumpen und Gebläse, vor allem aber die Rückkühlwerke. Daher müssen die solaren Kühl- bzw. Klimatisierungsanlagen hinsichtlich Stromeinsatz und zuverlässigem Anlagenbetrieb optimiert werden. Ziel ist die Reduzierung des Primärenergieverbrauchs von Anlagen zum solaren Heizen und Kühlen. Wesentliche Maßnahmen werden in der Entwicklung von optimierten System- und Regelungskonzepten für Gebäude- und Industrieanwendungen gesehen. Folgenden Maßnahmen sind zielführend:

- Entwicklung von optimierten System- und Regelungskonzepten für Gebäude- und Industrieanwendungen
- Entwicklung einer effektiven Rückkühleinheit
- Lebenszyklusanalyse der optimierten Konzepte

- Nachweis der Wirksamkeit der optimierten Konzepte durch Umsetzung an bestehenden Demonstrationsanlage

Maßgeblich durch die Impulse und Erkenntnisse aus der internationalen Zusammenarbeit im Rahmen der SHC Task 38 arbeiten die österreichischen Forschungspartner in nationalen Förderprojekten gemeinsam an der Technologieentwicklung zur solaren Kühlung bzw. Klimatisierung.

Wesentliche nationale Forschungsprojekte dazu sind:

- Haus der Zukunft plus - *Projekt SolarCooling Monitor*: Evaluierung Energieeffizienz und Betriebsverhalten von solarthermischen Kühlanlagen zur Gebäudekühlung in Österreich, FFG Projektnummer 822265
- Neue Energien 2020 - *Projekt Roadmap SK:* Entwicklung einer Technologie-Roadmap für solarthermische Kühlung in Österreich , FFG Projektnummer 819031
- Neue Energien 2020 - *Projekt SolarCoolingOpt* Primärenergetische Optimierung von Anlagen zur solaren Kühlung mit effizienter Anlagentechnik und innovativen Regelstrategien, FFG Projektnummer 825544

KNOW-HOW-TRANSFER

Know-how-Transfer – Austrian Institute of Technology

Aktuelle Vorträge und Veröffentlichungen im Bereich solare Klimatisierung von Seiten der Österreichisches Forschungs- und Prüfzentrum Arsenal Gesellschaft m.b.H.

- **Dong-Seon Kim**, “Analytic Modelling of a liquid desiccant dehumidifier”, 2nd International Conference ‘Solar Air-Conditioning’, Oct 18th/19th 2007, Tarragona, Spain

- **Dong-Seon Kim, Ivan Malenkovic**; “Dynamic Simulation of a Solar Cooling System using objectoriented Modelling”, 2nd International Conference ‘Solar Air-Conditioning’, Oct 18th/19th 2007, Tarragona, Spain

- **Tim Selke**, “Experience in planning, concepts and operation”, International Conference ‘Solar Cooling’, March 31st 2008, Vienna, Austria

- **Anita Preisler**, “Strategies for market penetration”, International Conference ‘Solar Cooling’, March 31st 2008, Vienna, Austria

- **Dong-Seon Kim**, “R&D activities on components and examples”, International Conference ‘Solar Cooling’, March 31st 2008, Vienna, Austria

- **A. Preisler**, T. Selke, Einsatz solare gestützter Klimatisierung am Beispiel ENERGYbase, 9. Internationales Symposium für thermische Sonnenenergienutzung, Solar 2008, 3.-5. September 2008, Gleisdorf, Österreich

- **Tim Selke**, *A. Preisler and U. Schneider*; ‘ENERGYbase – Sunny Office Future’, Eurosun 2008, 1st International Conference on Solar Heating, Cooling and Buildings, Lissabon, 7th to 10th October 2008, Book of Abstracts, Nr. 357

- **A. Preisler,** T. Selke, L. Sisó, A. LeDenn, R. Ungerböck, Case Study ROCOCO - Reduction of costs of Solar Cooling Systems, 1st International Conference on Solar Heating, Cooling and Buildings, Lissabon/ Portugal , 7th to 10th October 2008, Book of Abstracts, Nr. 187

- **S. Gosztonyi**, A. Preisler, ENERGYbase – Einsatz solar gestützter Klimatisierung, KinG – Seminar, Kompetenznetzwerk innovativer Gebäudetechnik, 28. Oktober 2008, Wien, Österreich
- **Tim Selke**, 'Visionen werden wahr: ENERGYbase – eine sonnige Bürozukunft'; Expertenforum Beton | Heizen + Kühlen mit Beton | Klimawandel fordert Baukonzepte; 28th Januar 2009; Graz, Austria
- **A. Preisler,** T. Selke, Solarthermische Kühlung in Österreich, Expertenworkshop im Rahmen des e2020 Projektes Roadmap SK – Einwicklung einer Technologieroadmap zur solarthermischen Kühlung in Österreich, 19.02.2009, Wien, Österreich
- **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 Conference on Solar Heating, Cooling and Buildings, Palermo/ Italy Sept 30th – Oct 3rd 2009, Proceedings Nr. 081
- **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

Weitere Aktivitäten:

- Zu einem aktuellen Bauvorhaben (Stand April 2011) in den Vereinigten Emiraten / Al Ain hat das AIT Expertise zur Solaren Kühlung eingebracht. Als Vertragspartner der IC Consulenten Ziviltechniker GmbH werden im laufenden Projekt 'Sheikh Zayed Desert Learning Center' SZDLC in Al Ain hohe Ziele hinsichtlich Nachhaltigkeit und Energieeffizienz im Rahmen einer LEED und Estidama Zertifizierung gesetzt. Das Österreichische Unternehmen S.O.L.I.D. Gesellschaft für Solarinstallation und Design mbH S.O.L.I.D plant und liefert für das SZDLC eine thermische Solaranlage für die Kühlung der Innenräume mit einer Kälteleistung von 350 kW. Für die Bereitstellung dieser Kälteleistung sorgt eine Fläche von ca. 1.100 m² Hochtemperaturkollektoren, welche neben dem Gebäude frei aufgestellt werden. Die Inbetriebnahme der Anlage soll 2011 erfolgen.

Organisation einer Konferenz:

- Am 31. März 2008 hat das AIT in Kooperation mit dem BMVIT; der IEA SHC TASK 38 und dem 6. Rahmenprogramm der EU eine internationale Konferenz zum Thema ‚Solar Cooling Systems' im TECHbase Vienna; Giefingasse 2, 1210 Vienna veranstaltet. Mit ausgewählten Experten als Referenten und einem marktnahen Programm kann diese Veranstaltung mit rund 120 Konferenzteilnehmern als großer Erfolg gewertet werden. Das Programm ist dem Anhang AIT5 zu entnehmen. Das Tagungsband ist online verfügbar (Stand Mai 2011):

  http://www.energiesystemederzukunft.at/edz_pdf/20080401_e2050_thermisch_kuehlen_tagungsband.pdf

Know-how-Transfer - Institut für Wärmetechnik TU GRAZ

Vorträge und Veröffentlichungen

- O. Kotenko, H. Moser, R. Rieberer, (2009) ‚FEASIBILTY STUDY OF ALTERNATIVE ABSORTPION HEAT PUMP PROCESSES' 3rd International Conference on Solar Heating, Cooling and Buildings, Palermo/ Italy Sept 30th – Oct 3rd 2009
  siehe Anhang 3.6

- Sparber, W., Thuer, A., Besana, F., Streicher, W., Henning, H.M., (2008) , Unified Monitoring Procedure and Performance Assessment for Solar Assisted Heating and Cooling Systems, Eurosun 2008, 1st International Conference on Solar Heating, Cooling and Buildings, Lissabon, 7th to 10th October 2008, Book of Abstracts, p. 318-319´

- Moser H., Rieberer R., (2008) "Second Law Analysis of a Laboratory Prototype of a Single Stage Ammonia / Water Absorption Heat Pump", International Sorption Heat Pump Conference, Seoul, Korea; p. AB 109 - 117

- Moser H., Rieberer R., (2008) "Thermodynamic Comparison of Different Designs of Single Stage Ammonia / Water Absorption Heat Pumps", IIR Gustav Lorentzen Conference on Natural Working Fluids; Copenhagen, Dänemark; p 583 - 590

- Moser H., Rieberer R., Dornstädter D., (2008): "Thermodynamic Analysis of a Small-Capacity Ammonia / Water Absorption Heat Pump for Heating and Cooling"; 9th Int. IEA Heat Pump Conference, 20 – 22 May 2008, Zürich, Switzerland

- Moser H., Rieberer R., (2007) SMALL-CAPACITY AMMONIA / WATER ABSORPTION HEAT PUMP FOR HEATING AND COOLING – FOR SOLAR COOLING", International Conference of Solar Air Conditioning; Tarragona, Spain; 18 – 19 October 2007,

- Streicher, W. (2006), Simulationstool zur Auslegung Solarer Kühlanlagen – Ergebnis aus der Task 25 der Internationalen Energie Agentur SHC, Vortrag im Rahmen des KinG Fachseminar für Gebäudetechniker, Wien, 5. Juli 2006

- Streicher, W., Franzke, U., Seifert, C. (2005), Solar Assisted Air Conditioning of Buildings – Computer Design Tool for System Design, International Conference on

Solar Air Conditioning, Kloster Banz, D-96231 Staffelstein, Editor: Otti- Technologie-Kolleg, Wernerwerkstr. 4, D-93049 Regensburg, P.248 – 252.

- Streicher, W. (2005), Solar Kühlen – Einführungsvortrag; Hintergründe und aktueller Forschungsstand, „Solar Kühlen“ Eco & Co-Firmentreffen am 9. JUNI 2005 im Rahmen der TEC-Days im MesseCenter Graz

- Neyer, Daniel; Streicher, Wolfgang; Weissensteiner, Thomas (2010), Practical experience of two small scale cooling plants and cost comparison to PV driven chillers. EuroSun 2010 - International Conference on Solar Heating, Cooling and Buildings. 28.09. - 01.10.2010, Graz. Saint Maur: OCS Associates (Europe), ISBN 978-3-901425-13-4, S. 88

- Weissensteiner (2010), Practical experience of two small scale solar cooling plants and cost comparison to PV driven compression chillers, Diplomarbeit am Institut für Wärmetechnik TU Graz, Dezember 2010, Graz, gesperrt bis mind. 12.2011

Weitere Aktivitäten:

Laufende Beratung von Kollektor- und Systemherstellern sowie von Gebäudeplanern und Architekten üb er die Einsatzmöglichkeiten aber auch Grenzen der solaren Klimatisierung.

Weiters war das IWT in Kontakt mit den Firmen Pink GmbH und Helioplus Energy Systems GmbH die in Österreich Absorptionswärmepumpen kleiner Leistung herstellen bzw. vertreiben. Mit der Fa. Pink wurden Gespräche bezüglich Komponentenentwicklung für Ammoniak/Wasser-Anlagen geführt die den Strombedarf von Absorptionswärmepumpen reduzieren sollen und es wurden mögliche Kooperationen diskutiert.

Know-how-Transfer
ASIC- Austrian Solar Innovation Center

Vorträge und Veröffentlichungen im Bereich solare Klimatisierung

- H. Focke. „Solares Klimatisieren – Konzepte und Erfahrungsberichte“, Tagung Effiziente Gebäudekühlung des Oberösterreichischen Energiesparverbandes, 8. Mai 2008 in Linz

- H. Focke, A. Preisler, G. Geissegger „Designing of a Technology- Roadmap for Solar Assisted Air Conditioning in Austria“, 3rd Int. Conference Solar Air Conditioning, Palermo, Italy 09.2009

- H. Focke, G. Steinmaurer: "Cooling load demand assessment - a key issue for economic operation of solar cooling systems", 2nd International Conference on Solar Air-Conditioning, , Tarragona, Spain 10.2007

- H. Focke: " Field report of a Solar Assisted Air Conditioning system in an office building located in Upper Austria", 2nd International Conference on Solar Air-Conditioning, , Tarragona, Spain 10.2007

- H. Focke: “Field report of a solar assisted air conditioning system in an office building located in Upper Austria”. WSED World Sustainable Energy Days, Wels, Austria 02.2009

- H. Focke. „Solares Klimatisieren – Konzepte und Erfahrungsberichte“, TECHNO LOG 10, Energieeffizienz – Mehr aus Weniger, veranstaltet von TIM bei WKO OÖ, 29. April 2010 in Wels

- H. Focke: “Field report of a solar assisted air conditioning system located in Upper Austria”. WSED World Sustainable Energy Days, Wels, Austria 02.2011

Weitere Aktivititäten

Hinsichtlich des Knowhow-Transfers war ASIC an folgenden Veranstaltungen aktiv beteiligt:

8. Mai 2008, Tagung „Effiziente Gebäudekühlung“
Ort: Redoutensäle, Promenade 39, 4021 Linz
Veranstalter: Oberösterreichischer Energiesparverband
Zielgruppe: Energieverantwortliche in Betrieben, Solartechnologie-Anbieter, Planer/innen, Installationsbetriebe, Energieberater/innen, etc.

Der Energieverbrauch für die Kühlung von Büro- und anderen Dienstleistungsgebäuden steigt von Jahr zu Jahr und übersteigt in manchen Fällen sogar den Energieverbrauch für die Beheizung in den Wintermonaten. Standen in der Vergangenheit die Optimierung von Heizanlagen und der Gebäudehülle für den Winter im Vordergrund, so rückt jetzt die Behaglichkeit im Sommer verstärkt in den Mittelpunkt. Aktive Gebäudekühlung sollte auf das absolut notwendige Maß reduziert werden, statt dessen sollte das Augenmerk auf die Vermeidung einer Überhitzung im Sommer mittels geeigneter Architektur und passiven Systemen gerichtet sein.

Rund 240 Personen informierten sich bei der Tagung über die Möglichkeiten, technischen Anforderungen und Innovationen im Bereich der Gebäudekühlung. Weiters wurden die Themen der Vermeidung sommerlicher Überhitzung und die Möglichkeiten passiver Kühlsysteme sowie Effizienzsteigerung bei aktiven Kühlsystemen diskutiert und Vorzeigeprojekte präsentiert.

29. April 2010, TECHNO LOG 10 „Energieeffizienz – Mehr aus Weniger“
Ort: Fh Campus Wels
Veranstalter: TIM bei WKO Oberösterreich
Zielgruppe: Planer und Anbieter von Heizungs- und Kältetechnik, Ziviltechniker und technische Büros
Inhalt
Im Block Solarthermie wurde von ASIC der 2- stündige Workshop „Solar unterstützte Kühlsysteme, Technologien und Rahmenbedingungen für einen sinnvollen Einsatz von solarem Kühlen und Kühlen mit Abwärme“ gehalten.
Die Unterlagen dazu finden sich auf: http://www.technolog.at/13_DEU_HTML.php

Seit dem Wintersemester 2010/ 11 werden im Master- Studiengang „Öko- Energietechnik“ die Lehrveranstaltungen Solares Heizen und Kühlen bzw. Solare Energieversorgungssysteme angeboten.

Der Master-Studiengang „Öko-Energietechnik“ bietet eine grundlegende Vertiefung in Themen der Umwandlung, der Verteilung und der optimierten, umweltfreundlichen Verwendung von Energie in Anlagen und Gebäuden. Mit den wählbaren Schwerpunkten „Solartechnik“ oder „Gebäudeoptimierung“ erfolgt eine spezielle Fokussierung auf Funktionsweise, Planung, Bau und Betrieb von energietechnischen Anlagen und Gebäuden. Projektorientiertes Lernen sorgt für den unmittelbaren Umsetzungs- und Praxisbezug und eröffnet die Möglichkeit, zusätzliche Eigenschwerpunkte zu setzen.

ASIC konnte im Rahmen dieser Lehrveranstaltung Expertenwissen aus dem SHC Task 38 und aktuell veröffentlichte Publikationen einbringen und somit eine Lehrveranstaltung mit neuesten Informationen und Erkenntnissen abhalten. Die Lehrveranstaltung besteht aus 30 LE Vorlesung und 15 LE Übung und ist in folgende Abschnitte gegliedert:

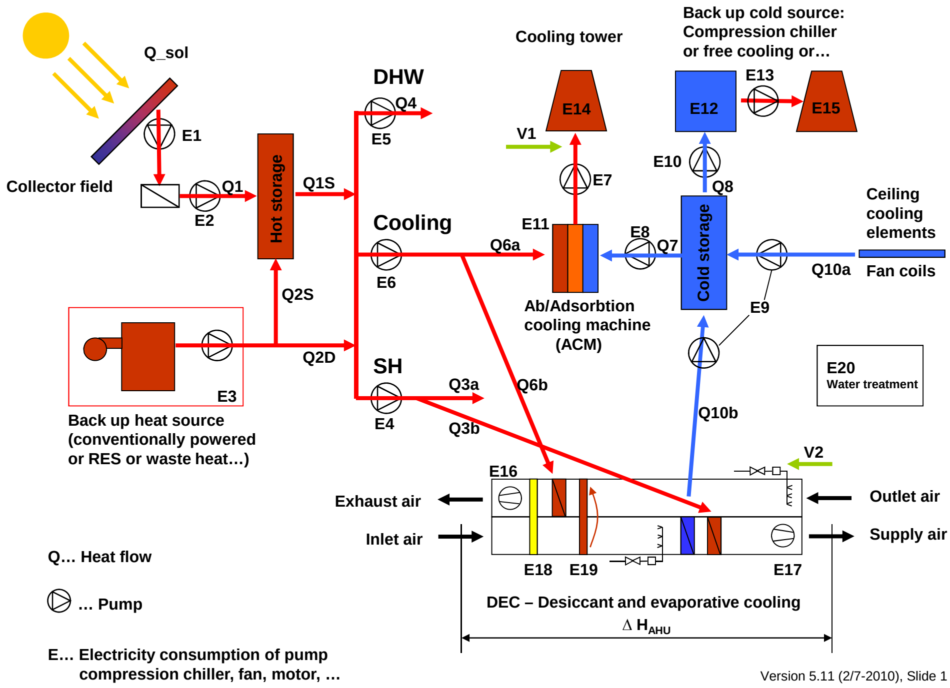
- Einleitung, Motivation
- Thermischer Komfort
- Grundlagen Lüftung- und Klimatechnik
- Technologien
- Maschinen
- Entscheidungsschemata
- Dimensionierung
- Wirtschaftlichkeit
- Ausblick

ANHANG

Austrian Institute of Technology

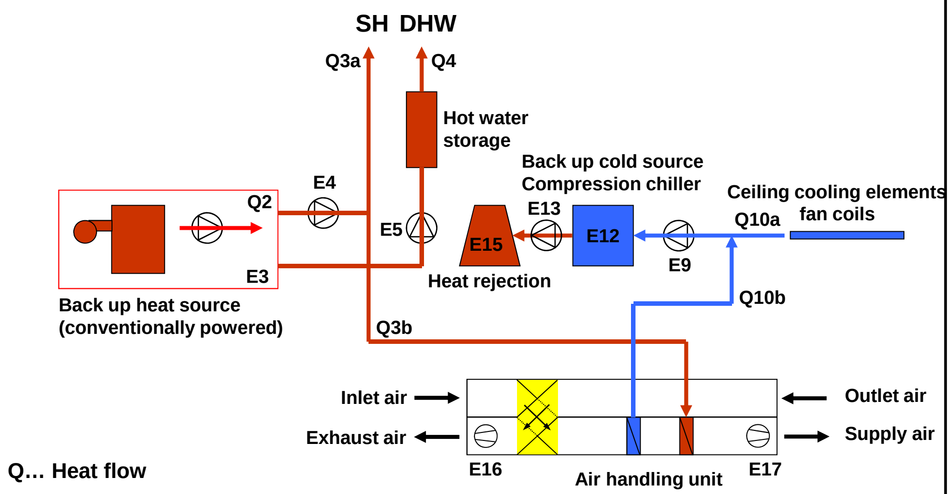
ANHANG AIT 1

IEA SHC TASK38 Monitoring Scheme for MA34 solare Adsorptionskaltwassererzeugung



Version 5.11 (2/7-2010), Slide 1

Definition of a conventional reference system



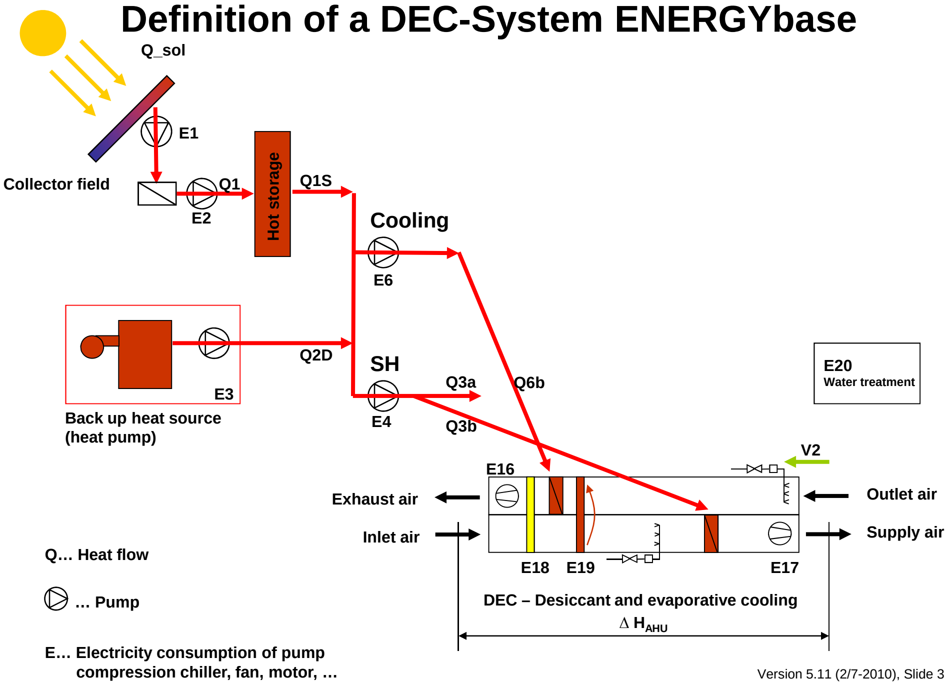
Q… Heat flow

… Pump

E… Electricity consumption of pump, fan, motor, …

Version 5.11 (2/7-2010), Slide 2

Definition of a DEC-System ENERGYbase



Version 5.11 (2/7-2010), Slide 3

ANHANG AIT 2

IEA SHC TASK38 – Enthalpie Berechnung zur solaren Klimatisierung im Gebäude ENERGYbase

This tool was elaborated within IEA SHC Task38 by:

Version: T38_MonProc_V5-11
released on July 2nd, 2010

UNIVERSITA' DEGLI STUDI DI PALERMO / Dipartimento di Ricerche Energetiche ed Ambientali (DREAM) / Palermo / Italy
Bettina Nocke, Pietro Finocchiaro (Group Leader: Prof. Ing. Marco Beccali)
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Task 38
Solar Air-Conditioning
and Refrigeration

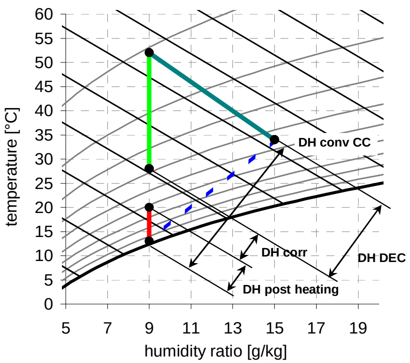
This file is aimed to calculate the different thermal energies in the Desiccant Cooling Process and in the Reference System (conventional AHU)
Please insert your measured values in the yellow columns! (with your time-step and for the whole month)
in the next worksheet "postheat - fixed t_supply_conv")

For a better validation of possible scenarios, 3 cases are considered:
1. **with post-heating** of supply air in the Reference AHU after the cooling to the dew-point for deumidification
up to a **FIXED** supply temperature (to be choosen individually for every plant)
2. **with post-heating** of supply air in the Reference AHU after the cooling to the dew-point for deumidification
up to the **IDENTICA**L supply temperature as measured in the DEC-AHU
3. **without post-heating**

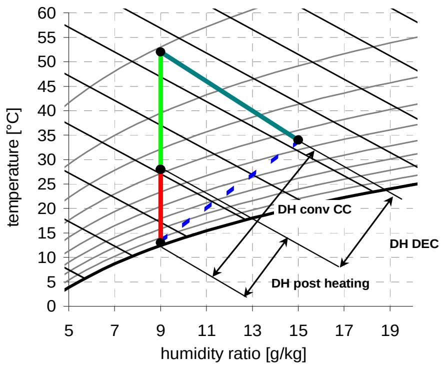
For the first and the second case you will find the algorith in the next 2 sheets.
For the third case, you can use the calculated value for m*dH for the conventional Cooling Coil.
The inserted value you will find here, are only an example (first days of July 2008!) Please replace and complete this values with your monitoring data!

If you need to change anything, just deactivate the worksheet protection, there is NO password.

ad 1) post heating up to a fixed supply temperature
(to be choosen individually for every plant)



ad 2) post heating up to the identical supply
temperature as masured in the DEC-AHU



Summary of calculation results:
Copy green field and paste values into line 96-106 in "3rdLvl" in "100702_T38_MonProc_V5-11.xls"

Month:		**July 10/LA01**	
with postheating, T_supply individually fixed:		**20**	°C
m*DH corr	kWh	**1130**	kWh
m*DH CC conv	kWh	**5702**	kWh
m*DH AHU conv	kWh	**4516**	kWh
m*DH AHU DEC cooling	kWh	**3386**	kWh
m*DH AHU DEC heating	kWh	**23**	kWh
m*DH postheat	kWh	**1186**	kWh
with postheating, Tsupply_conv = Tsupply_DEC:		**23,9**	°C ave
m*DH CC conv	kWh	**5702**	kWh
m*DH postheat	kWh	**2315**	kWh

DH AHU conv = DH DEC + DH corr

ANHANG AIT 3

IEA SHC TASK38 B3 Monitoring Report – SDEC ENERGYbase

1.1 Austria: ENERGYbase

Description of the application

Type of building Offices building
Location A- 1210 Vienna
In operation since 2008
System operated by Siemens Facility Management
Air-conditioned area ~ 5.000 m²
System used for space heating? Yes
System used for DHW preparation? No

General description of the system

The ENERGYbase building with 7.500 m² useful area is located in the 21st district of Vienna and was finished in July 2008. It is mainly used as an office building, but it also contains two universities of applied science and laboratory areas. The building and energy performance focuses on sustainable, innovative technologies as "Passivhaus" standard, ground coupled heat pumps, free cooling, solar cooling, building integrated Photovoltaic and ecological air humidification with plants. ENERGYbase is equipped with two identical air handling units "Desiccant Evaporative Cooling" (DEC) system with following key figures for the heating and cooling systems:

Technical data of the basic components

Desiccant cooling units

1. unit

Technology	Desiccant and evaporative cooling
Nominal capacity	~ 40 kW
Brand of cooling units	Robatherm
Cooling load subsystem	Central AHU
Dehumidification	Sorption wheel (Klingenburg SECO 1770)
Regeneration power	80 kW

2. unit

Technology	Desiccant and evaporative cooling
Nominal capacity - Nominal air flow	~ 40 kW
Brand of cooling units	Robatherm
Cooling load subsystem	Central AHU

Dehumidification	Sorption wheel (Klingenburg SECO 1770)
Regeneration power	80 kW

Solar thermal collectors fields

1. field

Collector type	Flat-plate collectors
Brand of collector	Sonnenkraft /MEA DESIGN
Collector area	285 m²
Tilt angle, orientation	32°C
Collector fluid	Water-Glycol 30%
Typical operation temperature	80°C
Heating load subsystem	Concrete core activation, central AHU

Technical data of optional components

Heat back-up system

1. unit

Technology	Electrical heating coil
Nominal capacity	80 kW
Heating load subsystem	Central AHU

2. unit

Technology	Electrical heating coil
Nominal capacity	80 kW
Heating load subsystem	Central AHU

3. unit

Technology	Heat pumps
Nominal capacity	2x160 kWth
Heating load subsystem	Concrete core activation

Heat storage system

1. unit

Number of units	1
Technology	Water tank
Storage capacity	15.000 l
Shared with heat back up systems	No
Type of connection	-

2. unit

Number of units	1
Technology	Water tank
Storage capacity	2.000 l
Shared with heat back up systems	Yes; with heat pumps
Type of connection	-

1.2 Austria: ENERGYbase

<u>Monitoring scheme</u>

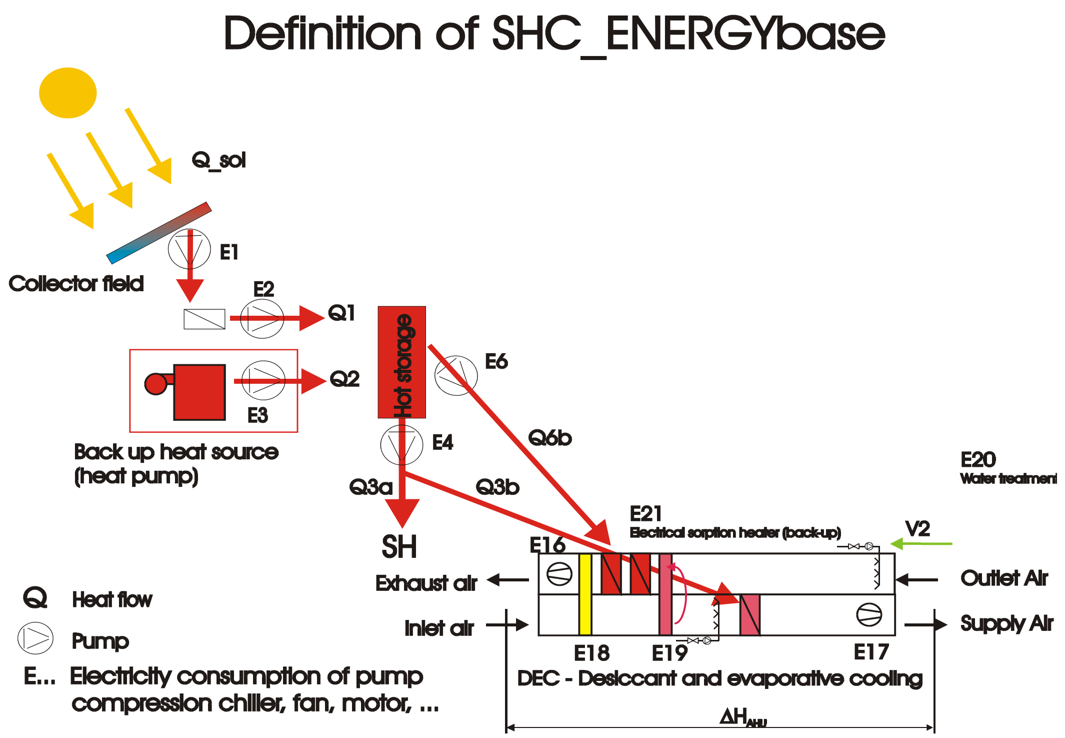


Figure 1: Monitoring scheme of DEC systems in ENERGYbase

<u>Monitoring data availability and reliability</u>

Monitoring data for the whole DEC system are available from 21st of September 2009 until August 2010. From January 2009 until 21st of September 2009 only part of the monitoring evaluation could be made. Therefore, the monitoring evaluation for 2009 starts with October.

Monitoring failures in the solar thermal system happened at following time periods in 2009:

3.10 - 7.10

Recording of regeneration: AHU 1 started with recording at 19.01.2009; AHU 2 started recording at 14.10.2009

The water consumption can only be stated for the whole year with ~ 200m³ for humidification in summer (adiabatic cooling) and winter (humidification of inlet air).

In 2010 the recording of monitoring data worked continuously.

Monitoring procedure

Monitoring procedure with data from October 2009 until September 2010

1. level

Output	Results	Units	Comments
COPel, y,tot	11,94	-	
COPel, y, overall	11,65	-	Enthalpy difference of AHU was recorded starting from 21st of September 2009
PERres, yearly	2,01	-	Renewable Energy source: electrically driven heat pumps (ε_RES=0,4; ηboiler_RES=3,2)
PERfossil, yearly	2,96	-	Fossil fuels: gas boiler (ε_fossil=0,9; ηboiler=0,9)
PERref, yearly	104,2	%	ΔH postheat is calculated now correctly
Costs Per Kilowatt	4.081	€/kW_cold	

2. level

Output	Results	Units	Comments
ηcoll,util, y	**21,6**	%	
ΔQsol, y	253.435	kWh	
SFhotstorage, y	33,8	%	
Qcoll_yield , y	244,9	kWh/m²	
Qloss_stge, y	16.431	kWh	
ηstge, y	76,5	%	
Qloss_sys, y	16.431	kWh	
ηsys, y	92,0	%	
Q*tot, y	64.233	kWh	
ηheat_solrad, y	19,9	%	
Qsolunex, y	258.989	kWh	

3. level

<table>
<tr><th colspan="3">Output</th><th>Results</th><th>Units</th><th>Comments</th></tr>
<tr><td rowspan="2">Calculation of Primary Energy Savings (f sav,shc)</td><td>Based on PER_RES</td><td rowspan="2">SPECIAL FOR DEC</td><td>49,53</td><td>%</td><td rowspan="2"></td></tr>
<tr><td>Based on PER_fossil</td><td>65,64</td><td>%</td></tr>
<tr><td colspan="3">Calculation of Fractional Solar Consumption - Solar Heating and Cooling (FSC-SHC) (for small tanks: one to two day capacity!!)</td><td>0,45</td><td></td><td></td></tr>
<tr><td colspan="3">Water treatment and consumption</td><td>1.000</td><td>kWh</td><td>~200m³/a</td></tr>
<tr><td colspan="3">COP electric for thermal cold production with chiller (without solar pumps and water treatment)</td><td>-</td><td></td><td>No chiller</td></tr>
<tr><td colspan="3">COP electric for thermal cold production with chiller AND solar pumps</td><td>-</td><td></td><td>No chiller</td></tr>
<tr><td colspan="3">COP electric just the chiller itself</td><td>-</td><td></td><td>No chiller</td></tr>
<tr><td colspan="3">COP thermal just the chiller</td><td>-</td><td></td><td>No chiller</td></tr>
<tr><td colspan="3">COP electric for heating and cooling DEC (without water treatment and Solar pumps)</td><td>8,12</td><td>-</td><td>October: 6,55
November: 8,51
December: 12,35
January: 18,41
February: 15,30
March: 12,78
April: 7,40
May: 1,62
June: 1,80
July: 2,67
August: 1,88
September: 1,88</td></tr>
<tr><td colspan="3">COP electric for heating and cooling DEC (but without water treatment)</td><td>7,98</td><td>-</td><td></td></tr>
<tr><td colspan="3">COP thermal DEC</td><td>0,51</td><td>-</td><td></td></tr>
</table>

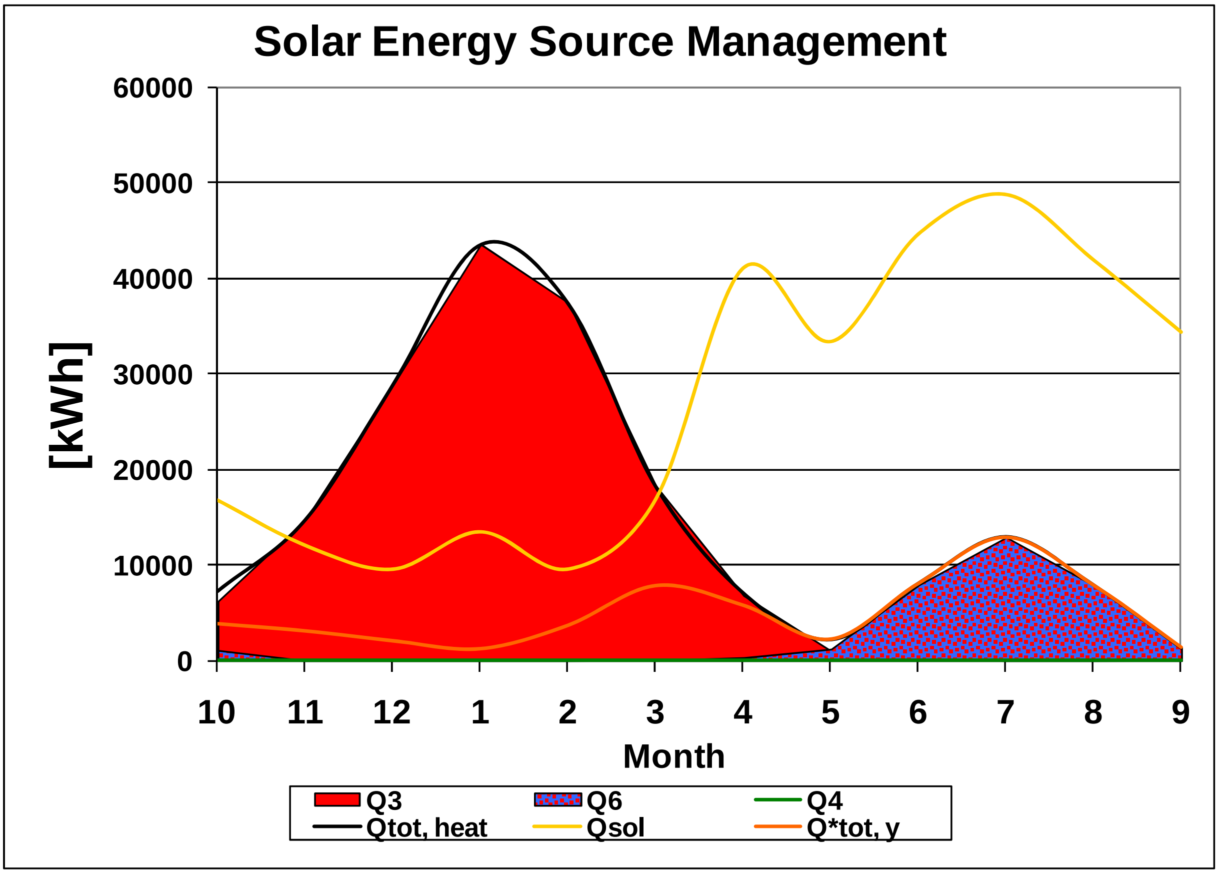


Figure 2: Solar energy source management October 2009 until September 2010

Special analysis of monitoring data

Since the monitoring recording contained many gaps in July and August 2009, the data of the 22nd and 23rd of September 2009 were selected for the evaluations represented here. During these two days the DEC system operated with the use of solar energy in order to regenerate the sorption wheel (DEC mode) for around 8 hours each day. A first comparison of measured temperature values of two different position indicate a substantial difference (up to 5 K) between the temperature sensors of the weather station (WS-10) at the ENERGYbase and of the inlet position of the fresh air stream into the two DEC systems (see Figure 3).

A comparison of the meteorological data of UBIMET[1] with the measured data at the ENERGYbase (Figure 3) shows, that the measurements of the weather station corresponded to those of UBIMET. Consequently this fact indicates an air interchange between the fresh air and

[1] Meteorological data in hourly values, measurement point: Donaufeld (approx. one kilometre distance from ENERGYbase), UBIMET GmbH, Vienna

the exhaust air stream. Since the exhaust air temperature was very high on these two days in the DEC mode (up to 51°C), it affected on a higher fresh air temperature up to 5 K.

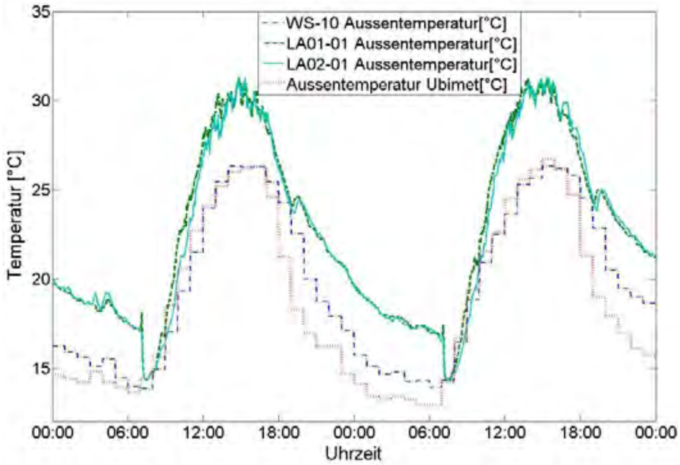


Figure 3: Comparison of measured outside air temperatures at ENERGYbase with UBIMET data; 22nd and 23rd of September 2009

In Figure 4 it can be recognized that after an initial overshooting of the regeneration air temperature the set value of 70°C (Klingenburg SECO 1770 sorption wheel) could be kept constant during the regeneration mode even in September.

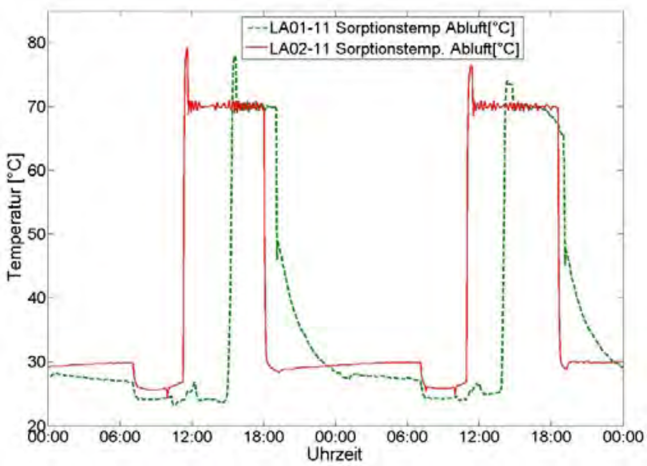


Figure 4: Exhaust air temperature after regeneration heating coil for both air handling units; 22nd and 23rd of September 2009

Lesson learnt

For the positioning of the fresh air stream and the exhaust air stream of DEC systems, special care is required. In case of unfavorable positioning of these openings, an admixture from exhaust air to fresh air can take place. This causes a significant heating up of the fresh air, particularly in the DEC systems with high exhaust air temperatures.

The COP electric evaluation showed that the whole DEC system works much more efficient in wintertime when the sorption wheel is also used for humidity recovery.

Reference persons on data

Contact Person: Anita Preisler

E-mail of contact person: anita.preisler@ait.ac.at

Name of the institution for the monitoring: AIT - Austrian Institute of Technology

Relevant literature

1. 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

ANHANG AIT 4

M. Jones, T. Selke “SURVEY OF CONTROL AND CONFIGURATION OF SOLAR HEAT DRIVEN CHILLER SYSTEMS” , EUROSUN 2010, 2nd International Conference on Solar Heating, Cooling and Buildings, Graz, Austria, 28th to 1st October 2010

SURVEY OF CONTROL AND CONFIGURATION OF SOLAR HEAT DRIVEN CHILLER SYSTEMS

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Abstract

Installed solar heat driven chiller (SHDC) cooling systems show a wide variety in both their configurations and control strategies, a variety beyond simple differences in applications and climates. Given the number of possible hydraulic circuit configurations, components, control strategies, and objectives, there are a large number of possible solutions in designing a solar heat driven cooling system. The design and operation of SHDC systems is further complicated by their interdependent and dynamic nature.

This paper presents an overview of SHDC systems with an emphasis on system control, drawn from an analysis of the configuration and control of 11 SHDC systems installed in Europe within Task 38 “Solar Air-Conditioning and Refrigeration” of the International Energy Agency Solar Heating & Cooling Programme [1]. The rated chiller capacity of the surveyed systems ranges from 7.5 to 300 kW chilled water with variations in application, auxiliary heating and cooling, storage strategies and control.

In terms of the secondary circuit control strategy (the driving heat circuit supplying the generator), most systems surveyed control the chilled water temperature by regulation of the driving hot water temperature from a hot water storage tank. Meanwhile, emerging literature and field experience are demonstrating that efficient system performance can be obtained by regulating the cooling water temperature by means of a variable speed drive on the cooling tower fan. This may represent a shift in the way these systems are designed and operated. Such control aspects and their interaction with system configuration will be discussed.

1 Introduction

A recent survey of world-wide solar cooling systems listed only 66 installed solar heat driven chiller (SHDC) based systems, demonstrating that field experience with these systems is relatively limited [2]. The variety of installed SHDC systems reflects designer’s exploration of the different ideas in the field of solar cooling. Within Task 38 “Solar Air-Conditioning and Refrigeration” of the International Energy Agency Solar Heating & Cooling Programme, an overview of solar air-conditioning systems was undertaken. The purpose of this activity (subtask B2 of Task 38), was to collect experiences and analyze both configuration and control of these systems. Results of this activity include contribution to the upcoming next edition of the Solar-Assisted Air-Conditioning in Buildings - A Handbook for Planners [3], as well as a final subtask report.

2 Control of Solar Cooling Systems

2.1 General control objectives

The control of solar cooling systems plays an important part in an overall facility energy management strategy, in order to minimize energy use and cost while maintaining internal comfort requirements. The control of a solar cooling installation is largely dependent on its configuration and application. The design of a solar cooling system, like the design of any HVAC system, is a trade-off between owning costs (primarily the initial capital expenditure), operating costs (maintenance and energy costs), and occupant comfort. The control of a solar cooling system largely influences operating costs and comfort levels. In general, the objectives of control include;

- Autonomous operation of the system
- Minimum cost of energy
- Maximum occupant comfort
- Safety
- Fault detection and correction

HVAC systems in buildings are typically controlled using a two-level control structure. The more general supervisory control level specifies set points and time dependent modes of operation for the lower local-loop control level, which attempts to meet the set points using the control actuators. The following discussion focuses on supervisory control strategies. The main goals of solar cooling control strategies are to;

- Avoid malfunctioning of both the overall system and the several technical components and as well to ensure a trouble-free operation
- Meet the cooling loads (i.e., guarantee certain indoor thermal comfort conditions in terms of relative humidity and temperature of indoor air)
- Minimize the primary energy usage per kilowatt produced chilled water or per cubic meter conditioned air

That means driving the implemented energy consumers (pumps, ventilators, solar collector, chiller etc.) of the overall system in its minimal energy consuming operation point and benefiting from maximal use of solar heat by using it as the major driving energy source for the thermally driven system.

2.2 Survey of scientific literature

Taking a broader perspective of the control issues, two strategies meeting two different objectives of control can be identified; a solar-guided strategy, and a cold-guided strategy [3]. The solar-guided strategy is characterized by a high chilled water demand such that the chiller operates whenever driving heat is available, maximizing the cold production at the cost of a variable chilled water temperature. The cold-guided strategy alternatively represents an active attempt by the controller to reach a chilled water temperature set-point, at the cost of reducing the chilling capacity. Regulation of the chilled water temperature by control of the cooling water circuit is an active area of research. Advantages of this strategy include reducing cooling tower electricity consumption, a major factor in determining system efficiency, and also water consumption [4]. A comparison of driving heat circuit and cooling circuit control is elaborated in [6].

2.3 Survey of systems

Within Subtask B of IEA Task 38, a survey was sent to experts responsible for designing and operating solar cooling installations. The survey contained 3 levels of contribution, from basic information, to more advanced details of, and experiences with, the control strategies employed. Table 1 lists the 11 SHDC systems whose designers and operators contributed an advanced level of detail in their experiences in design and control. Contributions varied in scope and detail of information, but the survey methodology included sections on general information, hydraulic scheme, control strategy, as well as general recommendations. These systems are a subset of a larger and more general survey within IEA Task 38 [7]. General conclusions are drawn in the areas of capacities and sizing, specific components chosen, and control strategies employed. Table 1 lists these systems, their locations, and their applications and chiller capacities.

Table 1 - Systems surveyed for control and configuration

City	Country	Lat	Long	Application	Application	Chiller Capacity
		-	-	-	-	kW
Perpignon[1]	France	42°42 N	2°53' E	Offices	Fan-coils	7.5
Rimsting[2]	Germany	47°53 N	12°20 E	Research		15
Rohrbach	Austria	48°34 N	13°59 E	Offices	Fan-coils	30
La Reunion	France	21°20 S	55°28 E	School	Fan-coils	30
Valladolid	Spain	41°32 N	4°45 W	Offices	Radiant floor	35
Gleisdorf[3]	Austria	47°06 N	15°42 E	Offices	Fan-coils	35
Sophia-Antipolis	France	43°37 N	7°33 E	Offices		35
Karlsbad	Germany	48°54 N	8°30 E	Offices	Fan-coils	54
Oberhausen	Germany	51°31 N	6°51 E	Offices	Fan-coils	58
Antalya	Turkey	36°53 N	30°42 E	Supermarket	Fan-coils	233
Bolzano	Italy	46°30 N	11°20 E	Offices		300

Notes

1 - Adsorption chiller
2 - Many various configurations possible
3 - Dual SDEC / SHDC installation

3 Results

Based on the completed survey contributions from the systems in Table 1, many various configurations were observed. Table 2 below summarizes the main aspects of configuration for these 11 systems based on the submitted hydraulic schemes and descriptions.

Table 2 - Survey of system configuration

City	Solar Field		Aux. Heating		Heat store	Aux. Cooling		Cold store	Heat Rejection
	m²	Type	kW	Type	m³	kW	Type	m³	-
Perpignon	25	FP			0.3			0.3	Closed
Rimsting	71	ETC / FP		Gas	2.0			1	Open / Closed
Rohrbach	120	FP	220	HP / Gas	8.0	100	VC	0.5	Open
La Reunion	90	FP2			1.5			1	Open
Valladolid	77.5	ETC / FP	235	Gas	8.0			1	Closed
Gleisdorf	302	FP			4.6		DEC	1	Open
Sophia-Antipolis	90	CPC / ETC	50	HP	0.3	50	HP	0.5	Open
Karlsbad	196	CPC			2.0			2	Closed
Oberhausen	108	ETC			6.7		VC	1.5	Open
Antalya	432	CPC			8.0		VC		Open
Bolzano	424	ETC	1031	Gas / Cogen	10.0	632	VC	5	Open

A description of the above configurations follows.

Solar field type

FP	Flat-plate solar thermal collector, single or double glazing
ETC	Evacuated tube collector
CPC	Concentrating parabolic collector

Auxiliary heating types

Gas	Natural gas boiler
HP	Heat pump
Cogen	Waste heat from electricity cogeneration unit

Auxilliary cooling types

VC	Vapor compression chiller
DEC	Solar desiccant cooling system
HP	Heat pump

Heat rejection types

Open	Open wet cooling tower
Closed	Closed wet cooling tower

In order to compare the systems, metrics based on the collector area and storage volumes, each divided by the chilled water capacity, can be employed. Therefore, the collector-capacity ratio is defined as the collector area divided by the chiller capacity. The collector-capacity ratio varied between 1 and 9 m^2/kW_{cold}. At the high capacity end, (> 200 kW cooling), the two systems show a relatively low ratio, near 2. All small systems vary across the entire range. However, the different collector field types must be considered in this metric, where a concentrating or evacuated tube collector will have an increased performance compared to a flat-plate collector. Similarly, the heat storage ratio and cold storage ratio vary between 0.01 to 0.25 m^3/kW_{cold} and 0.01 to 0.07 m^3/kW_{cold} respectively. Once again, the large scale systems show similar ratios at the lower range, with the smaller systems showing no trend.

Table 3 summarizes the control aspects related to each of the 11 surveyed systems, with a detailed description of each strategy following.

Table 3 - Survey of system control

City	System design	Solar start	Solar control		Chiller control	
Perpignon	Auton	Radiation, Diff. Temp.	On-Off	Diff. Temp	On-Off	Driving / Cooling temp
Rimsting	Assist		On-Off	Diff. Temp	Cooling fan	Driving / Cooling temp
Rohrbach	Assist	Diff. Temp	On-Off	Diff. Temp	Gen. mix	Driving Temp
La Reunion	Auton	Radiation	On-Off	Diff. Temp	On-Off	Driving Temp
Valladolid	Assist	Radiation	Proportional	Radiation	Gen. mix	Driving Temp
Gleisdorf	Auton		On-Off		Gen. mix	Driving Temp
Sophia-Antipolis	Assist		On-Off	Radiation	On-Off	Driving Temp
Karlsbad	Auton		On-Off		Gen. mix	Driving Temp
Oberhausen	Assist	Radiation	Proportional		Gen. mix	Driving Temp
Antalya	Assist		On-Off		Gen. mix	Driving Temp
Bolzano	Assist	Radiation	On-Off		Gen. mix	Driving Temp

System design

Auton	Autonomous - No back-up for heating or chilling.
Assist	Assisted - No backup present, or back-up present but not

	used or planned on being used.
Solar Start	
Diff. Temp.	The primary solar circuit circulation pump is activated in response to a set differential temperature.
Radiation	The instantaneous radiation on the collector field is measured and used to switch the primary circuit on and off.
Solar control	
On-Off	The simplest method of solar circuit temperature control, measure difference between tank and collector temperatures, and turn on the solar circuit circulation pump in response to a set differential temperature. Hysteresis is used to limit cycling.
Proportional	A more advanced control strategy taking advantage of a variable speed pump. The instantaneous temperature difference between the storage tank and collector field proportionally controls the mass flow rate of the solar heat circuit.
Diff. Temp	A temperature differential between tank and collector field is used to regulate the primary pump.
Radiation	The instantaneous irradiation on the collector field is measured and used to proportionally control the mass flow of the solar circuit. Stable temperature is main advantage. Empirical correlation between irradiation and mass flow is required.
Chiller Control	
On-Off	Simple on or off control of chiller depending on conditions.
Gen mix	Three way valve on generator for driving heat regulation.
Cooling fan	Vary the cooling temperature using the cooling tower fan.
Driving Temp	Control the generator temperature using a mixing valve or pump control.
Cooling temperature	Vary the cooling capacity using a variable speed fan on the cooling tower to control heat rejection temperature.

Solar circuit (primary loop) control was typically on-off with hysteresis. The secondary loop, which provides hot water from the storage tank to the absorption chiller, was controlled most commonly by a three way mixing valve and constant speed pump. Some systems have an on-off control only for regulating the hot water temperature. Two systems have the capability to use the cooling tower fan speed to regulate the cooling water and consequently the chilled water temperature. In the case of the Rimsting installation, a research installation with many possible control modes, a recommendation was made to use cooling tower fan speed regulation to improve system performance.

4 Conclusions

In a previous IEA TASK, IEA SHC Task 25, basic control strategies of SHDC systems were documented and described [3]. Nevertheless, a comprehensive documentation of practical experience and results in the design and operation of SHDC systems with applied control strategies was missing. Thus the activities of TASK 38 aim mainly to;

- Provide a comprehensive overview on applied SHDC system control
- Document and assess the experiences in the different demonstration projects
- Summarize important control issues as a result of this comprehensive work

This paper contains a selection and compilation of 11 case studies of applied control strategies of SHDC systems. Based on the Task 38 experts, practical experience regarding the applied control strategies and some selected general rules are stated which can guide a successful SHDC system design and operation.

Each SHDC system is more or less unique and only a few SHDC system providers offer comprehensive and recommended control strategies. The development of standard control strategies for the wide range of different SHDC system configurations is still a research task by many. Therefore, the effort required in order to develop and to implement a suitable control strategy of a SHDC system can be significant because there is no existing standard practice. Furthermore, it is obvious that the complexity of control strategy depends strongly on the complexity of the hydraulic scheme. Standard SHDC system configurations have been described, but local conditions and restrictions limit their applicability [3]. The control strategy can only be developed if the SHDC system configuration is decided and the comfort and system requirements (boundary conditions) are well determined.

Solar-assisted air-conditioning systems include a variety of components which need to be controlled efficiently in order to cover the comfort requirements and to meet expectations on the overall system efficiency. However, the implementation of control components such as actuators, sensors, and control system, increases the system costs and therefore has a significant influence on the economics of the system. Therefore the control devices should be selected in order to meet the highest possible system efficiency (primary energy saved, electric efficiency, etc.) with the lowest possible control effort (complexity, cost). In order to monitor the energy performance of a SHDC system it is highly recommended to implement the required hardware and software equipment. Normally the sensor equipment and measured data points of the implemented control system do not fulfil the requirement of an appropriate assessment of the energy SHDC system performance. This field equipment should be extended to meet a minimal monitoring level [1].

In existing control systems there are often different options to meet the same control task, but with different impact on the overall system performance. For example, the control options to manage the chilled water temperature strongly depend on the hydraulic system configuration and on the type of distribution system. Moreover, the whole control strategy should be very different if the system is mainly working at part load or full load conditions.

If a SHDC system with a comparatively low nominal COP is employed and a fossil-fuelled heat source is used as the back-up, a high solar fraction is necessary in order to achieve significant primary energy savings. An appropriate design of the solar system, i.e., suitable dimensioning of the solar collector and system-integrated energy storage, is necessary for this purpose. The control strategy should enable to use of solar heat as much as possible. Especially for solar-assisted air-conditioning with additional heat sources like cogeneration units, district heating, and boilers, the control priority should be using solar heat followed by the auxilliary heat sources.

Finally, a key target of the control strategy of a SHDC system is to minimise the primary energy use per produced kW of chilled water or conditioned air. That means driving the implemented energy consumers (pumps, ventilators, solar collector, chiller etc.) of the overall system at its minimal energy consuming operation point. Consequently, the control strategy has to be flexible and adaptable in order to optimise the overall SHDC system. The control system and software should enable easy adaptation of control algorithms, parameters and set values. In many of the above presented case studies, the applied control software and hardware is more or less open source.

Given the above discussion of the complexity of control and configuration based on practical experiences from 11 SHDC systems, it is clear that a comparison between all systems is difficult both in that each system is unique, and that there are not enough systems, or active contributions from experts managing the systems. Therefore, it is difficult to make a statistically important statement about an optimum configuration or control. This conclusion leads to the recommendation to first clearly define boundary conditions and objectives for success in the system design and operation, and to use computer simulation tools and current expert advice to achieve a successful SHDC system.

5 Acknowledgements

The authors wish to acknowledge the contributions received from the IEA Task 38 community, in particular those who submitted completed surveys for the 11 systems under consideration in this paper.

6 References

[1] International Energy Agency Solar Heating and Cooling Programme. Task 38 - Solar Air-Conditioning and Refrigeration. URL: http://www.iea-shc.org/task38/index.html. Accessed Jan 2010.

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[3] IEA SHC Task 25. *Solar-Assisted Air-Conditioning - A Handbook for Planners*. Springer-Verlag/Wien, 2004.

[4] Volker Clauß, Annett Kühn, and Felix Ziegler. A new control strategy for solar driven absorption chillers. In *2nd International Conference on Solar Air-Conditioning*, 2007.

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[6] Paul Kohlenbach. *Solar cooling with absorption chillers: Control strategies and transient chiller performance*. PhD thesis, Technischen Universitat Berlin, 2006.

[7] W. Sparber, A. Napolitano. *Practical experience on design and control from worldwide documented solar cooling systems*. Abstract submitted to EUROSUN 2010 conference.

ANHANG AIT 5

Agenda of the 1st International Conference ‚Sustainable Cooling Systems'

TECHbase Vienna, March 31st till April 1st 2008, Vienna/ Austria

Federal Ministry
for Transport,
Innovation and Technology

Programme – International Conference

Sustainable cooling systems

March 31st 2008
Solar cooling
TECHbase Vienna
Giefingasse 2, 1210 Vienna

April 1st 2008
Cooling with district heating
Media Tower
Taborstraße 1-3, 1020 Vienna

arsenal research
A Company of Austrian Research Centers

2050

e2050

The market on air-conditioning and cooling is anticipated to grow exponentially in the next decades, as the world-wide demand for building air-conditioning will definitely increase, also in Europe. It will thus be of utmost importance to develop high performance air conditioning and cooling systems based on renewable energy sources to counteract the significant rise in the use of fossil fuel and the associated climate change. Heating energy from thermal solar collectors and biomass-district heating systems can be used for thermally driven air-conditioning and cooling systems and have the potential to cover a large share of the growing cooling demand.

Solar-assisted cooling for buildings is close to a world wide market introduction. In the next few years a substantial cost reduction of this promising technology can be achieved by internationally coordinated Research & Development activities as well as the specific support of a broad market introduction. On the first conference day – **Solar-Assisted Cooling** – international experts and relevant market actors will present ongoing Research & Development activities, products currently available on the market and the potential of cost reduction and action plans to support a wider implementation of solar-assisted cooling systems in buildings.

Recently, the interest of district heating companies to use the surplus heating energy of the district heating grid during the summer for cooling applications in buildings has increased. On the second conference day – **Cooling with District Heating** – practical experience in design, operation and technical potential of such sustainable thermally driven cooling systems in district heating supplied regions will be presented. Furthermore solutions to overcome technical and economical barriers will be discussed.

Solar cooling

March 31st 2008
TECHbase Vienna
Giefingasse 2, 1210 Vienna

Cooling with district heating

April 1st 2008
Media Tower
Taborstraße 1-3, 1020 Vienna

www.e2050.at

Conference Programme

Monday, March 31st 2008

International Conference

Solar Cooling

Conference language is English with simultaneous translation in German

TECHbase Vienna
Giefinggasse 2
1210 Vienna

09:00 REGISTRATION

09:30 Welcome and Opening
Brigitte Bach, arsenal research
State Secretary Christa Kranzl, Federal Ministry for Transport, Innovation and Technology (requested)

Chair: Brigitte Bach, arsenal research

Solar cooling initiatives

10:00 ESTTP - European Solar Thermal Technology Platform "Strategic Research Agenda focusing Solar Cooling Technology"
Werner Weiss, AEE INTEC

10:15 Austrian Solar Thermal Technology Platform (ASTTP) and the Special Austrian Solar Cooling Programme
Erich Podesser, ASTTP

10:30 Introduction to Task 38 "Solar Air-Conditioning and Refrigeration" of the IEA Solar Heating & Cooling Programme and the topic of solar cooling
Hans-Martin Henning, Fraunhofer ISE

11:00 COFFEE BREAK

Solar cooling technologies

11:30 Overview of solar cooling technologies
Marcello Aprile, Politecnico di Milano

12:00 Existing solar cooling installations in Europe
Wolfram Sparber, EURAC

12:30 Experience in planning, concepts and operation
Tim Selke, arsenal research

13:00 LUNCH

Chair: Michael Hübner, Federal Ministry for Transport, Innovation and Technology

Market and economics of solar cooling systems

14:00 Economic and technical potential across Europe
Laura Sisó, AIGUASOL

14:30 **Targeting prices for next generation of solar cooling systems**
Reinhard Ungerböck, CONNESS

15:00 **Strategies for market penetration**
Anita Preisler, arsenal research

15:30 **COFFEE BREAK**

Future developments

16:00 **New concepts and promising technologies**
Uli Jakob, SolarNext AG

16:30 **Barriers, technology gaps and solutions**
Amandine Le Denn, TESCOL SA

17:00 **R&D activities on components and examples**
Dong-Seon Kim, arsenal research

17:30 **Conclusions**
Brigitte Bach, arsenal research

Tuesday, April 1st 2008

Workshop of the programme "Energy systems of tomorrow"

Cooling with district heating

MediaTower Wien
Taborstraße 1-3
1020 Vienna

conference language is German with simultaneous translation in English

09:00 **REGISTRATION**

09:30 **Welcome**
Brigitte Bach, arsenal research
Michael Hübner, Federal Ministry for Transport, Innovation and Technology

Chair: Theodor Zillner, Federal Ministry for Transport, Innovation and Technology

Cooling with small rural district heating networks: drivers and challenges

10:00 **Integration of thermal driven chillers in district heating networks: impacts on the network, case study Mureck**
Olivier Pol, arsenal research

10:30 **Cost effectiveness of the integration of absorption chillers in district heating networks, case study Mureck**
Reinhard Ungerböck, CONNESS

11:00 **COFFEE BREAK**

11:30 **Practical experience: absorption chiller in Güssing**
Richard Zweiler, reNet GmbH

12:00 **Integration of an absorption chiller in a biomass supported district heating network – technical and economic framework**
Ursula Eicker, zafh.net

12:30 **LUNCH**

13:30 **District heating networks in Austria: Potential, chances and risks from the point of view of a district heating operator**
Josef Füreder, Energie AG

14:00 **Small capacity thermal driven chillers – current developments, research topics and integration in district heating networks**
Werner Pink, Pink GmbH
Martin Stocker, Danfoss-Nopro GmbH

14:30 **COFFEE BREAK**

Chair: Brigitte Bach, arsenal research

Cooling with large urban district heating networks: drivers and challenges

15:00 **Combined heat, power and cooling plant in Fussach**
Alfred Hammerschmid, BIOS Bioenergiesysteme GmbH

15:30 **Potential for district cooling in large district heating networks, case study Vienna**
Adolf Penthor, Fernwärme Wien

Panel discussion: barriers, need of research and solutions

Introduction: Ivan Malenkovic, arsenal research
Chair: Ute Woltron, DER STANDARD

Josef Füreder, CEO Energie AG Oberösterreich Wärme GmbH
Alfred Hammerschmid, senior manager BIOS Energiesysteme GmbH
Ivan Malenkovic, arsenal research
Werner Pink, research & development Pink GmbH
Erich Podesser, consultant
Eberhard Reil, authorised representative Fernwärme Wien GmbH
Richard Zweiler, CEO reNet GmbH

17:30 **Conclusion and outlook**
Olivier Pol, arsenal research

Information and Registration:

Conference fee for both days:
EUR 400,– excl. VAT (until March 3rd 2008)
EUR 500,– excl. VAT (from March 3rd 2008)

Conference fee for one day (March 31st or April 1st 2008)
EUR 240,– excl. VAT (until March 3rd 2008)
EUR 300,– excl. VAT (from March 3rd 2008)

Registration and further conference information: www.arsenal.ac.at/cooling

Conference partner:

ENERGIE 2050 - An Initiative by the BMVIT

Austrian Ministry for Transport, Innovation and Technology BMVIT
Division of Energy- and Environmental Technologies
A-1010 Vienna, Austria, Renngasse 5
Head of Division: DI Michael Paula

www.e2050.at

ASiC – Austria Solar Innovation Center

ANHANG ASIC 1

H. Focke, A. Preisler, G. Geissegger „Designing of a Technology- Roadmap for Solar Assisted Air Conditioning in Austria“, 3rd Int. Conference Solar Air Conditioning, Palermo, Italy 09.2009

Designing of a Technology- Roadmap for Solar Assisted Air Conditioning in Austria

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Abstract

Aim of the project is the development of a technology roadmap for Solar Assisted Air Conditioning in Austria involving the relevant market player. The technology development should be described in short term, medium term, and long term goals as well as the connected market relevance for Austria and the economic development of this technology. The roadmap contains a survey of the initial position and a list of measures to be taken, developed by means of expert workshops and personal interviews. The project is currently in progress, but it figured out, that the classification of systems and applications and the allocated subsidies for the occurring differential costs play a very important role. Linked activities concerning solar thermal collectors, operating substances - cycles and heat rejection devices are the most necessary measures for the future of solar thermal cooling in Austria.

Introduction

Worldwide the cooling and Air Conditioning demand of buildings becomes more and more important. A current study[1] is predicting a dramatical increase on energy demand for cooling and Air Conditioning of buildings also for Austria: The electricity consumption for Air Conditioning in Austria was estimated by a so called bottom- up model. This model is based on the Air Conditioning equipment stock in the year 2030 on the one hand, and on the other hand on the specific energy consumption for active Air Conditioning. The model results in values of 970 GWh for the year 2020 and 1.875 GWh for the year 2030. Furthermore it was assumed that 30% of the above mentioned values for the year 2030 can be substituted by heat pumps with earth cold and further 30% can be substituted by solar cooling. Moreover, the higher electricity demand for cooling of buildings causes summer peaks, which may lead to high electricity prices and grid connected problems as black- outs. To work against this development on the one hand the cooling demand of buildings must be kept as low as possible, on the other hand the remaining cooling demand must be covered with alternative, environmentally friendly cooling technologies.

Aim of the project is the development of a technology roadmap for solar thermal cooling in Austria involving the relevant market player. The main content of the technology roadmap are the compilation of the actual position, identification of market potential, technology development and the necessary measures for it.

The project is carried out by a consortium consisting of Austria Solar Innovation Center – ASiC, Austrian Institute of Technology - AIT and Austrian Energy Agency - AEA.

ASIC is a non profit organisation, financially supported by the local government of upper Austria, the municipality of Wels and local companies. The R&D institute executes measurement and monitoring tasks in the fields of solar thermal systems and various research activities on experimental plants. ASiC covers the measuring and monitoring of solar thermal components and is accredited test center for the measurements on solar thermal collectors according to ÖNORM EN 12975-2.

The Austrian Institute of Technology (AIT), former arsenal research, division Sustainable Thermal Energy Systems, provides high-quality research services in the field of thermal energy systems. The strategic focus is on systems for providing thermal energy for the built environment, with the emphasis on solar thermal systems as well as thermal heat pump and ventilation technology.

The Austrian Energy Agency (AEA) is a non profit energy research and policy institution. Its mission is to promote rational use of energy and stimulation of renewable energy sources and of innovative technologies.

The project was further carried out with close collaboration to AEE- Institute for Sustainable Technologies and Podesser Consulting.

Methods/ Approach

The question - why another roadmap for solar thermal applications - can be answered with the following issues:

- To define clear targets on the way to solar assisted Air Conditioning:
  Technology Development - what is the position of Austria?
  Economic Development – resulting market relevance for Austria
  Energy Supply – when is something noticeable?
- To create a position paper and therefore a direction sign for political decisions concerning energy and technology

It’s only possible to hit one’s target if the target is well known!

The technology development should be described in short term, medium term, and long term goals as well as the connected market relevance for Austria and the economic development of this technology. Scenarios for a useful interaction with other sustainable thermal cooling technologies like cooling with district heating will be analysed to clarify the future position of solar thermal cooling in the Austrian energy supply. The market player of this technology go from component producer (solar thermal collector, ab-/ adsorption cooling machines, ventilation components, storage, control, etc.) business enterprises (hotels, breweries, laundries, supermarkets, etc.), building developer and consultancy engineers to research institutions, energy agencies and political decision maker. All of these groups will be involved in the development of the technology roadmap by expert workshops and interviews.

Figure 1 shows the work packages and the detailed milestones at a glance.

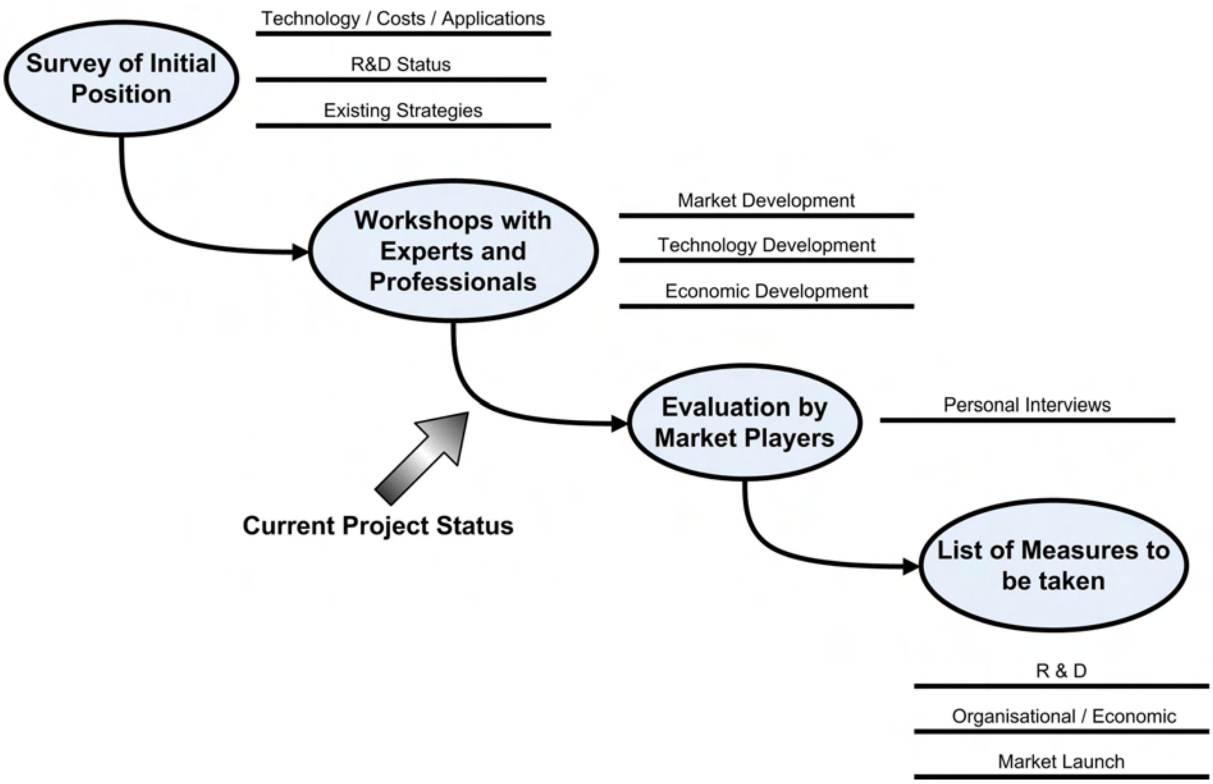

Figure 1: Work packages and milestones

Current results:

The active project work started in autumn 2008, managed by the Austrian Institute of Technology: The initial position concerning Technology which is commonly well known, and consist, in a general survey, of photovoltaic driven compression systems and solar thermal driven sorption systems (Absorption, Adsorption, DEC and Liquid DEC). The solar thermal driven sorption systems are mostly driven by flat plate collectors in connection with hot- or cold water storages. It should be pointed out that the actual costs in the work package "Survey of initial position" are taken from the EU- Project ROCOCO[2]: The investment costs, relating to electically driven compression systems vary from 300% to 625% for Absorption Systems, and 100% for DEC Sytems. The operating costs (20 years) vary from 25% to 210% for Absorption Systems, and 5% to 39% for DEC Sytems, also relating to electically driven compression systems. The current cost situation will be fundamental revised in the course of the Personal Interviews with the relevant market players in autumn 2009. Concerning applications the initial Position amounts 19 existing installations in Austria, mostly for supplying offices and seminar rooms, and mostly demonstration or pilot plants. Figure 2 shows the distribution of those installations regarding technology.

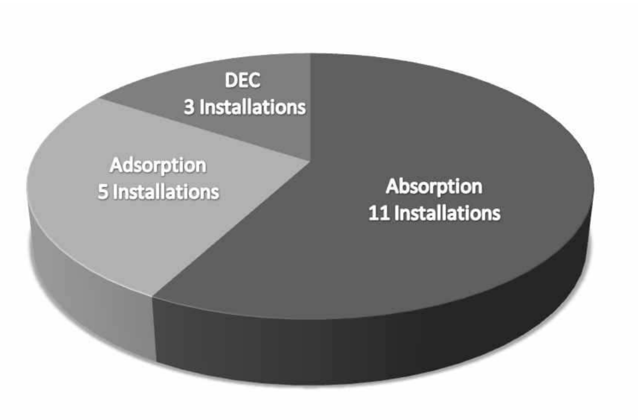

Figure 2: 19 existing installations in Austria

The already cited study[1] predicts a shift of only 9,5 % of the existing building stock in Austria. The dramatically increase on energy demand for cooling and Air Conditioning of buildings in Austria results in a huge potential for Solar Air Conditioning installations, especially for residential buildings. It should be mentioned, that the highest energy saving rations can be achieved with installations for solar thermal DHW preparation, solar assisted heating and Air Conditioning.

The energy consumption for Air Conditioning, known from southern Europe represents a new challenge for Austria. In some categories of service buildings (e.g. large office buildings, large hospitals etc.) air conditioning systems already exist also in Austria, but a broad market diffusion of the air conditioning units, also in residential buildings, is a new development.

The Initial Position relating to Existing Strategies is dominated by statistics concerning the solar thermal market in Austria: Compared to other countries Austria has a very diversified market: The installations which are currently under operation consist of 62% domestic hot water installations in single family houses, 26% installations for domestic hot water and space heating, plus 9% installations on multi-family houses respectively with district heating connection. About 1% of the cumulative solar thermal collector surface is installed in commercial and industrial applications. The installed solar thermal collector surface during the year 2007 consists of 47% installations for domestic hot water and 53% installations for domestic hot water and space heating.

Furthermore the technology Solar Air Conditioning has to be developed and deployed besides other new applications like active solar building, the active solar renovation, industrial applications up to 250 ℃ an d solar heat for district heating[3].

Before organising the expert workshops the project team agreed to define Central Questions for the categories Technology, Market Development and Subsidies.

Workshop 1 happened in February 2009 with 25 participants and the main ambition to define a target figure referring to each Central Question. Workshop 2, happened in May 2009 with 20 participants and the intention to complete details, to discuss cost impacts and to identify the position of the participating companies.

Figure 3 shows the defined Central Questions, the developed Objectives, the Measures and the expected Cost Impacts, which where defined in the past 2 expert workshops.

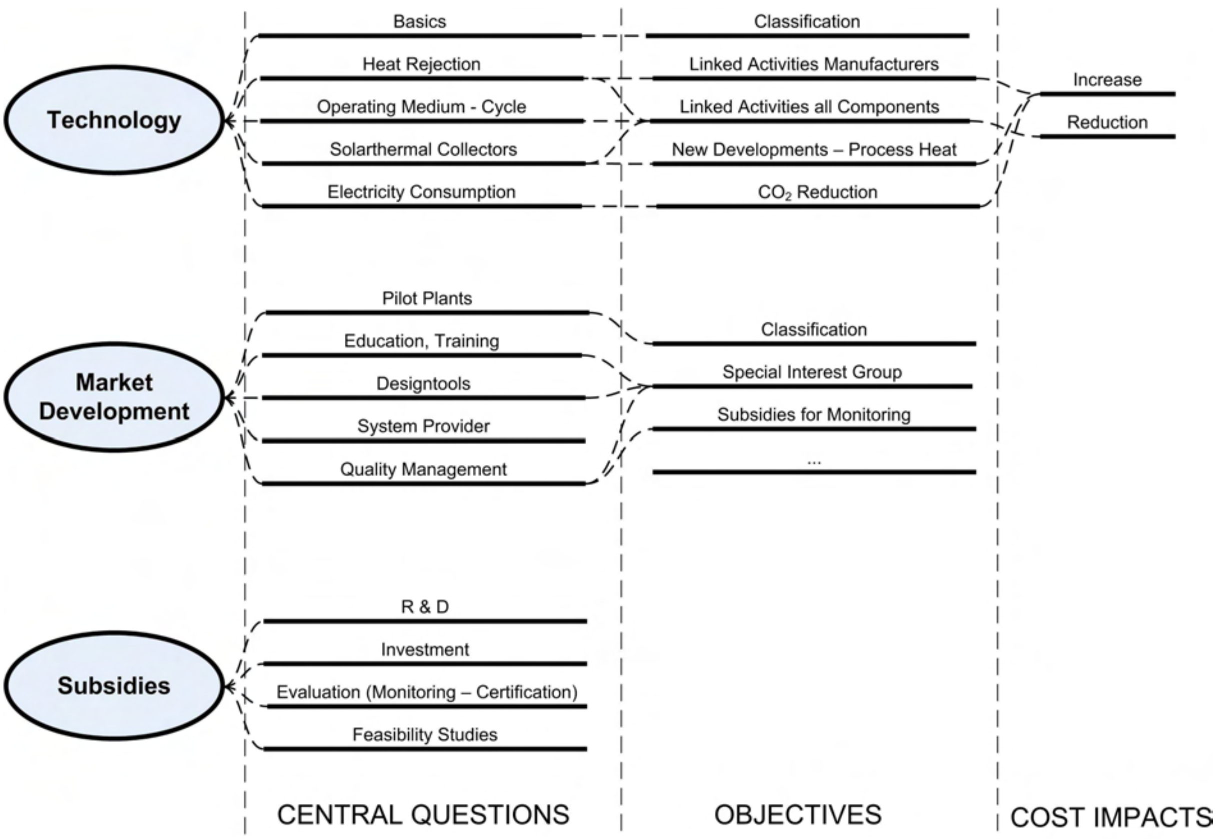

Figure 3: Current status relating to deliverables

Conclusions and Outlook

Currently there are hardly any comparable strategies for market penetration of this new technology available from other countries, but it can be noticed that more and more component producer of different branches change their product portfolio to products of the renewable energy sector.

It seems that one of the most important measures on the way to solar assisted air-conditioning is the Classification of systems and applications. Because of relatively low electricity prices the allocated Subsidies for the occurring differential costs play a very important role. The main target for these subsidies, on the other hand, is the significant reduction of CO_2 emissions. There is a huge potential for Solar Air Conditioning installations, especially for residential buildings, but in case of increasing prices for electricity the sector commercial Solar Assisted Cooling is the most promising future market in Austria. The technology roadmap outlines the possible short term, medium term and long term development of the Austrian market with the necessary measures for solar thermal cooling.

The work on the Roadmap should be finished by a publication in the beginning of 2010.

Acknowledgment

The project is financially supported by the Resarch- and Technologyprogram NEUE ENERGIEN 2020 from the Austrian Klima- und Energiefonds. The program is managed by the Austrian Research Promotion Agency (FFG) which is the national funding institution for applied industrial research in Austria.

References

[1] Haas et al, Wärme und Kälte aus Erneuerbaren 2030, Dachverband Energie-Klima, Wien 10/2007

[2] A. Preisler et al, EU-Projekt ROCOCO, Reduction of Costs of Solar Cooling Systems, TREN/05/FP6EN/S07.548855/020094, Brüssel, 3/2008

[3] European Solar Thermal Technology Platform (ESTTP), Solar Thermal Vision 2030, Brussels, 05/2006

ANHANG ASIC 2

H. Focke, G. Steinmaurer: "Cooling load demand assessment - a key issue for economic operation of solar cooling systems", 2nd International Conference on Solar Air-Conditioning, , Tarragona, Spain 10.2007

Cooling load demand assessment – a key issue for economic operation of solar cooling systems

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Abstract

Already existing buildings mostly do not correspond to the requirements for solar cooling technologies. The decision for solar cooling applications is often done on the basis of simulation tools. But the main challenge in the use of such simulation tools is the correct description of the internal loads, thus the description of the behaviour of human user of the building. An alternative procedure is the data collection of heating and cooling loads and environmental conditions during a selected period. The monitored data can result in a simulation with realistic cooling load characteristics to assess the economic efficiency of a solar cooling system.

In the presented project an application (mold design and manufacturing plant) will be analysed. The poster shows the results of the data monitoring for the cooling loads, the heating loads and the environmental conditions. Moreover a feasibility study, the comparison with common technologies, and the dimensioning of the solar cooling plant can be carried out based on the monitoring results and results in a decision-making-model for economic-efficient operation.

The presented results of monitoring and simulation show a vast simultaneity of solar irradiance and cooling loads. It turns out, that the huge water basin is a central component for a competitive solar cooling system. The results on this certain plant can be transferred to similar applications.

Introduction

The project has been carried out by “Austria Solar Innovation Center – AsiC”

ASIC is a non profit organisation, financially supported by the local government of upper Austria, the municipality of Wels and local companies. This R&D institute executes measurement and monitoring tasks in the fields of solar thermal systems and various research activities on experimental plants. ASiC covers the measuring and monitoring of solar thermal components and the detailed measurement of solar thermal collectors.

The considered plant is located in the company WIHO Hofbauer GmbH (Figure 1). This company acts as an expert in toolmaking, mold and die production as well as toll-manufacturing. Therefore more than 20 machines like machining centers, lathes, grinding machines and spark machines are located in a workshop on the scale of 2.500 m².

Figure 1: Building of company WIHO Hofbauer

Increasing cooling loads

Figure 2 shows the existing scheme for the heating and cooling supply of the plant. The system consists of 3 parts with cooling demand: an office with 250 m² (air conditioned with a fan coil system) and a factory workshop with 2.500m². Additionally a machinery with metal cutting machine tools and spark machines has to be chilled with constant low-temperature feed water. Now steadily increasing cooling demand makes it necessary to replace the old cooling unit by a greater one.
The cooling loads, mentioned in Figure 2, derive from state-of-the-art dimensioning, and yields typically oversized standard electric driven chiller. The cooling load is currently covered by an aligned 37 kW chiller using a 110m³ water basin. The hot water requirements, mainly for the showers, are rather marginal.

The Industrial spark machines need water quenches of constant 14˚C temperature for accurate operation. The cold water consumption for the office is negligible, but during hot periods in summer the interior air conditions in the workshop often become uncomfortable. The extension of the plant causes increasing cooling loads for the spark machines mainly during summertime, which could not be covered with the installed electric driven water chiller. Rising expenses for electric energy provoced the fundamental idea of the current project: to compare the economic efficiency of a standard electric driven chiller with an environmentally friendly solar driven chiller on the basis of the actual cooling demand.

Data monitoring

From summer 2006 to summer 2007 the meteorological data, the cooling loads and the consumption of electric energy have been acquired by a data monitoring system.

Additionally the heat consumption for hot water production and heating was monitored.

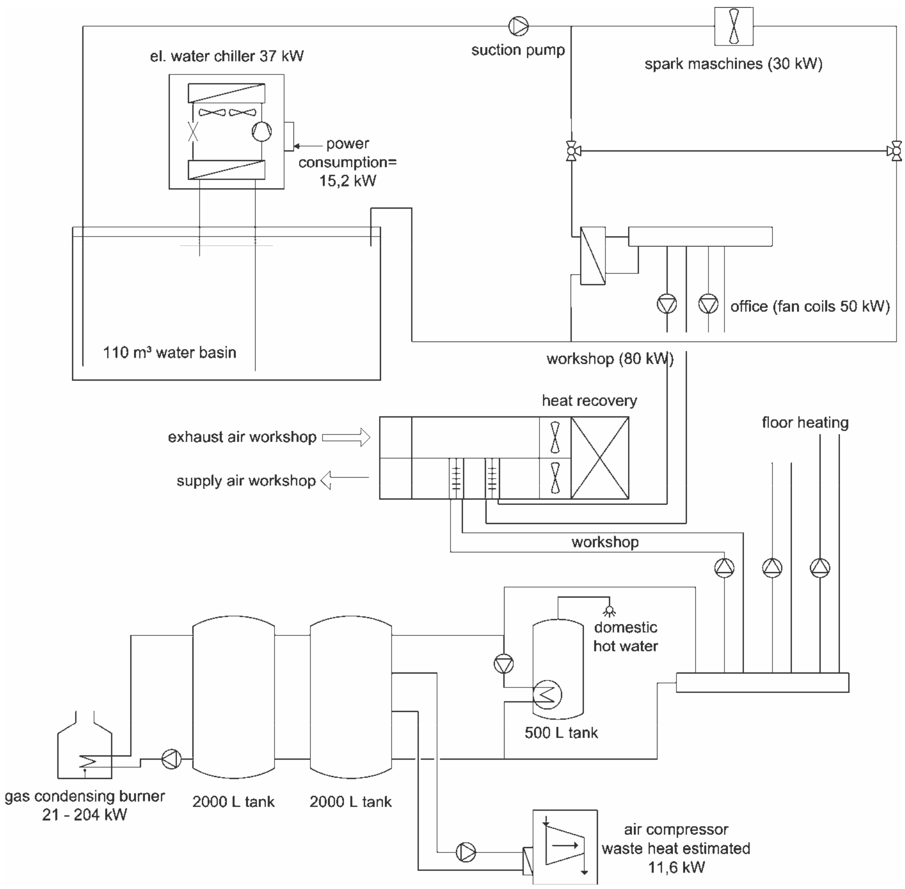

Figure 2: Hydraulic scheme of the plant

On the basis or the recorded data, a continuative feasibility study will demonstrate, if solar (assisted) cooling can be used efficiently. The following steps are simulation, designing and construction, followed by commissioning and monitoring of the realised solar cooling plant.

Figure 3 shows exemplarily the history of irradiance, ambient temperature and cooling loads during a period of hot weather in August 2006. It can obviously be seen in the graph, that there is a distinct correlation between the whole cooling load and the solar radiation during daytime, also a detailed statistical analysis of the monitored data yield to the same results.

The base load is mainly due to the amount of operating spark machines, and independent from the ambient conditions on a wide range. The peak load is caused principally by the factory workshop, and follows the course of solar radiation to a large extend. The temperature level seems to be suitable for a solar thermal driven water chillers. Furthermore numerous operating cycles (summer nights, wintertime and transit times) can be carried out by a wet cooling tower. It finally turns out, that the huge water basin is a central component for a competitive solar cooling system.

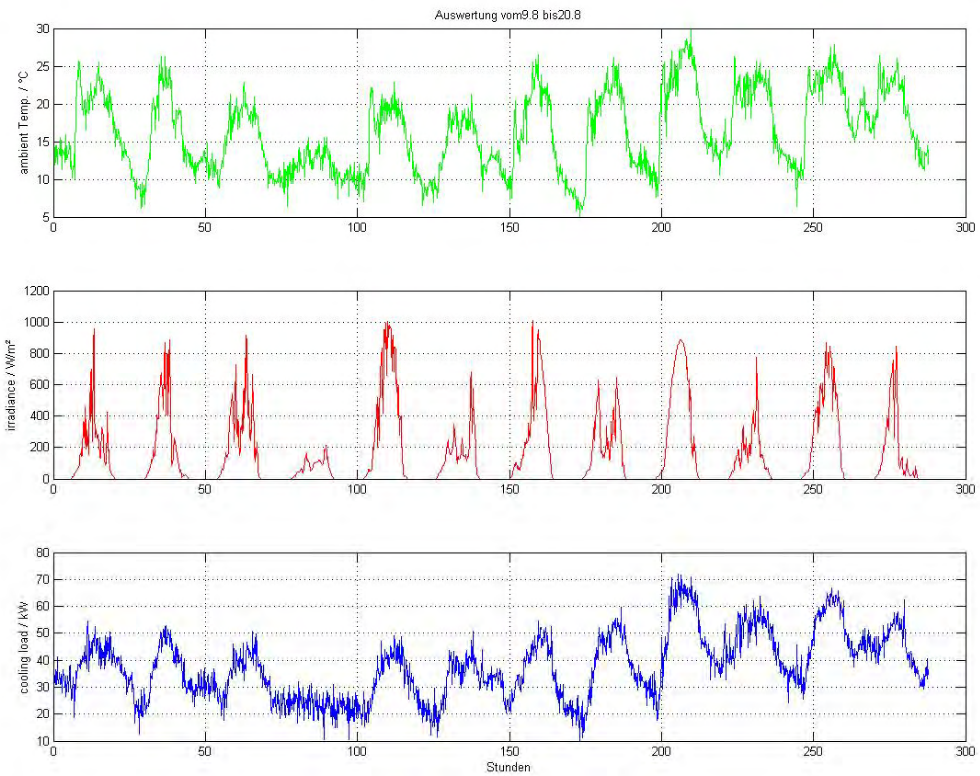

Figure 3: Sample data from the monitoring system

Simulation of the solar cooling plant

The simulation of the behaviour of the overall plant has been carried out with Matlab/Simulink ©. First results show a promising hybrid configuration with a 30 kW EAW SE 30 chiller and a 65 kW wet cooling tower.

The cooling tower is able to cover the cooling load over a period of more than half of the year. 30 % of the heat demand can be covered directly by solar energy.

Further details can be found on the poster as well as on www.asic.at.

Conclusions and Outlook

For integrating a solar assisted cooling system it is necessary to demonstrate the economic efficient operation of the overall plant. It can be shown, that in the considered example approx. 6.000€ for the cold production and 1.500€ for the heat generation can be saved by using a solarthermal plant.

It turns out, that a decision pro/con a solar assisted cooling system on the basis of economic efficiency can not be made in general. A realistic cooling load demand assessment is able to deliver a basis for decision-making.

Acknowledgment

The project is financially supported by the government of the Upper Austrian Energy-Technology-Programme ETP.

ANHANG ASIC 3

H. Focke: " Field report of a Solar Assisted Air Conditioning system in an office building located in Upper Austria", 2nd International Conference on Solar Air-Conditioning, ,Tarragona, Spain 10.2007

Field report of a Solar Assisted Air Conditioning system in an office building located in Upper Austria

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Abstract

The poster contains experiences from planning, realization and the operation of a solar assisted air conditioning system from a new office building owned by the regional government of Upper Austria. A solar thermal driven 30 kW Absorption chiller supports the electric driven main installation. The costs for the solar thermal cooling application are 9 times higher (per kW) than for the electric driven air conditioning unit, caused also by mistakes in the planning period. Problems with the control strategy can be seen in the monitoring data, which are not coherent available.

Objective

Within the Solar Heating and Cooling program, supported by the International Energy Agency (IEA) the Task 38 [1] is carried out aiming to promote solar assisted cooling systems. In order to define the state of the art of the existing large scale solar cooling systems, relevant data have been collected.

ASiC contributes with the monitoring data from an office building from the regional government of Upper Austria (Figure 1). The building was erected in the Year 2007, and settled in autumn 2008. The regional government of Upper Austria committed itself to sustainable energy production and use, by developing a comprehensive energy policy framework. So the government decided to implement the innovative technique solar thermal cooling in its own office building.

Figure 1: View of the office building

The collected monitoring data are conform to “Level 3” from the unified monitoring procedure, which was proposed within the IEA Task 38:

The three Levels are characterised by a progressive increase of accuracy of the monitoring leading to an increased number of sensors to be installed.
Level 1: Basic Information on primary energy, COP and costs
Level 2: Basic monitoring procedure (kept simple in sense of calculation and necessary monitoring hardware)
Level 3: Advanced monitoring procedure (more complex in sense of calculation and necessary monitoring hardware)

Facts

Location: Rohrbach, Upper Austria, Longitude: 13°5 9´, Latitude: 48°34´
Building use: Offices
Conditioned area: 2.860 m²

Solar collector field:
120 m² flat plate collectors
azimut: 0°, slope: 45°
2 heat storages, 4 m³ each one
Installation: Solar Combisystem for heating and cooling

The calculated heating and cooling loads are listed in Figure 2.

	Heating	Cooling
Load	182 kW (Floor)	39 kW (Fan Coils)
	35 kW (vent.)	69 kW (vent.)
Summary	217 kW	108 kW
Energy demand	110.000 kWh	70.000 kWh
	23 kWh/m²/a	14 kWh/m²/a

Figure 2: Heating and cooling loads

Thermally driven absorption chiller: 30 kW
Wet cooling tower
0,5 m³ cold storage
Electrically driven compression chiller: 100 kW
1,5 m³ cold storage
The solar thermal driven absorption chiller precools the supply of the electric driven chiller

30 kW heat pump (source: exhaust air)
2 gas boilers, 110 kW each one

The Monitoring system is included in the central building control system
Figure 3 shows the predicted solar yields for heating and cooling.

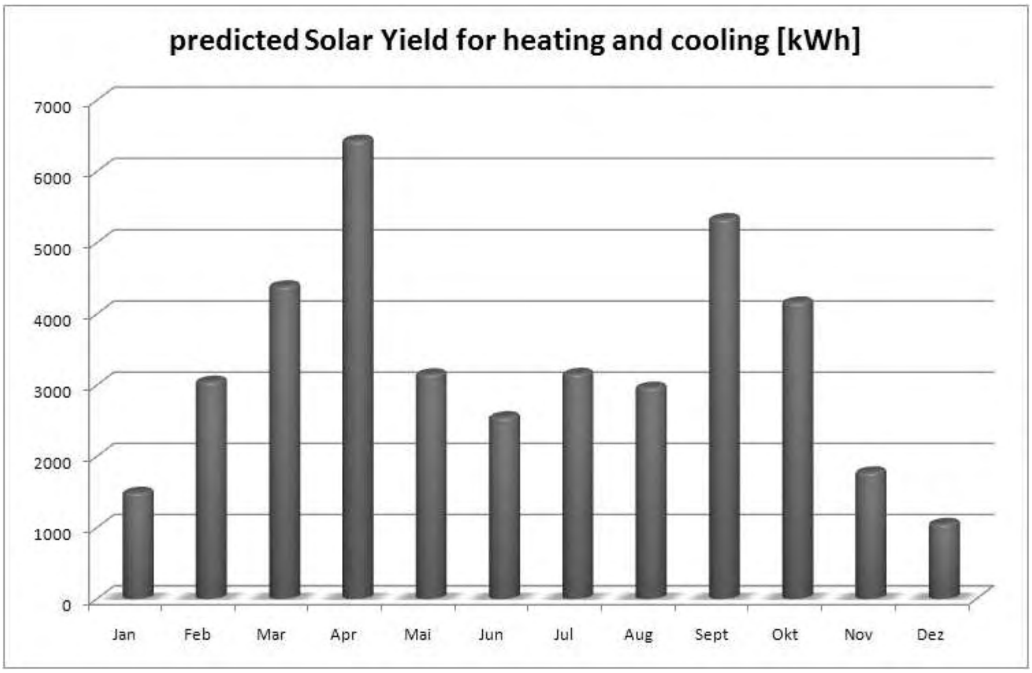

Figure 3: Predicted solar yields

Costs

30 kW solar driven cooling:
127.000 € excl. solar thermal installation, due to 4.233 €/kW
Figure 4 shows the detailed itemised costs for the solar thermal driven cooling installation.

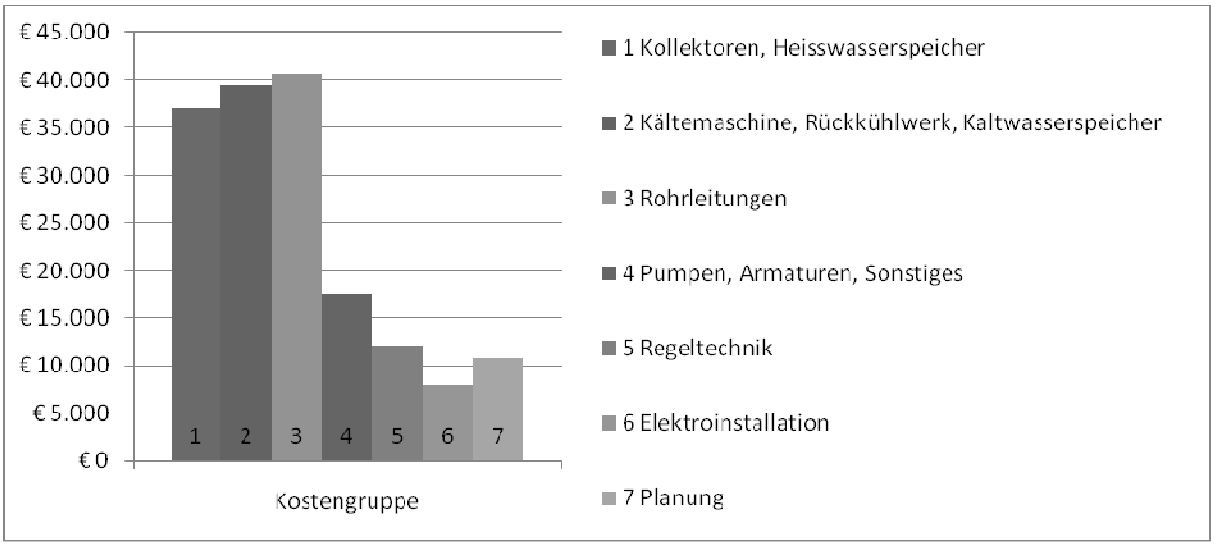

Figure 4: Costs for the solar thermal driven cooling installation

100 kW electric driven cooling:
47.000 €, due to 470 €/kW
Figure 5 shows the detailed itemised costs for the electric driven cooling application.

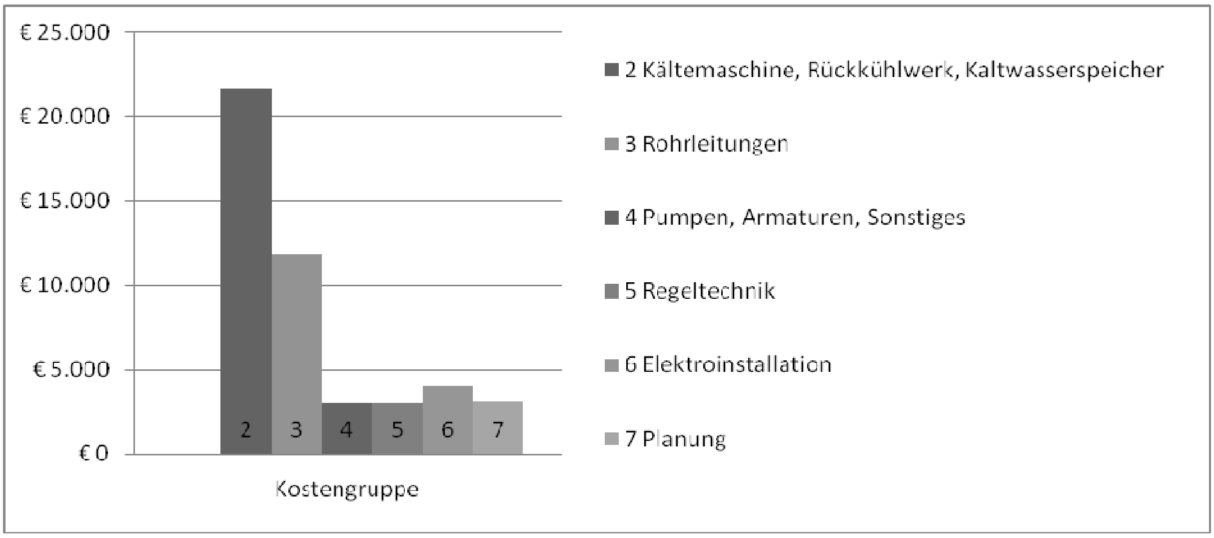

Figure 5: Costs for the electric driven cooling application

Results

The thermal driven chiller is located in the HVACR room in the cellar of the building. The position of wet cooling tower was chosen on the flat roof of the building (best practise for electric chillers): This results in high pressure losses in the piping and leads to a **overdesigned** pump and **extensive costs** for the heat rejection cycle.

The **COP measurement** of the electric driven chiller was realised supplementary (in order to make comparisons possible).

The **commissioning** was difficult and complicated, because the installation is not a pre-engineered system and there are no commissioning guidelines existing.

The **control strategy** is simple: Take what you can get from the sun for precooling, meet the demand with additional electric cooling.

The installation was finally put in operation in **September 2008**.
Because of several problems concerning control and the absorption chiller itself there are no coherent monitoring data for the summer 2009 available.

Conclusions

The key issues of further activity should contain:

- Electricity measurements for thermal driven chillers, pumps and fans (parasitic consumption)
- Sanitation of wet cooling towers (legionella)
- Water consumption of wet cooling towers
- Verification of Operation of thermal driven chillers in heat pump mode
- Definition of COP references for electric chillers (measurements under annual operation

In order to achieve these fundamental targets AIT, former arsenal research, ASiC and the Austrain Energy Agency established the Project “Roadmap for solarthermal cooling in Austria”.
Aim of the project is the development of a technology roadmap for solar thermal cooling in Austria involving the relevant market player. The main content of the technology roadmap are the compilation of the actual position, identification of market potential, technology development and the necessary measures for it. The technology development should be described in short term, medium term, and long term goals as well as the connected market relevance for Austria and the economic development of this technology [2].

References:
[1] www.iea-shc.org/task38
[2] www.klimafonds.gv.at

ANHANG ASIC 4

H. Focke: “Field report of a solar assisted air conditioning system in an office building located in Upper Austria”. WSED World Sustainable Energy Days, Wels, Austria 02.2009

DI Hilbert Focke, Dr Gerald Steinmaurer
Austria Solar Innovation Center Roseggerstraße 12 A- 4600 Wels Phone: +43 / 7242 / 9396 5563 **www.asic.at**

Field report of a solar assisted air conditioning system in an office building located in Upper Austria

A tremendous increase in the market for air- conditioning can be observed worldwide. Electrically driven chillers have reached a relatively high standard concerning energy consumption, but they still require a high amount of electricity and cause significant peak loads in electricity grids. It seems logical to apply solar energy for cooling purposes since in many applications, such as air conditioning, cooling loads and solar gains occur at more or less the same time.However, also in these sectors at least a coincidence between solar gains and load occurs at least on a seasonal level.
In general, solar assisted cooling can produce heat from solar radiation by solar thermal collector systems and employ thermally driven cooling processes. Thermally driven technology is of particular interest in case of applications where both cooling and heating is needed. In such cases a solar thermal collector can be used around the year for heating in winter and cooling in summer.

Objective:

Within the Solar Heating and Cooling Program, supported by the International Energy Agency (IEA) the Task 38[1] is carried out aiming to promote solar assisted cooling systems. In order to define the state of the art of the existing large scale solar cooling systems, relevant data have been collected.
ASiC contributes with the monitoring data from an office building from the regional government of Upper Austria.The buliding was built up in the year 2007, and settled in autumn 2008. The regional government of Upper Austria committed itself to sustainable energy production and use, by developing a comprehensive energy policy framework. So the government decided to implement the innovative technique solarthermal cooling in its own office building.

The collected monitoring data are conform to “Level 3” from the unified monitoring procedure, which was proposed within the IEA Task 38:

The three levels are characterised by a progressive increase of accuracy of the monitoring leading to an increased number of sensors to be installed.

Level 1:Basic Information on primary energy, COP and costs
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Level 3: Advanced monitoring procedure (more complex in sense of calculation and necessary monitoring hardware)

Facts:

- Location: Rohrbach, Upper Austria, Longitude: 13° 59´, Latitude: 48° 34´
- Building use: offices
- Conditioned area: 2.860 m²

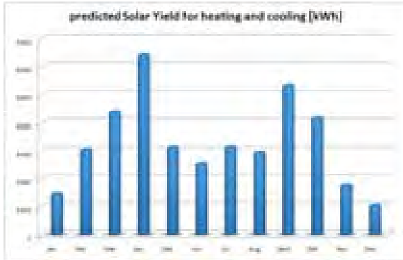
- Solar collector field: 120 m² flat plate collectors
- azimut: 0°, slope: 45°
- 2 heat storages, 4 m³ each one
- Installation: Solar Combisystem for heating and cooling

- calculated cooling load: 123 kW
- thermally driven absorption chiller: 30 kW
- wet cooling tower
- 0,5 m³ cold storage
- electrically driven compression chiller: 100 kW
- 1,5 m³ cold storage
- the absorption chiller precools the supply of the compression chiller
- Calculated heating load: 182 kW
- 30 kW heat pump (source: exhaust air)
- 2 gas boilers, 110 kW each one

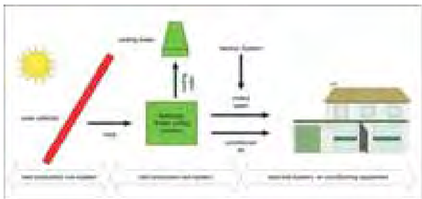
The Monitoring system is included in the central building control system.

Results:

The thermal driven chiller is located in the HVACR room in the cellar of the building. The position of wet cooling tower was choosen on the flat roof of the building (best practise for electric chillers):
➔ High pressure losses in the piping
➔ overdesigned pump for the recooling cycle



The COP measurement of the electric driven chiller was realised supplementary (in order to make comparisons possible).
The comissioning was difficult and complicated, because the installation is not a pre-engineered system and there are no comissioning guidelines existing.
The control strategy is simple: Take what you can get from the sun for precooling, meet the demand with additional electric cooling.
The installation was finally put in operation in September 2008.



Conclusions:

The key issues of further activity should contain:

- Electricity measurements of thermal driven chillers, pumps and fans (parasitic consumption)
- Sanitation of wet cooling towers (legionella)
- Water consumption of wet cooling towers
- Verification of operation of thermal driven chillers in heat pump mode
- Definition of COP references for electric chillers (measurements under annual operation)

In order to achieve these fundamental targets arsenal research, ASiC and the Austrain Energy Agency established the Project “Roadmap for solarthermal cooling in Austria”.
Aim of the project is the development of a technology roadmap for solar thermal cooling in Austria involving the relevant market player. The main content of the technology roadmap are the compilation of the actual position, identification of market potential, technology development and the necessary measures for it. The technology development should be described in short term, medium term, and long term goals as well as the connected market relevance for Austria and the economic development of this technology [2].

References:
[1] www.iea-shc.org/task38
[2] www.klimafonds.gv.at

Task 38
Solar Air-Conditioning and Refrigeration

ASiC
Austria Solar Innovation Center

ASiC is a non profit organisation, financially supported by the local government of upper Austria, the municipality of Wels and local companies. This R&D institute executes measurement and monitoring tasks in the fields of solar thermal systems and various research activities on experimental plants. ASiC covers the measuring and monitoring of solar thermal components and the detailed measurement of solar thermal collectors.

ANHANG ASIC 5

H. Focke: “Field report of a solar assisted air conditioning system located in Upper Austria”.

WSED World Sustainable Energy Days, Wels, Austria 02.2011

DI Hilbert Focke
Austria Solar Innovation Center Roseggerstraße 12 A- 4600 Wels Phone: +43 / 7242 / 9396 5563 **www.asic.at**

Field report of a Solar Assisted Air Conditioning system located in Upper Austria

The poster contains experiences from planning, realization and the operation of a solar assisted air conditioning system from a new office building owned by the regional government of Upper Austria. A solar thermal driven 30 kW Absorption chiller supports the electric driven main installation. The analysis of the monitored data delivers an average monthly electric COP of 3 and an average monthly thermal COP of 0.4. However the electric COPs concerning the electric backup compression chiller around the value 2.8 are also moderate.

Objective:

The presented installation is part of the project Solar Cooling Monitor, which is carried out in collaborative work of the research institutes AIT, ASIC, AEE INTEC, Univ. Innsbruck, and the company S.O.L.I.D.. The Project is funded by the Austrian Research Promotion Agency (FFG).

ASiC contributes with the monitoring data from an office building from the regional government of Upper Austria.The buliding was erected in the Year 2007, and settled in autumn 2008. The regional government of Upper Austria committed itself to sustainable energy production and use, by developing a comprehensive energy policy framework. So the government decided to implement the innovative technique solarthermal cooling in its own office building.

The collected monitoring data are conform to “Level 3” from the unified monitoring procedure, which was proposed within the IEA Task 38.

Facts:

- Location: Rohrbach, Upper Autria, Longitude: 13° 59´, Latitude: 48° 34´
- Building use: offices
- Conditioned area: 2.860 m²

- Solar collector field: 120 m² flat plate collectors
- azimut: 0°, slope: 45°
- 2 heat storages, 4 m³ each one
- Installation: Solar Combisystem for heating and cooling

- calculated cooling load:

	Heating	Cooling
Load	182 kW (Floor)	39 kW (Fan Coils)
	35 kW (vent.)	69 kW (vent.)
Summary	217 kW	108 KW
Energy demand	110.000 kWh	70.000 kWh
	23 kWh/m²/a	14 kWh/m²/a

- thermally driven absorption chiller: 30 kW
- wet cooling tower
- 0,5 m³ cold storage

- Electrically driven compression chiller: 100 kW
- 1,5 m³ cold storage
- the absorption chiller precools the supply of the compression chillerelectrically

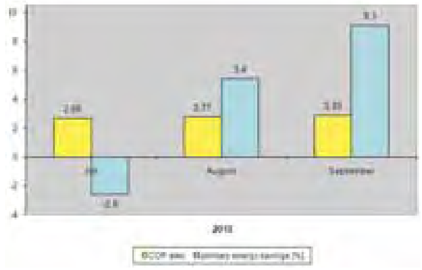
- 30 kW heat pump (source: exhaust air)
- 2 gas boilers, 110 kW each one

The installation was finally put in operation in September 2008. The Monitoring system is included in the central building control system.

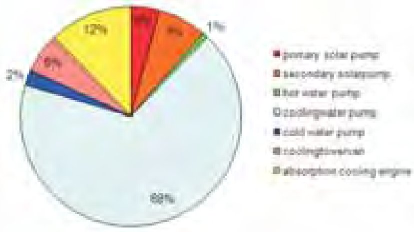
Results:

The analysis of the monitored data delivers an average monthly electric COP of 2,8 an average monthly thermal COP of 0,36 and an average monthly primary energy saving of 4%.

Monthly electric COP and primary energy saving of the Solar driven Cooling System for Summer 2010:



Fragmentation of the electricity consumption from the Solar driven Cooling System for August 2010:



The thermal driven chiller is located in the HVACR room in the cellar of the building. The position of wet cooling tower was choosen on the flat roof of the building (best practise for electric chillers):

➔ High pressure losses in the piping
➔ **overdesigned** pump for the heat rejection cycle

The **COP measurement** of the electric driven chiller was realised supplementary (in order to make comparisons possible). However the electric COPs concerning the electric backup compression chiller around the value 2.8 are also moderate.
The **comissioning** was difficult and complicated, because the installation is not a pre-engineered system and there are no comissioning guidelines existing.
The **control strateg**y is simple: Take what you can get from the sun for precooling, meet the demand with additional electric cooling.

Conclusions:

Intended aims for summer 2011:

- Revised control strategy for the absorption chiller
- Improvements concerning heat rejection (cooling tower)
- Increase of the cold water temperature in the building

The key issues of further activity should contain:

- Development of energy saving components for heat rejection
- Detailed description of operating conditions for absorption chillers
- Definition of COP references for electric chillers (measurements under annual operation)

ASiC is a non profit organisation, financially supported by the local government of upper Austria, the municipality of Wels and local companies. This R&D institute executes measurement and monitoring tasks in the fields of solar thermal systems and various research activities on experimental plants. ASiC covers the measuring and monitoring of solar thermal components and the detailed measurement of solar thermal collectors.

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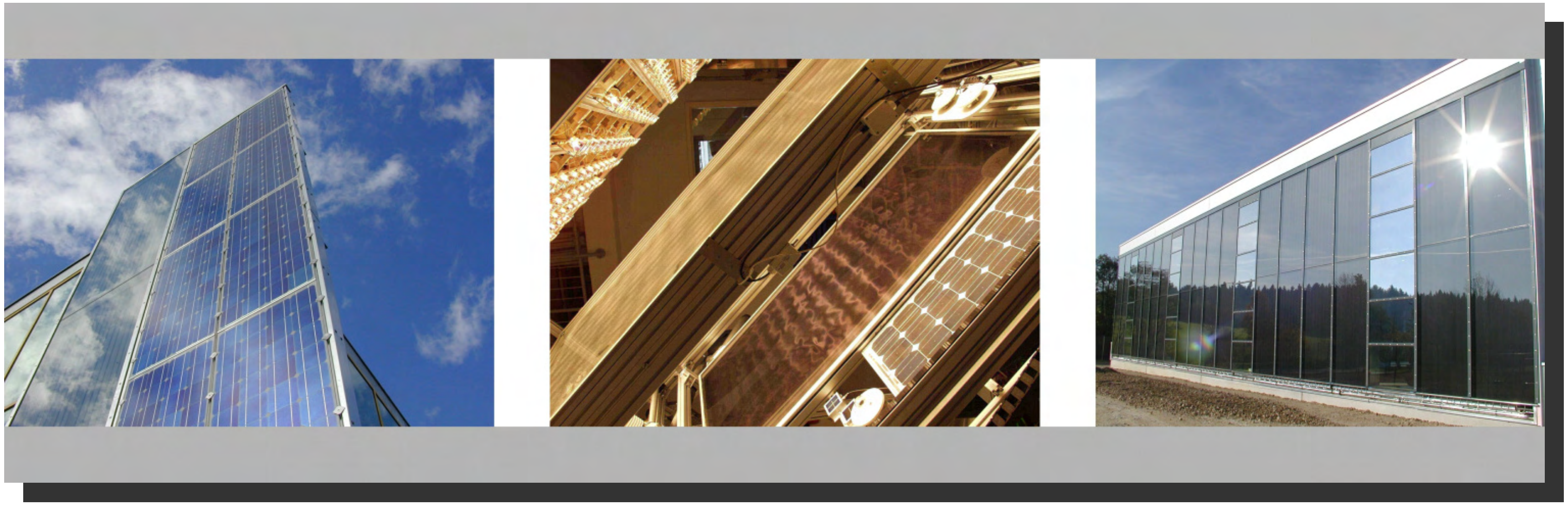
ANHANG ASIC 6

H. Focke, Presentation: Experiences from planning and commissioning a 30 kW installation, Expert Meeting Freiburg, 27.04.2009

Austria Solar Innovation Center



Experiences from planning and commissioning a 30 kW installation

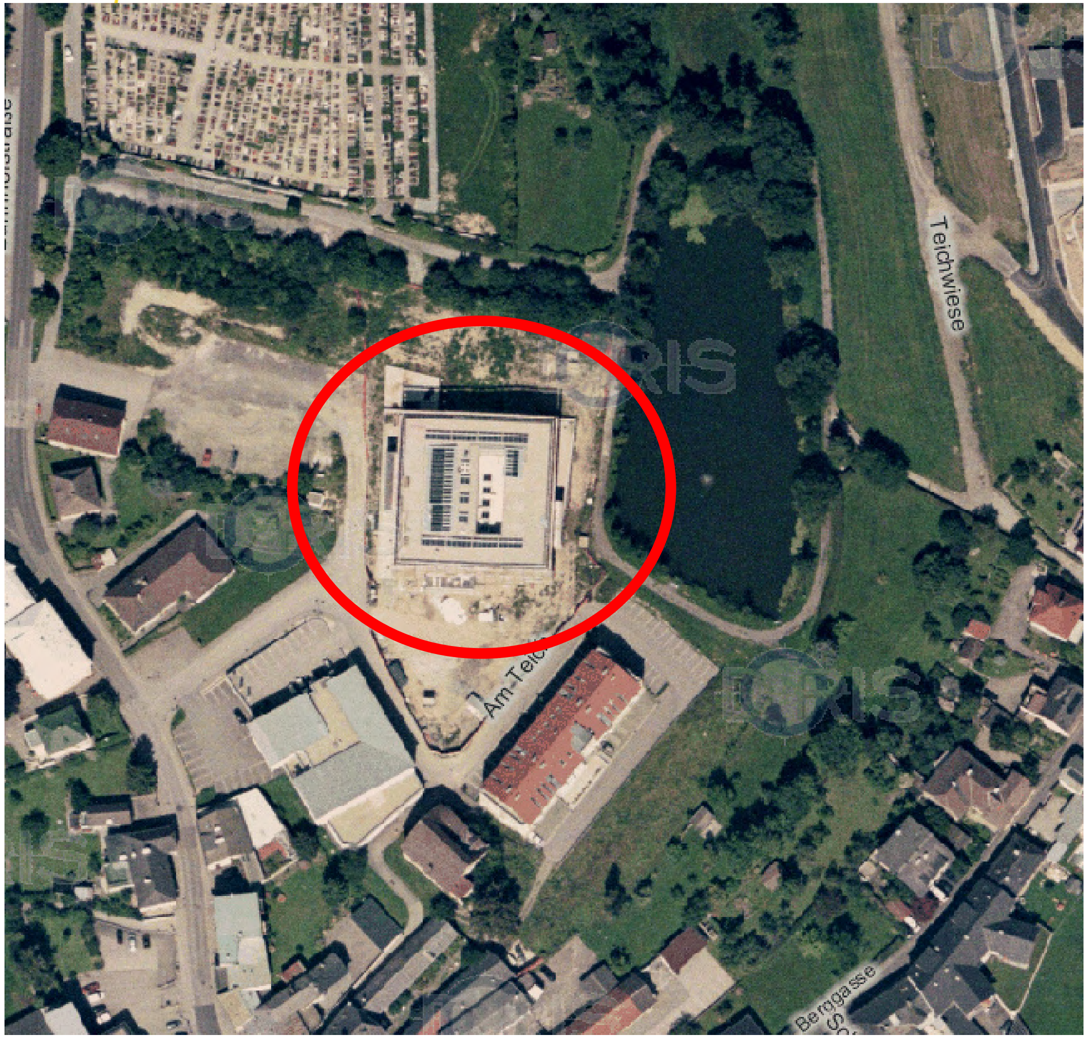
Content

- Where
- Why
- Installation
- Costs
- Experiences
- Monitoring System

Where


Bayern
Jihočeský
(Brünn) Brno
BH Rohrbach
Linz
Oberösterreich
Niederösterreich
Wien
Bratislava
Österreich
Tirol
Steiermark
Graz
Kärnten (Carinthia)
Trentino-Alto Adige
Ljubljana
Zagreb
Kroatien
Veneto
Rijeka
© 2009 Cnes/Spot Image
© 2009 Tele Atlas
Image © 2009 GeoContent
© 2009 Europa Technologies
Krk
Cres
Google
47°03'13.99" N 13°46'57.70" E
Höhe 1392 m
Sichthöhe 495.75 km

Where



LAND OBERÖSTERREICH

Site: northern part of Austria

Public funded building

Owner: LIG OÖ

The Building

Area: 4.880 m²

Conditioned area: 2.860 m²

Use: office building

4 floors

2 conference rooms

Why

- Calculation on summer behaviour
- Heating – Cooling Loads
- Use of Solar Energy – need of DHW 2.600 kWh
- 3 kW PV (feed in tariff)

  components not visible from entrance

Szenario	A (nothing)	B (AC + heat protect glazing)		C (AC + shading)	
St. room (Office) 20 m^2	Ti_{max}	$Kühllast_{max}$	$Kühllast_{max}$	$Kühllast_{max}$	$Kühllast_{max}$
	[℃]	[kW]	[W/m^2]	[kW]	[W/m^2]
South	35	1,90	95	0,35	18
West	34	2,70	140	0,45	23
North	30	0,80	40	0,13	7
East	32	1,60	80	0,18	9

- Calculation on summer behaviour
- Heating – Cooling Loads
- Use of Solar Energy – need of DHW 2.600 kWh
- 3 kW PV (feed in tariff)

  components not visible from entrance

	Heating	Cooling
Load	182 kW (Floor)	39 kW (Fan Coils)
	35 kW (vent.)	69 kW (vent.)
Summary	217 kW	108 kW
Energy demand	110.000 kWh	70.000 kWh
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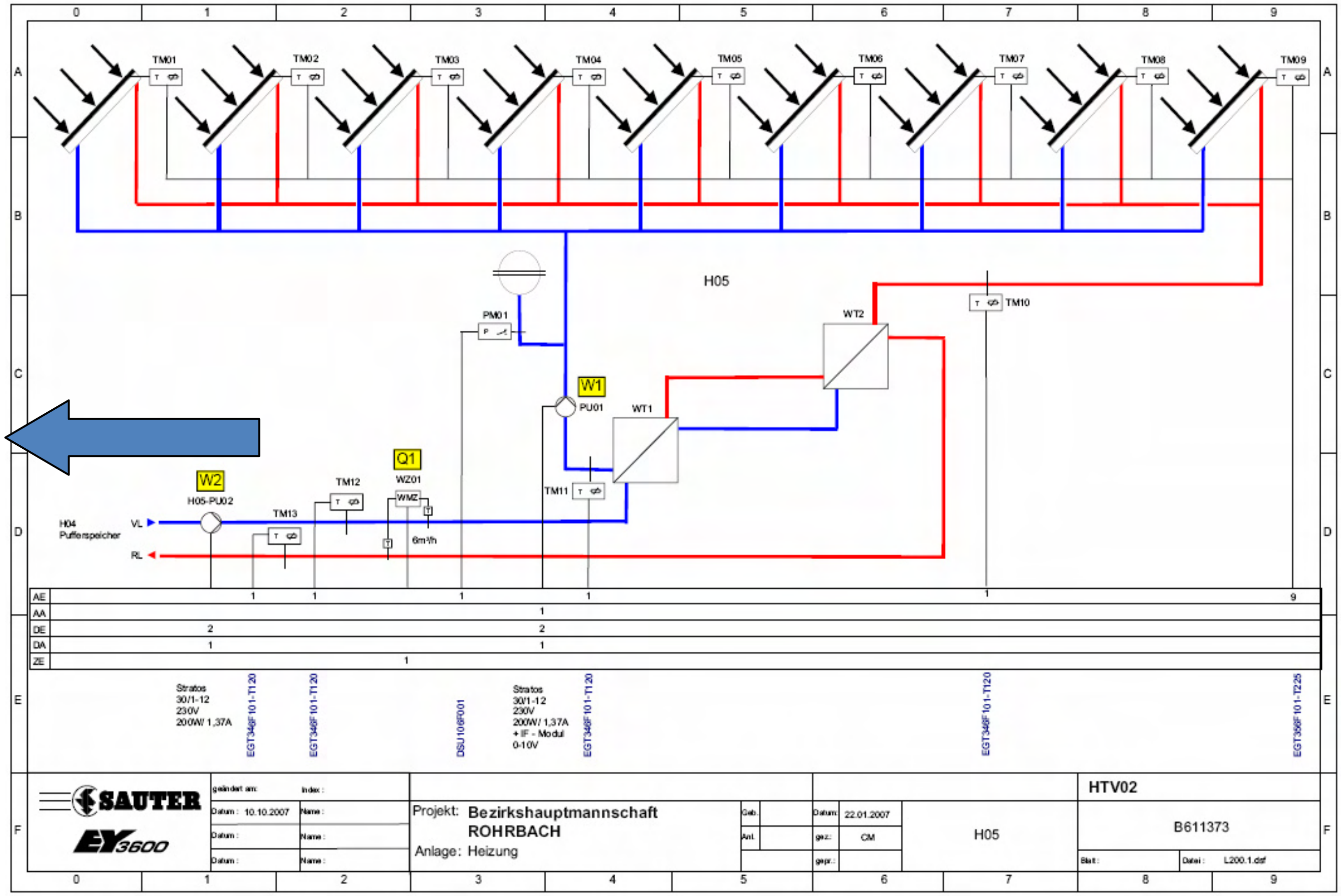
Installation

- Calculation on summer behaviour
- Heating – Cooling Loads
- Use of Solar Energy – need of DHW 2.600 kWh
- 3 kW PV (feed in tariff)

  components not visible from entrance
- Solarthermal system for heating and cooling (support)
- Dimensioning WITHOUT dynamic building simulation tool
- Advice: Detailed dimensioning

Installation

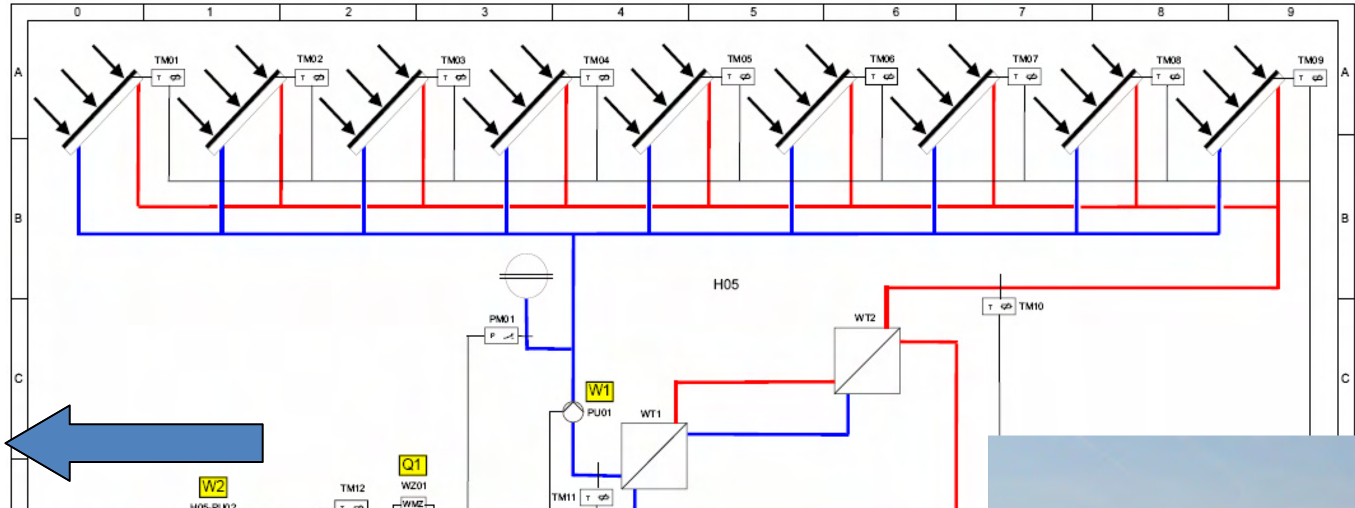


120 m² flat plate collectors

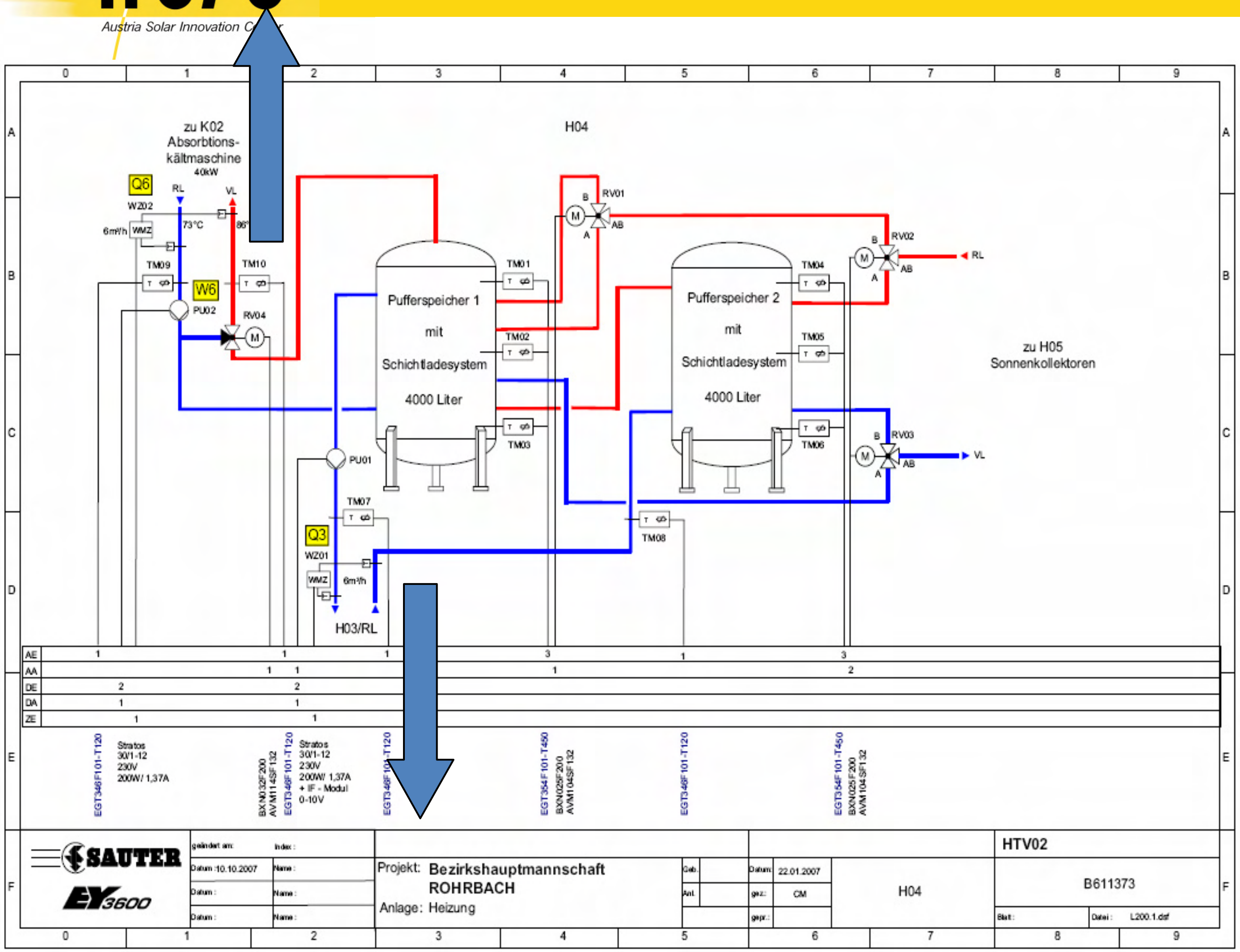
use of the max. area of the roof

slope: 45°

Installation



Installation



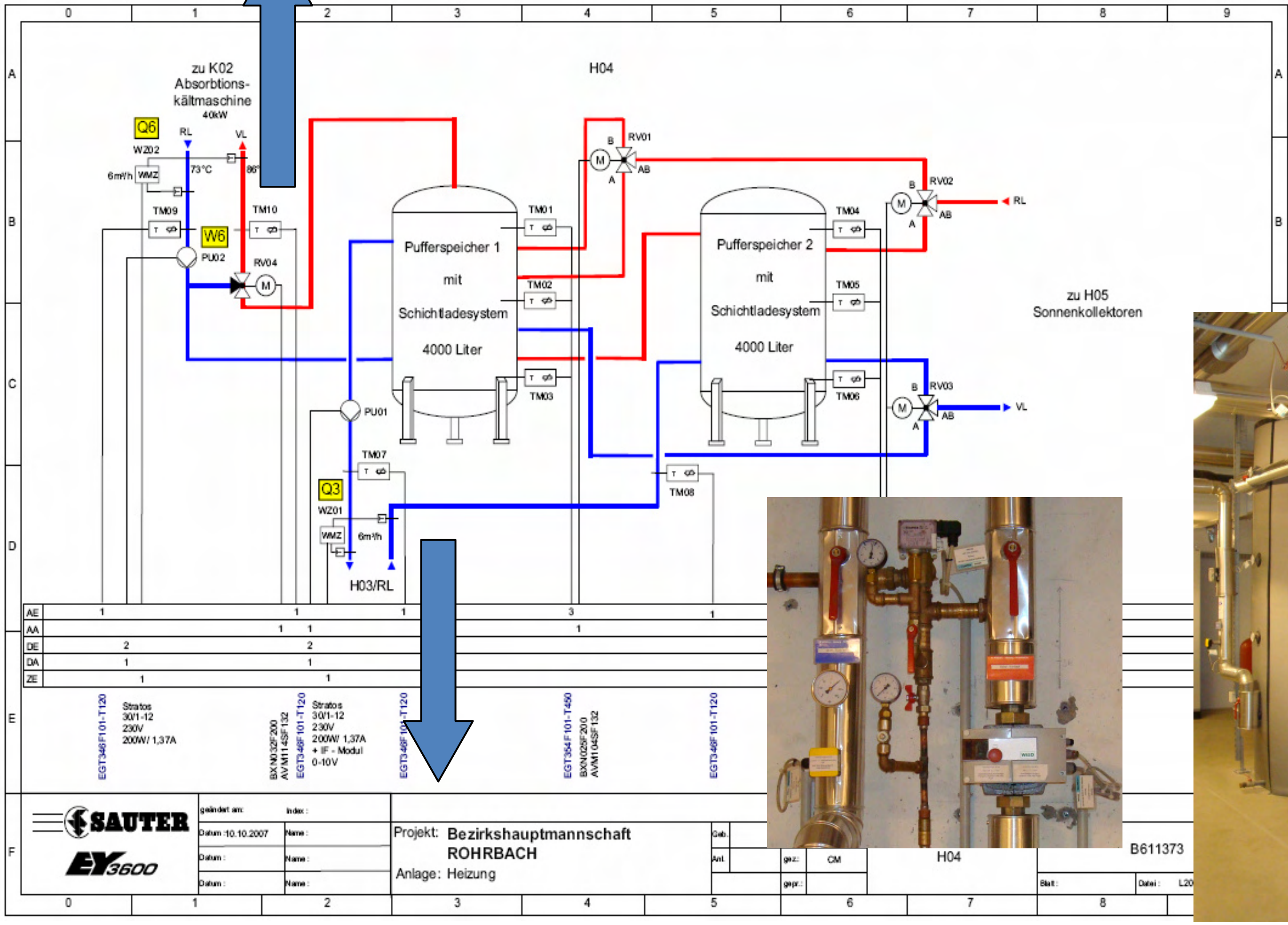
2 storages 4 m³

Nr. 1 – 2 levels

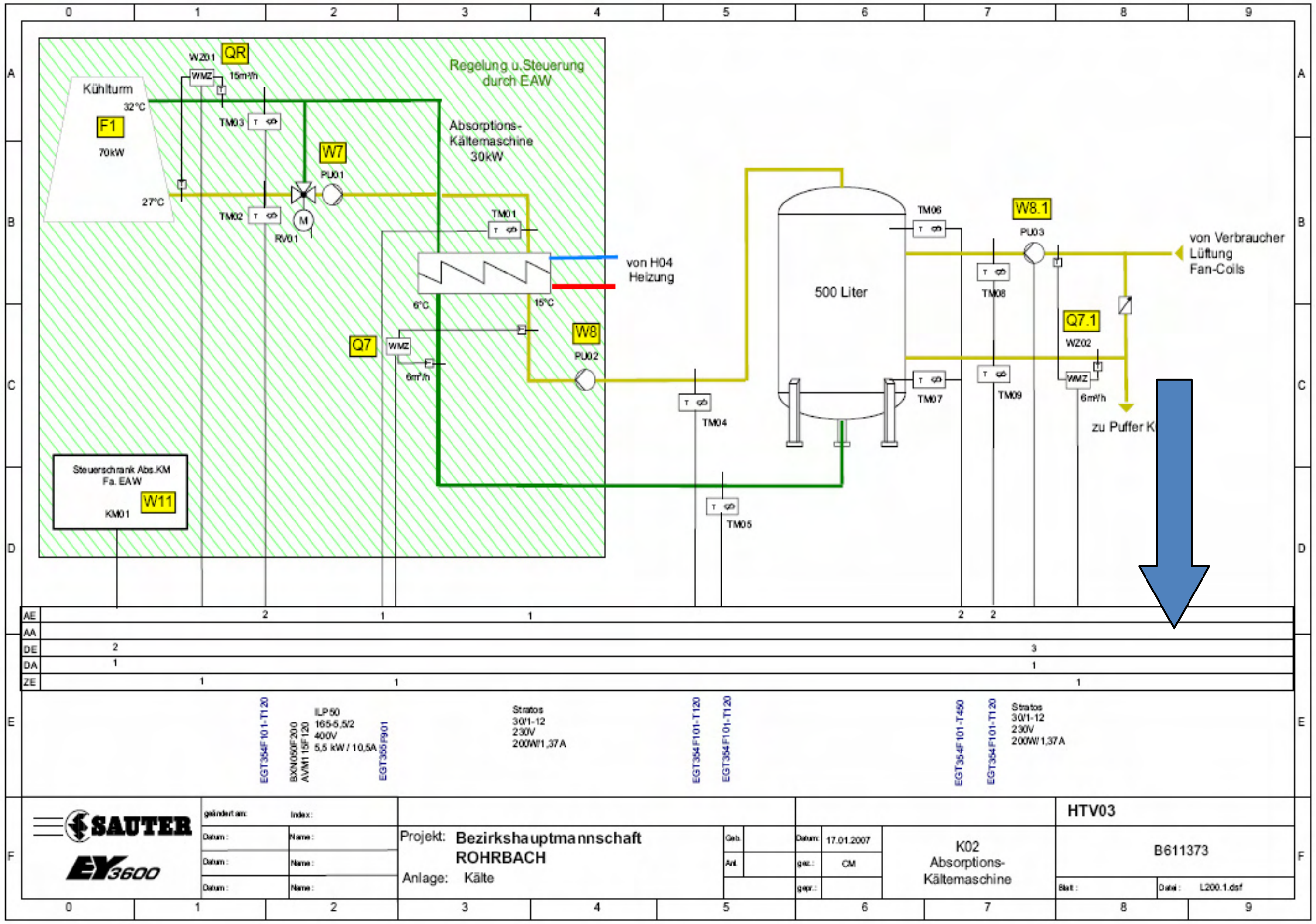
Nr. 2 additional

connections to heating system + absorption chiller

Installation
Installation



Installation

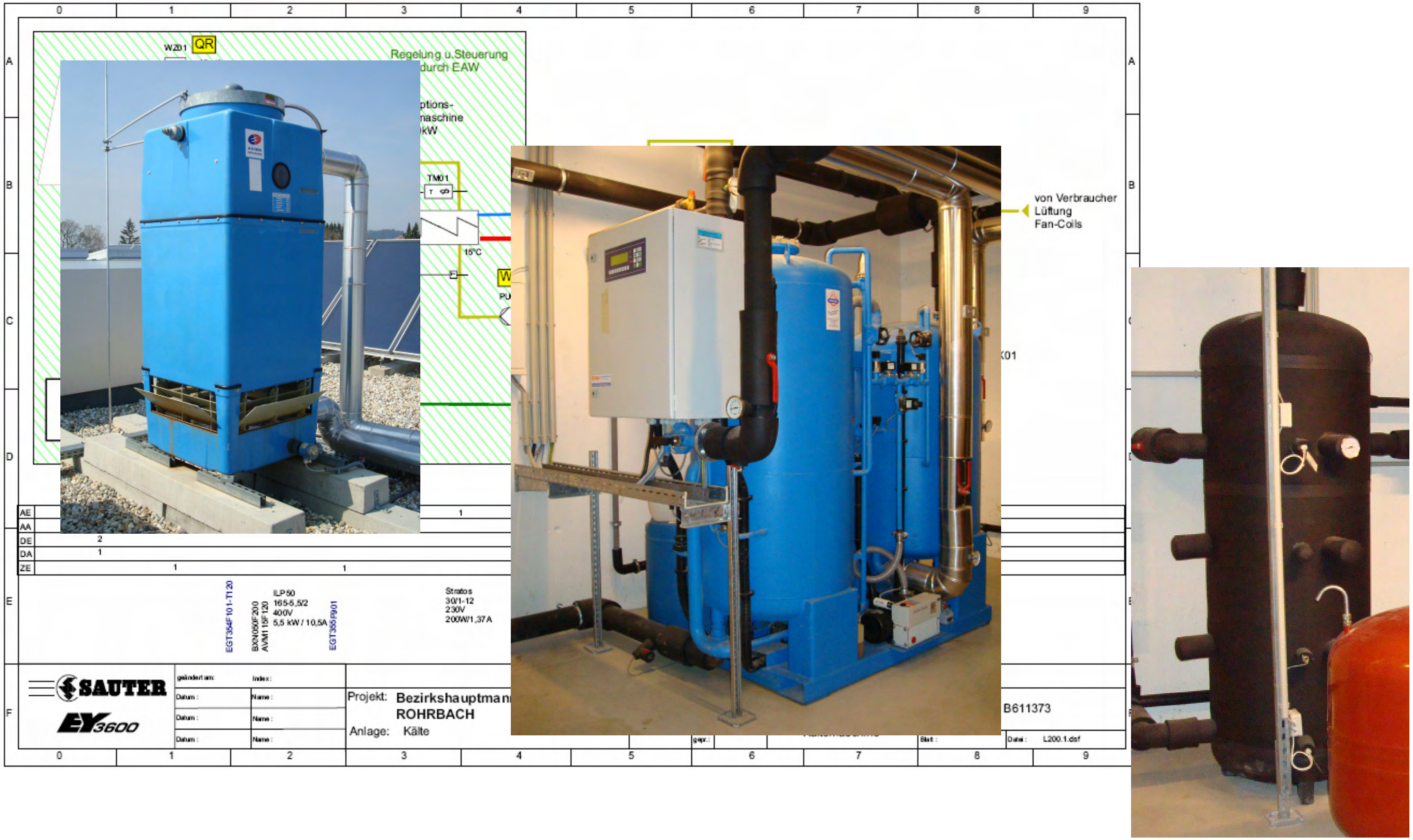


30 kW EAW Absorption Chiller

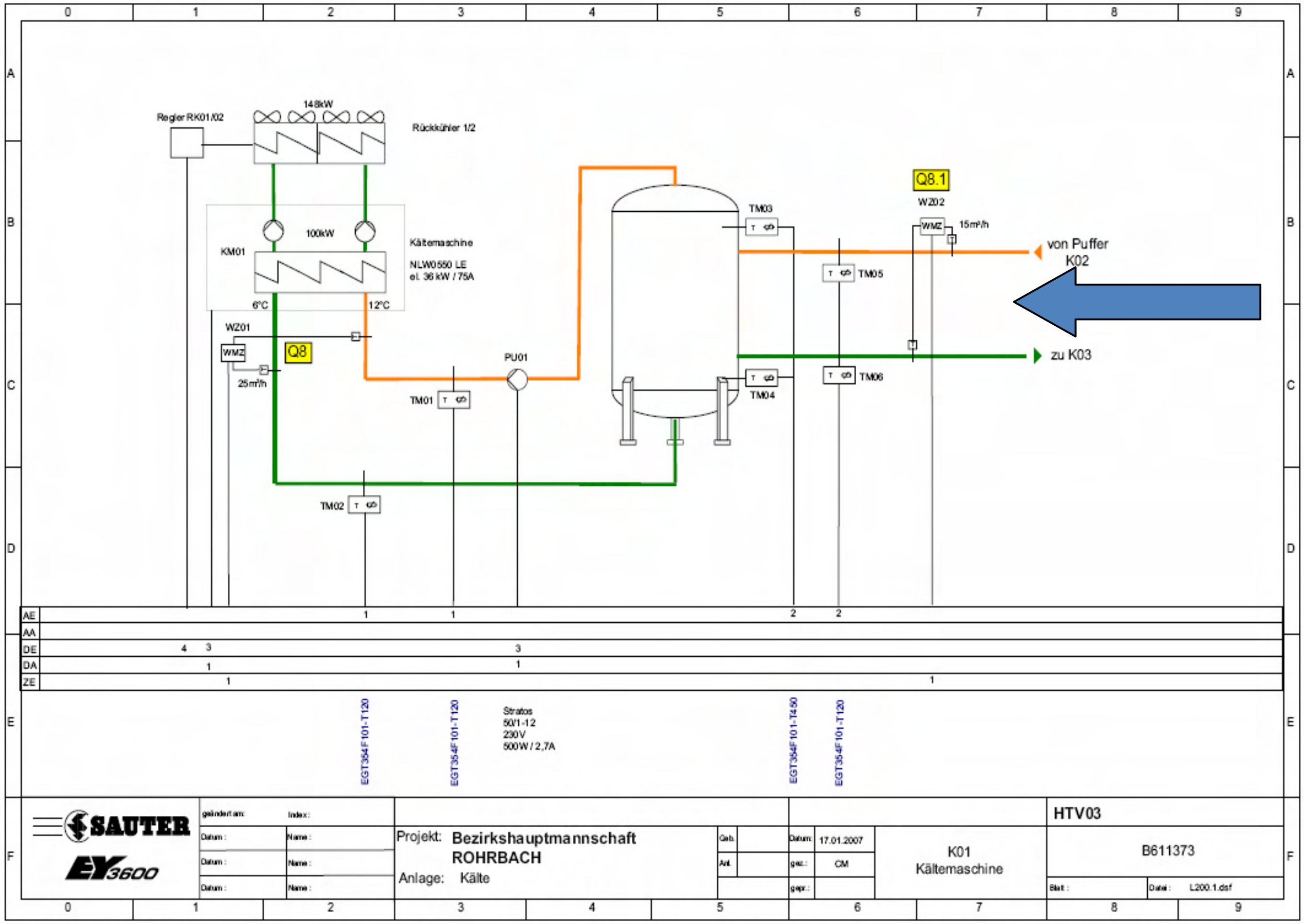
0,5 m³ cold storage

Wet cooling tower

Installation



Installation



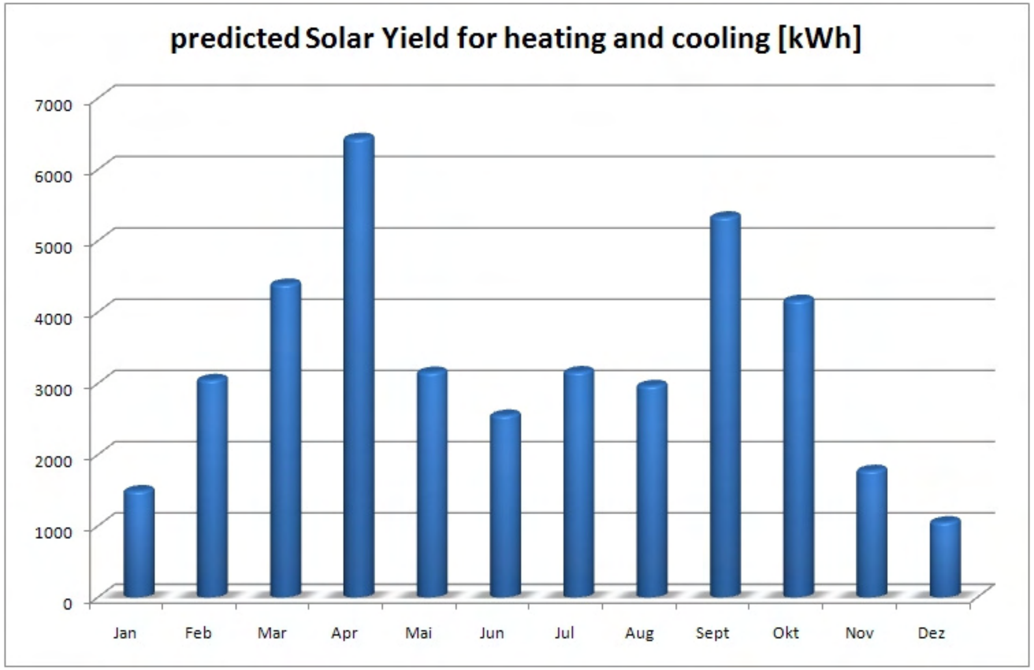
100 kW compression chiller

1,5 m³ cold storage

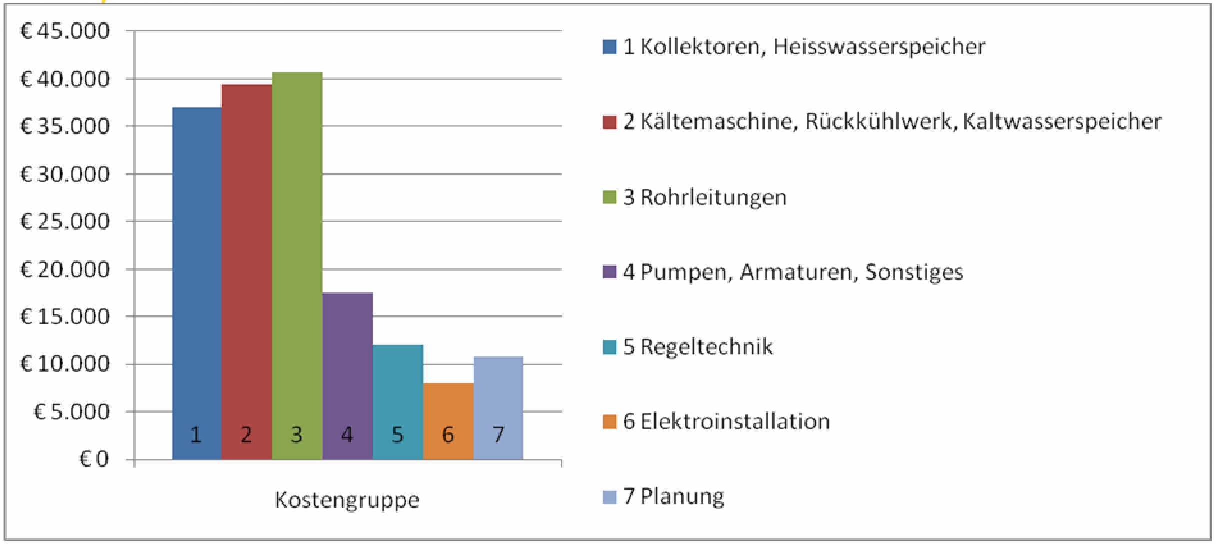
Absorption chiller pre- cools the supply of compression chiller

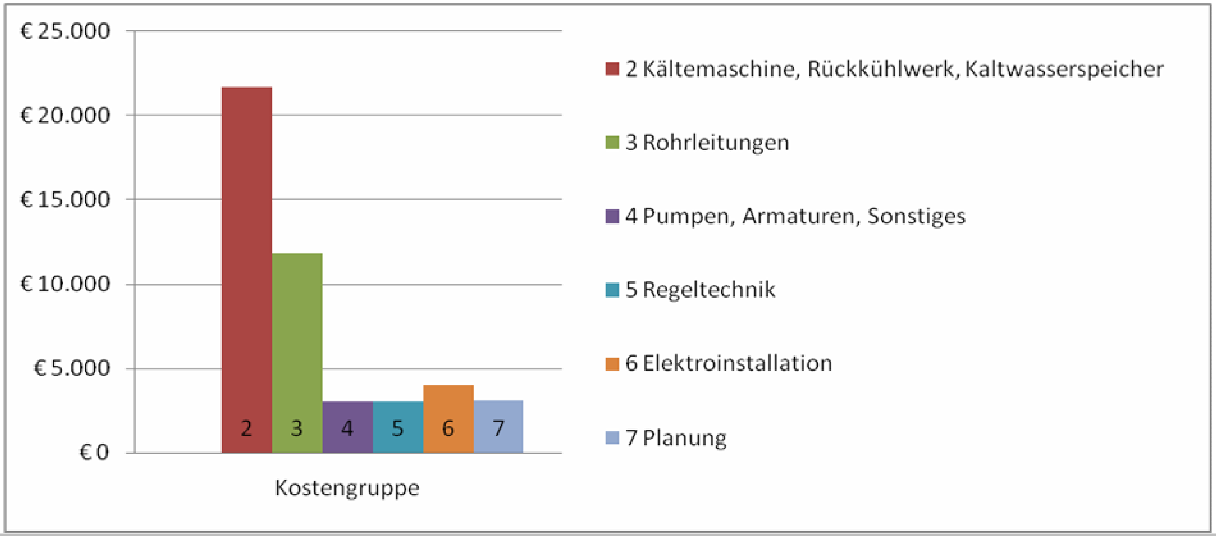
Installation

- Solar support for heating 25%
- Solar support for cooling 13%
- Solar yield: 39.550 kWh / year



Costs





Solar driven cooling:

127.000 € excl. Solarthermal installation

4.233 €/kW

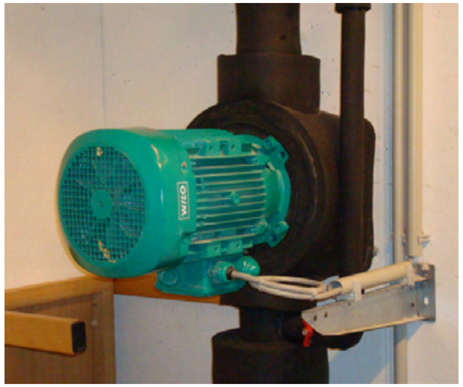
Conventional cooling:

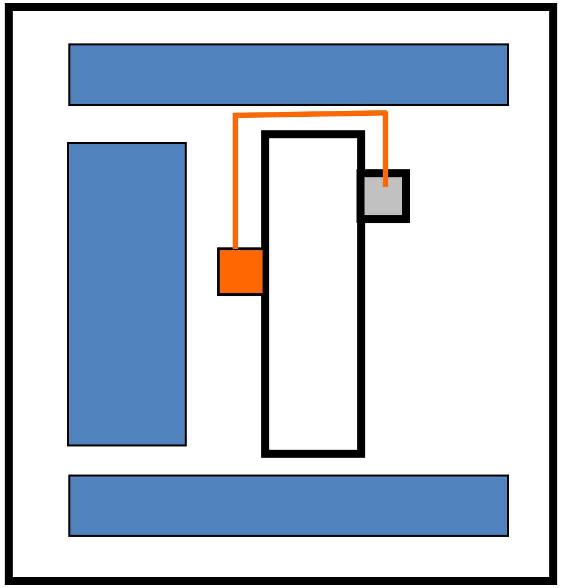
47.000 €

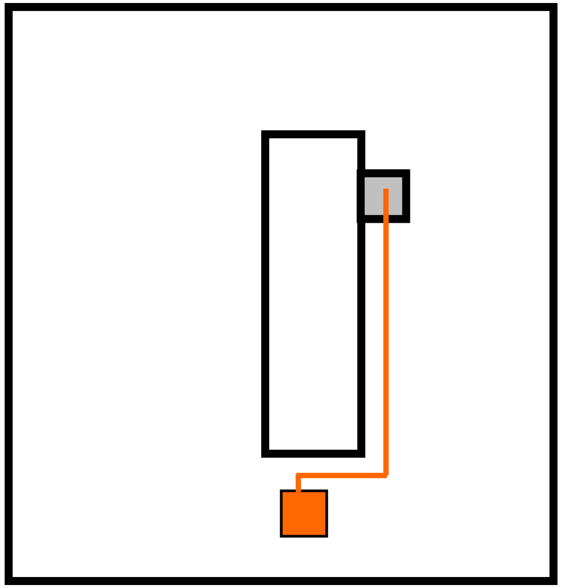
470 €/kW

Experiences

- energy efficient pumps, but esp. for re-cooling:
- very long distances (~200 m)
- 5 kW pump (2,5 kW in operation)







Control / Monitoring

Control:

- Simple
- Prevent part load operation of thermal driven chiller
- Problems in- between season (beginning of April)

Monitoring:

- Integrated in central building control system
- Level 3
- Includes COP of conventional chiller
- Data collection since April 2009

Thank you for listening!

Institut für Wärmetechnik, TU Graz

ANHANG IWT 1

Moser H., Rieberer R., (2007) SMALL-CAPACITY AMMONIA / WATER ABSORPTION HEAT PUMP FOR HEATING AND COOLING – FOR SOLAR COOLING", International Conference of Solar Air Conditioning; Tarragona, Spain; 18 – 19 October 2007

SMALL-CAPACITY AMMONIA / WATER ABSORPTION HEAT PUMP FOR HEATING AND COOLING - USED FOR SOLAR COOLING APPLICATIONS

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ABSTRACT

In contrast to large-capacity applications up to now small-capacity absorption heat pumping systems have difficulties to enter the market due to relatively low efficiencies, control issues, and high first cost. Recently small-capacity absorption chillers with the working pair water/lithium bromide have been developed and are already available on the market, but only a few installations of small-capacity ammonia/water applications can be found.

This paper describes the system layout, mathematical model and first experimental results of a small-capacity ammonia/water absorption heat pumping unit (cooling capacity 5 kW) which is designed for a wide operating range in order to allow operation with low cold water and/or high cooling water temperatures.

1. INTRODUCTION

By means of absorption heat pumps it is possible to lift heat from a low temperature level (cold water) to a medium temperature level (cooling water) whereas the driving energy for the process is mainly heat at a high temperature level (heating water). Depending whether the cold water or the cooling water (or both) of the heat pumping unit is used, the system may act as a cooling or a heating system, e.g. for residential cooling, refrigeration or residential heating systems. While the temperature lift which is the temperature difference between cold water and cooling water is small for a typical residential cooling application (e.g. 20 K), it can increase up to 50 K or more for heating and/or refrigeration. Recently different solar driven small capacity absorptions heat pumps for residential cooling has been developed and are available on the market. But up to now only a few ammonia/water prototype units are known which can be used at low cold water temperature, e.g. for refrigeration, and/or high cooling water temperature, e.g. in combination with a cooling tower at very hot weather conditions. Absorption heat pumps which are designed for residential heating can bridge this niche as they deal with lower cold water and higher cooling water temperature levels. The Figure 1 shows the log p vs. 1/T diagram for typical temperature levels for residential cooling and heating/refrigeration application.

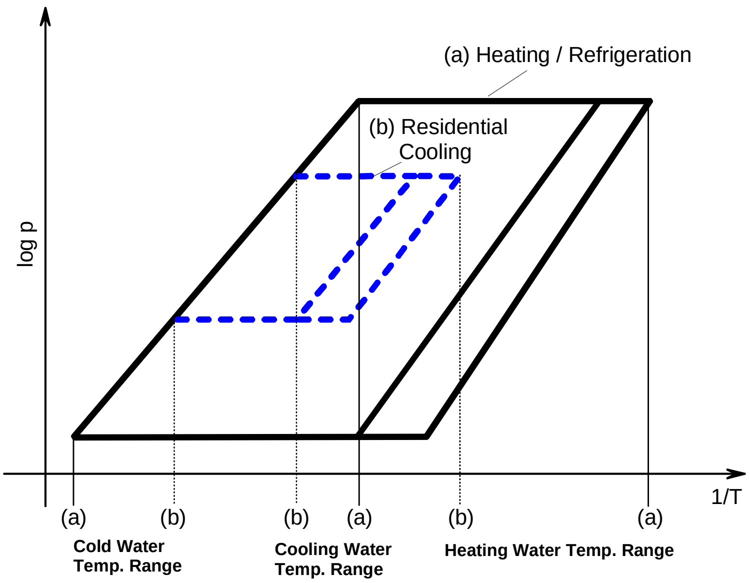

Figure 1 Log p vs 1/T Diagram for Residential Cooling and Heating Application

It is clearly seen from Figure 1 that for increased temperature lift of the application the required generator outlet temperature increases as well. That's why for applications for high temperature lifts standard flat plate solar collectors can not be used due to low efficiency at high operating temperature and other alternatives as vacuum collectors or concentrating solar collectors are necessary.

2. SYSTEM DESIGN

At the Institute of Thermal Engineering the development of a small-capacity single-stage ammonia/water absorption heat pumping unit for a wide operating range is going on. It allows both heating and cooling operation. The heat pump has been designed to operate with cold water temperatures from -10°C to 20°C and cooling water temperatures from 30°C to 50°C. It is indirectly driven by a heat carrier loop (water or thermo-oil) in order to allow the use of different heat sources like biomass, solar, process heat or others. The capacity modulated heat pump uses a variable speed controlled solution pump in order to avoid start and stop losses and the generator temperature is controlled to the optimum efficiency point (Fernández-Seara and Vázquez; 2001). For purification of the refrigerant a rectification column combined with a dephlegmator which is cooled by the rich solution has been developed. The Figure 2 shows the realised single-stage process.

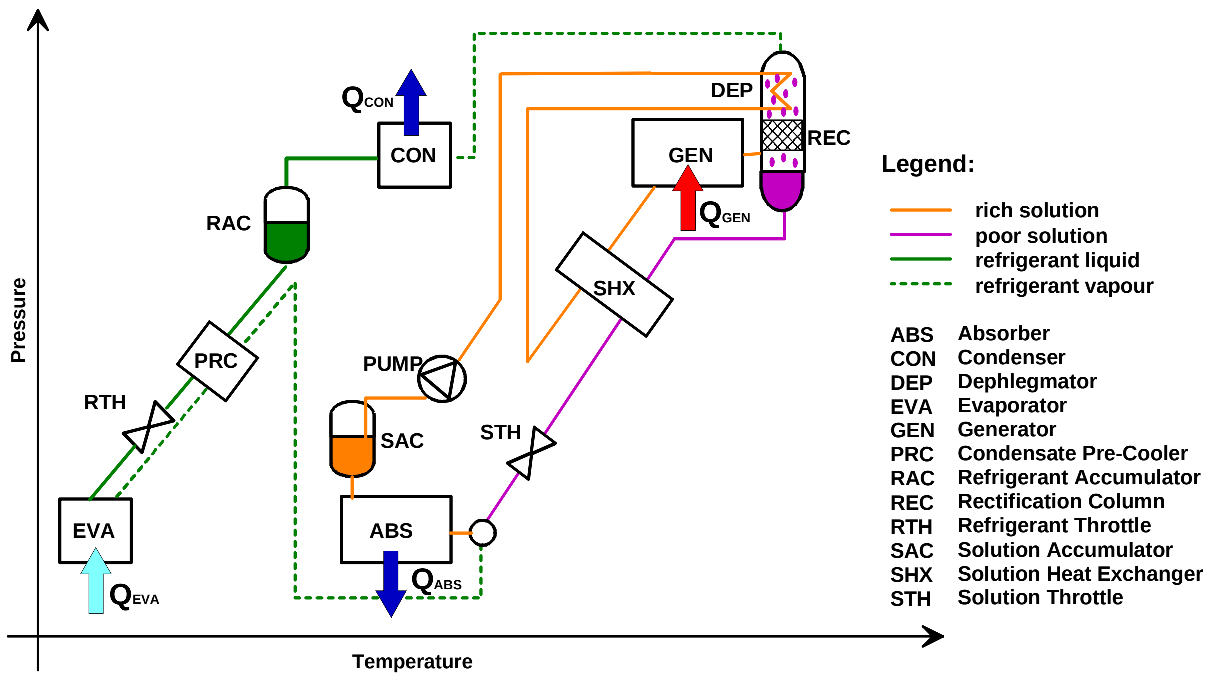

Figure 2 Realized single-stage process

For all heat exchangers – including absorber and generator – standard plate heat exchangers (PHEs) have been used. These PHEs offer very large heat transfer areas at compact dimensions, thus small temperature differences can be achieved and - compared to special designs - a considerable cost reduction is expected.

3. CALCULATION AND EXPERIMENTAL RESULTS

A mathematical model has been set up using the software program EES (2006) in order to study the effects of component behaviour, to perform sensitivity calculations and to compare the calculations to the experimental results. The model solves the energy-, mass-, and ammonia mass-balances for all components. For the internal heat exchangers efficiencies were considered and the influence of the rectification process on the dephlegmation capacity was calculated using the theoretical plate model (ponchon method) acc. to Herold and Pande (1996). Pressure drops in components and pipes were neglected.

A prototype unit of the absorption heat pump has been constructed, equipped with measurement instrumentation, and tested at different operating conditions.

Fig. 3 and 4 shows the calculation results for the "Coefficient of Performance" for cooling (COP_C) depending on the temperature lift (ΔT_{Lift}), in comparison to some results of the experiments for a generator heat input of 10 kW (full load) and 5 kW (part load).

The temperature lift and COP_C were calculated using the equations 1 and 2.

The calculation shows an approximately linear correlation between the COP and the temperature lift. The calculated values for different cooling water temperatures (inlet temperatures 28, 38 and 48°C) almost match at the same temperature lift. The calculated COP_C varies approximately between 0.75 at 10 K temperature lift and 0.4 at 50 K temperature lift.

The experimental results show a similar linear correlation of the COP_C with the temperature lift, but they are approximately 10% below the results of the mathematical model.

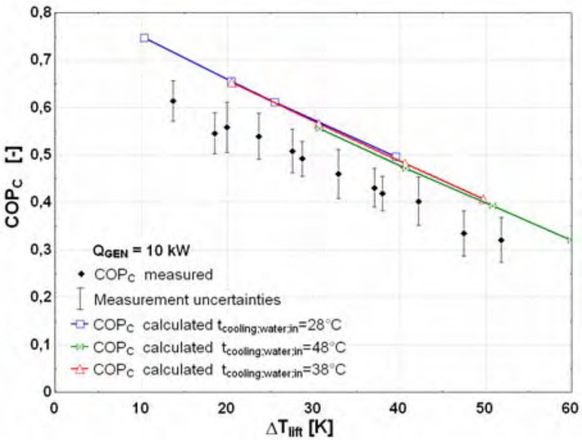

Figure 3 COP – calculated results vs. experiments at full load

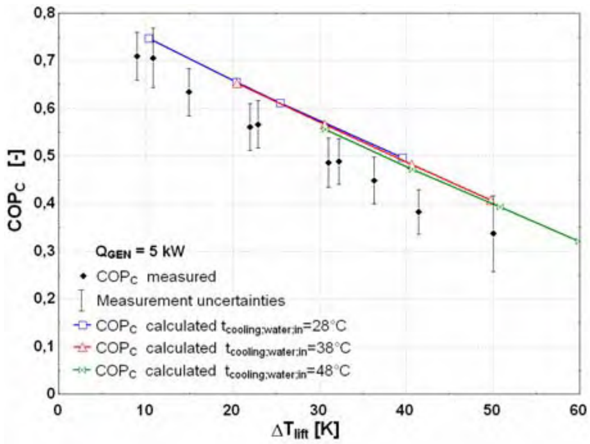

Figure 4 COP – calculated results vs. experiments at part load

$$COP_C = \frac{\dot{Q}_{EVA}}{\dot{Q}_{GEN}} \qquad (1) \; ; \qquad \Delta T_{Lift} = \frac{(t_{cooling;in} + t_{cooling;out})}{2} - \frac{(t_{cold;in} + t_{cold;out})}{2} \qquad (2)$$

The reasons for the deviation between the model and the experiments can possibly be assigned to the following subjects:

- The water fraction in the refrigerant seems to be slightly higher than calculated.
- In the evaporator the temperature difference between the cold water and the refrigerant is slightly higher than expected which could be caused by a non-uniform flow distribution in the plate heat exchanger.
- The temperature difference between the rich solution at the absorber outlet and the cooling water is slightly higher than expected which could also be caused by a non-uniform flow distribution in the plate heat exchanger.

Further optimization will be done to improve the heat pump performance and bring the experimental results toward the results of the mathematical model.

5. CONCLUSIONS

A small capacity ammonia/water absorption heat pump for a wide operating range has been developed and a prototype unit has been constructed and tested. The heat pump is designed to operate at cold water temperatures from -10°C to 20°C and cooling water temperatures from 30°C to 50°C. Furthermore, a mathematical model has been set up to calculate the process parameters and compare the results to the experiments.

The calculated COP_C shows an approximately linear correlation with the temperature lift and varies between 0.75 at 10 K temperature lift and 0.4 at 50 K temperature lift. The experimental results show a similar linear correlation of the COP_C with the temperature lift, but approximately 10% below the results of the mathematical model. In general the calculated and the experimental results are promising and further optimization will be done to improve the heat pump performance and bring the experimental results toward the results of the mathematical model.

ACKNOWLEDGEMENT

This work has been carried out under the framework of „Energy Systems of Tomorrow“, an initiative by the Austrian Federal Ministry for Transport, Innovation and Technology.

Furthermore the authors wish to thank our project partners, the companies “Heliotherm Wärmepumpentechnik Ges.m.b.H.”, “KWB - Kraft und Wärme aus Biomasse GmbH”, and “M-TEC Mittermayr GmbH” for there support and contributions.

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Herold K., E. Pande M., 1996, Counterflow Vapor-Liquid Exchange Processes Using Ammonia/Water, Int. Ab-Sorption Heat Pump Conference, Montreal, 481-488

ANHANG IWT 2

Moser H., Rieberer R., (2008) “Second Law Analysis of a Laboratory Prototype of a Single Stage Ammonia / Water Absorption Heat Pump”, International Sorption Heat Pump Conference, Seoul, Korea; p. AB 109 - 117

Cooling load demand assessment – a key issue for economic operation of solar cooling systems

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Abstract

Already existing buildings mostly do not correspond to the requirements for solar cooling technologies. The decision for solar cooling applications is often done on the basis of simulation tools. But the main challenge in the use of such simulation tools is the correct description of the internal loads, thus the description of the behaviour of human user of the building. An alternative procedure is the data collection of heating and cooling loads and environmental conditions during a selected period. The monitored data can result in a simulation with realistic cooling load characteristics to assess the economic efficiency of a solar cooling system.

In the presented project an application (mold design and manufacturing plant) will be analysed. The poster shows the results of the data monitoring for the cooling loads, the heating loads and the environmental conditions. Moreover a feasibility study, the comparison with common technologies, and the dimensioning of the solar cooling plant can be carried out based on the monitoring results and results in a decision-making-model for economic-efficient operation.

The presented results of monitoring and simulation show a vast simultaneity of solar irradiance and cooling loads. It turns out, that the huge water basin is a central component for a competitive solar cooling system. The results on this certain plant can be transferred to similar applications.

Introduction

The project has been carried out by “Austria Solar Innovation Center – AsiC”

ASIC is a non profit organisation, financially supported by the local government of upper Austria, the municipality of Wels and local companies. This R&D institute executes measurement and monitoring tasks in the fields of solar thermal systems and various research activities on experimental plants. ASiC covers the measuring and monitoring of solar thermal components and the detailed measurement of solar thermal collectors.

The considered plant is located in the company WIHO Hofbauer GmbH (Figure 1). This company acts as an expert in toolmaking, mold and die production as well as toll-manufacturing. Therefore more than 20 machines like machining centers, lathes, grinding machines and spark machines are located in a workshop on the scale of 2.500 m².

Figure 1: Building of company WIHO Hofbauer

Increasing cooling loads

Figure 2 shows the existing scheme for the heating and cooling supply of the plant. The system consists of 3 parts with cooling demand: an office with 250 m² (air conditioned with a fan coil system) and a factory workshop with 2.500m². Additionally a machinery with metal cutting machine tools and spark machines has to be chilled with constant low-temperature feed water. Now steadily increasing cooling demand makes it necessary to replace the old cooling unit by a greater one.
The cooling loads, mentioned in Figure 2, derive from state-of-the-art dimensioning, and yields typically oversized standard electric driven chiller. The cooling load is currently covered by an aligned 37 kW chiller using a 110m³ water basin. The hot water requirements, mainly for the showers, are rather marginal.

The Industrial spark machines need water quenches of constant 14˚C temperature for accurate operation. The cold water consumption for the office is negligible, but during hot periods in summer the interior air conditions in the workshop often become uncomfortable. The extension of the plant causes increasing cooling loads for the spark machines mainly during summertime, which could not be covered with the installed electric driven water chiller. Rising expenses for electric energy provoced the fundamental idea of the current project: to compare the economic efficiency of a standard electric driven chiller with an environmentally friendly solar driven chiller on the basis of the actual cooling demand.

Data monitoring

From summer 2006 to summer 2007 the meteorological data, the cooling loads and the consumption of electric energy have been acquired by a data monitoring system.

Additionally the heat consumption for hot water production and heating was monitored.

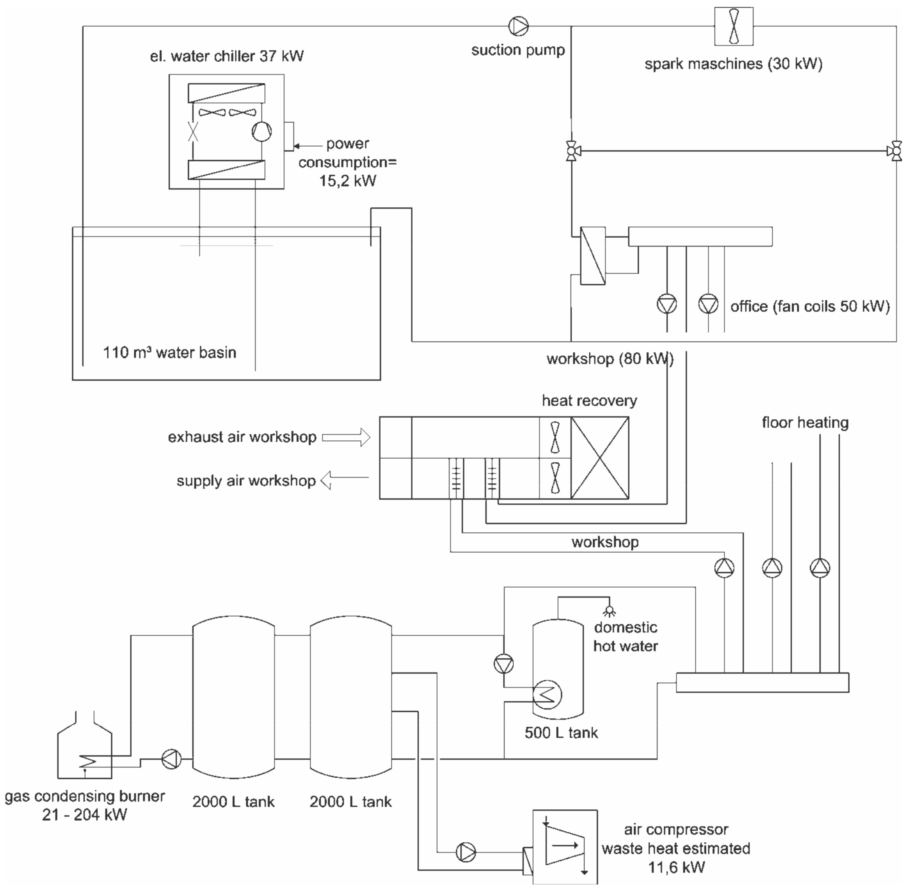

Figure 2: Hydraulic scheme of the plant

On the basis or the recorded data, a continuative feasibility study will demonstrate, if solar (assisted) cooling can be used efficiently. The following steps are simulation, designing and construction, followed by commissioning and monitoring of the realised solar cooling plant.

Figure 3 shows exemplarily the history of irradiance, ambient temperature and cooling loads during a period of hot weather in August 2006. It can obviously be seen in the graph, that there is a distinct correlation between the whole cooling load and the solar radiation during daytime, also a detailed statistical analysis of the monitored data yield to the same results.

The base load is mainly due to the amount of operating spark machines, and independent from the ambient conditions on a wide range. The peak load is caused principally by the factory workshop, and follows the course of solar radiation to a large extend. The temperature level seems to be suitable for a solar thermal driven water chillers. Furthermore numerous operating cycles (summer nights, wintertime and transit times) can be carried out by a wet cooling tower. It finally turns out, that the huge water basin is a central component for a competitive solar cooling system.

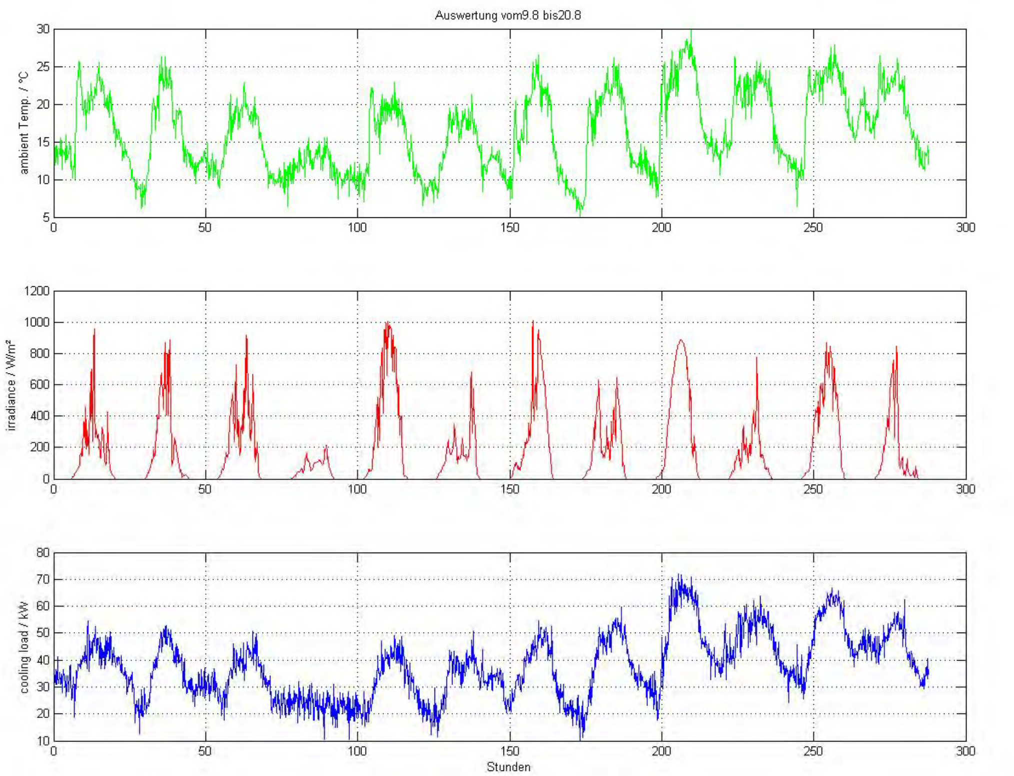

Figure 3: Sample data from the monitoring system

Simulation of the solar cooling plant

The simulation of the behaviour of the overall plant has been carried out with Matlab/Simulink ©. First results show a promising hybrid configuration with a 30 kW EAW SE 30 chiller and a 65 kW wet cooling tower.

The cooling tower is able to cover the cooling load over a period of more than half of the year. 30 % of the heat demand can be covered directly by solar energy.

Further details can be found on the poster as well as on www.asic.at.

Conclusions and Outlook

For integrating a solar assisted cooling system it is necessary to demonstrate the economic efficient operation of the overall plant. It can be shown, that in the considered example approx. 6.000€ for the cold production and 1.500€ for the heat generation can be saved by using a solarthermal plant.

It turns out, that a decision pro/con a solar assisted cooling system on the basis of economic efficiency can not be made in general. A realistic cooling load demand assessment is able to deliver a basis for decision-making.

Acknowledgment

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ANHANG IWT 3

Moser H., Rieberer R., (2008) “Thermodynamic Comparison of Different Designs of Single Stage Ammonia / Water Absorption Heat Pumps”, IIR Gustav Lorentzen Conference on Natural Working Fluids; Copenhagen, Dänemark; p 583 - 590

THERMODYNAMIC COMPARISON OF DIFFERENT DESIGNS OF SINGLE STAGE AMMONIA / WATER ABSORPTION HEAT PUMPS

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ABSTRACT

At the Institute of Thermal Engineering a small-capacity ammonia/water absorption heat pumping unit (AHP) with about 5 kW cooling capacity has been constructed and tested in the laboratory. In order to reduce the first costs of the AHP for all heat exchangers standard plate heat exchangers have been used. Due to the use of plate heat exchangers in both, the absorber and the generator the solution is passed through the components in co-current flow of liquid and vapour phase which leads to a thermodynamic disadvantage compared to especially designed components, i.e. expensive heat exchanger enabling counter-current flow.

In this work two thermodynamic models have been set up in order to calculate the thermodynamic penalty of a co-current flow in the generator compared to a counter-current flow and an exergy analysis has been carried out to identify irreversibilities associated with inefficient processes and to calculate the specific exergy loss of each component.

1. INTRODUCTION

In recent years small capacity absorption heat pumps (AHP) attract increasing research interest due to the possible use of renewable energy, e.g. biomass, solar or process heat for cooling applications. However, they are also considered to be used for residential heating application where usually the requirements regarding the temperature levels of the heat sources and the heat sink are different. In contrast to large-capacity applications, up to now small-capacity absorption heat pumping systems have difficulties to get into the market due to relatively low efficiencies, control issues, and high first costs.

At the Institute of Thermal Engineering a small-capacity ammonia/water absorption heat pumping unit for heating and cooling purpose has been developed and tested (Moser et. al., 2007). The heat pump has been designed to operate with cold water temperatures from -10°C to 20°C and cooling water temperatures from 30°C to 50°C. It is indirectly driven by a heat carrier loop (water or thermo-oil) in order to allow the use of different high temperature heat sources. For all heat exchangers – including absorber and generator – standard plate heat exchangers (PHE) have been used. PHEs offer very large heat transfer areas at compact dimensions, thus small temperature differences can be achieved. Furthermore PHEs are off-the-shelf products which offer a high prefabrication standard and compared to special designs a considerable cost reduction is expected. The capacity modulated heat pump uses a variable speed controlled solution pump in order to avoid start and stop losses and the generator temperature is controlled with respect to maximum efficiency (compare Fernández-Seara and Vázquez, 2001). For purification of the refrigerant a rectification column combined with a dephlegmator which is cooled by the rich solution has been developed.

In a conventional generator the liquid solution enters the heat exchanger on the top and leaves the heat exchanger at the bottom and the generated vapour flows upwards in counter-flow to the liquid and leaves the heat exchanger on the top. Using a standard plate heat exchanger for the generator the counter-flow of liquid and vapour requires large flow cross sections in order to avoid “flooding” of the heat exchanger. In the applied development the liquid and the generated vapour phases flows in co-current flow through the generator and are separated after the generator in the rectification.

The aim of this work is to investigate the thermodynamic penalty of a co-current flow in the generator compared to a counter-current flow and to identify potential for improvements of the applied design. Therefore a thermodynamic analysis of two different processes has been performed and compared to each other, including both the first and second law of thermodynamic. A previous comparison of co-current and a counter-current generator design has been presented by Moser et al, (2008). However, the calculation procedure has been extended and improved showing new results in this work.

2. SYSTEM DESCRIPTION

In Figure 1 the design and state points of the absorption heat pump cycle for the co-current generator design (left) and the generator section of the counter-current generator design (right) is shown. The process is described hereafter for the co-current generator design (left). Starting at the absorber the rich solution flows via the solution accumulator (SAC) to the solution pump where the pressure is increased to the high pressure level. Further the rich solution flows to the dephlegmator where it is heated by the refrigerant at the top of the rectification column and to the solution heat exchanger where it is preheated and possibly partly evaporated by the heat from the poor solution before it enters the generator. In the generator the rich solution is partly evaporated by the high temperature heat source. The two-phase flow leaves the generator and enters the liquid separator (LS) where it separates into liquid (poor solution) and vapour phase (refrigerant). The poor solution flows through the solution heat exchanger via the solution throttle to the absorber. In the solution throttle it expands to the low pressure level. The refrigerant flows in the rectification column upwards through the stages in counter flow to the liquid fraction (condensed refrigerant). The vapour leaving the rectification column on the top flows to the dephlegmator where it is partly condensed. The condensate flows back to the rectification column and the purified vapour streams to the condenser where it is totally condensed by rejecting the heat to the cooling water. After the condenser the refrigerant flows via the refrigerant accumulator (RAC) to the condensate pre-cooler where it is cooled by the refrigerant leaving the evaporator. After the pre-cooler the refrigerant expands through the refrigerant throttle to the low pressure level and enters the two-phase region and flows further to the evaporator. Receiving the heat from the low temperature heat source the refrigerant evaporates either partly or totally and flows to the condensate pre-cooler where the refrigerant is further evaporated or superheated and flows to the absorber. At the absorber inlet the refrigerant vapour and the poor solution are mixed in the mixing point (MP). The mixture flows further through the absorber where the refrigerant is absorbed and the heat is rejected to the cooling water.

On the right hand side of Figure 1 the generator section of the counter-current generator design is shown. It is assumed that the liquid and the vapour flowing in counter-current flow in the generator are in thermodynamic equilibrium. Except the generator no other changes have been assumed for the AHP-process.

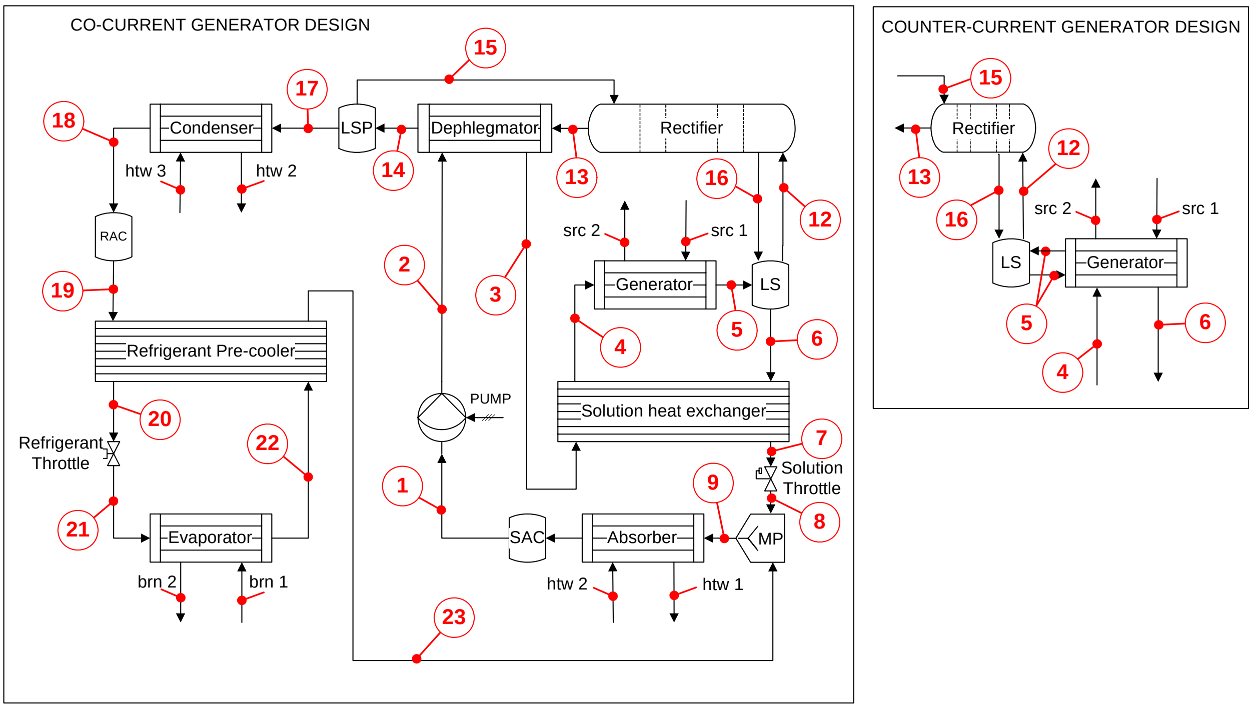

Figure 1 Design and State points of the absorption heat pump cycle for the co-current generator design (left) and the generator section of the counter-current generator design (right)

3. THERMODYNAMIC MODELS

Two different mathematical models have been developed using the software package EES (2006) in order to study the effects of component behaviors, to perform calculation of variations by varying the external heat source and heat sink temperatures and to compare the results to the experimental results.

3.1. Fluid Property Data

The thermodynamic properties of ammonia/water mixtures were calculated using the procedure „NH3H2O“ (Ibrahim and Klein, 1993), which is implemented in EES. The external circuits are the cold water circuit for the low temperature heat source, the cooling water circuit for the medium temperature heat sink and the heating water circuit for the high temperature heat source. For the external circuits the specific heat capacities are calculated from T_1 to T_2 (at constant pressure) using Eq. 1, the enthalpy differences are calculated using Eq. 2 and the entropy differences are calculated acc. to Eq. 3 (Lüdecke and Lüdecke, 2000).

$$c_p = a \cdot T^2 + b \cdot T + c \tag{1}$$

$$\Delta h^{1\rightarrow 2} = \frac{a}{3} \cdot \left(T_2^3 - T_1^3\right) + \frac{b}{2} \cdot \left(T_2^2 - T_1^2\right) + c \cdot \left(T_2 - T_1\right) \tag{2}$$

$$\Delta s^{1\rightarrow 2} = \frac{a}{2} \cdot \left(T_2^2 - T_1^2\right) + b \cdot \left(T_2 - T_1\right) + c \cdot \ln\left(\frac{T_2}{T_1}\right) \tag{3}$$

The used coefficients are given in Table 1. The values for the cold water circuit (brine, $50\%_{Mass}$ Glysantine and $50\%_{Mass}$ Water) are derived from BASF (2001). The values for the cooling water have been determined using the EES-procedure “Water”. For the heating water circuit thermal oil has been considered and the specific heat capacity has been derived from VDI Wärmeatlas (1997) for “BP Transcal N”.

Table 1 Coefficients for the calculation of c_p, Δh and Δs

Description	a	b	c	validity [K]
Cold Water (brine)	-2.4673E-05	2.0468E-02	-0.5101	243 < T < 383
Cooling Water (water)	1.2505E-05	-7.9884E-03	5.4548	273 < T < 423
Heating Water (thermal oil)	0	3.4131E-03	0.9825	273 < T < 590

3.2. First law analysis

For the thermodynamic analysis of the process two steady state models for the co-current and counter current generator design have been set up. The models solve energy, overall mass and ammonia mass balances for all components. The effect of the rectification process on the water fraction of the refrigerant and on the dephlegmation capacity was calculated using the theoretical plate model (ponchon method) acc. to Herold and Pande (1996). For this two theoretical plates has been assumed. In order to compare the results to another, the coefficient of performance for cooling (COP_C) is calculated using Eq. 4. The external temperature lift - which is the temperature difference between the low temperature heat source and the medium temperature heat sink - is calculated using Eq. 5. $\Delta\xi$ means the concentration difference between the rich solution and the poor solution (Eq. 6). The internal heat exchanger efficiencies (ε) have been considered to be constant acc. to Eq. 7. Where $\dot{Q}_i$ means the actually transferred capacity and $\dot{Q}_{i,max}$ the maximum possible capacity determined by the inlet temperatures.

$$COP_C = \frac{\dot{Q}_{EVA}}{\dot{Q}_{GEN} + P_{PUMP}} \tag{4}$$

$$\Delta T_{LIFT} = \frac{T_{htw;1} + T_{htw;3}}{2} - \frac{T_{brn;1} + T_{brn;2}}{2} \tag{5}$$

$$\Delta\xi = \xi_{rso} - \xi_{pso} \tag{6}$$

$$\varepsilon = \frac{\dot{Q}_i}{\dot{Q}_{i,max}} = \text{const.} \tag{7}$$

In order to solve the system of equations different assumptions and the definition of boundary conditions for the external circuits are necessary. Table 2 summarizes these assumptions, which have mainly been derived and validated by the recalculation of the process using experimental results of the prototype application.

Beside the temperature differences the UA-value - which is the product of the coefficient of heat transfer and the heat transfer area - of the dephlegmator has been assumed to be constant and has been calculated using Eq. 8. Because of the fact that the specific heat capacity of the refrigerant along the dephlegmator is not constant the heat exchanger has been subdivided into 10 parts with the same capacity of transferred heat. For each part the specific heat capacity has been assumed to be constant and the thermodynamic mean temperature differences have been calculated using Eq. 9. The average value of these temperature differences has been calculated using Eq. 10 were N is the number of subdivided parts.

$$UA_{DEP} = \frac{\dot{Q}_{DEP}}{\Delta T_{LOG,DEP}} \tag{8}$$

$$\Delta T_{LOG,DEP,i} = \frac{(T_{ref,i} - T_{rso,i}) - (T_{ref,i+1} - T_{rso,i+1})}{\ln\left(\frac{(T_{ref,i} - T_{rso,i})}{(T_{ref,i+1} - T_{rso,i+1})}\right)} \tag{9}$$

$$\frac{1}{\Delta T_{LOG,DEP}} = \frac{1}{N} \cdot \sum_{i=1}^{N} \frac{1}{\Delta T_{LOG,DEP,i}} \tag{10}$$

Table 2 Assumptions and boundary conditions used for the process calculation

State point	Value	Unit	Description
Q_{GEN}	10	kW	Capacity of the generator
m_{brn}	0,35	kg/s	Cold water mass flow rate (brine)
m_{htw}	0,5	kg/s	Cooling water mass flow rate (water)
m_{src}	0,35	kg/s	Heating water mass flow rate (thermal oil)
t_E	17	°C	Temperature of the environment
ΔT_1	4	K	Temperature difference between t_1 and $t_{htw;2}$
ΔT_{1_sat}	2	K	Temperature difference between saturation temp. at point 1 and t_1
ΔT_{18}	4	K	Temperature difference between t_{18} and $t_{htw;3}$
ΔT_{21}	5	K	Temperature difference between $t_{brn;2}$ and t_{21}
ΔT_{GEN}	14	K	Logarithmic mean temperature difference in the generator
$\Delta T_{22\text{-}21}$	6	K	Temperature difference between t_{22} and t_{21}
$UA_{DEP,co\text{-}current}$	200	W/K	UA-value of the dephlegmator for the co-current generator design
$UA_{DEP,counter\text{-}current}$	100	W/K	UA-value of the dephlegmator for the counter-current generator design
$q_6, q_{15}, q_{16}, q_{18}$	0	-	Vapor fraction at state points 6, 15, 16 and 18
q_{12}, q_{13}, q_{17}	1	-	Vapor fraction at state points 12, 13 and 17
ε_{SHX}	0,88	-	Heat exchanger efficiency of solution heat exchanger
ε_{PRC}	0,5	-	Heat exchanger efficiency of refrigerant pre-cooler
η_{PUMP}	0,5	-	Isentropic efficiency of solution pump

3.3. Second law analysis

According to the first law of thermodynamics all forms of energy are convertible in one another. But different forms of energy have different potential to produce useful work, thus the second law of thermodynamics limits the possible energy conversion. The second law analysis of the absorption heat pump have been performed by determining the exergy losses of each process component acc. to Aphornratana and Eames (1995), the procedure is described hereafter.

A reversible process is one that can be reversed completely and after reversing no change remains in the system or its surroundings. All real processes are not reversible due to e.g. friction, heat transfer, hysteresis, mixing or diffusion processes. Exergy is defined as the maximum possible reversible work that can be obtained from a source with respect to the surrounding environment and is given for work in Eq. 11, for heat in Eq. 12, and for a mass flow in Eq. 13, if changes in the potential and kinetic energy can be neglected. Heat and work interaction can be defined in terms of exergy or exergy change (loss), which always decreases during real irreversible processes. For each process (component) the exergy loss can be calculated applying Eq. 14.

$$\dot{E}_P = P \tag{11}$$

$$\dot{E}_Q = \left(1 - \frac{T_{ENV}}{T}\right) \cdot \dot{Q} \tag{12}$$

$$\dot{E}_M = \dot{m} \cdot \left[h - h_E - T_E \cdot (s - s_E)\right] \tag{13}$$

$$\Delta\dot{E} = \sum \dot{E}_P + \sum \dot{E}_Q + \sum \dot{E}_M \tag{14}$$

In the following heat losses to the environment have been neglected and except in the solution pump no work is supplied, thus only the mass flows are to be considered acc. to Eq. 14. The exergy loss of the four main components absorber, generator, condenser, and evaporator can be divided into one part which is assigned the heat transfer between the external circuits and the internal process flows (this part can be influenced by the component size), and another part which is assigned to the internal irreversibilities, e.g. from mixing processes (this part can not be influenced by the component size).

In order to calculate the exergy losses due to the heat transfer it is necessary to calculate a theoretical temperature profile of the external heat sources and heat sinks where the minimum approach temperature in the component is zero. For this theoretical temperature profile the heat transfer area of the component has to be infinite. Figure 2 shows as an example the different temperature profiles in the condenser.

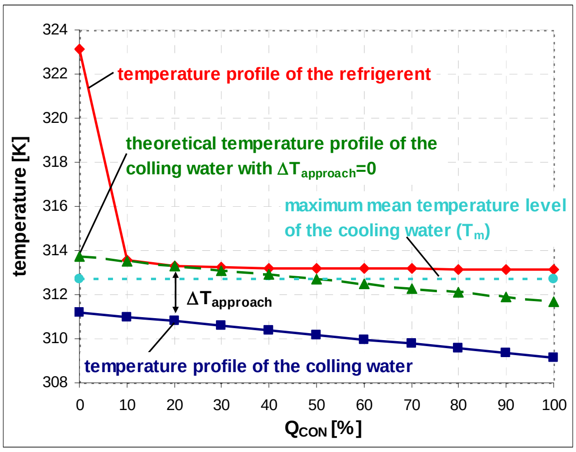

Figure 2 Temperature profile in the condenser and illustration of the minimum approach temperature and the theoretical temperature level of the cooling water

Assuming constant specific heat capacities of the external fluids the thermodynamic mean temperature level of the theoretical temperature profile can be calculated as shown in Eq. 15. Depending whether the heat is supplied to, or rejected from the absorption heat pumping process by the component, the exergy loss due to the heat transfer is calculated using the Eq. 16 for the generator & evaporator, and using Eq. 17 for the absorber & condenser. The exergy loss due to internal irreversibilities is the difference between the total exergy loss of the component and the exergy loss due to the heat transfer.

$$T_m = \frac{T_{in} - T_{out}}{\ln\left(\frac{T_{in}}{T_{out}}\right)} \tag{15}$$

$$\Delta\dot{E}_{HT} = \Delta\dot{E}_{ext;in} - \Delta\dot{E}_{ext;out} - \dot{Q} \cdot \left(1 - \frac{T_E}{T_m}\right) \tag{16}$$

$$\Delta\dot{E}_{HT} = \Delta\dot{E}_{ext;in} - \Delta\dot{E}_{ext;out} + \dot{Q} \cdot \left(1 - \frac{T_E}{T_m}\right) \tag{17}$$

4. RESULTS AND DISCUSSION

Figure 3 shows the comparison of some results for the COP_C of measured values of the prototype application and of the thermodynamic model (with t_{21} = -10°C, compare Fig. 1, and $\Delta\xi$ = 0.1) for the co-current generator. The COP_C decreases with increasing temperature lift approximately linearly from about 0.7 at 15 K down to approximately 0.4 at 45 K temperature lift. The thermodynamic model shows a similar characteristic but with approximately 0.05 higher COP_C values compared to the measurements.

Figure 4 shows one result for the COP_C depending on the generator outlet temperature of the poor solution (t_6) for four different load cases with evaporation/condensation temperatures of 10/30, 0/30, 10/50 and -10/50 °C and varying $\Delta\xi$ starting at 0.05 up to 0.15. The temperature of the poor solution at the generator outlet increases from about 70°C to over 170°C and the COP_C decreases from about 0.75 to about 0.4 for the 4 load cases. The deviation of the results between the co-current and counter-current generator (compare Fig. 1) increases with increasing generator temperature whereas the difference is small for a small $\Delta\xi$ and becomes higher with increasing $\Delta\xi$. The results for the co-current generator show a maximum of the COP_C at $\Delta\xi$ = approx. 0.1 for all load cases while for the counter-current generator the COP_C has a maximum at $\Delta\xi > 0.15$.

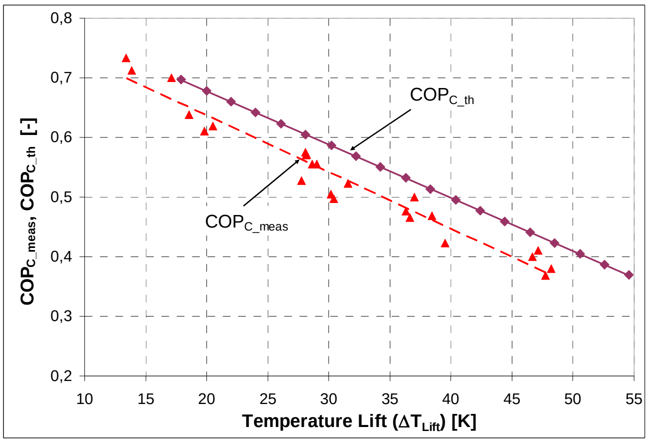

Figure 3 Comparison of the COP_C of the measurements and of the thermodynamic model for the co-current generator design depending on the temperature lift.

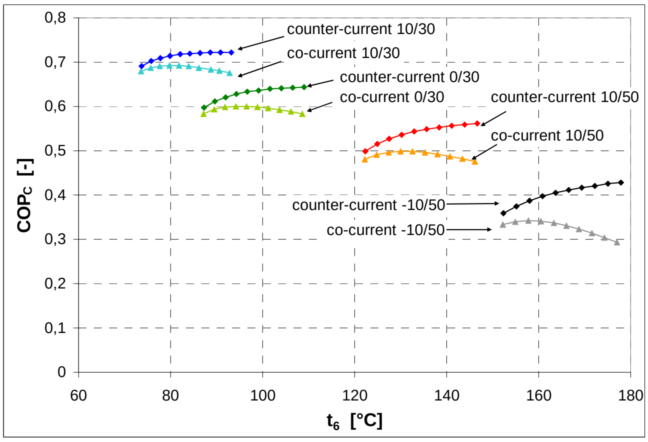

Figure 4 COP_C depending on t_6 for the co-current and counter current flow generator design for four load cases with evap./cond. temp. of 10/30, 0/30, 10/50 and -10/50 °C and varying $\Delta\xi$ = 0,05 to 0,15.

In order to gain a more detailed insight to the origin of the process losses the exergy loss of each process component has been calculated for both generator designs. Figure 5 shows the simulation results for an evaporation temperature of -10°C and varying condensation temperature (10 up to 50°C) for co-current generator design (left) and counter-current generator design (right). The total exergy loss of the AHP increases for both designs with increasing temperature lift. At a small temperature lift (10 K) both designs have a total exergy loss of about 1.32 kW. At a large temperature lift (50 K) the AHP with the co-current generator design has a total exergy loss of about 1.95 kW and the counter-current generator design of about 1.83 kW.

Comparing the component losses of the two designs to each other it can be seen that - except the dephlegmator, rectification column and solution heat exchanger - the component losses are very similar. Particular the exergy losses of the dephlegmator and the rectification are significantly higher at high temperature lift for the co-current design. This is mainly caused by the higher temperature and water content of the refrigerant vapour entering the rectification column for the co-current design. Thus also the capacity of the dephlegmator is significantly higher for the co-current design. The exergy loss of the solution heat exchanger is lower for the co-current design. Because of the preheating of the rich solution in the dephlegmator the solution heat exchanger has a higher capacity for the counter current design, which leads to higher exergy loss.

The exergy losses in the "main" components - which are the absorber, generator, condenser and evaporator - have been divided into one part which originates from the heat transfer and one part which is caused by internal losses. The absorber causes approx. 30% of the total exergy loss at low and high temperature lifts whereas the internal losses dominate. This is mainly caused by the much higher gradient of the temperature profile in the process flow compared to the gradient of the temperature profile in the cooling water flow. This means that increasing the absorber component size – which affects the exergy losses due to heat transfer - would have a minor effect on the total exergy losses. Lowering the cooling water mass flow rate would decrease the internal exergy losses in order to achieve similar gradients of the temperature profiles in the component. However, this would also increase the mean temperature level of the absorber and influences the losses of other components which maybe decrease the system efficiency.

The generator causes approx. 30% of the total exergy loss of the absorption heat pump at a low temperature lift and approx. 10% at a high temperature lift. Due to the fact that the gradients of the temperature profiles of the rich solution and the thermal oil in the generator are similar to each other, the exergy losses are mainly caused by the heat transfer and can be decreased be increasing the component size or improving the heat transfer. In the evaporator and condenser the absolute value of the exergy losses decrease with increasing temperature lift which is mainly caused by the decreasing component capacity.

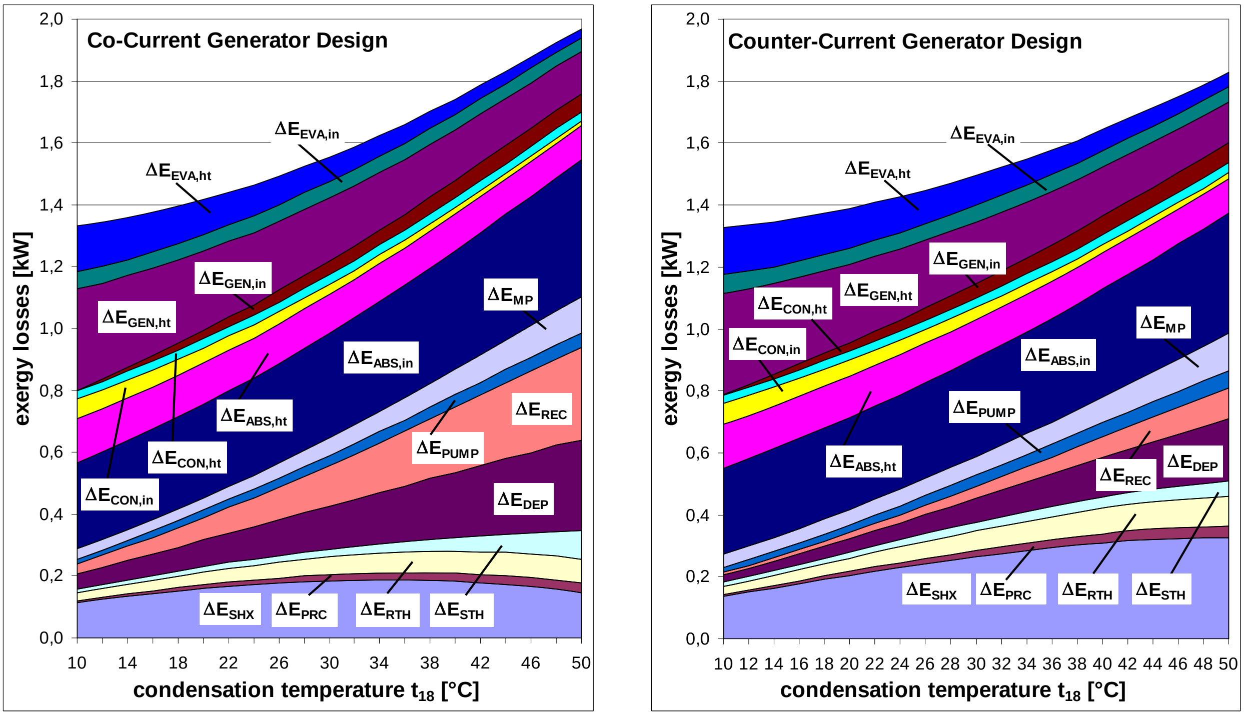

Figure 5 Exergy losses for an evaporation temperature of -10°C and varying condensation temperature (10 up to 50°C) for $\Delta\xi = 0.1$

5. SUMMARY

A thermodynamic analysis of two single stage NH_3/H_2O absorption heat pumping units with a co-current and a counter-current generator design has been presented. As expected, the models show higher efficiencies for counter-current generator design. The deviation between co-current and counter-current generator design increases with increasing generator temperature and concentration difference between rich and poor solution. In order to determine the magnitude of the irreversibilities which occur in the different components an exergy analysis has been performed. The results show, that due to the increased rectification and dephlegmation effort the exergy losses of these components are significantly higher at the co-current generator design, particular at high temperature lifts. In order to minimize the exergy losses the improvement of the heat transfer in the generator and solution heat exchanger seems to be most promising. Even though the absorber causes approx. 30% of the total exergy losses it has a lower potential for improvement, because only a small part of the component exergy loss can be affected by the heat transfer.

Acknowledgement
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NOMENCLATURE

c_p	Specific heat capacity	[kJ/(kg K)]
$\dot{E}$	Exergy	[kW]
ε	Efficiency	[-]
h	Specific enthalpy	[kJ/kg]
$\dot{m}$	Mass flow rate	[kg/s]
p	Pressure	[kPa]
P	Mechanical power	[kW]
$\dot{Q}$	Thermal capacity	[kW]
q	Vapour fraction	[-]
s	Specific entropy	[kJ/(kg K)]
t	Temperature	[°C]
T	Temperature	[K]
ξ	Ammonia mass fraction	[-]

Subscripts

E	environment
ext	external
i	index
in	inlet
meas	measurement based model
out	outlet
th	theoretical model

brn	Cold water (brine)
htw	Cooling water (water)
pso	Poor solution
ref	Refrigerant
rso	Rich solution
src	Heating water (thermal oil)

ABS	Absorber
CON	Condenser
DEP	Dephlegmator
EVA	Evaporator
GEN	Generator
LS	Liquid separator downstream GEN
LSP	Liquid separator downstream DEP
MP	Mixing point upstream ABS
PUMP	Solution pump
PRC	Condensate precooler
RAC	Refrigerant accumulator
REC	Rectification column
RTH	Refrigerant throttle
SAC	Solution accumulator
SHX	Solution heat exchanger
STH	Solution throttle

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ANHANG IWT 4

Moser H., Podesser E. (2009) “Hygienic Aspect of Small Wet Cooling Towers”, IEA SHC Task38 Technical report of subtask C

Task 38
Solar Air-Conditioning
and Refrigeration

Hygienic Aspect of Small Wet Cooling Towers

A technical report of subtask C

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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).

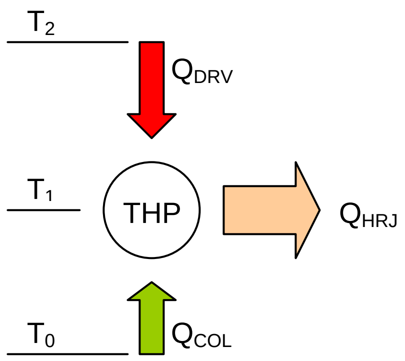


Figure 1-1: Sketch of heat flows and temperature levels for a thermally driven heat pump

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)

$$COP_C = \frac{Q_{COL}}{Q_{DRV}} \qquad \textit{Eq. 1}$$

$$\frac{Q_{HRJ}}{Q_{COL}} = 1 + \frac{1}{COP_C} \qquad \textit{Eq. 2}$$

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.

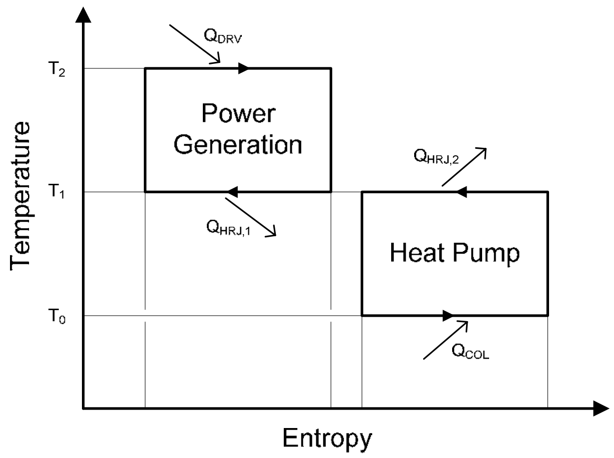


Figure 1-2: *Carnot cycles of a thermally driven heat pump*

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 $T_2 = 80°C$ and of the cooling load $T_0 = 5°C$ 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°C. 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°C and decreases below 0.4 for 40°C.

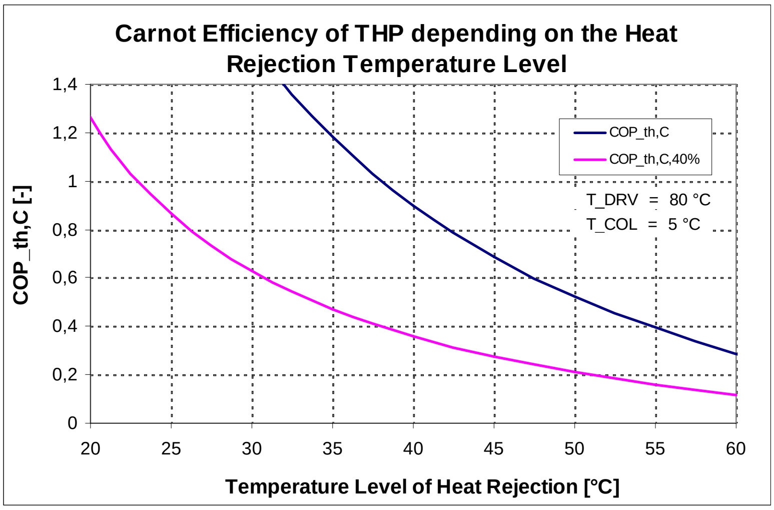


Figure 1-3: Carnot efficiency for different temperature levels of the heat rejection system (T_{DRV}=80°C, T_{COL}=5°C)

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.

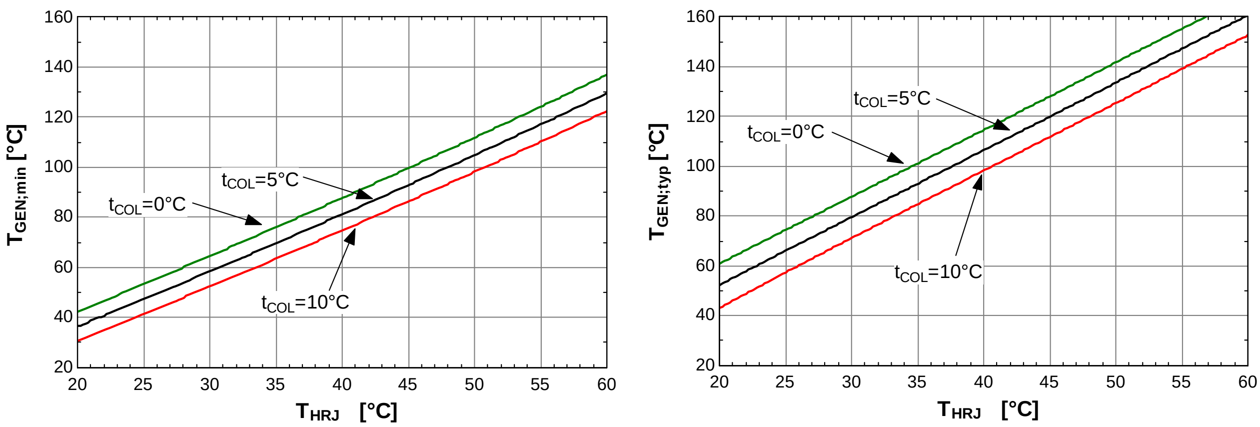


Figure 1-4: Thermodynamic calculation of the minimum (left) and a typical (right) temperature level of driving heat depending on heat rejection temperature level.

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°C and a heat rejection temperature of 30°C the minimum temperature level of the driving heat is at approx. 60°C. If the heat rejection temperature increases to 50°C the minimum driving temperature level has to be increased to approx. 105°C 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.

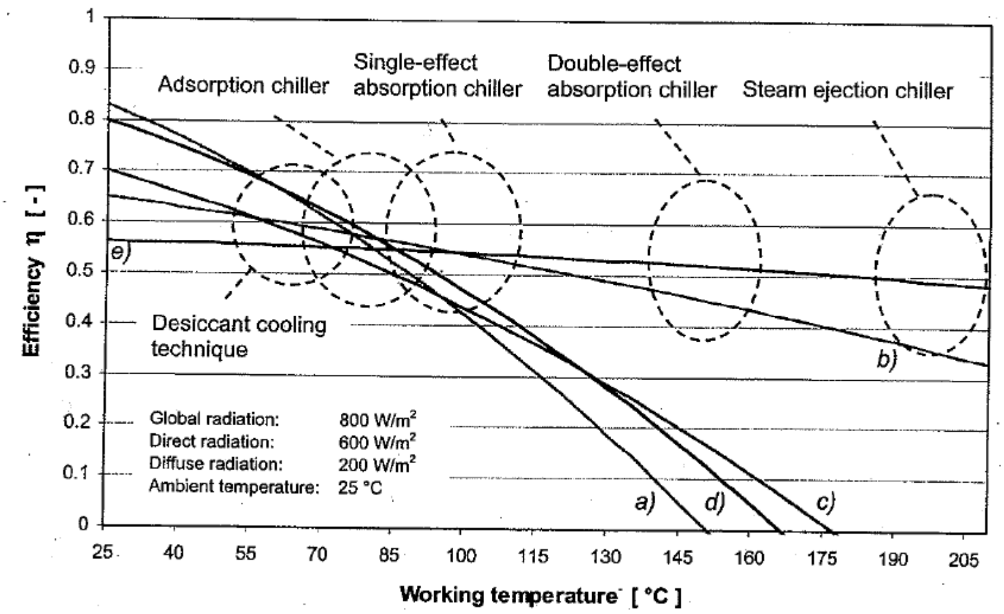


Figure 3: Comparison of different collectors at 800 W/m². a) Single glazed flat-plate with AR, b) Evacuated tube collector of the Sydney type, c) CPC flat-plate with Teflon foil, d) Flat-plate with double AR-glazing and inert gas filling, e) Small parabolic trough (under development; only the fraction 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 (ΔΔt) 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.

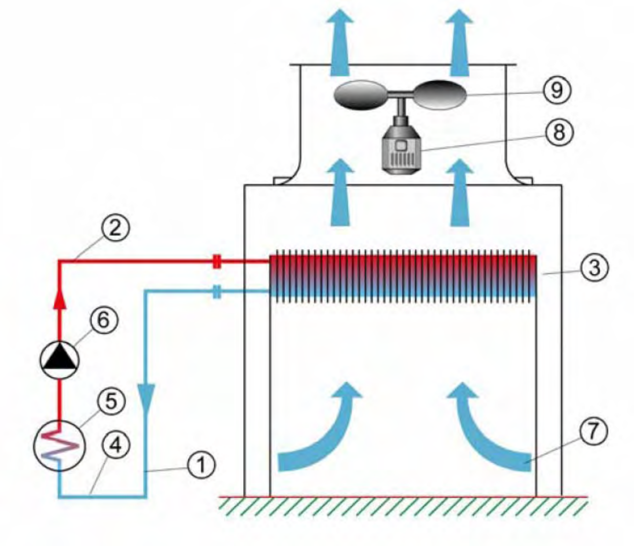
1.3.1 Dry Cooler

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.

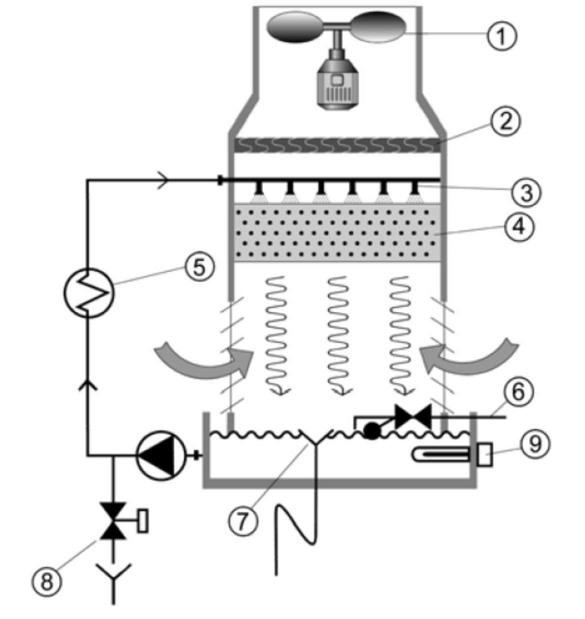
1.3.2 Wet Cooling Towers

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

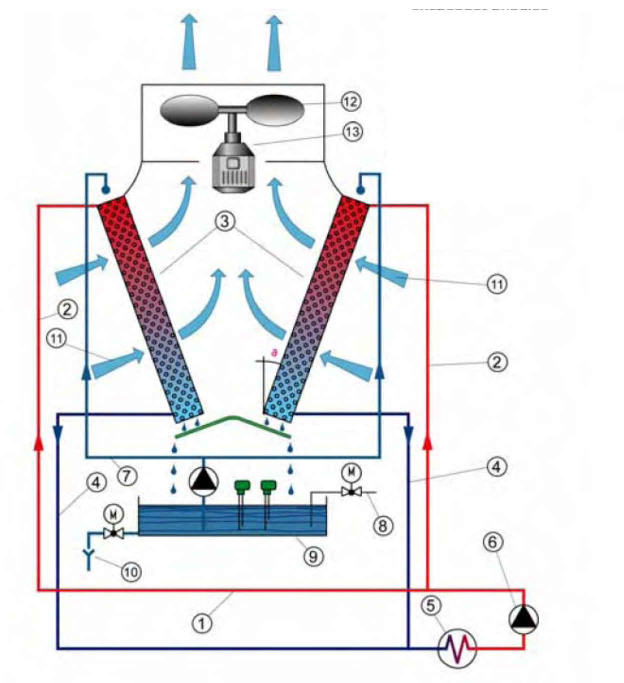
Figure 1-7: Sketch of a wet cooling tower (Jaeggi, http://www.guentner.ch/pdfs/Evaluation of Aircooled Cooling Systems.pdf, 26.03.2009)

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.

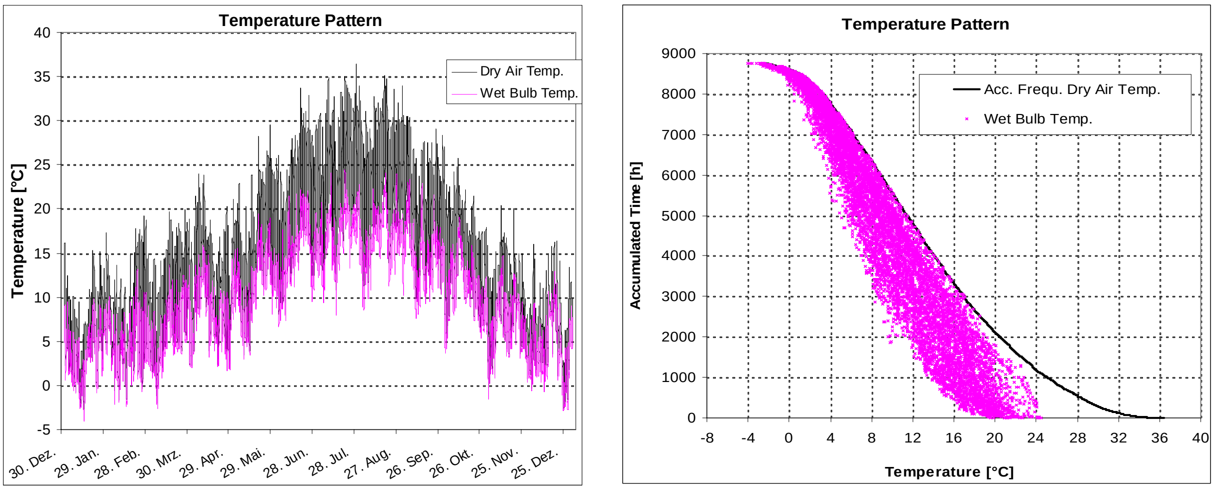


Figure 1-9: Dry and wet bulb temperature on hourly basis for one year of Frankfurt (left) and annual temperature distribution curve for dry bulb temperature with dedicated wet bulb temperatures (right).

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.

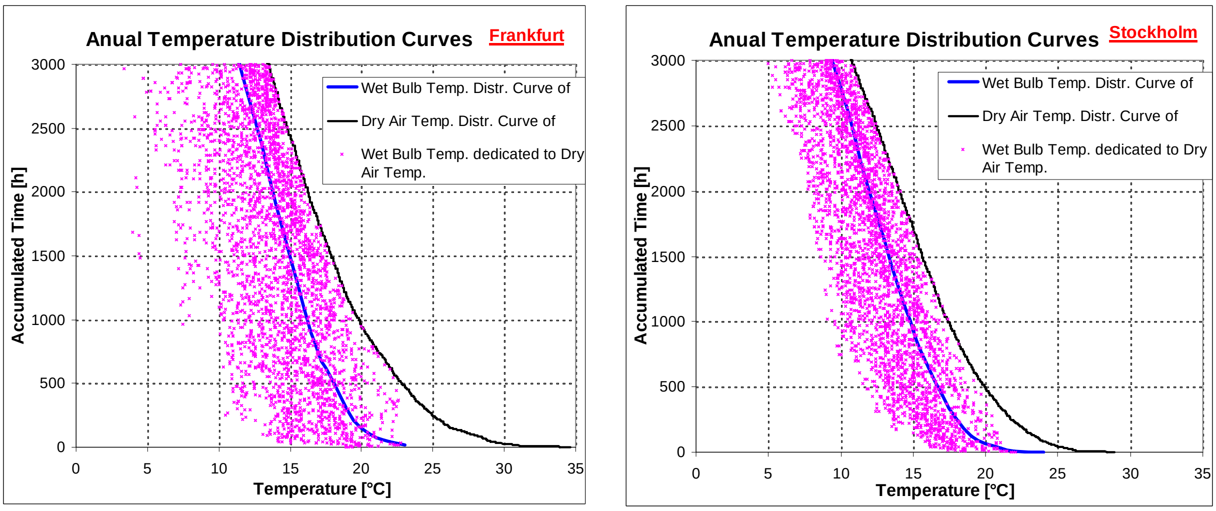


Figure 1-10: Annual dry air temperature distribution curve with dedicated wet bulb temperatures and annual wet bulb temperature distribution curve for Frankfurt (left) and Stockholm (right)

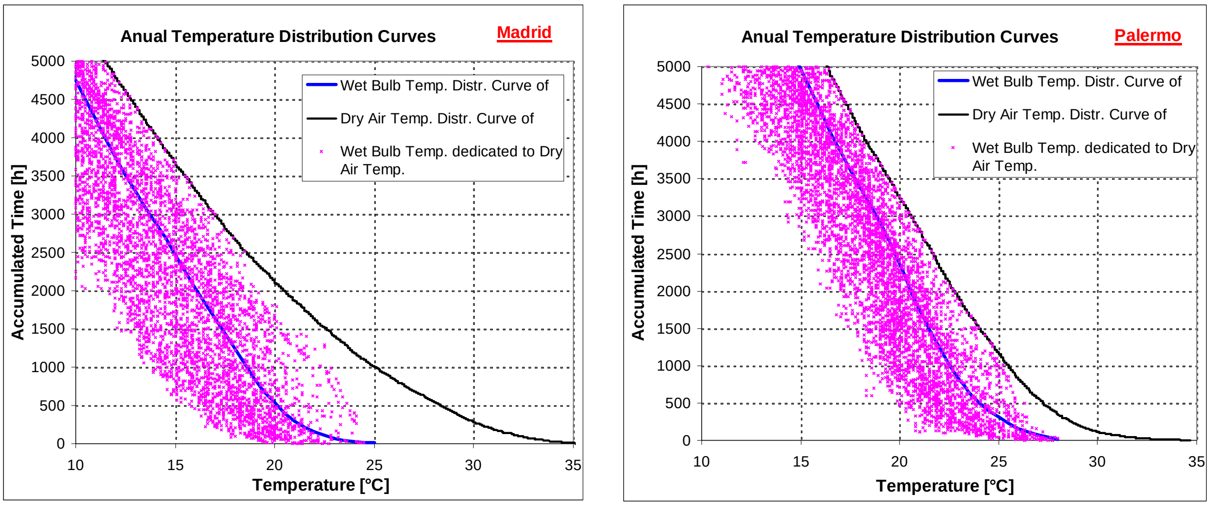


Figure 1-11: Annual dry air temperature distribution curve with dedicated wet bulb temperatures and annual wet bulb temperature distribution curve for Madrid (left) and Palermo (right)

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.

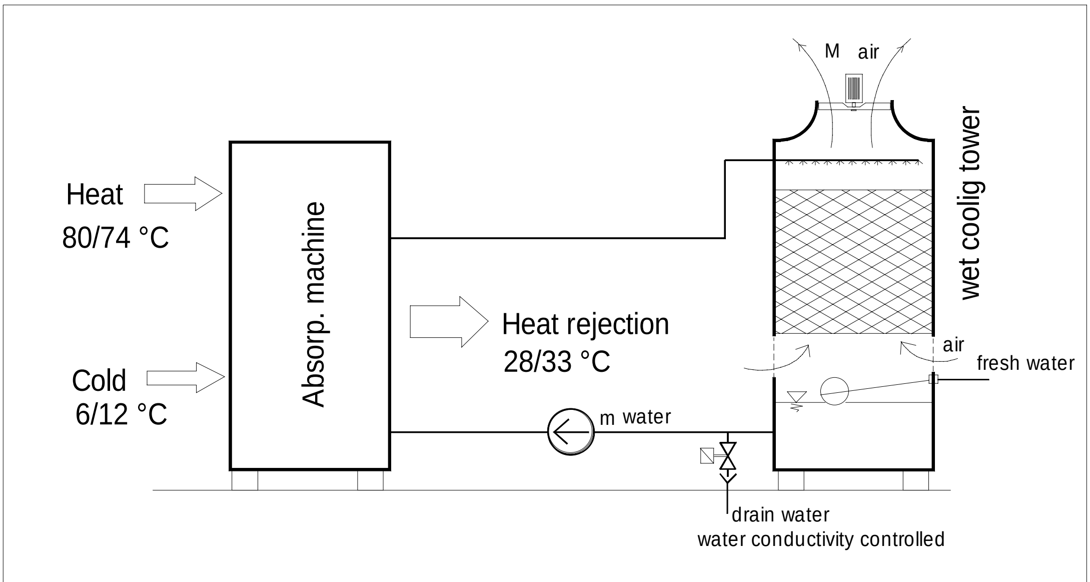


Figure 3-1: Principle of a thermal driven cooling with wet heat rejection

3.2 State of Air and Temperatures Inside the Cooling Tower

The following equation describes the heat balance at the wet cooling tower.

$$m * c * (\vartheta w1 - \vartheta w2) = M * (h1 - h2) \tag{3-1}$$

<u>Legend</u>

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.

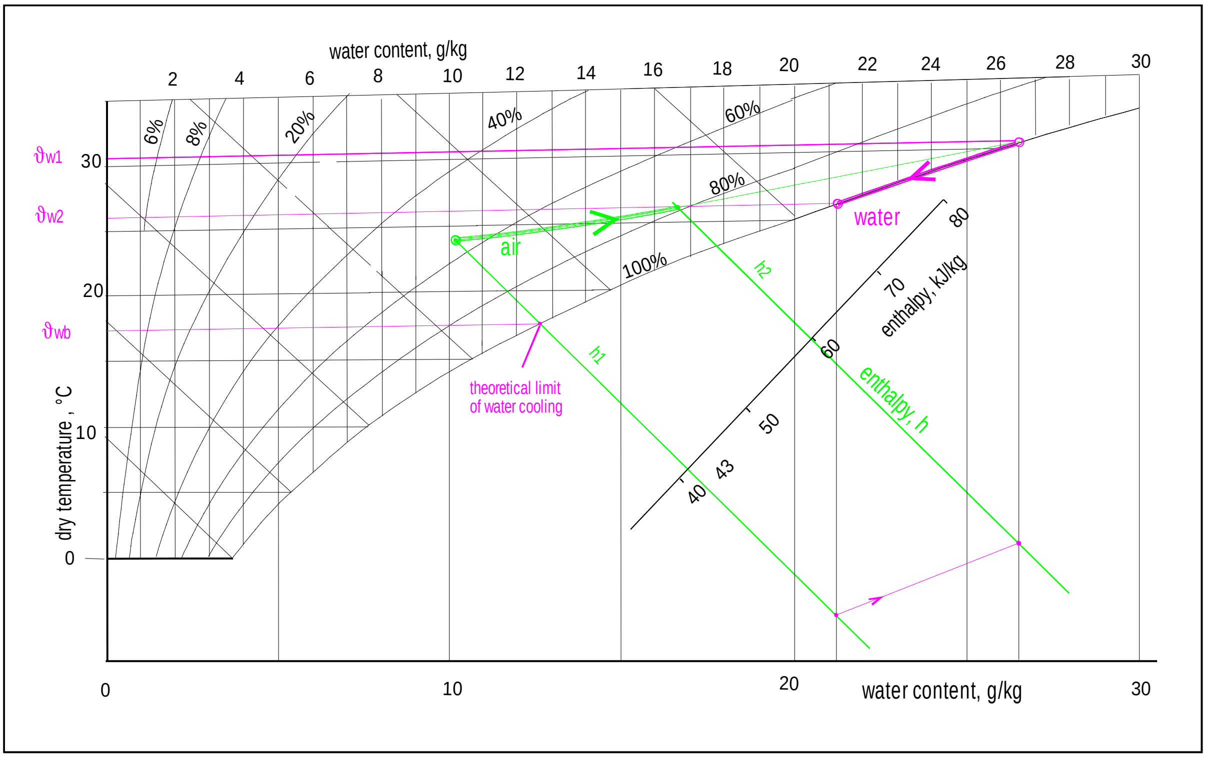


Figure 3-2: States of the air and temperatures of the cooling water inside of the cooling tower

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.

$$\eta = \frac{\vartheta w1 - \vartheta w2}{\vartheta w1 - \vartheta wb} \tag{3.2}$$

η ………… water cooling gradation …………………… -

ϑwb ……..wet bulb temperature ………………………°C

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 $\vartheta\ w1 = 33$ °C, 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 °C. With the aid of the wet bulb temperature and Figure 3-3 now the relative minimum quantity of air can be found.

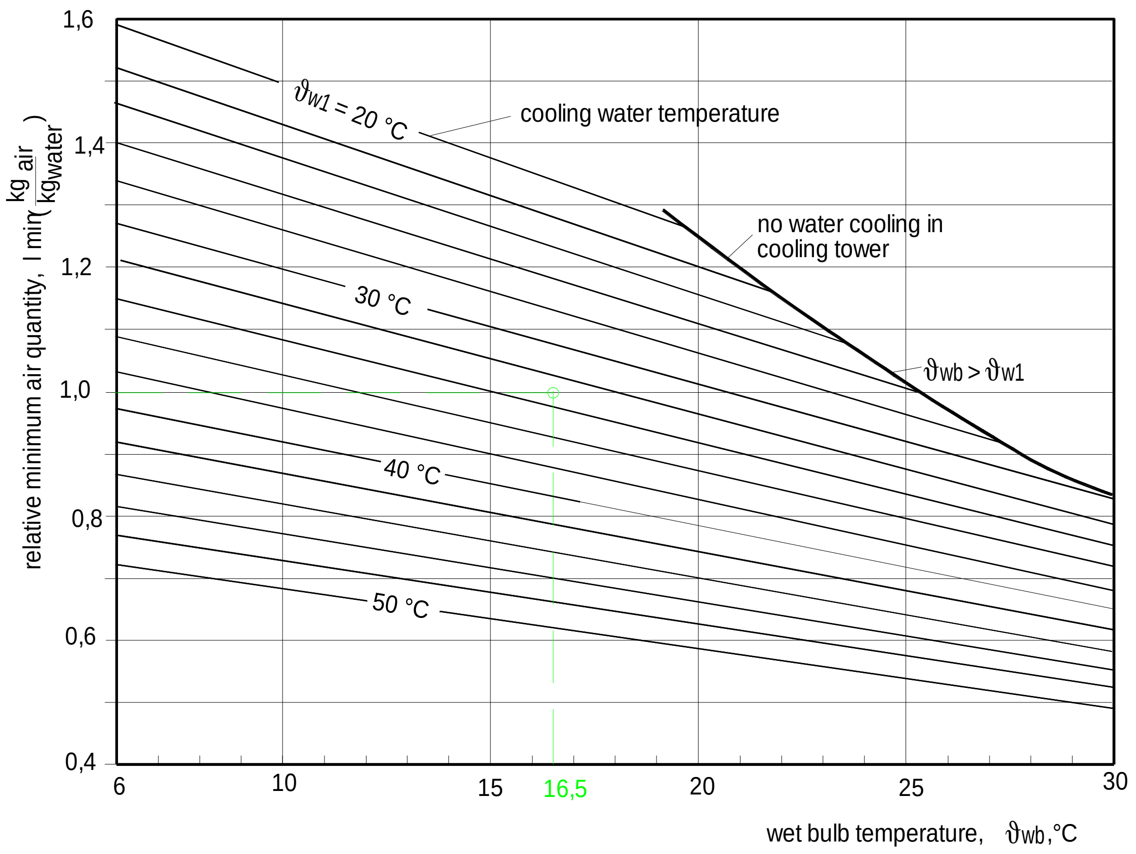


Figure 3-3: Relative minimum quantity of air (Recknagel Sprenger, 1997/98)

The relative minimum air quantity is determined with l_{min} = 1,0

For the further calculations the equations below are used:

$$l_{\min} = \frac{M_{\min}}{m} \tag{3.3}$$

$$l_0 = \frac{M}{m} \tag{3.4}$$

$$\lambda = \frac{l_0}{l_{\min}} \tag{3.5}$$

$l_{\min}$ ……………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

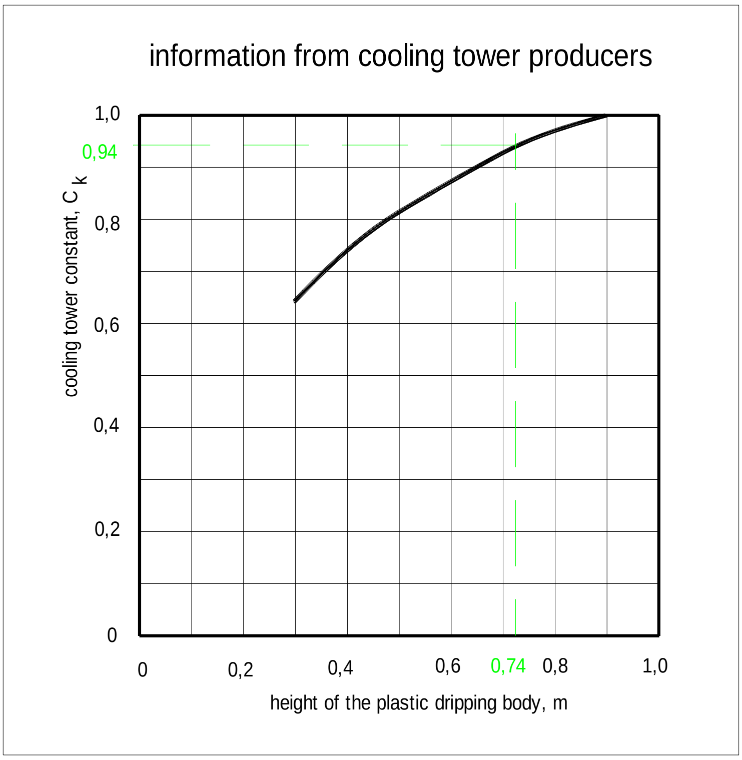


Figure 3-4: Cooling tower constant as a function of the height of the dripping body (on the basis of Recknagel Sprenger, 1997/98)

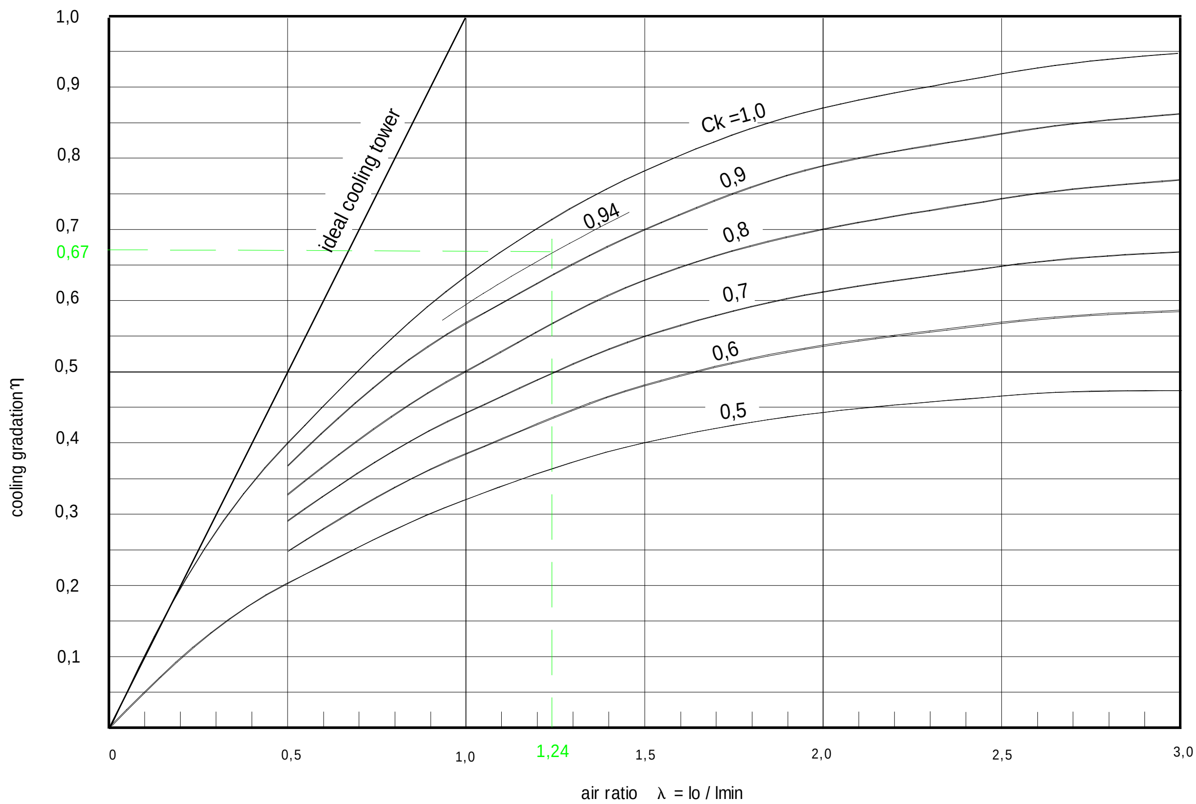


Figure 3-5: Cooling gradation η as a function of the air ratio with the parameter of the cooling tower constant (Recknagel Sprenger, 1997/98)

3.3 Example

Cooling water:	ϑ_{w1} = 33 °C
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 °C
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°C Disinfection range
- 60 °C 90% of Legionella die within 2 minutes
- 20 - 50°C Legionella growth range
- 35 - 46°C Ideal growth range
- < 20 °C 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.

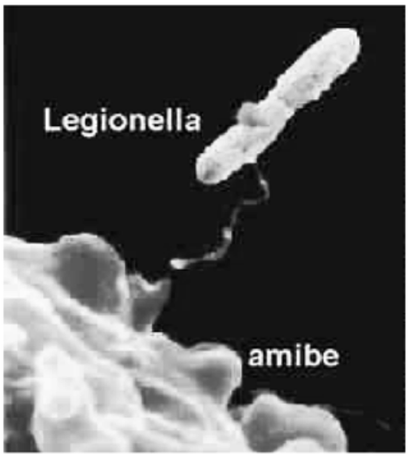


Figure 4-1 Picture of a Legionella growing on a amoeba (Aquaprox, 2007)

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)

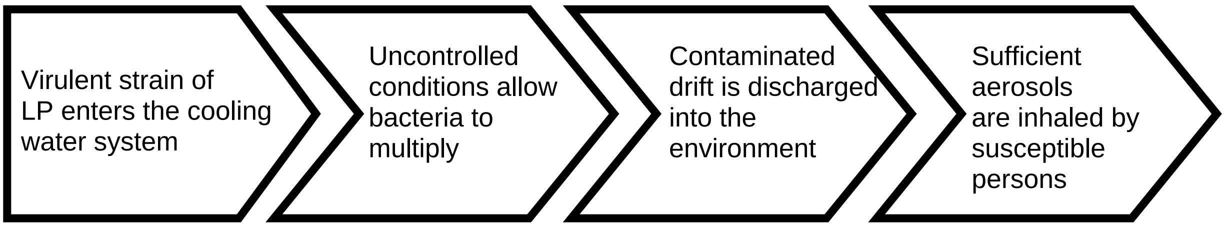


Figure 5-1: Chain of Events for the outbreak of Legionnaires' Disease associated with a cooling tower (EUROVENT 9/5, 2002))

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.

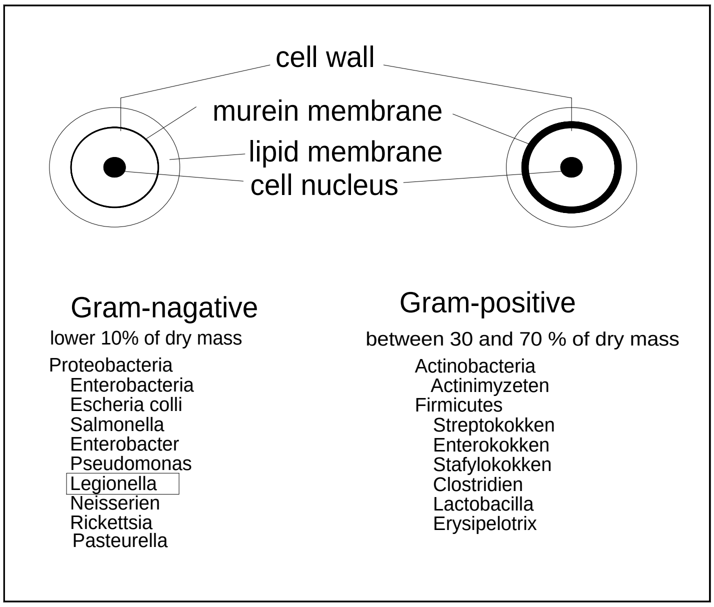


Figure 5-2: Principle construction of Gram positive and Gram negative bacteria.

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)

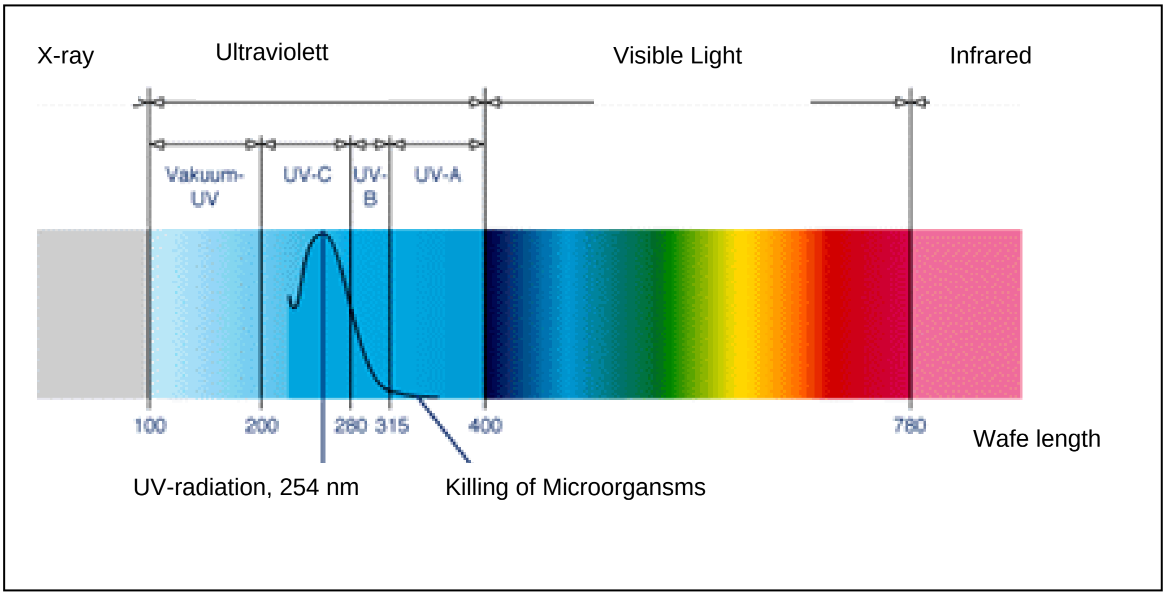


Figure 5-3: Position of the UV-C radiation in the frequency band from 100 to 750 nm

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.

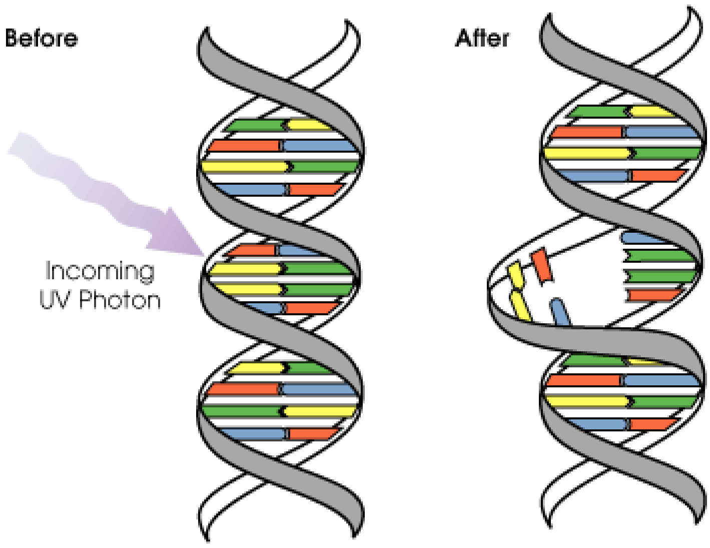


Figure 5-4: Schematic shape of the Nuclein Acid (DNA) before and after radiation treatment (http://waterecotechnology.com/?page_id=414, 09.12.2009)

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.

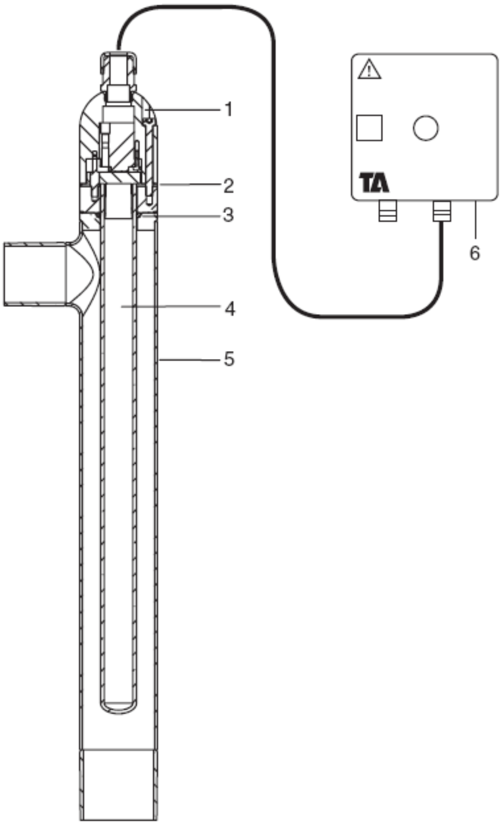


Figure 5-6: UV-C disinfection facility for integration in the installation (http://www.hydroteam.gr/pdf/TA-Aqua.pdf, 09.12.2009)

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, which 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 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.

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.

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Unified Monitoring Procedure and Performance Assessment for Solar Assisted Heating and Cooling Systems

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Abstract

In the present paper a monitoring procedure for Solar assisted Heating and Cooling (SHC) systems is presented. The procedure is subdivided in three different levels of complexity, allowing a first evaluation with a limited set of sensors or a precise evaluation with a full monitoring system. The procedure has been elaborated within a project of the International Energy Agency. The application of the procedure should allow a direct comparison of the functioning of different solar cooling systems and of simulation results. Further information can be found as well on the related task homepage [5].

Keywords: Solar heating and cooling, monitoring procedure, performance assessment

1. Introduction

Monitoring of installed solar assisted cooling systems represents a fundamental tool in order not only to optimise the monitored system it self, but as well to draw conclusions for the optimisation of the 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 assisted cooling systems. In fact today, to the knowledge of the authors, only around 300 solar assisted cooling systems are installed world wide, and most of these show different designs and surrounding conditions [1][2][3].

In order to allow a clear comparison between monitoring results of different installed systems, and as well between measured and simulated values, a comprehensive and unified monitoring procedure is required.

The purpose of the present work is the presentation of such a monitoring procedure which has been developed within the frame of the international perennial years project – Task 38 – under the umbrella of the International Energy Agency (IEA) Solar Heating and Cooling Program [5]. Within Task 38 in the years 2008 and 2009 14 small scale (< 20 kW cooling capacity) and 12 large scale systems are planned to be monitored. The procedure should not be restricted to the IEA activities and the mentioned systems but support future monitoring activities in general.

For the elaboration of the monitoring procedure several boundary conditions have been considered. On one hand the procedure defines minimum requirements to be respected by all running monitoring projects, on the other hand detailed data and single energy flows should be measured where feasible in order to acquire a possibly complete picture of the functioning and to allow detailed analysis. The procedure should allow a comparison between different systems and as well permit to draw (with the results) a learning curve over the coming years on the cost development of installed solar assisted heating and cooling systems.

In order to respect these requirements a three step monitoring procedure will be presented in the following, differing in level of detail, necessary hardware, measured data and complexity. DEC systems are included only in the third level calculation

2. Monitoring procedure

In order to have a common starting point for the single monitoring levels a basic scheme and a reference system had to be defined. In the following Figure 1 and Figure 2 the proposed reference solar assisted heating and cooling (SHC) system including the detailed energy fluxes for full monitoring and the conventional reference system are shown:

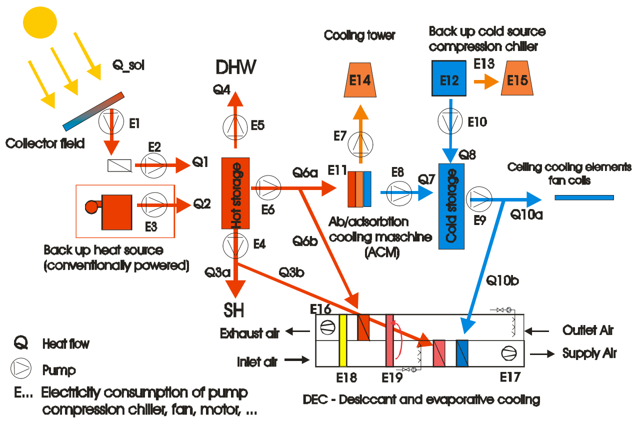

Figure 1. Proposed reference solar heating and cooling system including the single energy fluxes (SHC Max System)

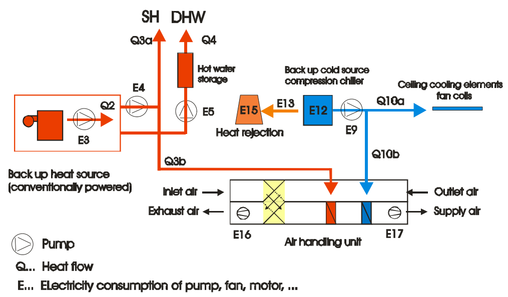

Figure 2. Proposed schematics of the conventional reference systems including the single energy fluxes

2.1. First level

The first level of the procedure permits to acquire basic information on cost and performance (on primary energy level) of the system with a limited number of sensors. Within this level 4 heat flow meters and the electricity counters for the measurement of the electricity consumption of the overall system is required. In Figure 3 the scheme with the measured energy fluxes is shown.

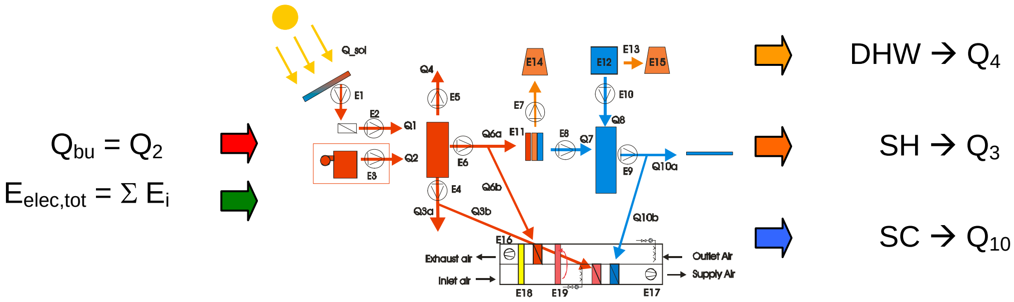

Figure 3. 1st level monitoring scheme including measured inputs and outputs of the solar assisted cooling system

The primary energy ratio of the solar assisted cooling system can be calculated as shown in (Equation 1:

$$PER = \frac{(Q_3 + Q_4 + Q_{10})}{\dfrac{Q_2}{\varepsilon_{fossil} \cdot \eta_{boiler}} + \dfrac{E_{elec,\,tot}}{\varepsilon_{elec}}} \qquad \text{(Equation 1)}$$

Where the heat and electricity fluxes are measured while the primary energy conversion factors for heat and electricity from fossil fuels have been set with the following values, being based on European Directives [4], Task 25 and Task 32 [7]:

- ε_{elec} = 0.4 (kWh of electricity per kWh of primary energy)
- ε_{fossil} = 0.9 (kWh of heat per kWh of primary energy)
- η_{boiler} = 0.95 (boiler efficiency)

In general it has to been stated that the conversion and performance factors given in the present paper are based on literature and discussion agreements but have to be considered only as proposals for calculation and comparison as they depend on different aspects such us country specifications, technology and component size. The Primary Energy Ratio of a reference system (see Figure 2) can be calculated as shown in (Equation 2:

$$PER_{ref} = \frac{Q_3 + Q_4 + Q_{10}}{\dfrac{(Q_3 + Q_4)}{\varepsilon_{fossil} \cdot \eta_{boiler,ref}} + \dfrac{Q_{10}}{SPF_{ref} \cdot \varepsilon_{elec}} + \dfrac{E_{elec,\,tot\_ref}}{\varepsilon_{elec}}} \qquad \text{(Equation 2)}$$

In this equation the heat and electricity fluxes are again the measured values. Within the calculation of the electricity consumption of the reference system the consumption of the pumps of solar circuit loop and of the absorption chiller have to be subtracted according to Figure 2 and as shown in (Equation 3.

$$E_{elec,\,tot\_ref} = E_{elec,\,tot} - (E_1 + E_2 + E_6 + E_7 + E_8 + E_{10} + E_{11} + E_{14} + E_{18} + E_{19}) \qquad \text{(Equation 3)}$$

The electricity consumption of the auxiliary E3 has to be corrected as described with (Equation 15, since the auxiliary in the reference has to deliver much more heat. The primary energy conversion factors for heat and electricity from fossil fuels have been listed before, the Seasonal Performance Factor (SPF) of the reference compression chiller has been set to:

- SPF_{ref} = 2.8 (compression chiller efficiency of the reference system)
- $\eta_{boiler,\,ref}$ = 0.95 (reference boiler efficiency)

From the financial point of view the overall cost per installed cooling capacity can be calculated following

Specific solar assisted cooling installation cost = $\dfrac{Cost(€)}{kW_{cold}}$ (Equation 4)

Cost (€) includes the costs of all components shown in Figure 1 minus the appliances deployed in the corresponding conventional reference system (Figure 2) such us back up heating / cooling system and eventually installed cogeneration systems.

2.2. Second level

The second step consists within deeper partial monitoring of single parts of the system with an increased number of sensors in respect to the 1st level. In fact within this level 2 heat counters and a pyranometer have to be added to the 4 heat counters and the total electricity counters already present in the 1st level. The monitoring for this level is concentrating mainly on the solar thermal energy management (see Figure 4).

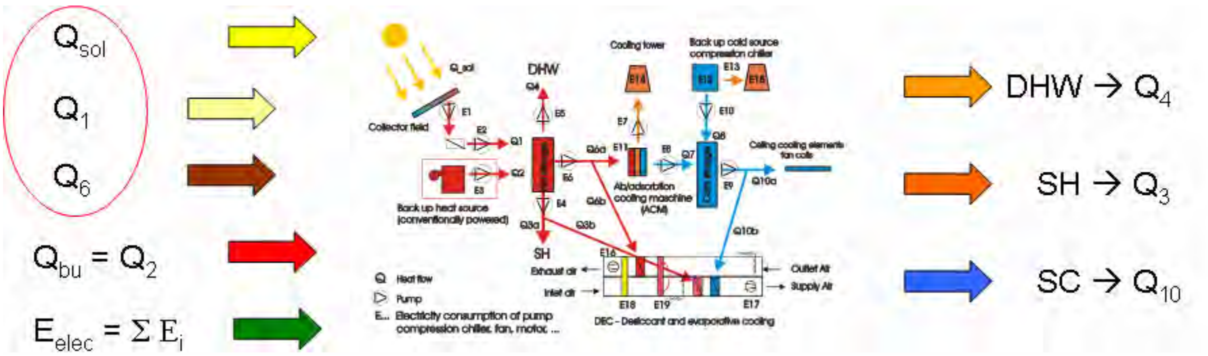

Figure 4. 2nd level monitoring scheme including the measured inputs and outputs of the solar assisted cooling system

In the following several equations are shown which allow to calculate the amount of solar energy which the solar assisted cooling system is not able to exploit because of different losses such us within the solar collectors, heat exchangers or the storage.

The amount of solar energy that is not exploited because of losses in the solar loop, heat exchangers and collectors is calculated by the (Equation 5 - (Equation 6:

$$\eta_{coll,net} = \frac{Q_1}{Q_{sol}} \quad \text{(Equation 5)}$$

$$\Delta Q_{sol} = Q_{sol}(1-\eta_{coll}) \quad \text{(Equation 6)}$$

The amount of thermal energy that is not exploited because of losses in the storage is calculated by the (Equation 7 - (Equation 8:

$$\eta_{storage} = \frac{Q_3 + Q_6 + Q_4}{Q_1 + Q_2} \quad \text{(Equation 7)}$$

$$Q_{loss,storage} = (Q_1 + Q_2) - (Q_3 + Q_6 + Q_4) \quad \text{(Equation 8)}$$

Finally, the amount of available solar energy that is not used in the thermal loop of the solar assisted cooling system ($Q_{solar,\ unex}$) is calculated by the (Equation 9 - (Equation 13:

$$SF = \frac{Q_1}{\left(Q_1 + Q_2\right)}$$ (Equation 9)

$$Q_6^* = SF \cdot Q_6$$
$$Q_3^* = \ldots$$ (Equation 10)

$$Q^* = Q_6^* + Q_3^* + Q_4^*$$ (Equation 11)

$$\eta_{Heat} = \frac{Q^*}{Q_{sol}}$$ (Equation 12)

$$Q_{solar,unex} = Q_{sol} - Q^*$$ (Equation 13)

2.3. Third level

The third level consists in a full system monitoring based on the method of "fractional energy saving" as part of the FSC method which was elaborated in the IEA SHC Task 26 for solar combi systems and extended in the IEA SHC Task 32 for solar heating and cooling systems [7]. This file outlines the energy-flux components required to characterize the performance of solar heating and cooling systems with this method. The FSC method as such is not explained in this document, readers interested in a tutorial on the FSC method are referred to the cited literature [8] [9] [10].

$$f_{sav,shc} = 1 - \frac{\dfrac{Q_{boiler}}{\varepsilon_{fossil} \cdot \eta_{boiler}} + \dfrac{E_{el}}{\varepsilon_{elec}} + \dfrac{Q_{cooling,missed}}{SPF_{ref} \cdot \varepsilon_{elec}}}{\dfrac{Q_{boiler,ref}}{\varepsilon_{fossil} \cdot \eta_{boiler,ref}} + \dfrac{E_{el,ref}}{\varepsilon_{elec}} + \dfrac{Q_{cooling,ref}}{SPF_{ref} \cdot \varepsilon_{elec}}}$$ (Equation 14)

(Equation 14 defines the "fractional solar heating & cooling savings" ($f_{sav,shc}$) in terms of:

- energy consumption attributed to auxiliary devices required for the solar heating/cooling system. (numerator)
- energy usage allocated to a reference system with no solar energy-input (denominator)

The crux of the mater is to come up with a practical definition as to define the reference system, having in mind that the only accessible measurement object is the building with the SHC system. The proposed strategy is to derive the electricity consumption of the reference system "$E_{el,ref}$" from the SHC system, by adequate modifications in the measurement data analysis, following the below outlined scheme:

- define a maximal equipped SHC system (see Figure 1, labelled "SHC_max system")
- the appliances deployed in the corresponding conventional reference system are depicted graphically in Figure 2 by skipping all solar assisted equipment from the SHC_max system.
- All thermal and electrical energies of the conventional reference system can be determined from measurements conducted in the SHC_max system using the following assumptions.
- All thermal energies (hot/cold) supplied by solar in the SHC-system are fully substituted by conventional heat/cold production in the reference system.
- Electricity consumption of pumps for DHW+SH and cold supply is equivalent in the SHC-systems and the reference systems.

For the measured solar heating and cooling system (numerator in (Equation 14) the energies according to Figure 1 are as following: The measured boiler energy supplied to the system Q_{boiler} is equal to Q2 and the additional cooling provided from the compression chiller $Q_{cooling,\ missed}$ is equal to Q8. The electricity consumption E_{el} is the sum of all electricity consumer except the compression chiller: $E_{el} = (\sum E_i)$-E10-E12-E13-E15.

For the reference system (denominator in (Equation 14) the following calculations have to be done:

For the boiler the ratio of electrical energy to thermal energy is identical in the SHC-system and the reference-system.

$$\alpha_{SHC} = \frac{\mathrm{E}_{\mathrm{el,boiler}}^{\mathrm{SHC}}}{\mathrm{Q}_{\mathrm{boiler,SHC}}^{\mathrm{thermal}}} = \frac{E_3}{Q_2} \approx \alpha_{ref} \qquad \text{(Equation 15)}$$

Reference storage heat losses according to IEA SHC Task 26 and with reference to ENV 12977-1 (2000):

$$\mathrm{Q}_{loss,ref} = 0.00016 * \sqrt{0.75 * V_D} * \left(T_T - T_a\right) * 8760 \qquad \text{(Equation 16)}$$

$Q_{loss,ref}$	… Reference storage heat losses [kWh/a]
V_D	…Average daily hot water consumption [Liter/day]
T_T	…Set point temperature of the hot water tank [°C], 52.5°C is used for this in Task 26
T_a	…Ambient temp. around the hot water tank [°C], 15°C is used for this in Task 26

The reference boiler energy supplied to the system therefore is:

$$\mathrm{Q}_{\mathrm{boiler,ref}} = \mathrm{Q}_{\mathrm{SHC}}^{\mathrm{SH}}(\mathrm{Q3}) + Q_{\mathrm{SHC}}^{\mathrm{DHW}}(\mathrm{Q4}) + Q_{\mathrm{ref}}^{\mathrm{loss}} \qquad \text{(Equation 17)}$$

On inspection of Figure 2 the electrical energy consumption in the ref. system sums up to:

$$\mathrm{E}_{\mathrm{ref}}^{\mathrm{el}} = \mathrm{E}_{\mathrm{ref}}^{\mathrm{DHW,el}}(\mathrm{E5}) + \mathrm{E}_{\mathrm{ref}}^{\mathrm{SH,el}}(\mathrm{E4}) + \mathrm{E}_{\mathrm{ref}}^{\mathrm{c\text{-}supply}}(\mathrm{E9}) + \mathrm{E}_{\mathrm{ref}}^{\mathrm{el,boiler}} + \mathrm{E}_{\mathrm{ref}}^{\mathrm{el,vent}} \qquad \text{(Equation 18)}$$

Where the electrical consumption of the boiler in the reference system is given through:

$$\mathrm{E}_{\mathrm{ref}}^{\mathrm{el,boiler}} \approx \alpha_{SHC} * \mathrm{Q}_{\mathrm{boiler,ref}} = \alpha_{SHC} * (\mathrm{Q}_{\mathrm{SHC}}^{\mathrm{SH}}(Q3) + \mathrm{Q}_{\mathrm{SHC}}^{\mathrm{DHW}}(Q4) + \mathrm{Q}_{\mathrm{ref}}^{\mathrm{loss}}) \qquad \text{(Equation 19)}$$

Fans’ electricity consumption for the conventional ventilation system is calculated based on the measured electricity consumption of the desiccant cooling system (SHC_max,) and corrected by the ratio of theoretical design pressure losses of the conventional ventilation system to the desiccant cooling system (based on datasheet of the DEC system). The electrical consumption of the two fans of the ventilation system can be estimated by:

$$\mathrm{E}_{\mathrm{ref}}^{\mathrm{vent,el}} = \mathrm{E}_{\mathrm{SHC}}^{\mathrm{DEC,el}} \cdot f(\Delta P^{REF}, \Delta P^{DEC}) \qquad \text{(Equation 20)}$$

Considering that for each fan of the system, the electrical power is given by (with V in [m3/s] and ΔP in [Pa]):

$$E_{Fan} = \frac{\Delta P \cdot \dot{V}}{\eta}[W] \qquad \text{(Equation 21)}$$

Assuming the same η and flow rates for the fans of the reference and DEC system, the electricity power of the reference system can be calculated as following:

$$\mathrm{E}_{\mathrm{ref}}^{\mathrm{vent,el}} = \mathrm{E}_{\mathrm{SHC}}^{\mathrm{DEC,el}} \cdot \frac{\left(\Delta P_{\sup ply}^{REF} \cdot \dot{V}_{\sup ply} + \Delta P_{return}^{REF} \cdot \dot{V}_{return}\right)}{\left(\Delta P_{\sup ply}^{DEC} \cdot \dot{V}_{\sup ply} + \Delta P_{return}^{DEC} \cdot \dot{V}_{return}\right)} \quad \text{(Equation 22)}$$

All the ΔP are known for the DEC system. The ΔP for the reference Air Handling Unit (AHU) can be estimated considering in the calculation only the components used, assuming that normally the pressure losses of each component of the DEC Air Handling Unit are known from the manufacturer of the AHU.

These calculations result in the definition of the electrical consumption of the reference system in terms of data measured in the SHC-system:

$$\mathrm{E}_{\mathrm{ref}}^{\mathrm{el}} = \mathrm{E}_{\mathrm{SHC}}^{\mathrm{DHW,el}} + \mathrm{E}_{\mathrm{SHC}}^{\mathrm{SH,el}} + \mathrm{E}_{\mathrm{SHC}}^{\mathrm{c\text{-}supply,el}} + \alpha_{SHC} * \mathrm{Q}_{\mathrm{boiler,ref}} + \mathrm{E}_{\mathrm{ref}}^{\mathrm{vent,el}} \quad \text{(Equation 23)}$$

$$\mathrm{E}_{\mathrm{ref}}^{\mathrm{el}} = \mathrm{E5} + \mathrm{E4} + \mathrm{E9} + \alpha_{SHC} * \mathrm{Q}_{\mathrm{boiler,ref}} + (E16 + \mathrm{E17}) * f(\Delta P) \quad \text{(Equation 24)}$$

Finally $\mathrm{Q}_{\mathrm{cooling,ref}}$ is equal to the measured “cold” production SHC system:

$$\mathrm{Q}_{\mathrm{cooling,ref}} = \mathrm{Q}_{\mathrm{SHC}}^{\mathrm{cold}} = Q_{\mathrm{ACM}}(\mathrm{Q7}) + Q_{\mathrm{bup}}(\mathrm{Q8}) + Q_{\mathrm{DEC}} - \mathrm{Q10b} \quad \text{(Equation 25)}$$

Where $\mathrm{Q}_{\mathrm{DEC}}$ is: Thermal cooling energy delivered from DEC system in the SHC-system in terms of enthalpy difference between ambient air and supply air (latent and sensible heat has to be taken into account!)

The Seasonal Performance Factor SPF for the compression chiller in the reference system can not be known exactly. The following possibilities are proposed:

- $\mathrm{SPF}_{\mathrm{meas}}$ measured SPF in the monitored SHC system
- $\mathrm{SPF}_{\mathrm{ref}}$ proposed SPF for a chiller in the reference system: $\mathrm{SPF}_{\mathrm{ref}} = 2.8$

3. Expected results

The target from the presented monitoring procedure is to have a common base for the monitoring of solar assisted heating and cooling systems, allowing a comparison of the performance of different systems and allowing the elaboration of a learning curve of the following years.

3.1. Expected results – 1st level

The main results within the first level are the

- *PER* … primary energy ratio of the installed solar assisted heating and cooling system ((Equation 1)
- PER_{ref} … primary energy ratio of an assumed reference system working under the same conditions as the installed SHC system ((Equation 2).

Herewith the annual savings of primary energy through the utilisation of the SHC system can be expressed.

Further through the calculated cost per installed cooling capacity ((Equation 4), it will be possible in the following years to draw a learning curve for SCH systems.

3.2. Expected results – 2nd level

The main result within the second level is the calculation of the solar thermal energy which could not be exploited by the SHC system – this shows the “efficiency” of the SHC system ((Equation 12 and (Equation 13, see Figure 5)

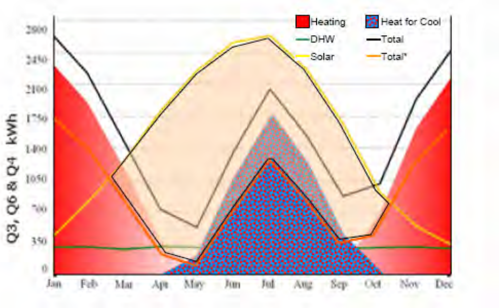

Figure 5: The single energy fluxes are shown in dependence of the months, further the high lighted area shows solar thermal energy which could not be exploited by the SHC system (in beige, Q*)

3.3. Expected results – 3rd level

The main result within the third level is given by the fractional solar heating & cooling savings ($f_{sav,shc}$) which is defined by the (Equation 14. It presents a comparison of the primary energy need of the installed SHC system with a reference system, which is defined based on the real needs for heating and cooling.

4. Acknowledgments

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[2] H. M. Henning, 2005, ”Solar Air-Conditioning R&D in the framework of the international Energy Agency (IEA)”, 1st International Conference Solar Air-Conditioning, Bad Staffelstein.
[3] W. Sparber et al, 2008, “Modeling of a Solar Combi Plus System – Framework and Hydraulic Scheme Proposals”, Ninth International Symposium Gleisdorf Solar, 3rd – 5th September 2008, Gleisdorf, Austria
[4] Directive 2006/32/EC of the European Parliament and of the Council of 5 April 2006 on energy end-use efficiency and energy services and repealing Council Directive 93/76/EEC, L114/76, Annex II, footnote 3
[5] http://www.iea-shc.org/task38/index.html
[6] H. M. Henning (ed), “Solar-assisted air conditioning and buildings, a handbook for planners”, IEA SHC Task 25, Springer Wien New York
[7] http://www.iea-shc.org/task26/index.html; http://www.iea-shc.org/task32/index.html; http://www.iea-shc.org/task25/index.html
[8] T. Letz, 2002, "Validation and background information the FSC procedure", A technical report of subtask A, IEA-SHC Task 26, http://www.iea-shc.org/outputs/task26/A_Letz_FSC_method.pdf
[9] W. Weiss (ed), 2003, “Solar Heating Systems for Houses – A Design Handbook for Solar Combisystems ” IEA-SHC Task 26, James & James Ltd London, pp 125-154
[10] R. Heimrath, M.Haller, May 2007, “The Reference Heating System, the Template Solar System”, IEA SHC - Task 32 Project Report, Institute of Thermal Engineering University of Technology Graz Austria, pp 28-30 ; available from the task 38 download area.

ANHANG IWT 6

Sparber, W., Thuer, A., Besana, F., Streicher, W., Henning, H.M., Präsentation: IEA Task 38 “proposal for a unified monitoring procedure for solar heating and cooling system“

Task 38
Solar Air-Conditioning
and Refrigeration

IEA Task 38 “Proposal for a unified monitoring procedure for solar heating and cooling system

by

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Introduction / Monitoring Procedure

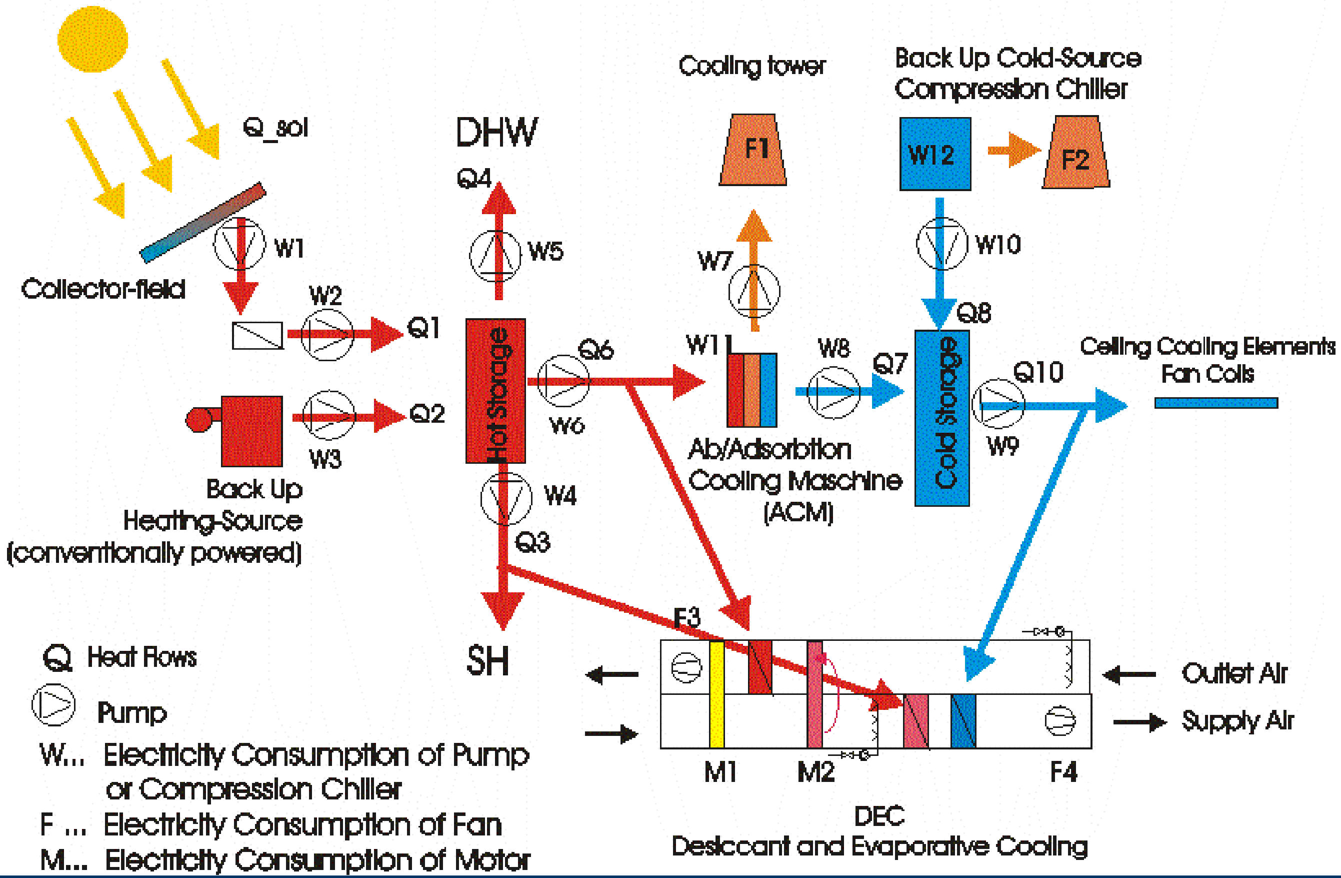
As discussed within the IEA Task 38 Meeting in Barcelona in Autumn 2007 a three steps program on information for monitored systems is proposed:

1. Level: Basic Information on primary energy COP, Solar Fraction and Costs
2. Level: Basic monitoring procedure (kept simple in sense of calculation and necessary monitoring hardware)
3. Level: Advanced monitoring procedure (more complex in sense of calculation an necessary monitoring hardware)

Reference scheme

In the following a scheme is presented (called SHC_max system) which is the base scheme where the single energy flows are nominated and shown.

Definition of SHC_max System



AEE INTEC

TU Graz

Proposal of accuracy

Accuracy of the monitoring system

- Temperature sensors → $\pm 0.21°C$
- Mass Flow meter → 0.25% of full span
- Electricity counter → Class 0,5 (1%)

Measurements every:
- 1st level ?
- 2nd level ?
- 3rd level ?

1. Level: Basic Information on Primary Energy COP and Costs

To define a minimal level for a comparison between solar cooling systems regarding primary energy efficiency and economical value

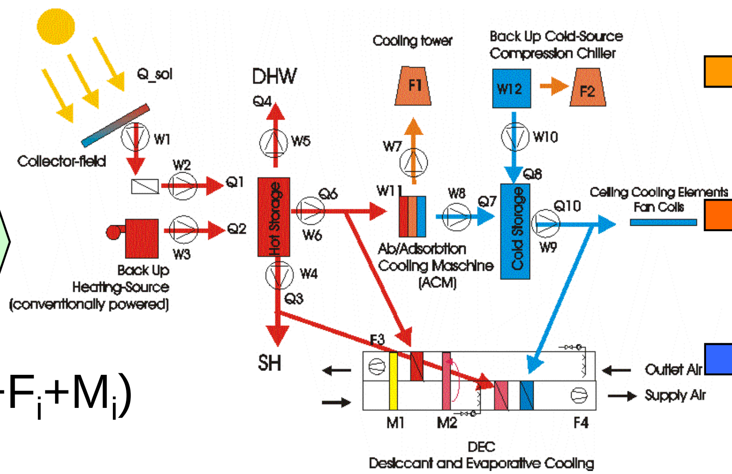
A limited set of sensors is required (4 mass flow meter and total electricity counter);

Basic Information on primary energy COP

We propose the utilization of fixed fossil based conversion factors for kWh_{th} and kWh_{el}, already included within the Handbook of Task 25:

$PE_{th} \rightarrow \quad \varepsilon_{fossil} \quad Q_{bu} = Q_2$

$PE_{elec} \rightarrow \quad \varepsilon_{elec} \quad E_{elec} = \Sigma\,(W_i+F_i+M_i)$



DHW → Q_4

SH → Q_3

SC → Q_{10}

Basic Information on primary energy COP

The primary energy COP is defined as:

$$COP_{PE} = \frac{\left(Q_4 + Q_3 + Q_{10}\right)}{\frac{Q_{bu}}{\varepsilon_{fossil}} + \frac{E_{elec,tot}}{\varepsilon_{elec}}} \quad (1)$$

Following the "Solar-Assisted Air-Conditioning in Buildings" by H.-M. Henning, two primary energy conversion factors for heat and electricity from fossil fuels:

ε_{elec} = 0.36 (kWh of electricity per kWh of primary energy)

ε_{fossil} = 0.9 (kWh of heat per kWh of primary energy)

Basic Information on primary energy COP

Reference household energy use

52%$_{PE}$ → Refinery ($\varepsilon = 0.9$) → 47% → Oil/N.gas Boiler ($\eta = 0.9$) → 42% → Space Heating + Domestic Hot Water

losses

Primary Energy [%]

$$COP_{PE}^{ref} = \frac{\left(SH_{\%} + DHW_{\%} + SC_{\%}\right)}{PE\%_{fossil} + PE\%_{elec}} = 0.82 \quad (2)$$

losses

11%$_{PE}$ → Power plant ($\varepsilon = 0.36$) → 4% → Comp. Chiller ($\eta = 2.5$) → 10% → Space Cooling

Source: US DOE RECS Survey 2000

Basic Information on Costs

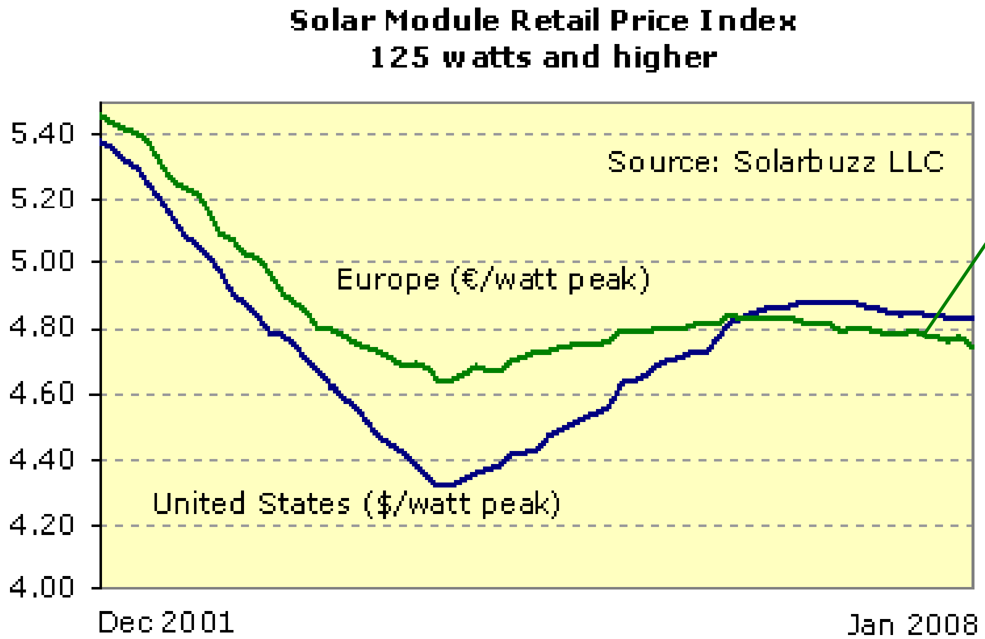
The cost is defined per chilled capacity:

$$\frac{Cost(€)}{kW_{cold}} \tag{3}$$

we propose to include within the cost factor the whole solar cooling system costs (collectors, pipes, buffers, ab/adsorption, cooling tower, ...) but without the cost for the back up heater (boiler, cogeneration, ..) and without the cost for the ev. back up cooling (compression chiller).

Basic Information on Costs

PV market as an example



$$\frac{Cost(€)}{kW_{peak}}$$

2. Level: Basic monitoring procedure

In the following a method is presented that has the aim to suggest to technicians what is the **minimum monitoring devices** to be installed permitting a certain analysis of the exploited potential of the system. In the second level the main attention is paid to the solar thermal system especially to the **hot storage buffer management.**

For this reason the method is:

- As simple as possible;
- A limited set of sensors is required (6 mass flow meter, pyranometer and total electricity counter);
- Output: Selected diagrams & indexes

Aim of the method: how much of the available solar energy (W/m²) in the system location does reach the final application: DHW, SH, SC (in terms of solar heat supplied to thermally driven chiller).

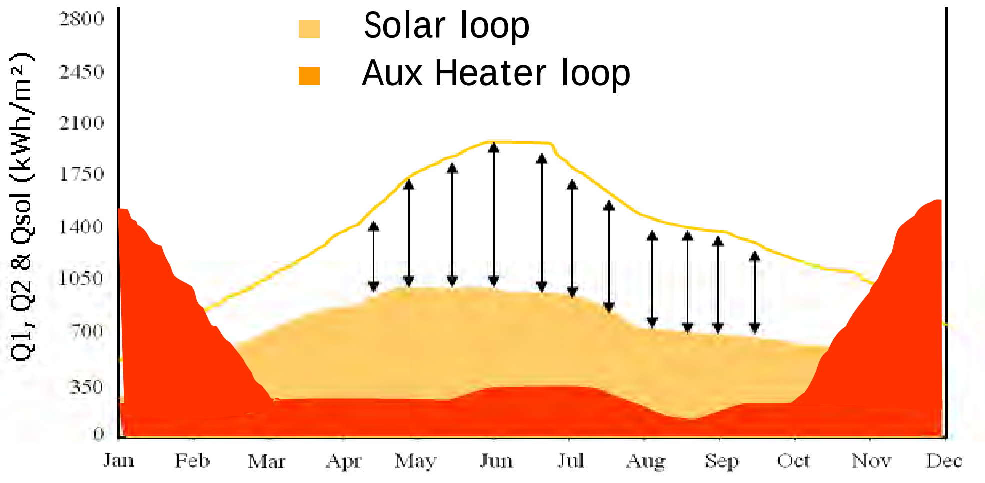
a) Basic monitoring procedure

Sensor	Measurement	Output
• 1 Pyranometer	Tilted solar radiation →**Qsol,m (monthly)** →**Qsol (yearly)**	Monthly/Yearly energy available per unit of **solar collector surface** [kWh/m²]
• 2 PT100 • 1 Volume flow meter	Solar collector energy →**Q1,m (monthly)** →**Q1 (yearly)**	Montly/Yearly energy delivered by the collector per unit of **solar collector surface** [kWh/m²]
• 2 PT100 • 1 Volume flow meter	Auxiliary heater energy →**Q2,m (monthly)** →**Q1 (yearly)**	Montly/Yearly energy delivered by the heater per unit of **conditioned surface** [kWh/m²]

a) Basic monitoring procedure

What is the difference between solar radiation [kWh] and the produced heat [kWh]

Diagram 1



Indexes & value

$$\eta_{coll} = \frac{Q_1}{Q_{sol}} \quad (4)$$

$$\Delta Q_{sol} = Q_{sol}(1 - \eta_{coll}) \quad (5)$$

$$SF = \frac{Q_1}{(Q_1 + Q_2)} \quad (6)$$

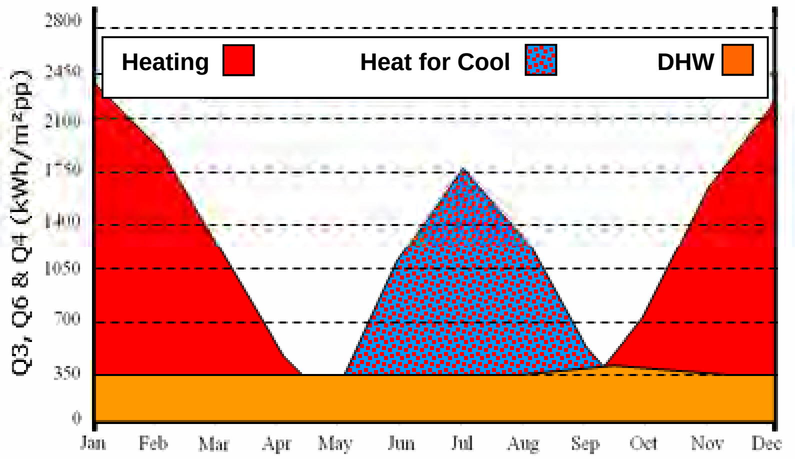
b) Basic monitoring procedure

Sensor	Measurement	Output
• 2 PT100 • 1 Volume flow meter	Supplied energy to the thermally driven chiller **→Q6,m (monthly)** **→Q6 (yearly)**	Monthly/Yearly energy supplied per unit of **conditioned surface** [kWh/m²]
• 2 PT100 • 1 Volume flow meter	Supplied energy to the Space Heating **→Q3,m (monthly)** **→Q3 (yearly)**	Montly/Yearly energy supplied per unit of **conditioned surface** [kWh/m²]
• 2 PT100 • 1 Volume flow meter	Supplied energy to the Domestic Hot Water **→Q4,m (monthly)** **→Q4 (yearly)**	Montly/Yearly energy supplied per **person** [kWh/pp]

b) Basic monitoring procedure

Final utilization of heat (Heating, cooling and domestic hot water) and how efficient is the heat management ?

Diagram 2



Indexes & value

$$SHCF = \frac{Q_6}{Q_6 + Q_3} \quad (7)$$

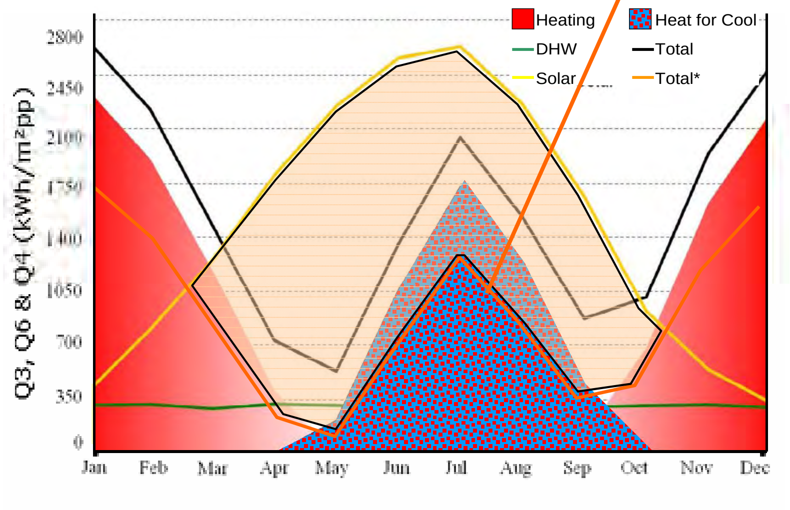
$$Q_{loss} = (Q_1 + Q_2) - (Q_3 + Q_6 + Q_4) \quad (8)$$

$$\eta_{storage} = \frac{Q_3 + Q_6 + Q_4}{Q_1 + Q_2} \quad (9)$$

c) Basic monitoring procedure

Which amount of the available solar energy is unexploited in the system?

Diagram 3 $Q^* = Q_6^* + Q_3^* + Q_4^*$



Indexes & value

$$Q_6^* = SF \cdot Q_6$$

$$Q_3^* = \ldots \qquad (10)$$

...

$$\eta_{Heat} = \frac{Q^*}{Q_{sol}} \qquad (11)$$

$$Q_{solar,unex} = Q_{sol} - Q^* \qquad (12)$$

3. Level: Advanced monitoring procedure

This file outlines the energy-flux components required to characterize the performance of solar heat and cooling systems with the **FSC method** as developed within IEA SHC Task 26 and extended in Task 32

The FSC method as such is not explained in this document. Readers interest in a tutorial on the FSC method are referred to the below cited literature.

A) T.Letz , 2002, "Validation and background information the FSC procedure", A technical report of subtask A, IEA-SHC Task 26, http://www.iea-shc.org/outputs/task26/A_Letz_FSC_method.pdf

B) Werner Weiss (ed), 2003, "*Solar Heating Systems for Houses – A Design Handbook for Solar Combisystems* " IEA-SHC Task 26, James&James Ltd London, pp 125-154

C) R.Heimrath,M.Haller, May 2007, “The Reference Heating System, the Template Solar System” , IEA SHC - Task 32 Project Report, Institute of Thermal Engineering University of Technology Graz Austria, pp 28-30 ;
available from the task 38 download area

$$f_{sav,shc} = 1 - \frac{\frac{Q_{boiler}}{\eta_{boiler}} + \frac{W_{el}}{\eta_{el}} + \frac{Q_{cooling,missed}}{SPF_{ref} \cdot \eta_{el}}}{\frac{Q_{boiler,ref}}{\eta_{boiler,ref}} + \frac{W_{el,ref}}{\eta_{el}} + \frac{Q_{cooling,ref}}{SPF_{ref} \cdot \eta_{el}}} \qquad (13)$$

Equation (13) defines "fractional solar heating & cooling savings" ($f_{sav,shc}$) in terms of:

a) energy consumption attributed to auxiliary devices required for the solar heating/cooling system. (numerator) → E_{aux}

b) energy usage allocate to a reference system with no solar energy-input (denominator) → E_{ref}

The fraction in equation (13) can be rationalized as a "normalization "of energy usage incurred with the solar assisted heating & cooling system to usage stemming from a non solar heating & cooling set up (reference system).

The crux of the mater is to come up with practical definition as to define the reference system, having in mind that the only accessible measurement object is the building with the SHC system

The proposed strategy is to derive the electricity consumption of the reference system "$W_{el,ref}$" from the SHC system, by adequate modifications in the measurement data analysis, following the below outlined scheme.

1) define max. equipped SHC system (labeled "SHC_max system") made up by the following sub systems:
   - solar collectors + heat storage + boiler
   - heat distribution system (space heating, domestic hot water)
   - solar driven cooling system including cooling tower
   - compression chiller as cooling back-up system
   - desiccant cooling/ dehumidification System
   - cold distribution system

Figure 1 gives a graphical account of the elements including heat/cold flows.

Definition of SHC_max System

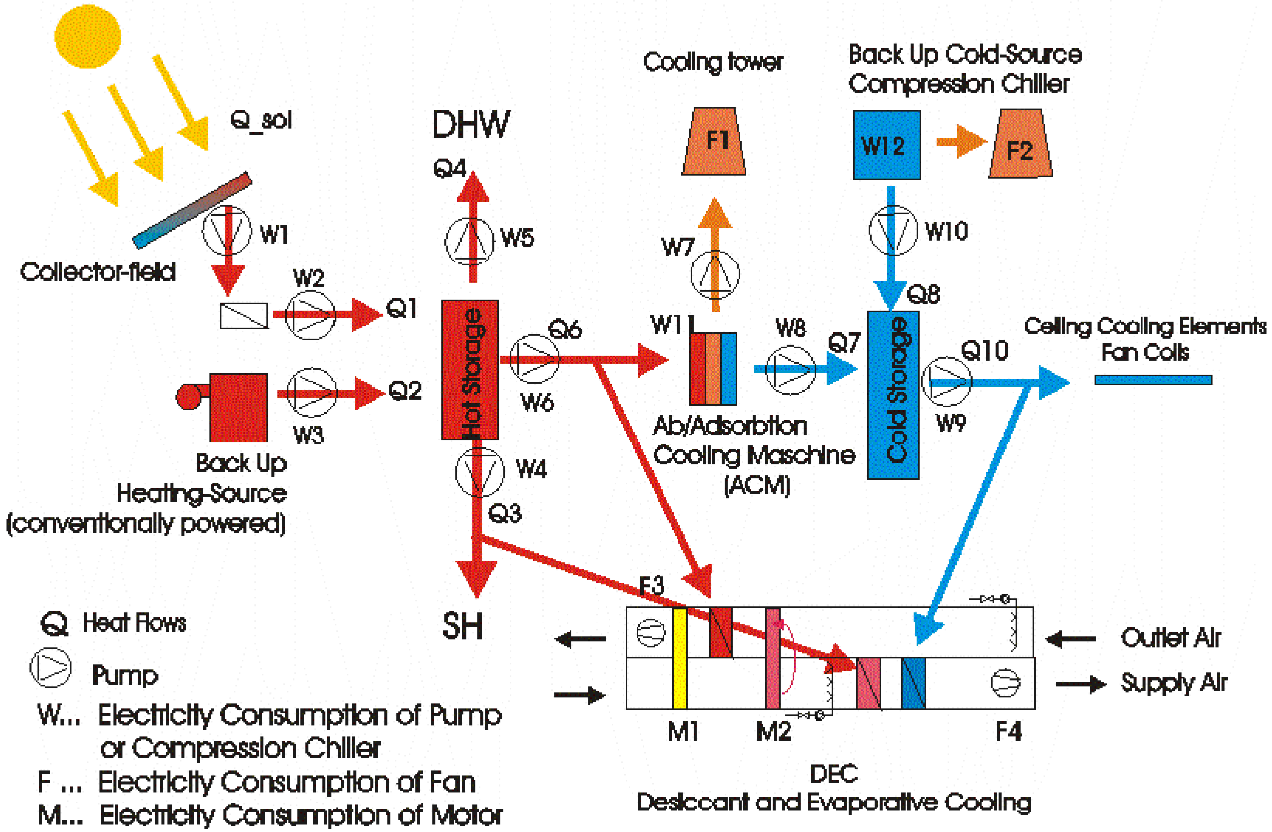


Fig. 1 Schematics of SHC_max system

2) The appliances deployed in the corresponding conventional reference system are depicted graphically in figure 2. All electrical appliances of the SHC_max system are listed in table 1.

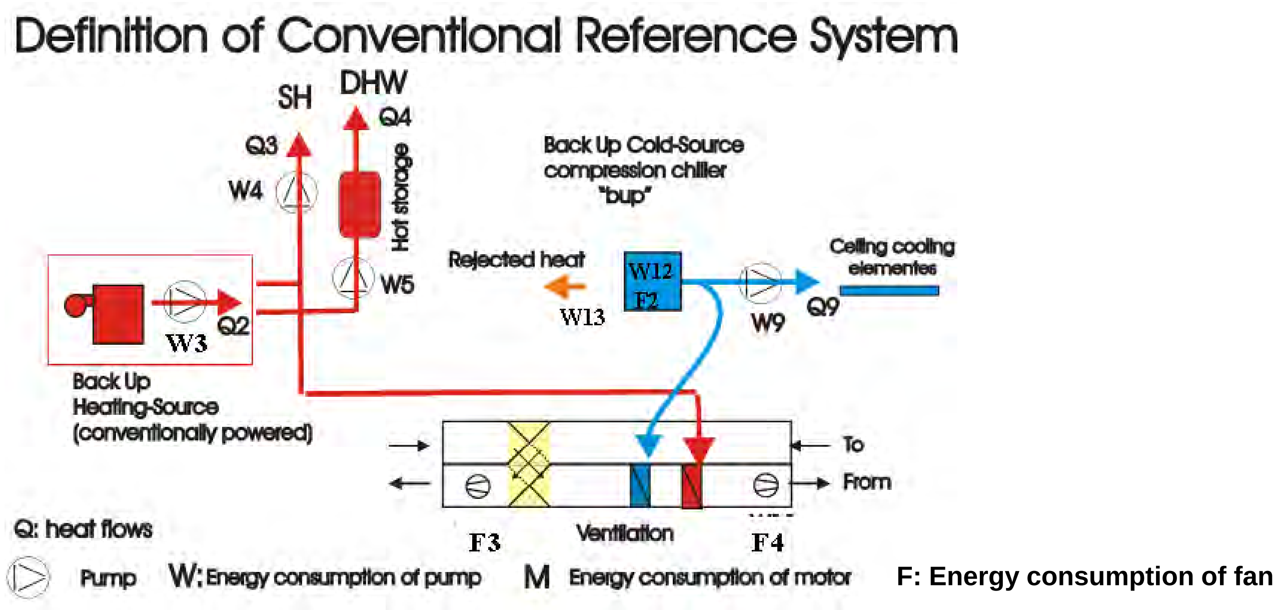


Fig. 2 Schematics of the conventional reference system

Electricity consumers	***SHC System***	***Reference System***	***Remarks***	***Labeling in Fig.1// Fig.2***
Heating System				
pump collector field (primary)	yes	no		W1
pump collector field (secondary)	yes	no		W2
pump boiler hot-storage (incl. internal boiler consumption)	yes	yes		W3
pump hot-storage SH	yes	yes		W4//W4
pump hot-storage DHW	yes	yes		W5//W5
Cooling System				
pump hot-storage cooling maschine	yes	no		W6
pump cooling maschine - cooling tower	yes	no		W7
pump cooling maschine (ACM) - cold-storage	yes	no		W8
pump cold distribution - ceiling cooling	yes	yes		W9//W9
pump back up source - cold storage	no/yes	yes	included in W12 ?	W10
fan; cooling tower	yes	no		F1
absorption/adsorption cooling maschine	yes	no		W11
compression chiller (back up system)	yes/no	yes		W12
pump compression chiller to fan (back up system)	no/yes	yes	included in W12 ?	W13
fan; compression chiller (back up system)	no/yes	yes	included in W12 ?	F2//F2
Desiccant cooling/ dehumidification System				
fan escaping air	yes	yes		F3//F3
fan supply air	yes	yes		F4//F4
motor for desiccant wheel	yes	no		M1
motor for heat recovery wheel	yes	no		M2
Thermal Energies	***SHC System***	***Reference System***	***Remarks***	***Labeling in Fig.1// Fig.2***
solar thermal energy output to hot storage	yes	no		Q1
boiler thermal output	yes	yes		Q2
SH consumption (conventional)	yes	yes		Q3
SH consumption (ventilation system)	yes/no	yes		Q3
DHW consumption	yes	yes		Q4
hot storage input to DEC-system (sorption regeneration)	yes	no		Q6
hot storage input to cooling machine (ACM)	yes	no		Q6
cold output acm to cold-storage	yes	no		Q7
cold output back-up system to cold-storage	yes	no		Q8
cold storage output to cold-distribution	yes	no		Q10
cold storage to DEC-system	yes	no		Q10

Table 1: List of electrical appliances and thermal energies in SHC and conventional REF.-system

3) the “conventional reference system” is designed by skipping all solar assisted equipment from the SHC_max system (fig2).

All thermal and electrical energies of the conventional ref. system can be determined from measurements conducted in the SHC_max system given the following assumptions:

- All thermal energies (hot/cold) supplied by solar in the SHC-system are fully substituted by conventional heat/cold production in the reference system
- Electricity consumption of pumps for DHW+SH and cold supply is equivalent in the SHC-systems and the ref. systems.

$$W_{ref}^{DHW,\ el}(W5) = W_{SHC}^{DHW,\ el}(W5)$$
$$W_{ref}^{SH,\ el}(W4) = W_{SHC}^{SH,\ el}(W4) \qquad (14)$$
$$W_{ref}^{c\text{-}supply}(W9) = W_{SHC}^{c\text{-}supply}(W9)$$

- For the boiler the ratio of electrical energy to thermal energy is identical in the SHC-system and the reference-system.

$$\alpha_{SHC} = \frac{W_{el,boiler}^{SHC}}{Q_{boiler,SHC}^{thermal}} \approx \alpha_{ref} = \frac{W_3}{Q_2} \qquad (15)$$

3) continued

➢ Fans' Power consumption for desiccant cooling system (SHC_max,fig.1) and conventional ventilation system (ref. system,fig.2) is set equal. It is recommended to verify this assumption for each project under consideration.

Consecutively each term in formula (13) is outlined resting upon the definition of the

- SHC_max system (Fig1)
- Conventional reference system (Fig2)

and the associated equipment listed in table 1.

TERM: $\frac{Q_{boiler}(Q2)}{\eta_{boiler}}$

Quantity / Units	Definition	Remark
Q_{boiler}[kWh]	thermal energy load of the auxiliary boiler	
η_{boiler}	mean annual efficiency of the auxiliary boiler	

TERM: $\dfrac{Q_{boiler,Ref}(Q2)}{\eta_{boiler,Ref}}$

Quantity / Units	Definition	Remark
$Q_{boiler,Ref}$ [kWh]	defined as : $Q_{SH}+Q_{DHW}+Q_{loss,ref}$ Q_{SH} = energy demand space heating (Q3) Q_{DHW}= energy demand domestic hot water (Q4) $Q_{loss,ref}$= heat losses reference storage tank	for $Q_{loss,ref}$ see next slide further details to $Q_{loss,ref}$: lit-ref. B, pp:129
$\eta_{boiler,Ref}$	mean annual efficiency of the reference boiler	mean annual efficiency of 85% is recommended

Reference storage heat losses according to Task 26 and with reference to ENV 12977-1 (2000):

$$Q_{loss,ref} = 0.00016 * \sqrt{0.75 * V_D} * (T_T - T_a) * 8760$$

$Q_{loss,ref}$	Reference storage heat losses [kWh/a]
V_D…	Average daily hot water consumption [Liter/day]
T_T…	Set point temperature of the hot water tank [°C]: 52.5°C is used for this in Task 26
T_a…	Ambient temperature around the hot water tank [°C]: 15°C is used for this in Task 26

$$\text{TERM:} \quad \frac{W_{el}}{\eta_{el}}$$

Quantity / Units	Definition	Remark
W_{el}[kWh]	W_{el} is determined by adding up all present electrical consumers in the SHC system under investigation. Table1 (see also fig.1) is seen as the master reference-list for possible electrical devices.	If any of the listed cooling/heating appliance (table1) is not present in the investigated SHC system this appliance should obviously be omitted in calculating W_{el}
η_{el}	annual electricity generation efficiency	defined to be : $\eta_{el} = 40\%$

If W12 is used in this term, the $\text{TERM:} \quad \frac{Q_{\text{cooling, missed}}}{SPF_{ref} \cdot \eta_{el}}$ in (13) has to be set to zero!

$$\text{TERM:} \quad \frac{W_{ref}^{el}}{\eta_{el}}$$

Quantity / Units	Definition	Remark
W_{el}^{ref} [kWh]	Is the electrical energy consumption of all consumers as consecutively described and finally defined in (19)	
η_{el}	annual electricity generation efficiency	defined to be : η_{el} = 40%

On inspection of fig.2 the electrical energy consumption in the ref. system sums up to:

$$W_{ref}^{el} = W_{ref}^{DHW,el}(W5) + W_{ref}^{SH,el}(W4) + W_{ref}^{c\text{-}supply}(W9) + W_{ref}^{el,boiler} + W_{ref}^{el,vent} \qquad (16)$$

With (15) the electrical consumption of the boiler in the reference system is:

$$W_{ref}^{el,boiler} \approx \alpha_{SHC} * Q_{boiler,ref} = \alpha_{SHC} * (Q_{SHC}^{SH} + Q_{SHC}^{DHW} + Q_{ref}^{loss}) \qquad (17)$$

or

$$W_{ref}^{el,boiler} \approx \alpha_{SHC} * Q_{boiler,ref} = \alpha_{SHC} * (Q3 + Q4 + Q_{ref}^{loss})$$

Further consumption stems from the two fans of the ventilation system

$$W_{ref}^{vent,el} = W_{SHC}^{DEC,el} \qquad (18)$$

$W_{SHC}^{DEC,el}$: Electricity consumption of fans (F3/F4) in the desiccant cooling system

Inserting (14), (17) and (18) into (16) results in the definition of the electrical consumption of the ref. system in terms of data measured in the SHC-system

$$W_{ref}^{el} = W_{SHC}^{DHW,\ el} + W_{SHC}^{SH,\ el} + W_{SHC}^{c\text{-}supply,\ el} + \alpha_{SHC} * Q_{boiler,\ ref} + W_{SHC}^{Dec,\ el} \quad (19)$$

or

$$W_{ref}^{el} = W5 + W4 + W9 + \alpha_{SHC} * Q_{boiler,\ ref} + F3 + F4$$

$$\text{TERM:} \quad \frac{W_{el}}{\eta_{el}}$$

Quantity / Units	Definition	Remark
W_{el}[kWh]	W_{el} is determined by adding up all present electrical consumers in the SHC system under investigation. Table1 (see also fig.1) is seen as the master reference-list for possible electrical devices.	If any of the listed cooling/heating appliance (table1) is not present in the investigated SHC system this appliance should obviously be omitted in calculating W_{el}
η_{el}	annual electricity generation efficiency	defined to be : $\eta_{el} = 40\%$

If W12 is used in this term, the $\text{TERM:} \quad \frac{Q_{cooling,\ missed}}{SPF_{ref} \cdot \eta_{el}}$ in (13) has to be set to zero!

$$\text{TERM:} \quad \frac{Q_{cooling,ref}}{SPF_{ref} \cdot \eta_{el}}$$

The delivered thermal energies of cooling in the SHC-system are (see fig.1)

$$Q_{SHC}^{cold} = Q_{ACM} + Q_{bup} + Q_{DEC} \qquad (20)$$

Q_{ACM} = Thermal cooling energy delivered from the ab/adsorption cooling machine in the SHC-system (Q7)

Q_{bup} = Thermal cooling energy delivered from the back up cooling machine in the SHC-system (Q8)

Q_{DEC} = Thermal cooling energy delivered from DEC system in the SHC-system in terms of enthalpy difference between ambient air and supply air (latent and sensible heat has to be taken into account!).

The thermal energy of cooling in the reference system must be the same:

$$Q_{cooling,\ ref} = Q_{SHC}^{cold} = Q_{ACM} + Q_{bup} + Q_{DEC} \qquad (21)$$

TERM: $$\frac{Q_{cooling,missed}}{SPF \cdot \eta_{el}}$$

$Q_{cooling,missed}$ is defined as the energy quantity of cold provided by a back up cold source in order to compensate for an insufficient cold supply of the solar assisted cooling system.
This term is only applicable if a backup system is installed as assumed in the SHC_max system (Fig.1, Q8).

The seasonal performance factor SPF can be:

SPF_{meas} measured SPF in the monitored SHC system (prefered)

SPF_{ref} assumed SPF as it is used for the reference system: $SPF_{ref} = 2.8$

η_{el} annual electricity generation efficiency; defined to be : $\eta_{el} = 40\%$

Annex 1

As to determine "$f_{sav,shc}$", *heat quantities* ("Q") are to be measured – typically using a heat meter. The underlying operating mode is illustrated and recommended as measurement prescription.
Using thermodynamic quantities Q can be written out as follows:

$$\mathbf{Q = m\,(h1 - h2)} \qquad (22)$$

Q = quantity of heat
m = mass of the heat carrier
h1 = specific heat for the forward flow at temperature T1
h2 = specific heat for the return flow at temperature T2

As can be seen from the equation, the quantity of heat cannot be directly measured but only indirectly using other physical quantities. Since the heat meter measures the volume rather than the mass and the temperature difference rather than the difference in enthalpy, it is necessary to simply the equation given, which in practice is therefore:

$$\mathbf{Q = V * \Delta\Theta * K} \quad (23)$$

V = volume of the heat carrier
$\Delta \Theta$ = temperature difference between the forward flow and return flow
K = heat coefficient; taking into account the temperature-dependent specific density and specific heat with regard to the actual temperature in the flowmeter.

Appraisal of equation (23) – especially the correction calculation via the heat coefficient K is typically carried out by the electronics of the deployed heat meter.

In case of pumps running with constant power the *electrical energy consumption* (W) is quantified by multiplying the power of the device under test with the operating time (Δt).

e.g. pump

$$W_{pump} = P_{pump} * \Delta t \quad (24)$$

The more general case with varying device performance is handled with deploying an electric meter.

Data to be delivered

1. Primary energy COP (1)
2. Cost per chilled capacity (3)
3. Diagram 3 and unexploited solar energy value (12)
4. Fractional solar heating & cooling savings (13)
5. Alex (??)

ANHANG IWT 7

Weissensteiner Thomas, Präsentation: Measurement results of the solar cooling plant Coolcabin / Solid GmbH in Graz, Austria, expert meeting Palermo, 28.09.2009

Measurement results of the solar cooling plant COOLCABIN / SOLID GmbH in Graz, Austria

Thomas Weissensteiner

SOLID solarinstallation+design

SHC SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning and Refrigeration

Content

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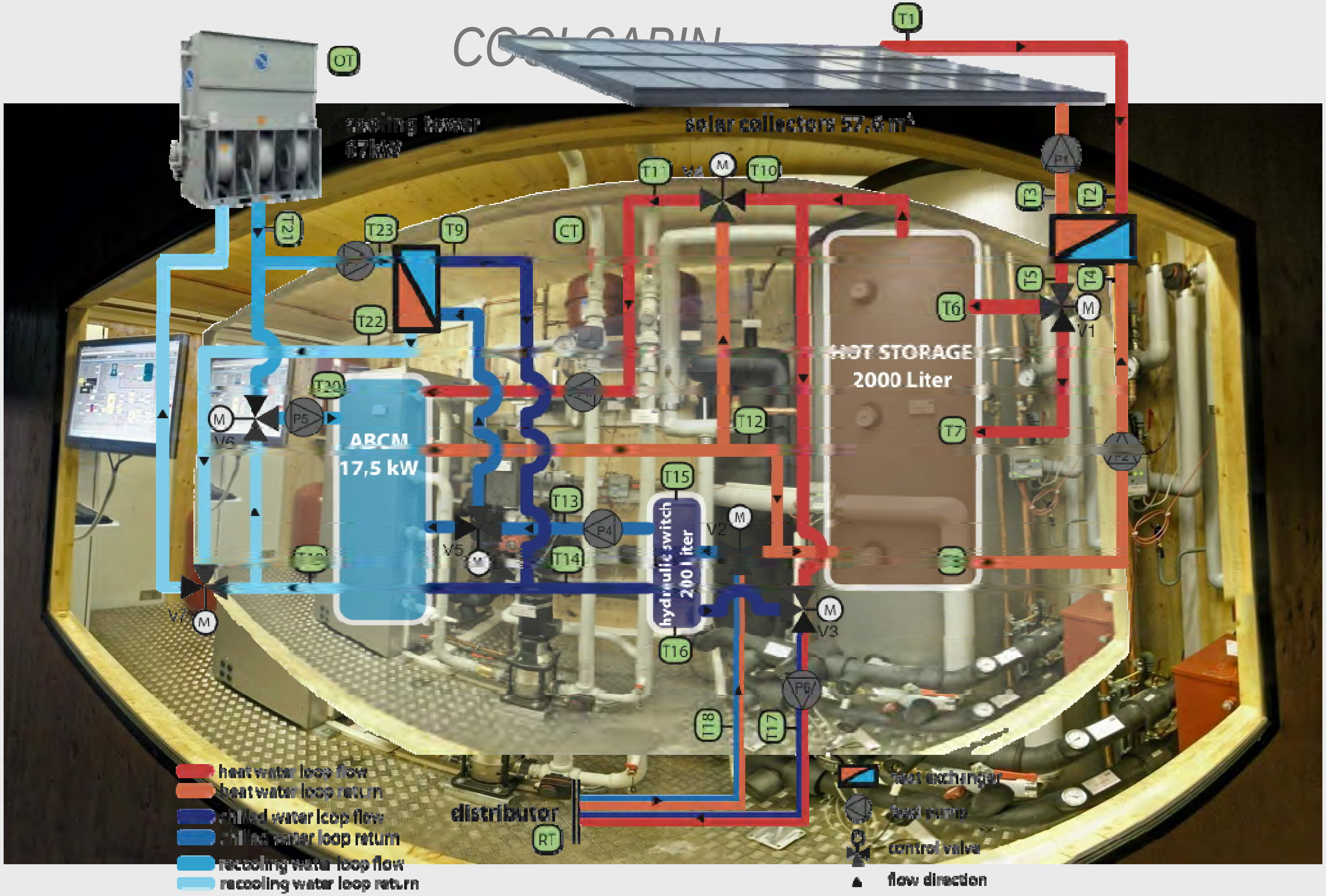
Monitoring
Results
Free cooling
Conclusion

- Introducing the COOLCABIN
- Monitoring Strategy
- Measurement results summer 2009
- Free cooling results
- Conclusion

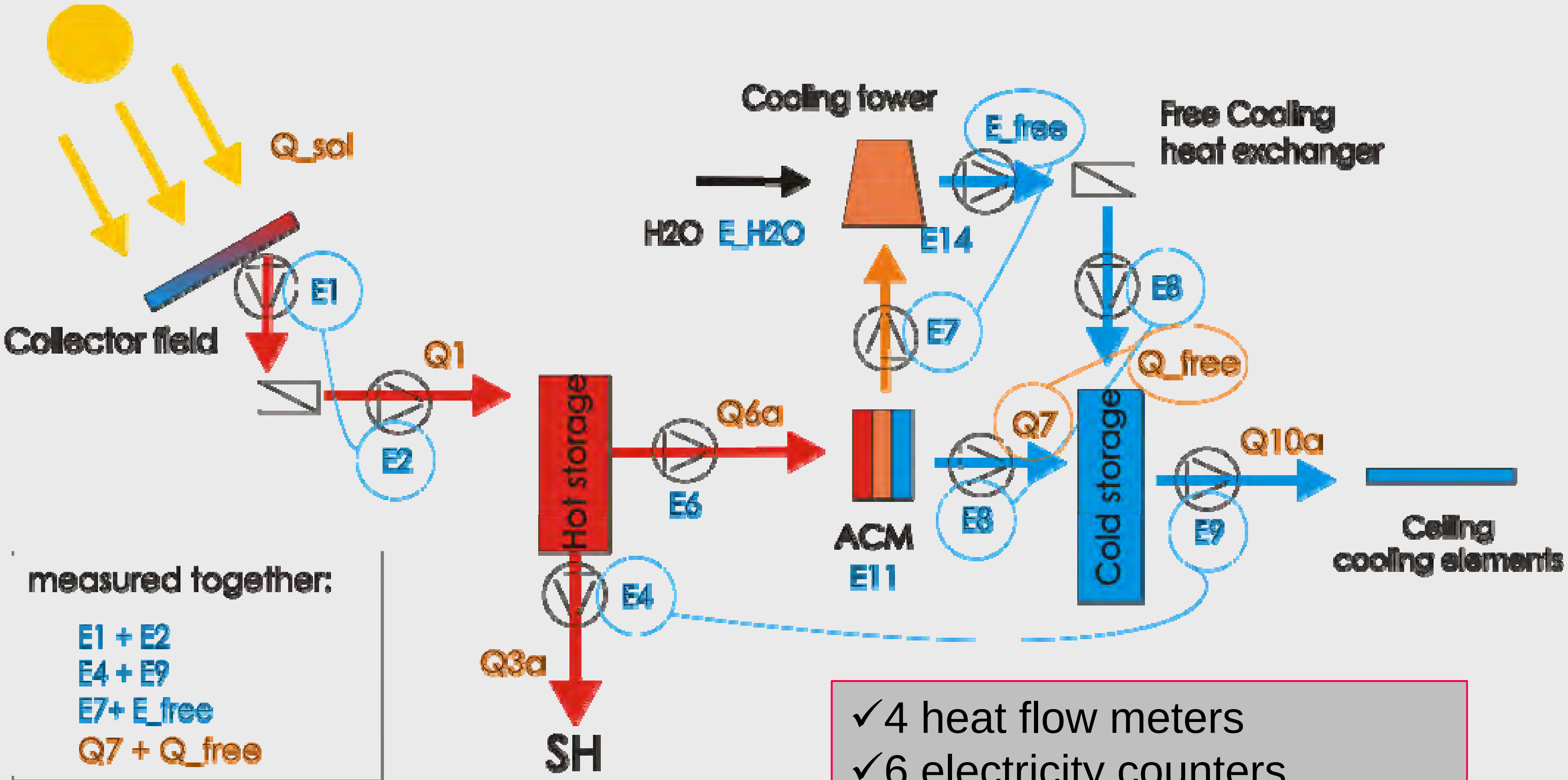
COOLCABIN


Cooling tower: 67 kW
Office:
570 m²
S
Collectors:
58 m²
11°
Absorbtion CM:
17.5 kW

COOLCABIN


OT
solar collectors 57,6 m²
T1
T2
T3
T4
T5
T6
T7
T8
T9
T10
T11
T12
T13
T14
T15
T16
T17
T18
T20
T21
T22
T23
CT
RT
P1
P2
P4
P5
P6
V1
V2
V3
V5
V6
V7
M
HOT STORAGE
2000 Liter
ABCM
17,5 kW
hydraulic switch
200 l iter
distributor
heat water loop flow
heat water loop return
chilled water loop flow
chilled water loop return
recooling water loop flow
recooling water loop return
control valve
flow direction

Monitoring strategy



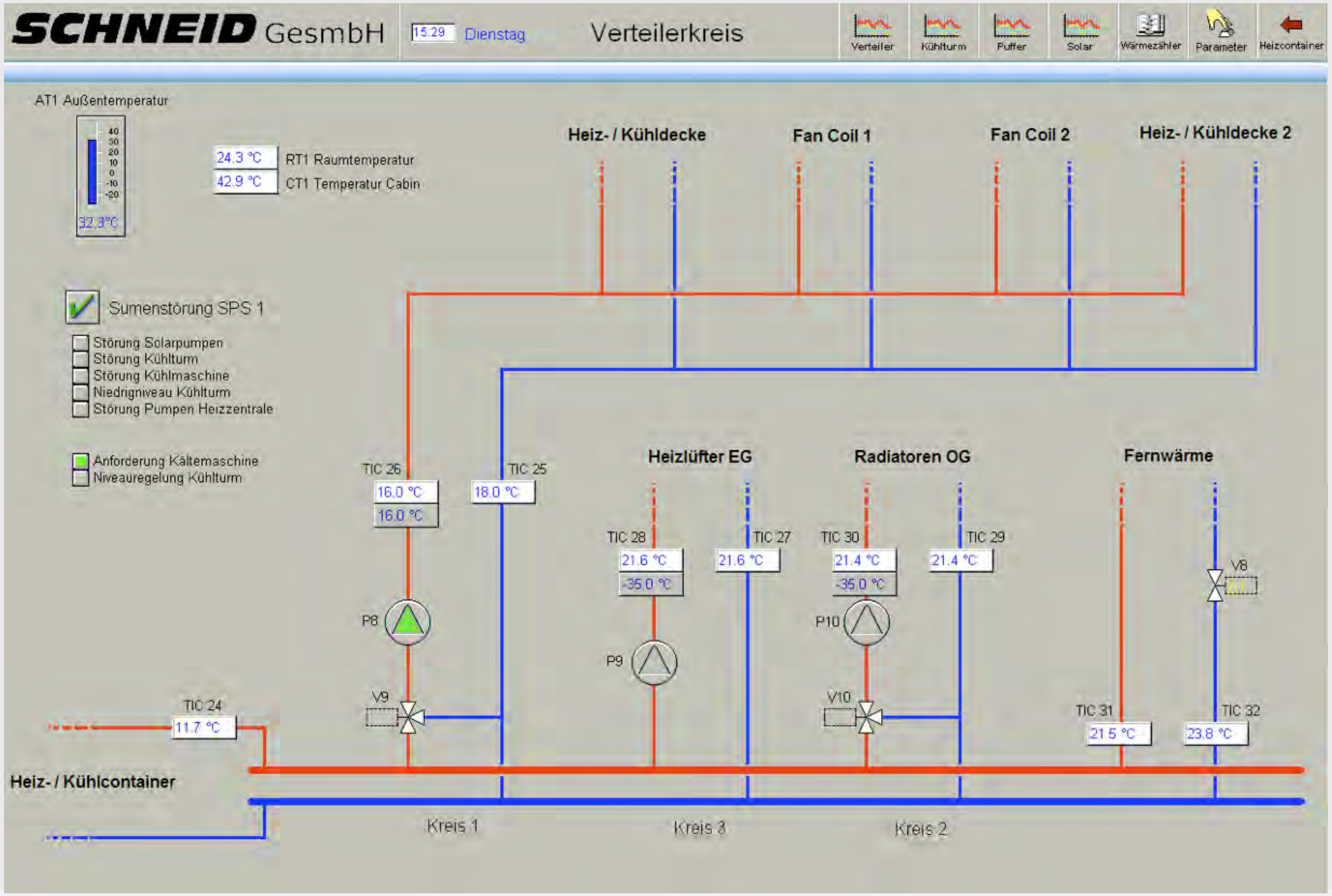
measured together:

E1 + E2
E4 + E9
E7+ E_free
Q7 + Q_free

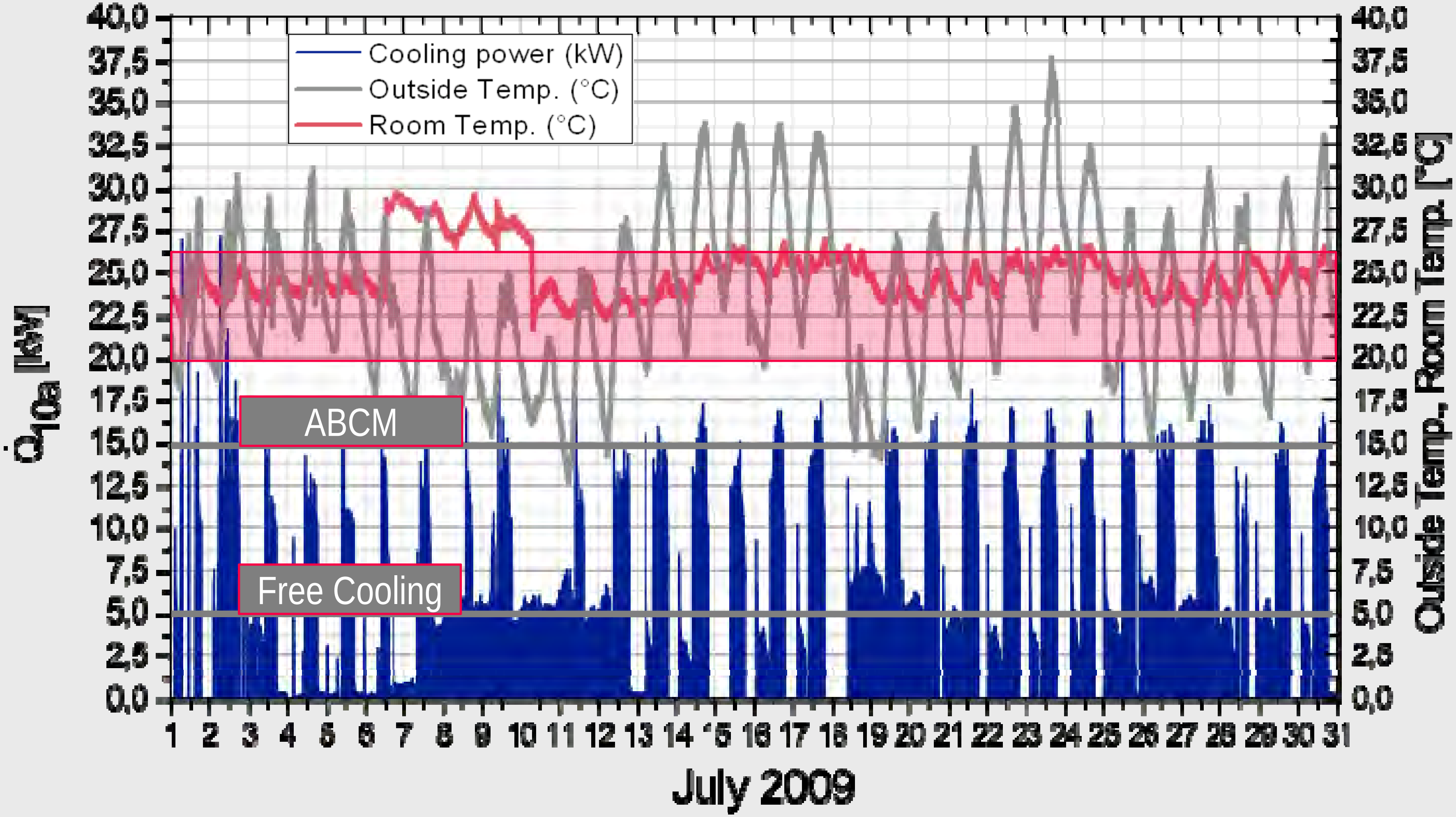
✓4 heat flow meters
✓6 electricity counters
✓Level 3 Monitoring

Monitoring Screen

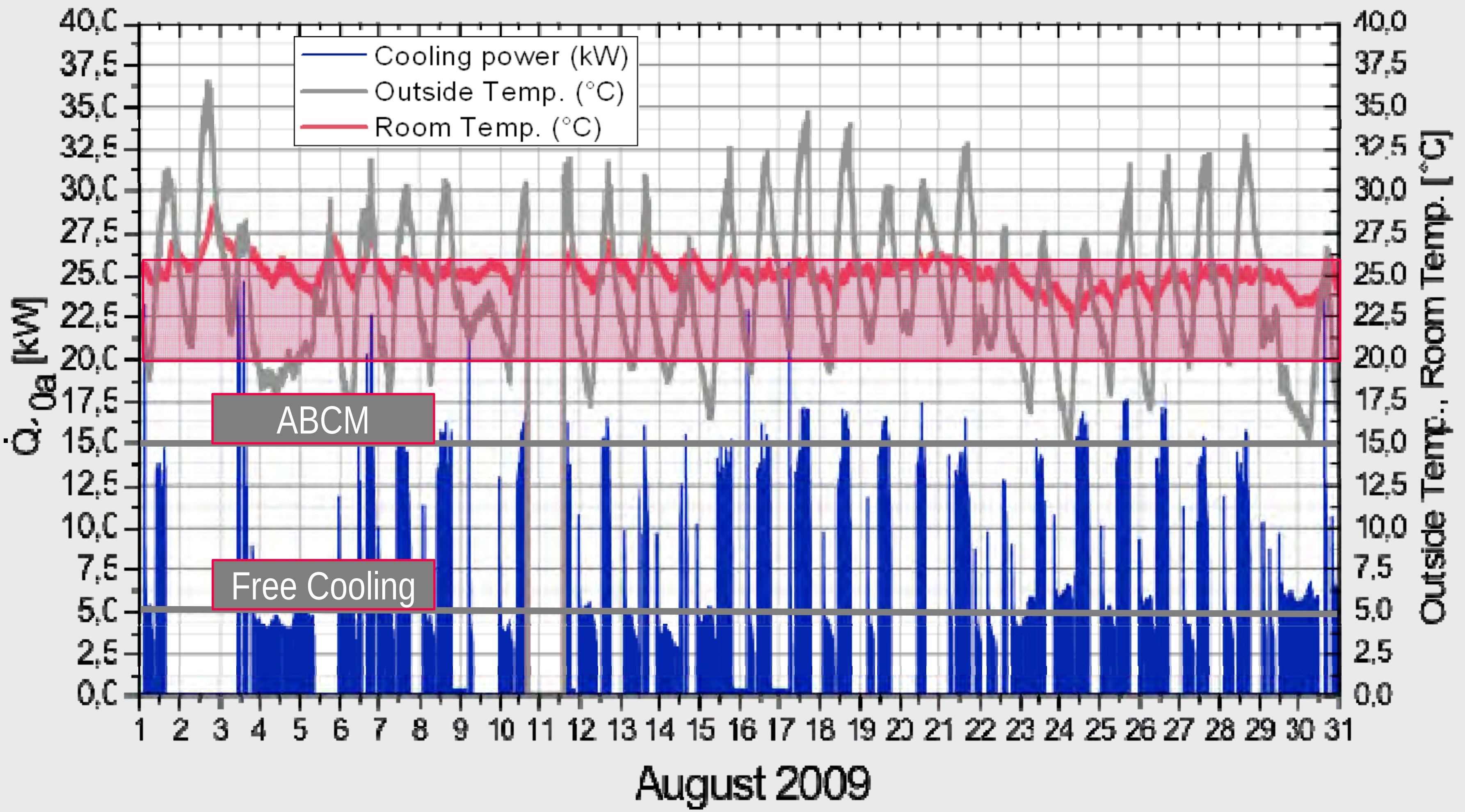
Introduction
COOLCABIN
Monitoring
Results
Free cooling
Conclusion



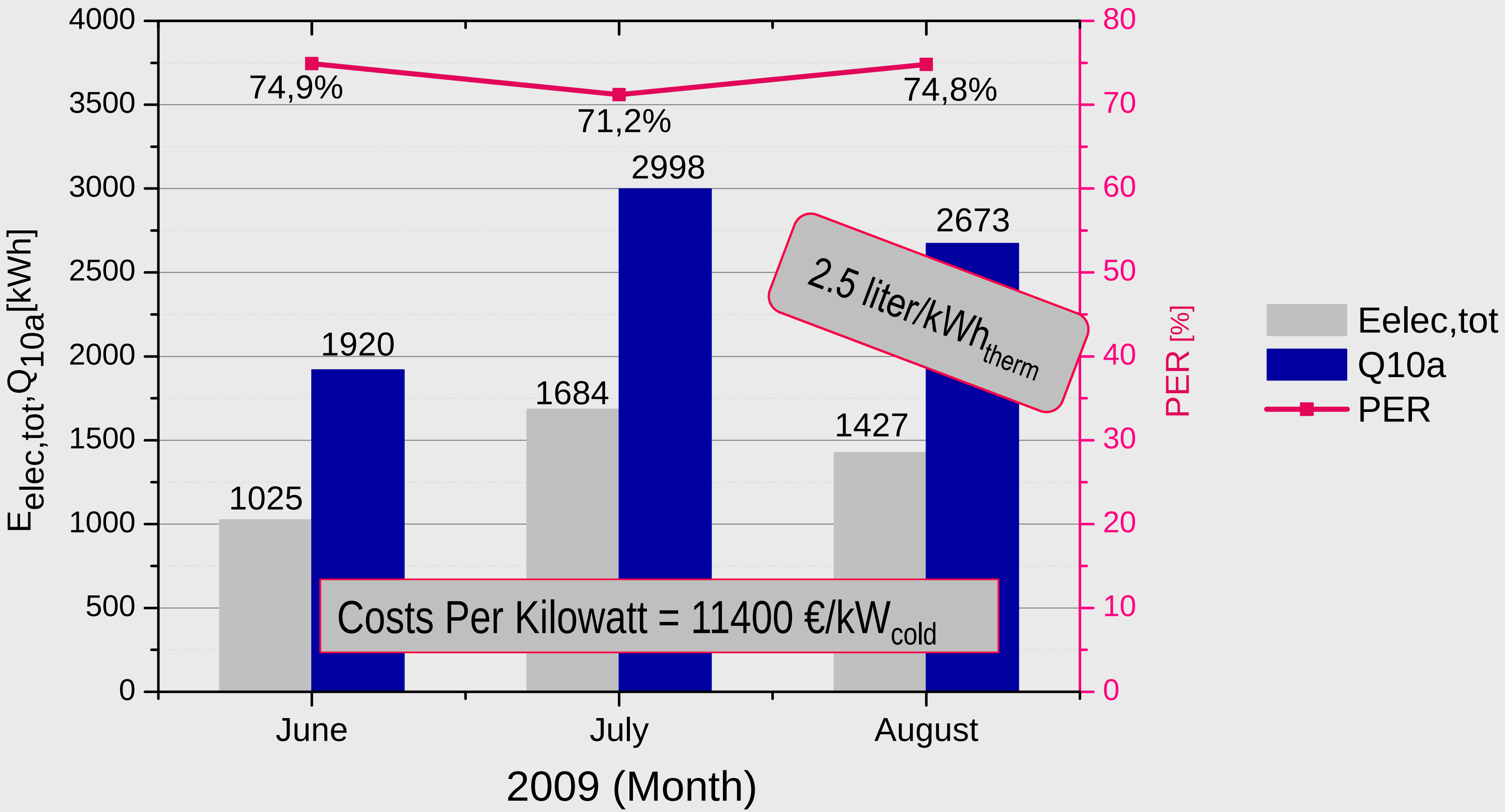
Measurement results



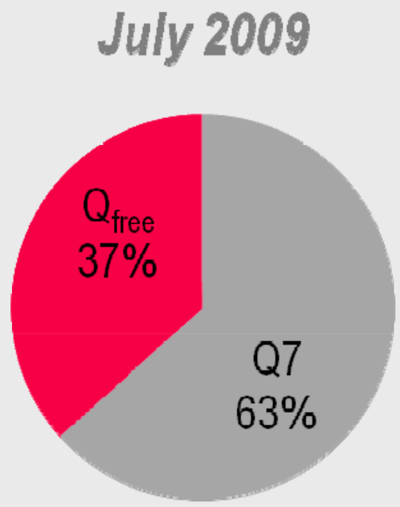
Measurement results

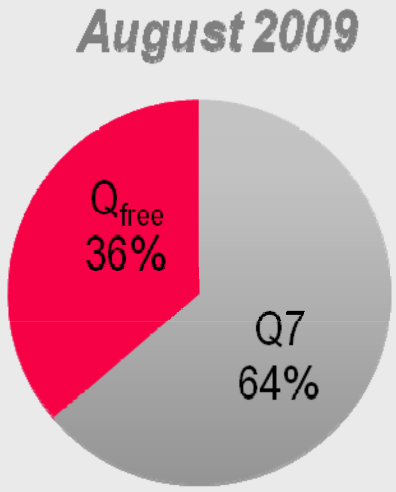


Primary Energy Ratio (PER)



Free cooling

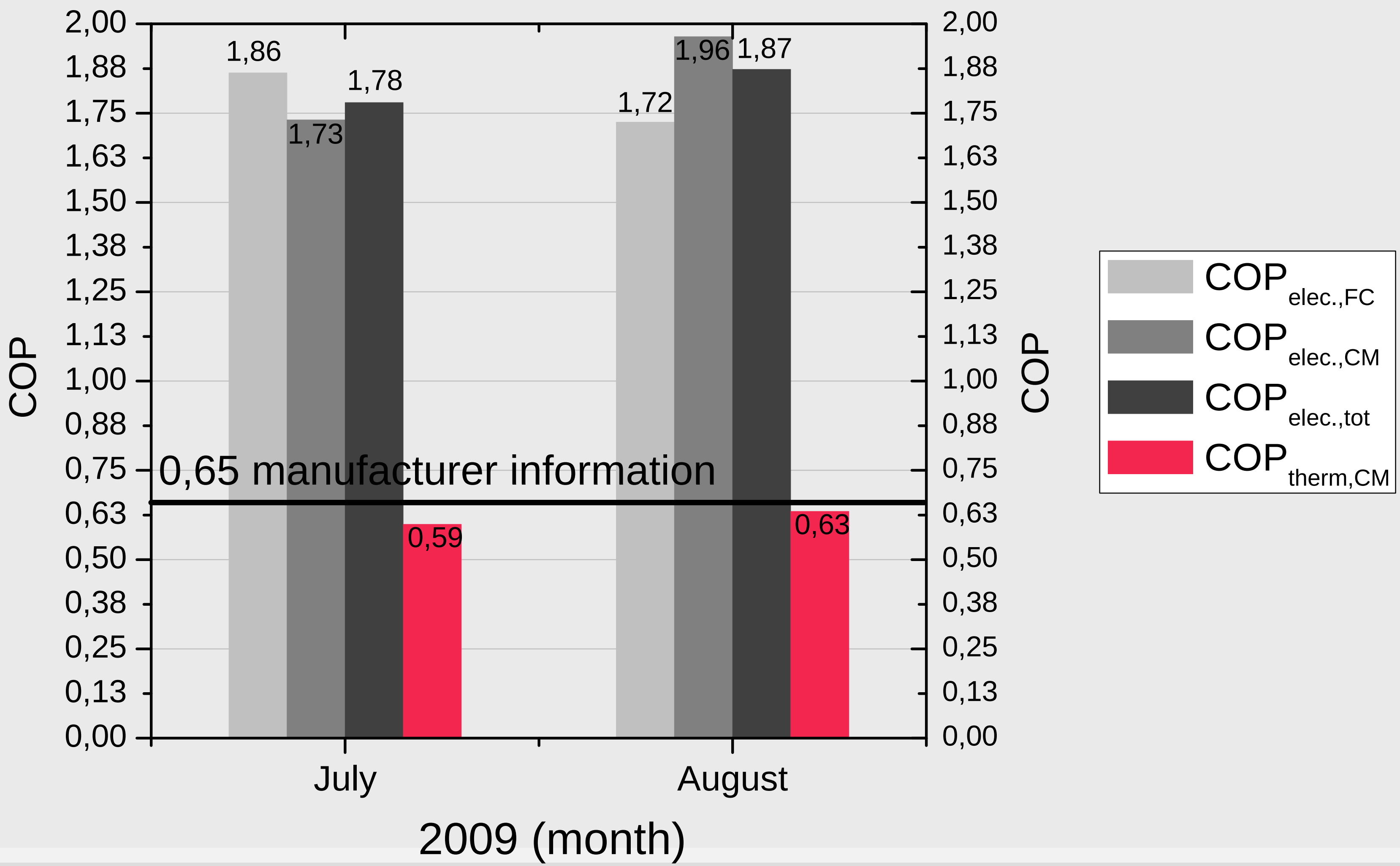




Q10a	Q7	Q_{free}
2998.0 kWh	1898,5 kWh	1099,5 kWh
100%	63.33%	36.67%

Q10a	Q7	Q_{free}
2673.0 kWh	1702.0 kWh	971.0 kWh
100%	63.67%	36.33%

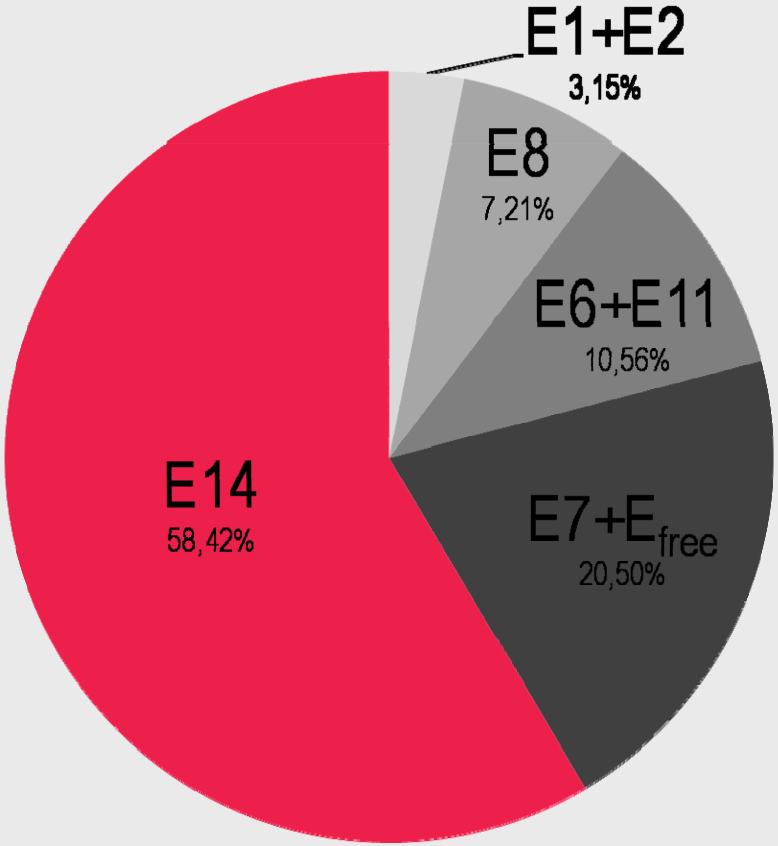
COP


COP
2,00
1,88
1,75
1,63
1,50
1,38
1,25
1,13
1,00
0,88
0,75
0,63
0,50
0,38
0,25
0,13
0,00
1,86
1,73
1,78
0,59
1,72
1,96
1,87
0,63
0,65 manufacturer information
July
August
2009 (month)
COP elec.,FC
COP elec.,CM
COP elec.,tot
COP therm,CM

Conclusion

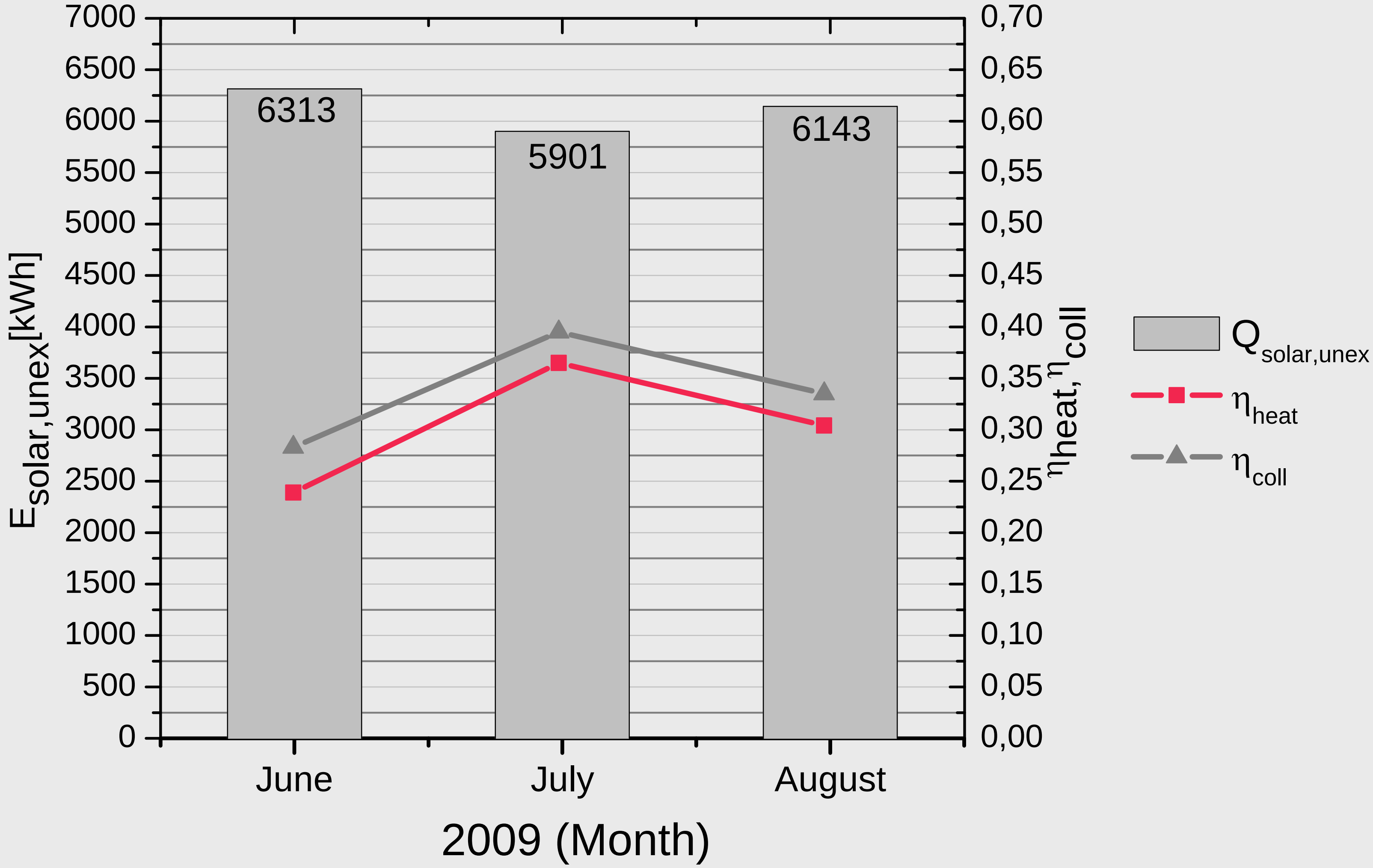
- The Coolcabin has been monitored over the summer 2009
- ~1/3 of the cooling load was done with free cooling
- whole system was running very reliable
- Optimizing operations after commissioning is extremely important for an efficient process
- Optimization potential in control system
- will be monitored summer 2010

$E_{elec.,tot}$ August 2009



E1+E2	45,0 kWh
E8	103,0 kWh
E6+E11	151,0 kWh
E7+E$_{free}$	293,0 kWh
E14	835,0 kWh
$E_{elec.,tot}$	1427,0 kWh

Solar efficiencies



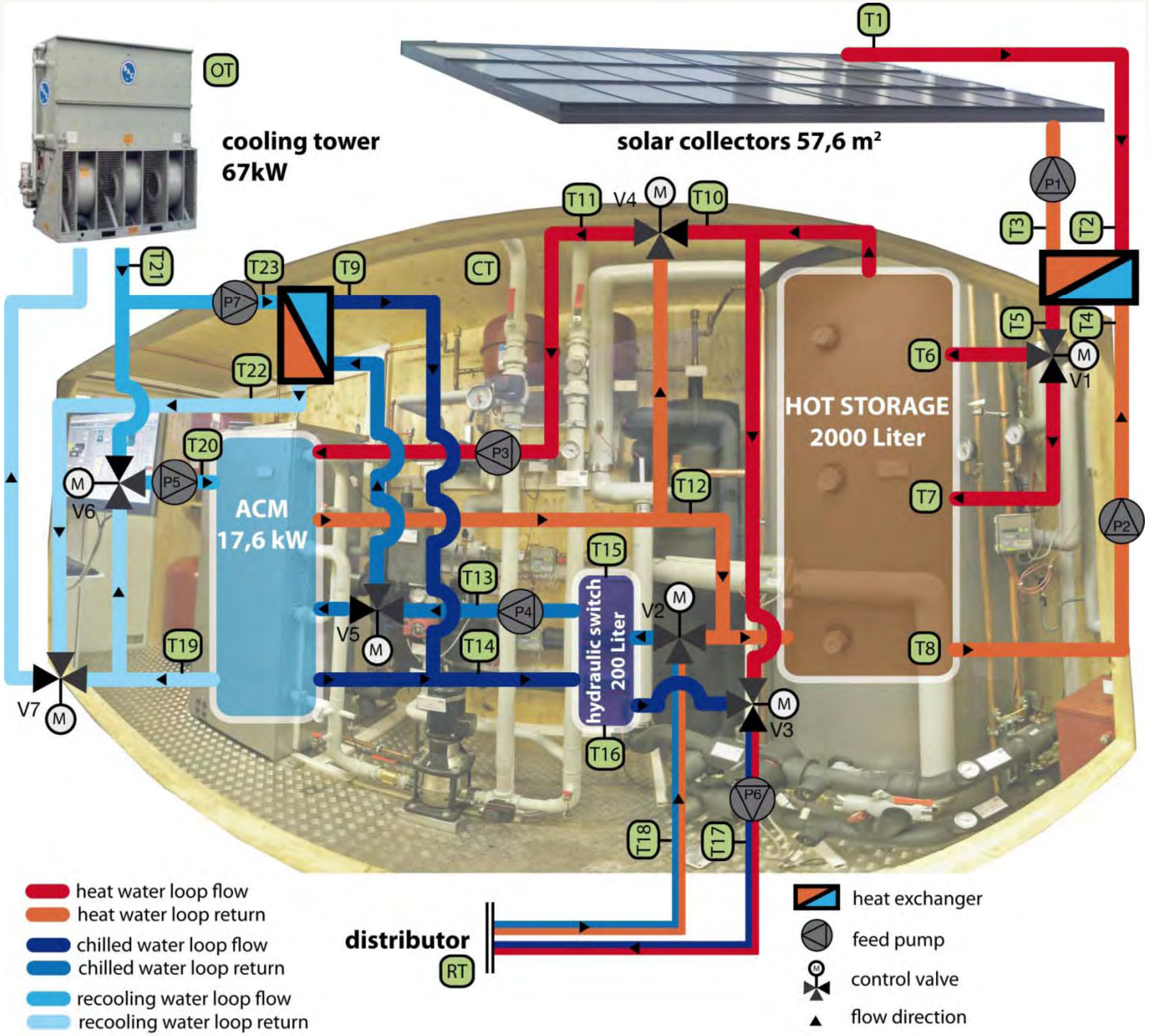
ANHANG IWT 8

Neyer Daniel, Präsentation: Coolcabin – Results 2010, expert meeting Graz, 27/28.09.2010

Coolcabin - Results 2010

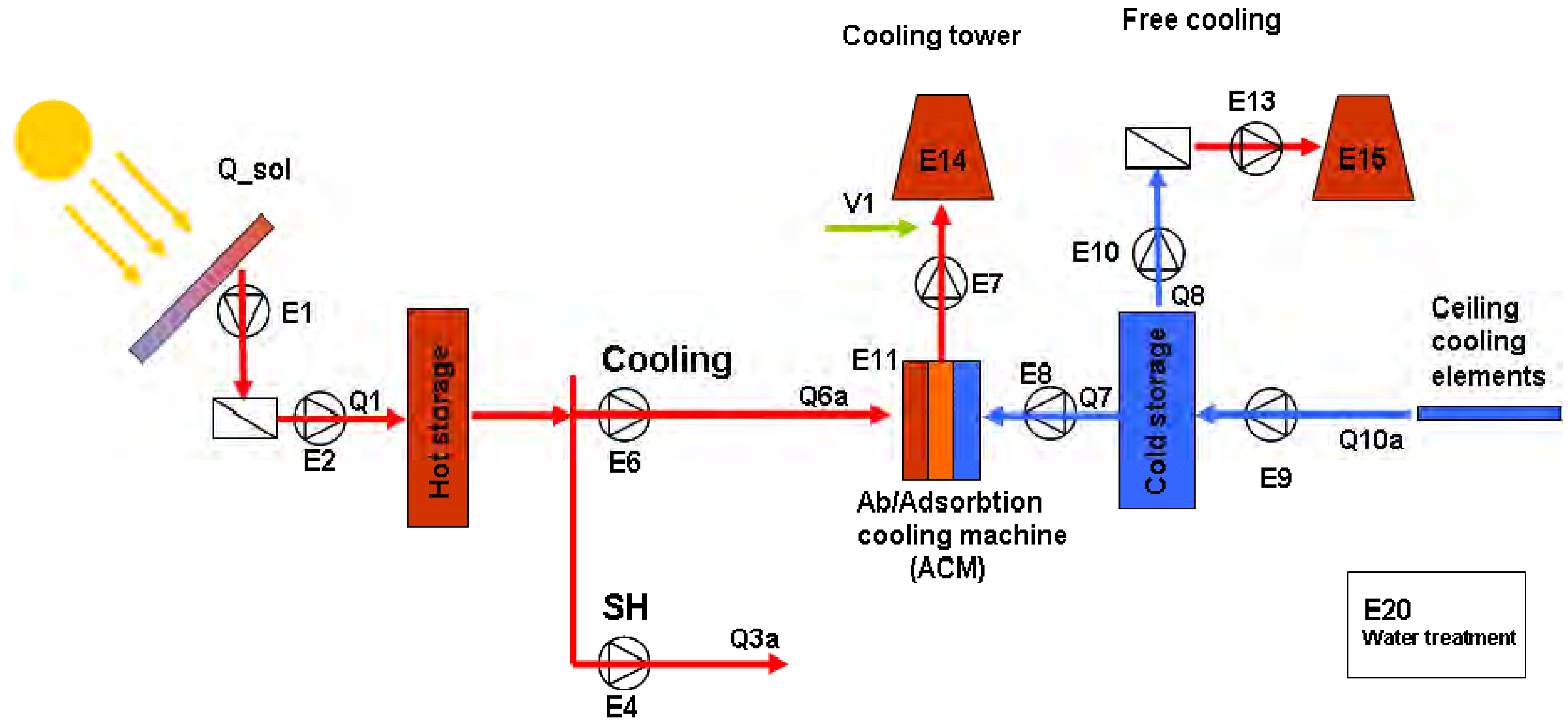
Daniel Neyer

Contents

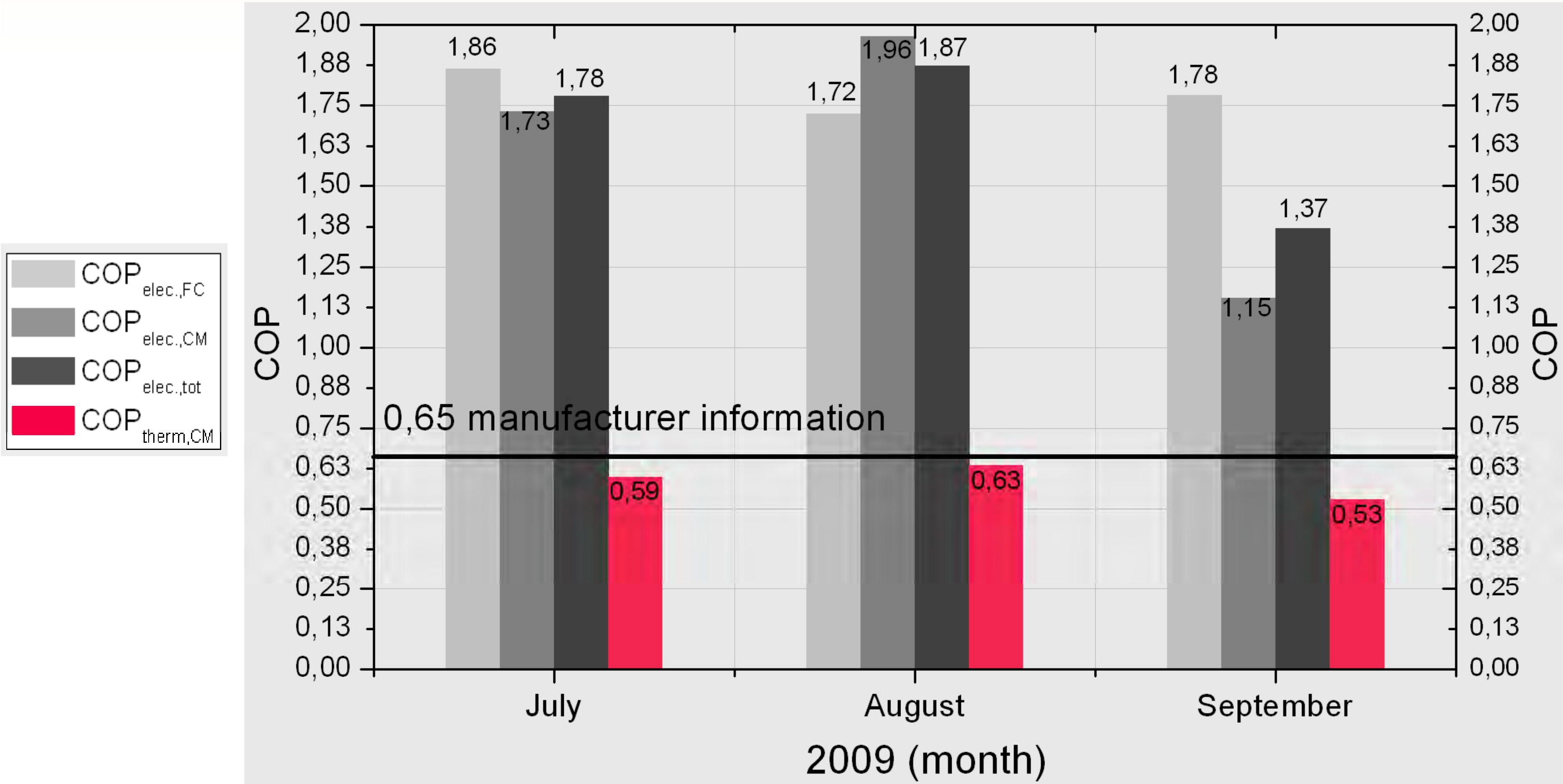


source: Thomas Weissensteiner

- Level III
- Since June 2009
- …



Results summer 2009



source: Thomas Weissensteiner

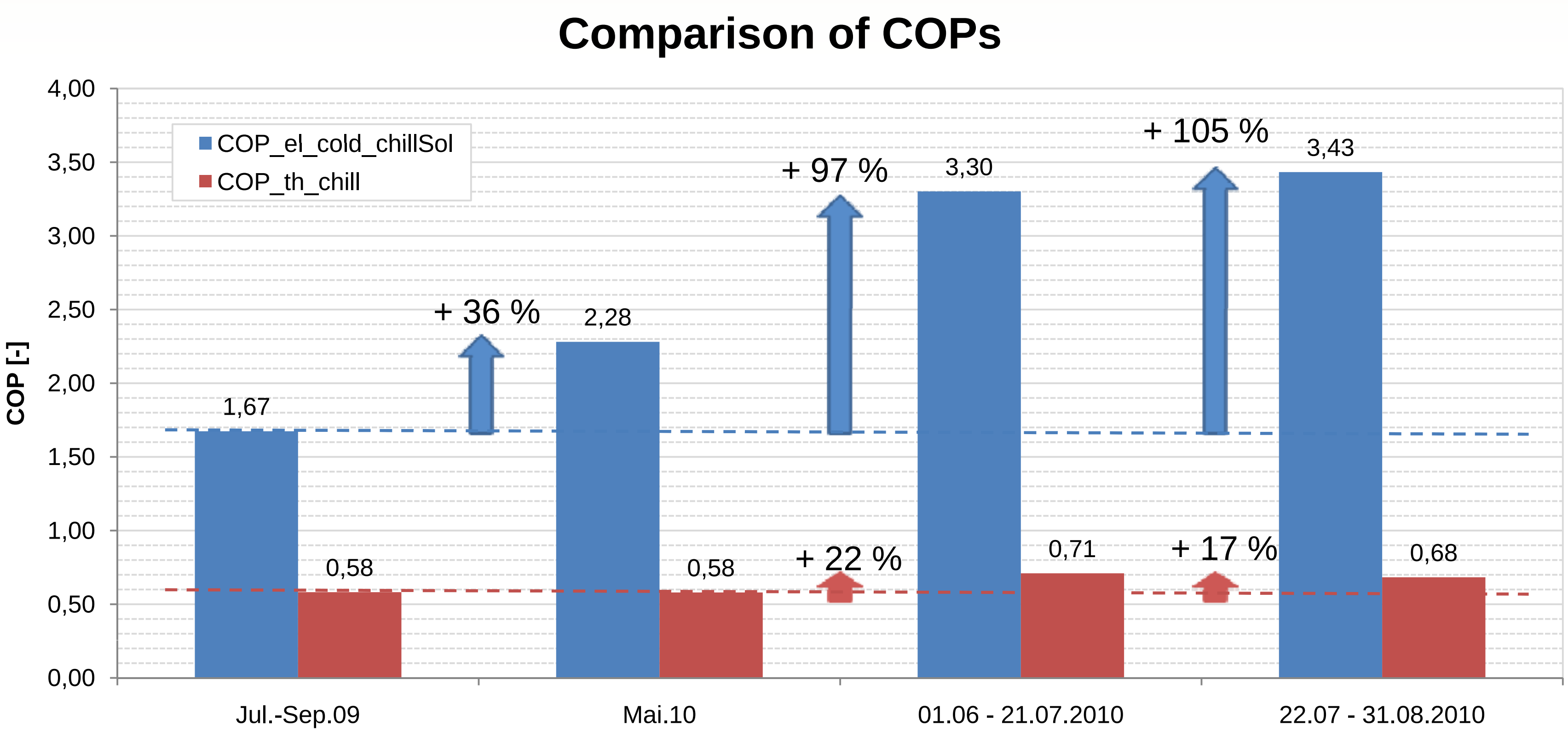
Improvements

- Shut-off Free Cooling Mode
- Service / maintenance of cooling tower

- Control parameters
  - Set temperatures (supply-temp., starting temp. storage)

- Control system
  - Cooling tower ventilator – speed controlled

Results 2009 vs. 2010



maintenance

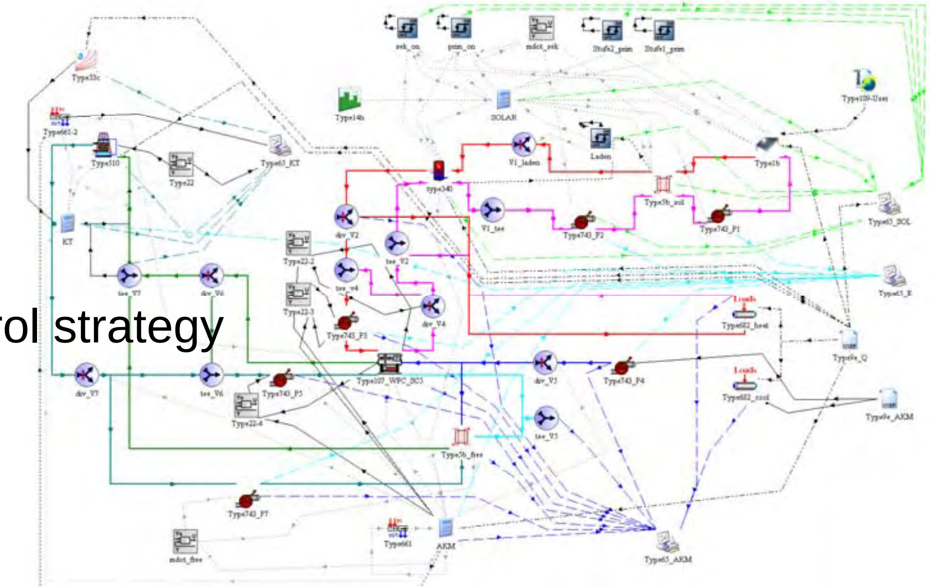
set temperatures
supply temp. 17°→ 15°
start temp. 75°→ 88°

speed control cooling tower fan
recooling temp. 29° → 31°
start temp. 88°→ 80°

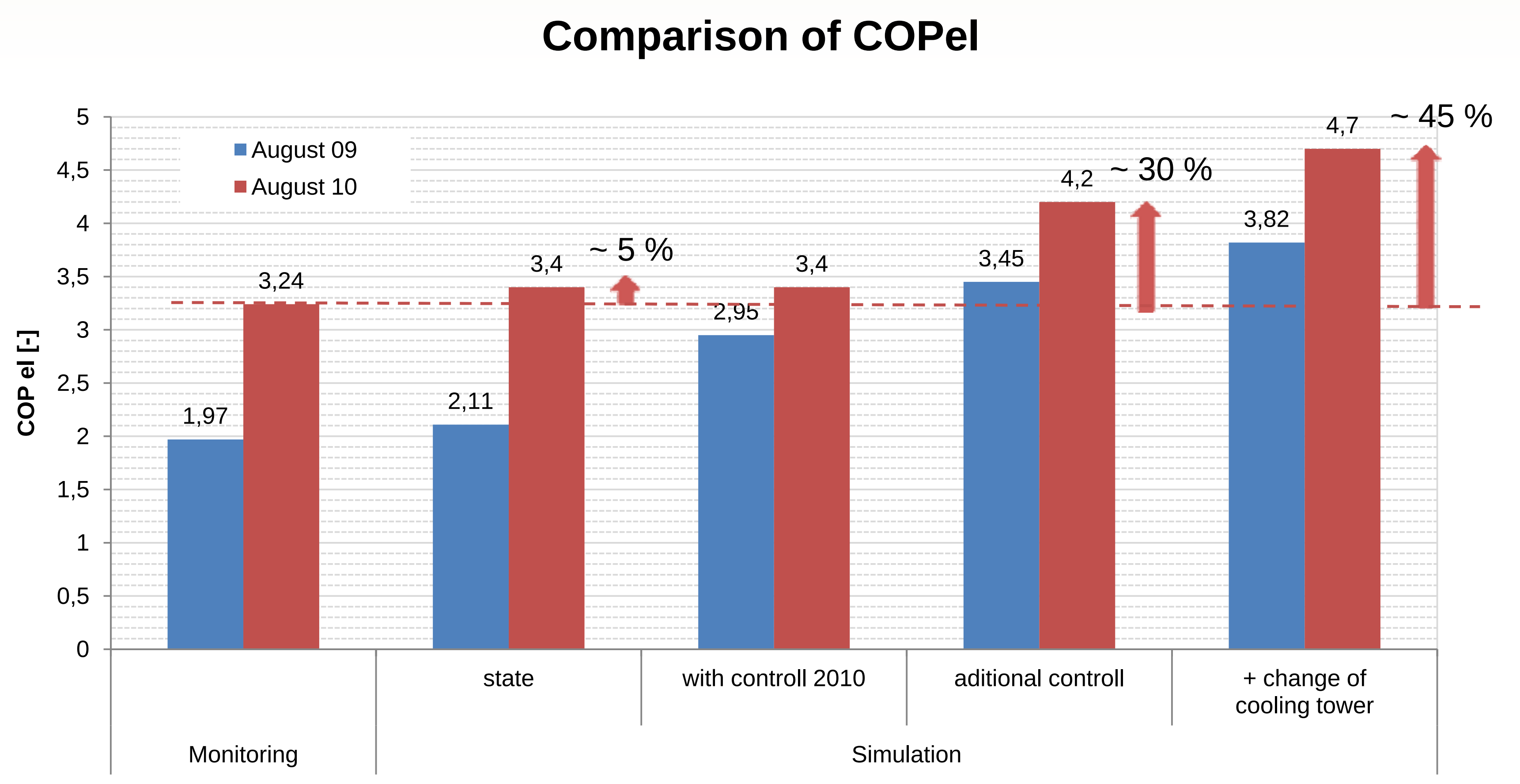
Simulation

- TRNSYS
- Simplified models (Characteristic curve models)
- Validation of single models (±10% deviation)
- Assembling and implementation of control strategy
- Load-file approach
- Only August

- 4 steps
  - Situation 2009
  - Situation 2010
  - Further improvements of control strategy
  - + change of cooling tower
    (no validated data)



Simulation Results



Results for 2009 includes Free Cooling!

Free Cooling

- summer 2009 Ø COPel of 1.8
- Autumn09/Spring10 Ø COPel of 12.4
- → strongly depending on weather conditions

- Shut-off May 2010

- according to Simulation daily COPel up to 11
- More results with free floating approach

- we'll reactivate Free Cooling with better control strategy

Conclusion

- Improvements
  - COPel +105%, COPth +17%
  - Higher loads due to weather condition + lower supply temp.
  - Service and maintenance is important (descaling?).
  - Simple changes of parameters show a huge effect
  - Speed control of cooling tower fan - small effects (effect without set temp.?)

→ relatively high improvements, absolute values still not optimal (COPel 3.4)

- Further possibilities of improvements?
  - Including water pump of cooling tower in control strategy? (Solar Cooling Opt)
  - Speed control of recooling pump? (Solar Cooling Opt)

- Simulations-Results
  - Satisfying but still improvable
  - Potential for improvement up to COPel of 4.5
  - Free Cooling Mode should be activated again

Thanks for your attention

ANHANG IWT9

Weissensteiner T., Streicher W., Präsentation: thermal cooling, PV compression cooling and pure compression cooling – an economic comparison on the basis of two real plants, expert meeting Aarhus, 27.04.2010

Thermal cooling, PV compression cooling and pure compression cooling - an economic comparison on the basis of two real plants

27.04.2010

Thomas Weissensteiner, Wolfgang Streicher

Solar Cooling -
More than Solar thermal ??

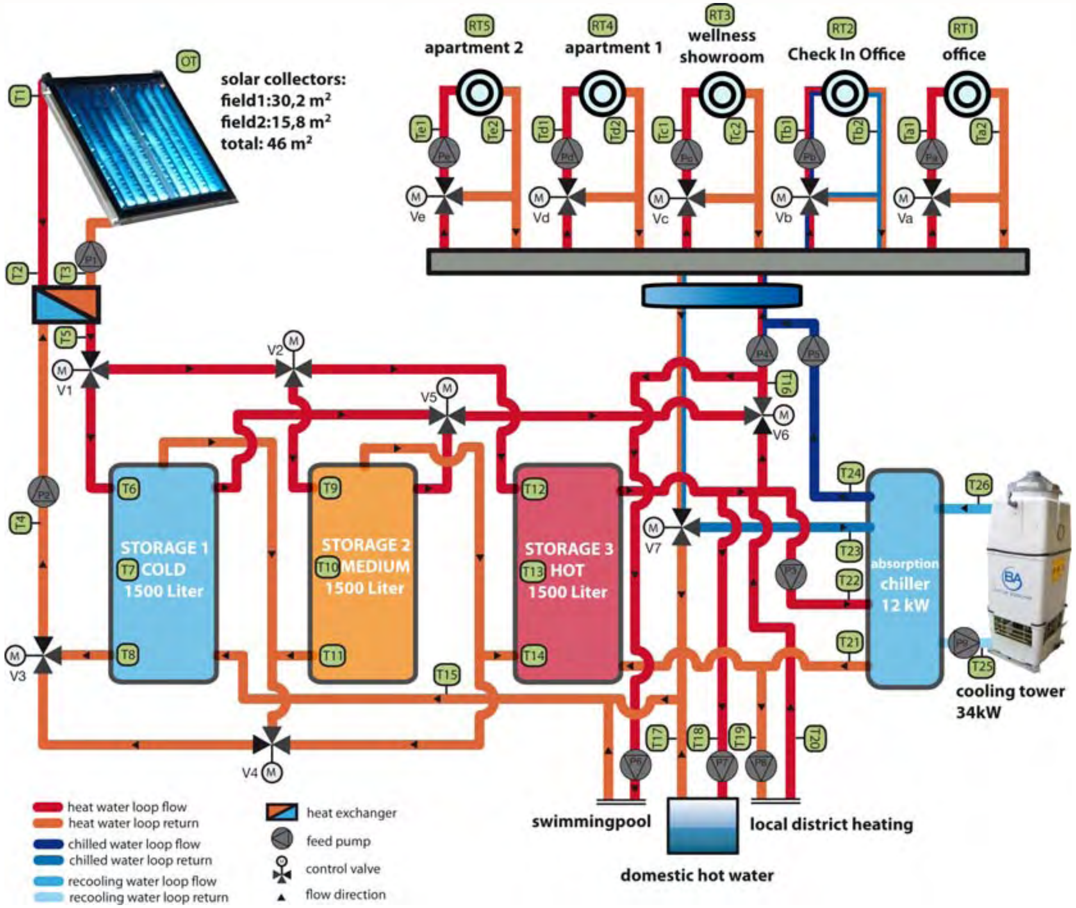
Solar Cooling

- **Solar Thermal Cooling**
- **Photovoltaic driven Compression Cooling**

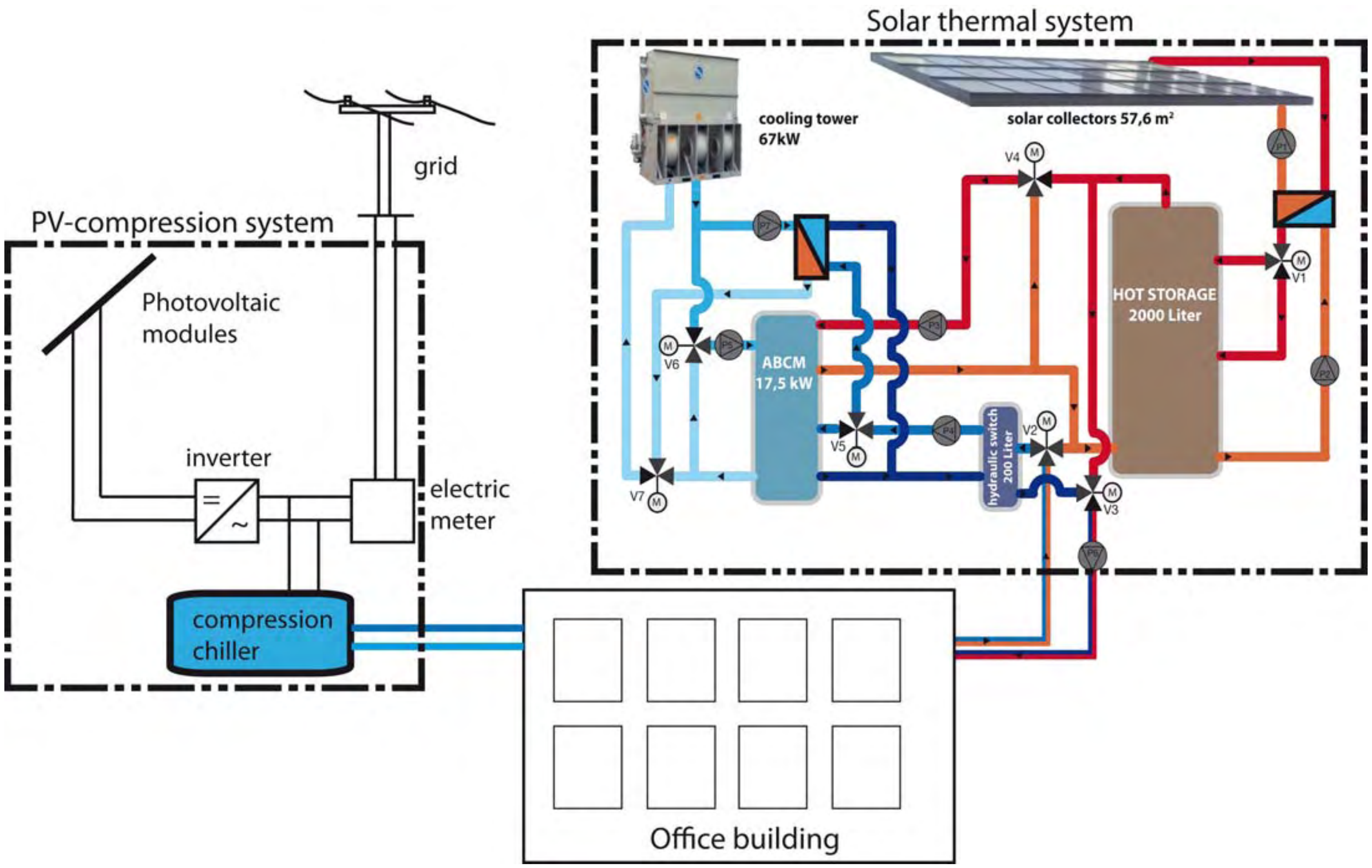
To be compared with:

- **Conventional Compression Cooling**

The „Bachler“ plant



System definition of solar thermal and PV-compression cooling systems in the cost comparison (example Solid plant).



Boundary conditions general

- Efficiencies are estimated after completed optimization
- Gross prices including all taxes, dues, insurance, planning and installation costs
- No subsidies,
- Feed in tariffs for surplus solar thermal in district heat = 1: 1
- Feed in tariff for surplus electricity from PV = 1:1
- No costs for area needed for plant

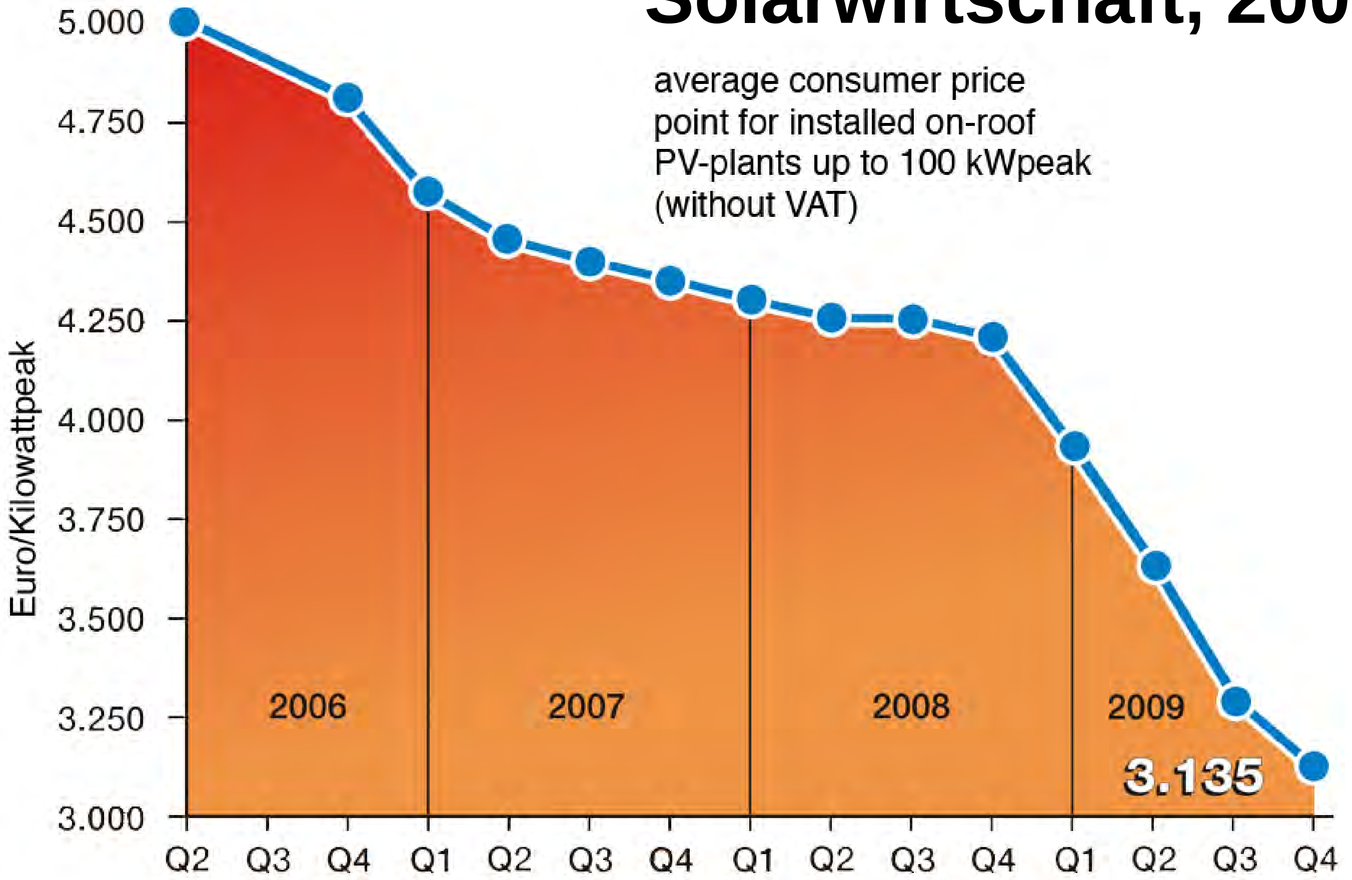
- Monthly Calculation

Boundary conditions
Solar Thermal

Table 1: boundary conditions and assumptions for the solar thermal systems

Solar thermal cooling	Solid Coolcabin	Pink Bachler	units
Absorption chiller power	17.6	12	kW_{cold}
Solar collectors area	58	46	m^2
Thermal COP	0.63	0.6	-
Electrical COP	5	5	-
Investment (inkl 20 %VAT)	10,500	6500	€/kW
Rate of interest	5	5	%
Term of the loan	15	15	a
Ø district heat prize	0.0918	0.0918	€
Maintenance costs	30	30	€/month
Electricity tariff	0.1507	0.1507	€

Average consumer price point for installed on-roof PV-plants up to 100 kWp (without VAT) (Bundesverband Solarwirtschaft, 2009)



Boundary Conditions PV-Compression Cooling

Table 1: boundary conditions and assumptions of the PV-compression chiller systems

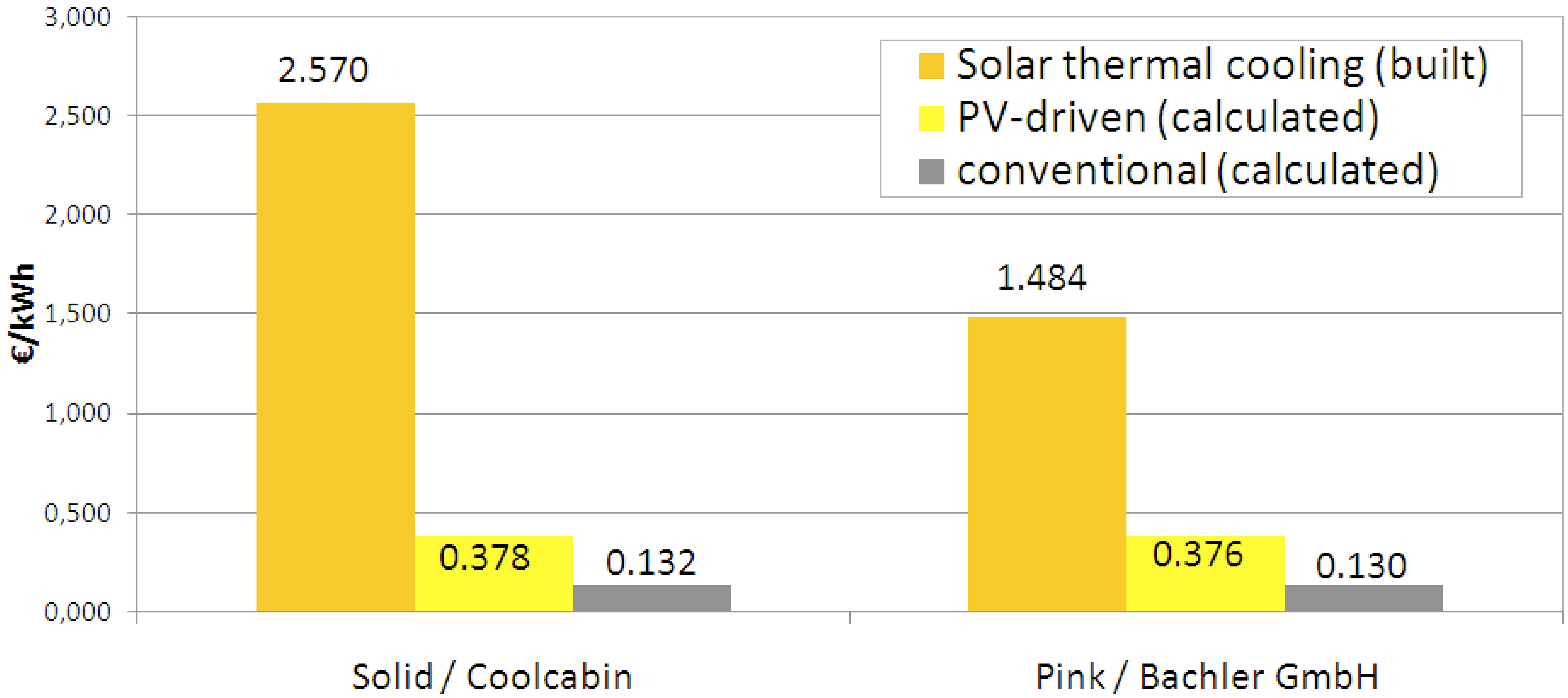
PV-compression cooling	Solid Coolcabin	Pink Bachler	units
Compression chiller power	17	12	kW_{cold}
PV-panels area	58.5	40,5	m²
Power PV-panels	6.5	5	kW_{peak}
Chiller COP	2.8	2.8	-
Rate of interest	5	5	%
Term of the loan	15	15	a
Maintenance costs	22	17	€/month
Electricity tariff	0.1507	0.1507	€
Investment costs PV	3915.6		€/kW_{peak}
Investment costs Chiller	300		€/kW_{cold}

Monthly Calculation

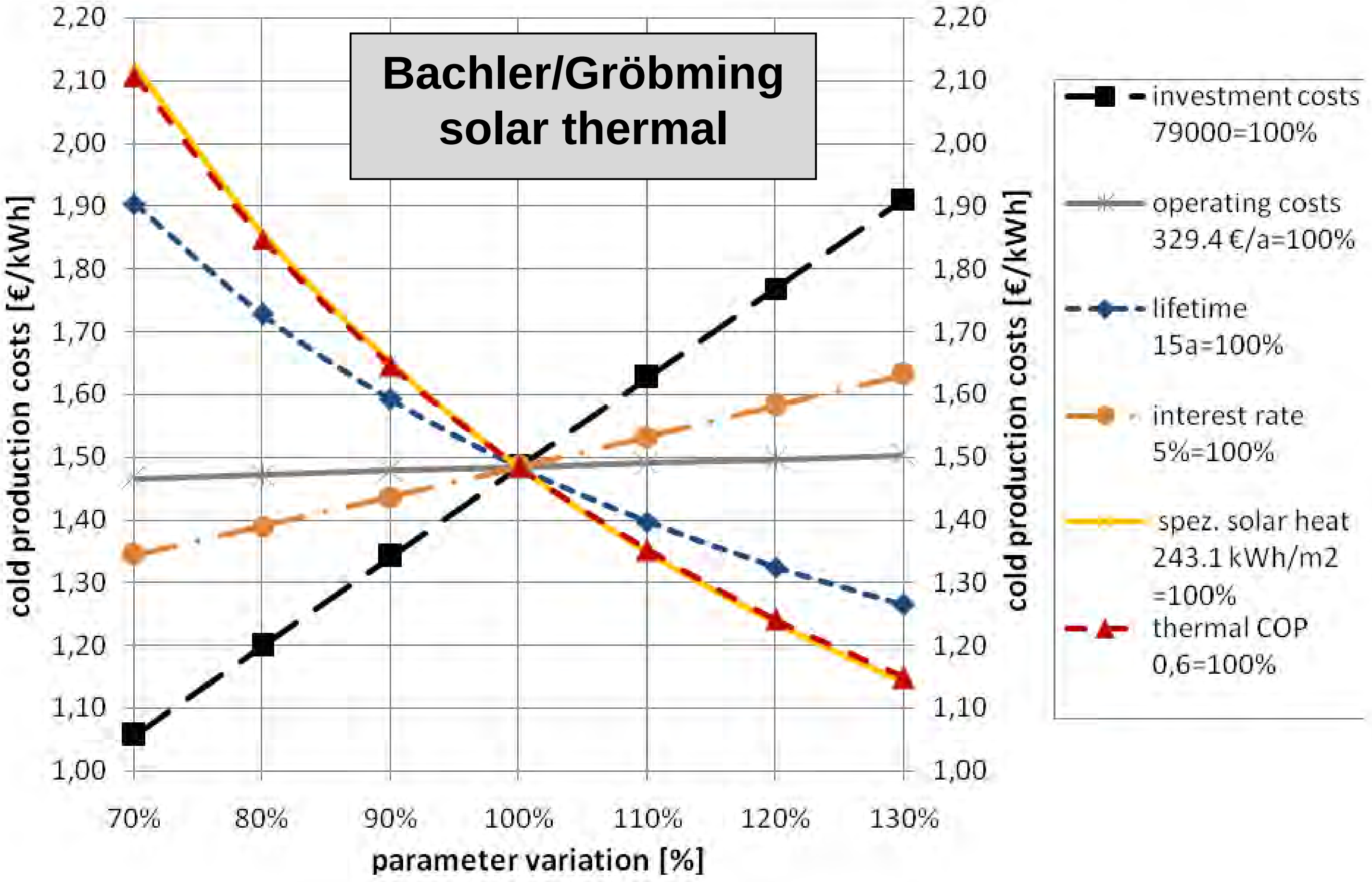
Table 1: Monthly cost calculation of the solar thermal cooling plant in Gröbming

Pink solar thermal	specific usable solar heat	usable solar heat *ush*	effective produced cold load *cl*	heat revenues hr_m	Auxiliary electricity costs ac_m	Operating costs oc_m	Investment costs ic_m	Total costs tc_m
unit	kWh/m^2	kWh	kWh	€	€	€	€	€
January	4.5	207.0	0	19.00	-1.55	-12.55	-634.25	-646.8
February	2.8	128.8	0	11.82	-0.96	-19.14	-634.25	-653.4
March	8.5	391.0	0	35.89	-2.93	2.96	-634.25	-631.3
April	20.7	952.2	0	87.41	-7.14	50.27	-634.25	-584.0
May	33.0	1518.0	910.8	0	-27.32	-57.32	-634.25	-691.6
June	34.2	1573.2	943.9	0	-28.32	-58.32	-634.25	-692.6
July	58.6	2695.6	1617.4	0	-48.52	-78.52	-634.25	-712.8
August	46.2	2125.2	1275.1	0	-38.25	-68.25	-634.25	-702.5
September	21.8	1004.9	602.9	0	-18.09	-48.09	-634.25	-682.3
October	10.1	463.2	0	42.52	-3.47	9.05	-634.25	-625.2
November	1.7	78.2	0	7.17	-0.59	-23.40	-634.25	-657.7
December	1.0	46	0	4.22	-0.35	-26.12	-634.25	-660.4
total	243.1	11183.3	5350.1	208.06	-177.50	-329.44	-7611.00	-7940.5

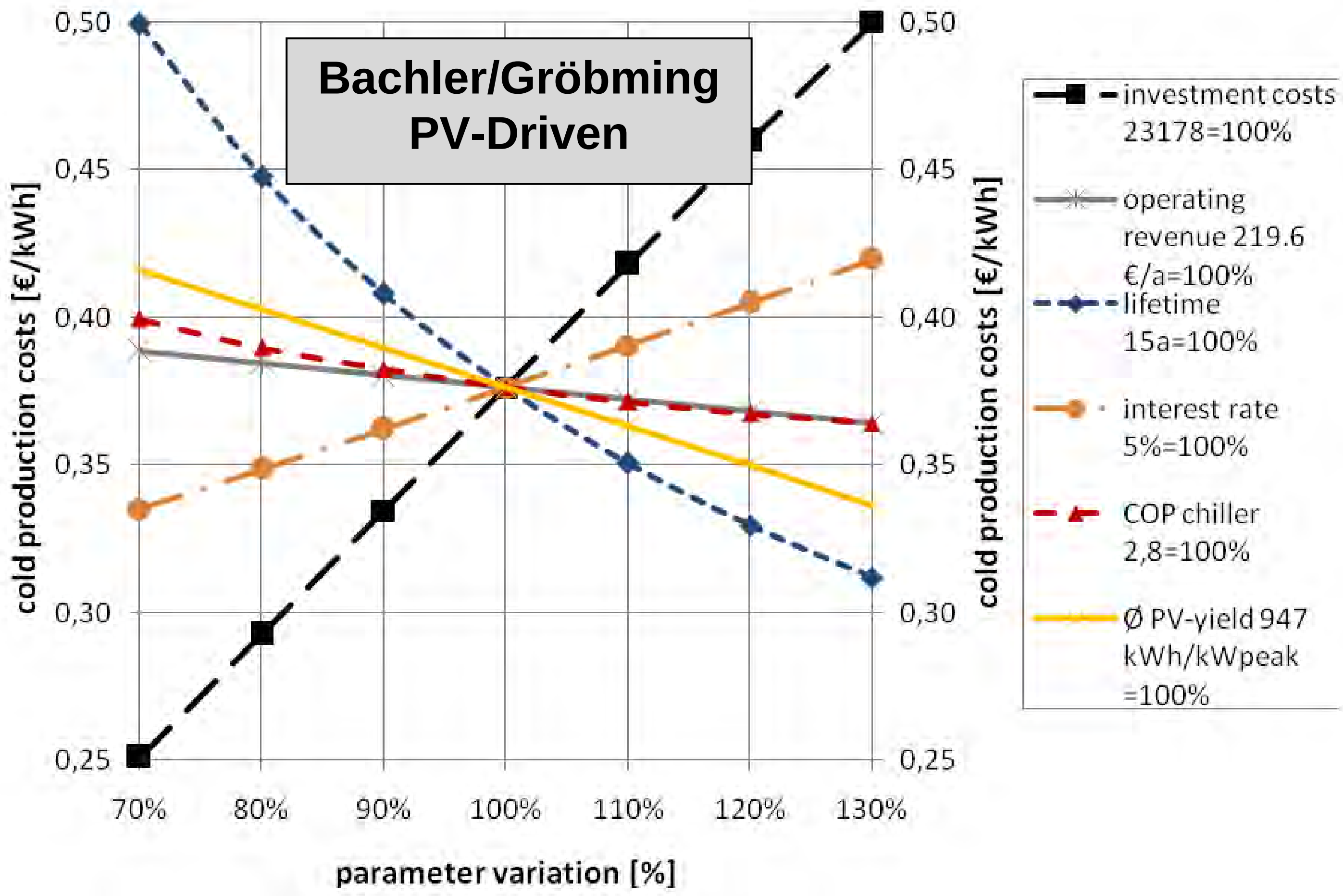
Results


3,000
2,500
2,000
1,500
1,000
0,500
0,000
€/kWh
2.570
0.378
0.132
1.484
0.376
0.130
Solar thermal cooling (built)
PV-driven (calculated)
conventional (calculated)
Solid / Coolcabin
Pink / Bachler GmbH

Parameter variation analysis of the solar thermal cost calculation at the Bachler Plant



Parameter variation analysis of the solar thermal cost calculation at the Bachler Plant



Conclusion

- Solar thermal cooling is very expensive in the moment (10 times of conventional and 4 times more the PV-drive cooling)
- This may be better in southern climates with more cooling (but feed in tariffs were calculated for all surplus heat)
- Of course, solar thermal is still prototyping for the system, but most of the components are already standard
- PV-driven compression is still about 3 times more expensive than conventional cooling
- Discussion on PV-driven compression should not be neglected

To Dos !!!!

- Improvement of the electrical COP
- Investment costs will be the main challenge for a real market

ANHANG IWT10

Neyer, Daniel; Streicher, Wolfgang; Weissensteiner, Thomas (2010), Practical experience of two small scale cooling plants and cost comparison to PV driven chillers. EuroSun 2010 - International Conference on Solar Heating, Cooling and Buildings. 28.09. - 01.10.2010, Graz. Saint Maur: OCS Associates (Europe), ISBN 978-3-901425-13-4, S. 88

Practical experience of two small scale cooling plants and cost comparison to PV driven chillers

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Abstract

This work describes the measurement results and a cost analysis of two small scale solar thermal cooling plants located in Austria. All monitoring results refer to IEA SHC Task 38 level 3. The measurements were taken during the cooling period of summer 2009.

Most of the monitoring results are not as high as expected; especially the electrical coefficient of performance (COP) could not reach the estimated values. The highest total monthly electrical COP of plant 1 could only reach a value of 1.87. For plant 2 an average electrical COPs of 3.09 were measured between August and September 2009.

A cost comparison between the two solar thermal cooling plants and photovoltaic (PV) driven compression chillers was accomplished. The results show 4 to 7 times higher cold production costs (€/kWh) of the solar thermal plants compared to the PV driven systems and even 11 to 19 times higher production costs compared to the conventional compression chiller variant.

These facts show that the economical performances of these two solar thermal systems in Austria are currently not competitive to other cooling systems.

1. Introduction

The two solar thermal cooling plants are monitored and analyzed following the monitoring procedure of IEA SHC Task 38 level 3. Main results of this method are Energy fluxes, total electrical consumption, thermal and electrical COPs and Primary Energy Ratio.

The first system (plant 1) includes a 17.6 kW absorption cooling machine running with lithium-bromide, a 2’000 litre hot storage and 57.6 m^2 flat-plate collectors. Bypassing the absorption chiller and just using the cooling tower for free cooling can be realised. There is no backup for the cooling task installed. The cold water is distributed through radiant ceilings in order to cool the 573.5 m^2 office area.

Plant 2 is a retrofitting to an existing solar heating system and was realised in 2007. The cooling load is supplied to an office building. The solar cooling system has no conventional backup. It includes a 12 kW ammonia water absorption chiller and three 1’500 litre hot storage tanks. The flat-plate collectors (46m^2) are partly façade integrated partly mounted on the ground.

2. Practical Experience and Monitoring Results

During summer 2009 practical experience was made with these two small scale solar thermal cooling plants. Each component fulfils its function, but the assembled systems have significant technical difficulties to reach the expectations. The most important results are shown in Table 1.

Table 1: Calculated results of the solar thermal cooling plants

	electrical COP	thermal COP	time period
plant 1	1.8	0.6	Aug.2009-May2010
plant 2	3.2	0.55	Aug.2009-May2010

2.1. Plant 1

From July to September 2009 a thermal coefficient of performance of 0.6 was obtained. About 1/3 of the cooling demand was delivered by free cooling and 2/3 by solar thermal cooling. The electrical coefficient of performance for free cooling (COPelec) was 1.8 and for solar cooling 1.6. The entire coefficient of performance of the whole cooling period was approximately 1.8 and between Oct. 09 and April 10 about 8.2 for free cooling. No solar thermal cooling was obtained in winter time.
The whole system is running reliable but with a poor overall performance. Especially the electrical consumption is very high. The chiller was running very reliable throughout the whole summer. The chiller reaches the expected power level and an adequate COP. Free cooling mode is activated mainly in night times but runs with poor efficiency. The highest electrical loads, 75% of the whole electrical consumption, are needed by cooling tower and pump. A reduction in electrical consumption can be achieved by installation of a speed-regulated fan, improvements of the control system and by small changes of the hydraulics.

2.2. Plant 2

Between August, 21st and September, 30th a thermal COP of 0.565 could be reached. At the same time the electrical COP ranged from 0.5 to 5 and achieved 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.
When the plant was running, it worked as expected. Nevertheless the average values over a longer time period are still improvable. In the beginning of the cooling season cycling 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 with the measurement and control system occurred. The wet cooling tower worked very well. 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 seems to be overloaded and not clearly arranged.

Following suggestions are made to improve this solar thermal cooling plant:

- adapting the control system in order to make clear division between heating and cooling periods
- including the cooling cycle in the DHW priority rule to shut off the absorption chiller when DHW is tapped and avoid a short circuit in the cooling circuit
- to match the inertia system and to use the concrete core activation properly the control system should include a time delay or a dead band to shut off the cooling circuit
- rising the starting temperature in the control strategy to avoid a cycling behaviour of the chiller

2. Cost comparison

2.1. Structured boundary conditions and assumptions

PV driven solar cooling plants are compared to these two existing plants, where most of the data is known (e.g. investment- and operation costs). Therefore the PV solar cooling plant is dimensioned in a way, to produce the same cooling load as the two plants. Ongoing a sensitivity analysis will be shown in order to find crucial assumed parameters.

The price of photovoltaic systems decreased approximately 37% from the beginning of 2006 until the fourth quarter of 2009 with a strong downward trend in the last four quarters. Similar numbers are shown for Austria in [1]. The prices include the PV-modules, the inverter, miscellaneous components as well as the planning and installation on site.

The system boundary for the cost comparison is shown in Figure 1. Inside this boundary all components, which are related to the cold production, are taken into account. The distribution system is not included in the comparison. Neither costs of monitoring as installed in the current plants are included. All prices are overall gross prices including all taxes, dues, insurance, planning and installation costs.

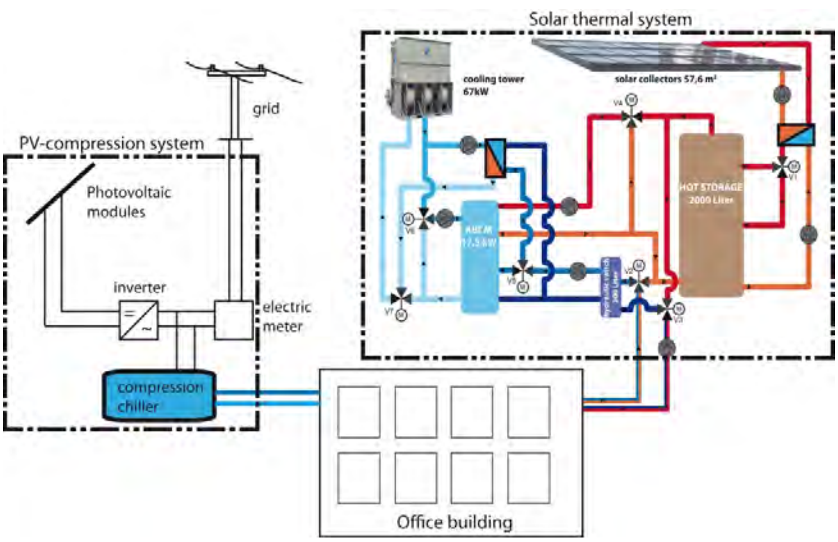

Fig. 1. System Boundary for thermal and PV driven plants

The analysis is based on the annuity method assuming that the plant is financed through a private loan. All the plant specific gaining such as extra produced electricity in the case of the PV driven system or the useful solar heat for space heating (SH) or domestic hot water (DHW) at the solar thermal plant are subtracted from the operating costs. No governmental subsidizes are included. The feed-in tariffs are equal to electricity costs, no investment subsidies are considered. Neither, needed roof area for the collectors nor installation spaces for the plants are taken into account but are compared and discussed.

The used values related to the solar thermal plants are taken out of the measurements from summer 2009. For the PV plant up-to-date costs and yield values were collected [1].

2.2. Solar thermal cooling plants

Table 2 summarizes the assumptions of the two solar thermal plants. The assumed thermal COPs are average expected values for the next summer after optimization of the system. Also the electrical COPs are expected values for the oncoming summer (2010). These values are linked to the existing plants and refer to the practically experiences of the Task 38 members [7].

Table 2. Boundary conditions and assumptions for the solar thermal systems

Solar thermal cooling	Plant 1	Plant 2	Units	References
Absorption chiller power	17.6	12	kW_{cold}	-
Solar collectors area	58	46	m²	-
Thermal COP	0.63	0.6	-	-
Electrical COP	5	5	-	-
Rate of interest	5	5	%	-
Term of the loan	15	15	a	-
Ø district heat prize	0.0918	0.0918	€	[2]
Maintenance costs	30	30	€/month	[1]
Electricity tariff	0.1507	0.1507	€	[3]

2.3. Photovoltaic solar cooling plants

Table 3 shows the assumptions and boundary conditions for the photovoltaic based system. The PV area and power values are calculated in order to reach at least the same cooling load as the solar thermal cooling plant. The rated COP of the compression chiller is assumed with 2.8 following the IEA Task 38 [4] taking part loads and practical conditions into consideration.

Table 3. Boundary conditions and assumptions of the PV-compression chiller systems

PV-compression cooling	**Plant 1**	**Plant 2**	**Units**	**Reference**
Compression chiller power	17	12	kW_{cold}	-
PV-panels area	58.5	40.5	m²	-
Power PV-panels	6.5	5	kW_{peak}	-
Chiller COP	2.8	2.8	-	[4]
Maintenance costs	22	17	€/month	[1]
Investment costs PV	3915.6		€/kW_{peak}	[1]
Investment costs Chiller	300		€/kW_{cold}	[5]

2.4 general assumptions

Measured results of the two small-scale solar cooling plants are implemented into the cost comparison. The calculation of the cost is done on a monthly basis, investment costs are calculated with the annuity method.

By definition the cooling system is working between May and September. In the heating season the solar plants are used either to support the SH and DHW production or to produce electricity in case of the photovoltaic panels. The monthly usable solar heat, which is not used for cooling, is calculated as heat revenue. Equally additionally produced electricity is calculated as electricity revenue. The auxiliary electricity costs of the solar thermal cooling plants are calculated through the electrical COP of the plant and the electricity tariff. In times were the cooling plant is not working an auxiliary electricity consumption of 5% of the produced usable heat was assumed for the solar plant.

3. Results and sensitivity analysis

For plant 1 cold production costs of 2.57 €/kWh_{cold} were calculated. Plant 2 has cold production costs of 1.48 €/kWh_{cold}. This difference results out of the diversity of initial investment costs. All results are compared and shown in Figure 2.

The specific average photovoltaic yield for Germany between 2004 and 2009 [6] was assumed as a pessimistic yield for the PV panels. The compression chiller was dimensioned for the same cooling power as the absorption chiller. For the PV driven compression cooling system at plant 1 cold production costs of 0.38 €/kWh were calculated. The PV variant at plant 2 reaches nearly the same price with the smaller plant. Figure 2 illustrates the annual cold production costs for the two monitored systems compared with PV-driven and conventional cooling systems.

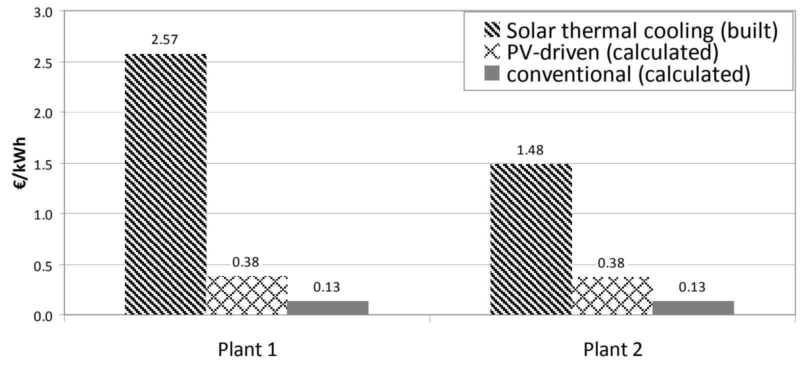

Fig. 2. Cold production costs for the two monitored systems compared with PV driven and conventional cooling systems

The results show huge cost differences between the solar thermal and the other two cooling systems. In case of plant 2 the solar thermal produced kWh_{cold} costs nearly 4 times more than the kWh_{cold} produced with the PV driven solar cooling system. The cold productions costs are approximately 11 times higher for solar thermal cooling system compared to the conventional compression cooling system. The conventional cooling costs in this figure were calculated with the same boundary conditions as the PV driven compression chiller, excluding the photovoltaic costs and revenues. Figure 2 shows that the first generation of small scale solar thermal cooling plants in Austria are not economically competitive to comparable renewable cooling technologies such as the PV driven compression chiller systems.

The next two figures (Figure 3 and 4) show a cost-sensitivity analysis of the solar thermal and the PV driven cooling plant for plant 2. Initial points of this sensitivity analysis are real variables including relatively high investment costs and poor specific solar gains including storage losses.

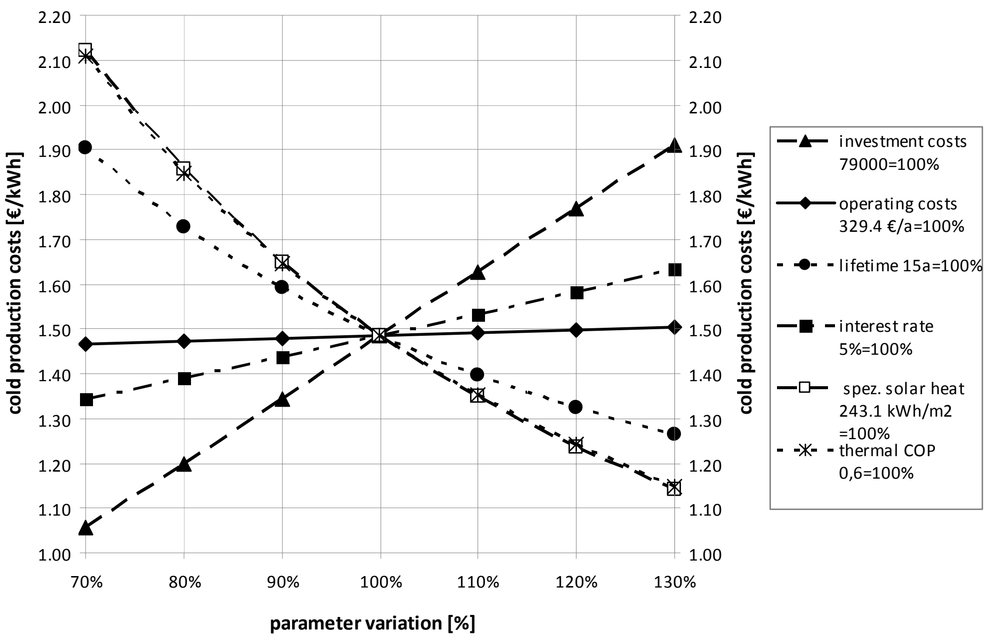

Fig. 3. Parameter variation analysis of the solar thermal cost calculation of plant 2

For the solar thermal part the specific usable solar heat, the thermal COP of the absorption chiller and the investment costs are the most important values. The operating costs and the interest rate have hardly any influence to the cold production costs. All sensitive parameters are measured or known values. Therefore the calculation can be rated as quite feasible.

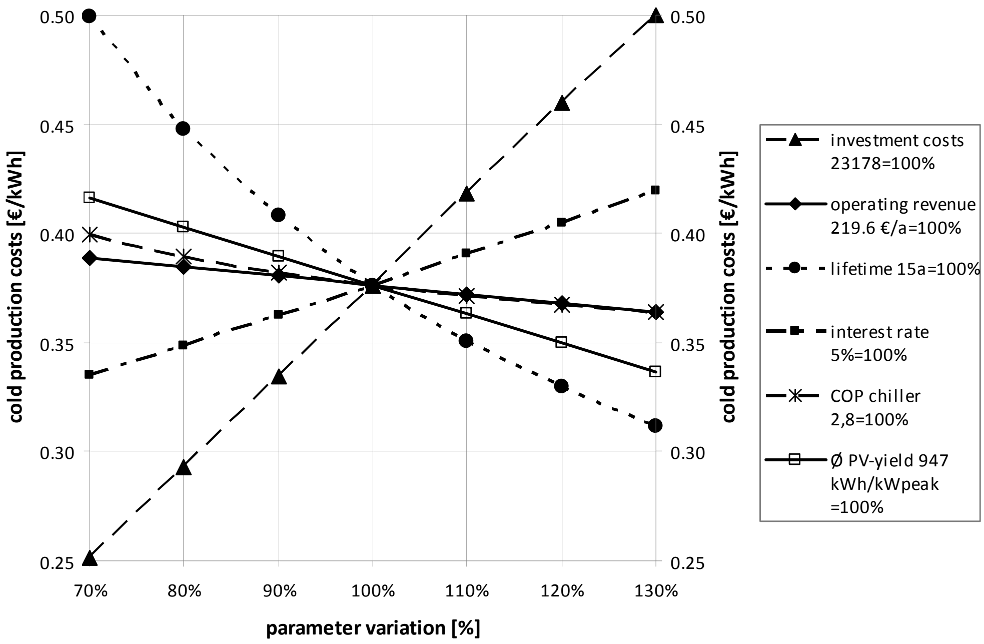

Fig. 4. Parameter variation analysis of the PV-driven cost calculation at plant 2

In case of the PV driven system the most influencing parameters are investment costs and lifetime. The same lifetime has been chosen for all variants so it has no influence to the comparison. The investment costs are assumed based on profound source values but should be taken with caution because of the extensive mounting system at plant 2.

The sensitivity analysis shows that low system losses and therefore higher specific usable solar heat, as well as higher thermal COPs, are crucial for solar thermal cooling plants. In this work only results of the parameter variation of plant 2 are shown. The variations of the plant 1 parameters show similar result.

5. Conclusion

Both plants were running reliable within the monitoring period started in August 2009 up to now. Nevertheless, the overall performance is quite poor mainly due to mistakes in hydraulics, design and control systems. The plants show a huge potential for optimization:

- coordinate design with focus on electrical consumption and high useable solar gains
- optimization of the hydraulic systems
- advanced control systems

The main assumptions for the cost comparison were not including any governmental subsidies, a lifetime of 15 years for both plants, a rate of interest of 5 %, a COP of the compression chiller of 2.8, an electricity tariff of 0.15 €/kWh, district heat cost of 0.09 €/kWh and assumed maintenance costs between 17 and 30 €/month. On the basis of the average consumer price the investment cost for the PV modules were calculated. The results show 4 to 7 times higher cold production costs (€/kWh) of the solar thermal plants compared to the PV driven systems and even 11 to 19 times higher production costs compared to the conventional compression chiller variant. In Figure 3 and 4 results of the parameter variation are shown for plant 2. The most important parameters are the investment costs, the solar yield as well as the thermal COP of the absorption chiller.

The cost comparison shows that the economical performances of these two solar thermal systems in Austria are not competitive to other cooling systems. Taking into account the high optimisation potential of both plants and the high investment costs general statements are only valid for the trends of the crucial parameters. For more general statements this cost comparison should be done with various and more advanced plants. . In case of bigger systems cost reduction and rising competitive capability can be expected.

The two plants will be monitored during this summer (2010). Therefore the implementation of the suggested changes, as well as the further monitoring of both plants, is important. Within a national Austrian Solar Cooling project (SolarCoolingMonitor), which started in November 2009, the two plants will be simulated in order to compute possible physical changes of the plants. A comparison and systematic analysis of the monitoring results measured last summer to the ones of this summer will bring more detailed results in order to estimate further improvement and optimization potentials.

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