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IEA HEAT PUMP CENTRE

NEWSLETTER
VOL. 32
NO. 4/2014

Innovative Technology

Heat Pumps - A key technology for the future

IEA **Heat Pump** CENTRE
NEWSLETTER

Volume 32 - No. 4/2014

In this issue

Heat Pump Centre Newsletter, 4/2014
At the recent IEA Heat Pump Conference, a number of papers described new and innovative technology, covering new sources, components, refrigerants, control strategies, and more. The topic of this issue of the HPC Newsletter is Innovative technology. Similarly to the previous issue, a few of the papers from the conference have been selected, this time covering some of the innovative aspects mentioned above. In addition, and also from the conference, we are given a regional overview from the Asia/Oceania region.

Enjoy your reading!

Johan Berg, Editor

Heat pump news

Regulars

Topical articles

COLOPHON

Published by IEA Heat Pump Centre
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Language editing: Angloscan Ltd.

IEA OECD heat pump centre

Foreword

Roger Nordman
Technical Editor of the IEA HPC Newsletter

New heat pumping technology knowledge – delivered!

Every three years, the heat pump community gathers to participate in the IEA Heat Pump Conference. This year, the conference took place in beautiful Montreal, Canada. Over 300 researchers, manufacturers, consultants and policy-makers shared knowledge and listened to the latest findings regarding research, as well as current market and policy developments. One of the key success factors with the IEA conference is the mixed audience, and the width of areas covered, from technology to market, which gives new perspectives for all participants.

This year was no exception, and even if heat pumping technology is sometimes considered a mature technology, the broad spectrum of research topics and results shows that substantial improvements are still possible, both regarding components, units, systems and application areas.

Some areas for which especially interesting results were presented are, to my mind, hybrid heat pumps, high-temperature heat pumps for industrial applications, control strategies for higher efficiency and efficient operation in smart grids and, finally, large systems for use in cities or very large building complexes.

As you can see, picking winners is not an easy task, but isn't that the beauty with such a conference?

In this issue of the Newsletter, you can read three papers that were presented at the conference, namely "Optimization of control strategies – Switching between passive cooling and reversible heat pump" by Franziska Bockelmann et al., "Techno-economic evaluation of combining heat pump and mechanical vapour compression for the production of low pressure steam from waste heat" by Marc-André Richard et al., and "Development of a high efficient heat pump system using seawater heat source and exhaust energy with the automatic decontamination device" by Jong-Taek Oh et al. These papers describe interesting ways to expand the operating window, increase efficiency and combine sources to get the most out of the heat pump system.

In addition to these papers, the full set of conference proceedings is available for purchase from the Heat Pump Centre. This compilation is probably the best Christmas gift you can buy your fellow heat pump friends, so don't hesitate to check in at *www.heatpumpcentre.org*.

We hope to see you all at the next IEA heat pump conference, to be held in 2017, location to be decided!

Message from the Chairman

Sophie Hosatte-Ducassy,
Stepping-down Chairman,
Executive Committee (ExCo)
of the Heat Pump Programme

After ten years as chairman of the IEA Heat Pump Programme, I decided it was time for me to move on and let someone new lead the Programme. Elections took place at the last Executive Committee meeting recently held in Germany and a new chairman was appointed.

I truly enjoyed working with the committee during my tenure as chairman. I appreciated the interesting, lively and sometimes heated discussions which resulted in an active and vigorous programme. The number of activities realized brought about an increased worldwide interest in the programme. We can list approximately 15 Annexes covering the whole spectrum of activities from research to deployment, 4 conferences (Las Vegas, Zurich, Tokyo and Montreal), the establishment of the prestigious Ritter-von-Rittinger award, and 4 new member countries (South Korea, Italy, Finland and Denmark).

During this period, the energy context has changed towards a more favorable environment for heat pump use: the awareness of environmental impacts (in particular GHG emissions) associated with energy production and use has increased; more renewable energy has been installed; smart-grid is under development; and electricity decarbonization has become a priority. In parallel, new regulations for the replacement of synthetic refrigerants have been approved. As a predominantly electricity-driven technology, and because of their high performance compared to conventional heating technologies, heat pumps are positioned as a preferred solution.

For air conditioning, heat pumps are already a well-established technology. However, due to the increasing demand for increased comfort, particularly in developing countries, there is a need to further increase their performance, reduce their environmental impacts, and address peak load issues. In the building sector, the move towards near- or net-zero energy buildings makes heat pumps an unavoidable solution. Heat pumps will reduce the energy demand and the size, and therefore the cost of renewable energy systems. The industrial sector needs technology solutions to increase its competitiveness; heat management with heat pumps is an attractive solution.

I am confident that, under the new leadership, the Heat Pump Programme will continue to grow and remain a major reference for the use of heat pumps in the world, thus contributing to important reductions in energy use and GHG emissions.

I would like to thank my colleagues, the member delegates, for their support and confidence during these years, and the Heat Pump Centre for their considerable and valuable work in supporting the Programme. I am not breaking all ties with the Programme since I will remain active as the Canadian delegate.

I wish all the best to the new chairman, Stephan Renz, and vice-chairman, Antonio Bouza.

With best regards,
Sophie Hosatte-Ducassy

IEA Heat Pump Programme News

Stephan Renz elected new Chairman of the HPP

At the recent meeting of the Executive Committee (ExCo) of the Heat Pump Programme, Mr. Stephan Renz, Switzerland, was elected new Chairman for the next three years. He succeeds Ms. Sophie Hosatte, Canada, in this role.
Further, the ExCo also elected Mr. Antonio Bouza, USA, to succeed Mr. Claus Börner, Germany, as the Vice-Chairman.

The HPC wishes to thank Ms. Hosatte and Mr. Börner for excellent cooperation. We also wish to congratulate Mr. Renz and Mr. Bouza, and are looking forward to fruitful cooperation.

General

Building performance must deliver occupant comfort at reduced costs

Building performance must deliver occupant comfort at reduced costs, say UK industry experts at the Chartered Institution of Building Services Engineers (CIBSE) Conference. CIBSE's Leadership in Building Performance Conference and Exhibition event, held at the QEII Conference Centre in October brought together experts from across the built environment to discuss the challenges and opportunities of delivering building performance. A recurring theme was that whilst carbon targets and environmental concerns are important, they should not be met by compromising financial performance or occupant comfort.

Ed Gray, Head of Carbon and Energy for Marks and Spencer, CIBSE's 2014 Carbon Champion of the Year, outlined the clear business rationale behind the retailer's 'Plan A' scheme for energy efficiency, which is expected to bring savings of 145 million pounds in 2013/14.

But he made clear that savings are not being made at the expense of customer satisfaction and that it is "vital to create a positive store environment which improves the health and wellbeing of our staff and customers".
Source: www.racplus.com

U.S. DOE Updates National Reference Standard for Commercial Buildings to 90.1-2013

Following preliminary analysis that ASHRAE/IES's 2013 energy efficiency standard contains energy savings over the 2010 standard - 8.5 percent source energy savings and 7.6 site energy savings - the U.S. Department of Energy (DOE) has issued a ruling that establishes the 2013 standard as the commercial building reference standard for state building energy codes.

DOE attributes the greater energy savings to improvements in ANSI/ASHRAE/IES Standard 90.1-2013, Energy Standard for Buildings Except Low-Rise Residential Buildings, related to several areas, including better lighting, fans, commercial refrigeration, boilers and controls.

The determination means that states are required to update their codes to meet or exceed the 2013 standard within two years. Currently, states must meet or exceed the 2010 standard, which serves as the commercial building reference standard for state building energy codes under the federal Energy Conservation and Production Act.

"ASHRAE is pleased with this ruling from the DOE, recognizing the energy savings measures in the standard," ASHRAE President Tom Phoenix said. "Standard 90.1 was an original cornerstone in our efforts to improve building performance, and we continue to strive to increase its efficiency in the future."
Sources: www.ashrae.org

Fifteen ways to cut carbon

BSRIA has provided tips on reducing carbon emissions, following the Intergovernmental Panel on Climate Change's (IPCC) Synthesis Report highlighting the need to reduce carbon use.

According to BSRIA, non-domestic buildings are responsible for around a fifth of the UK's total carbon emissions. As the majority of existing buildings will still be in use by 2050, it argued that improving their energy performance was key in helping the country to meet its emission reduction targets.

BSRIA has compiled a list of 15 low-cost or no-cost energy efficiency measures that building operators could implement.
Source: www.hvnplus.co.uk and www.bsria.co.uk

Renergy award to Dr. Miara from ISE Fraunhofer

Dr. Marek Miara, Head of Group Heat Pumps at Fraunhofer has been awarded with the 2014 RENERGY award in the category "Outstanding Personality in the field of renewable energy and energy efficiency". The awards took place in September 2014 during the 4th International trade fair and Conferences for Renewable Energy and Energy Efficiency (Renexpo 2014) in Warsaw.
Source: www.ehpa.org

Heat islands

Large cities create their own climate in both winter and summer - the so-called urban heat island (UHI) - and this has a significant impact in estimating summer and winter loads for heating and air conditioning.

Summer temperatures in the UK and northern Europe are forecast to increase in the not so distant future, so that by 2040 at least half the summers will experience temperatures similar to the heat wave of 2003, when death rates soared in the United Kingdom. The maximum impact in future will be in the larger cities.

The new *CIBSE guidance TM49 and the accompanying Design Summer Years for London* enable designers to analyse the summer performance of their buildings and investigate the impact of urban macroclimatic factors and climate change when carrying out overheating risk assessments for buildings in London.
Source: www.ejarn.com and JARN, September 25, 2014

Conservation First - Electricity Rate Increases Second!

The Government of Québec has issued an Order in Council reminding the Québec Energy Board that it must consider specific economic, social and environmental factors when deciding on future electricity rate increases for the year 2015-2016.

More specifically, the government directed the Board to consider the limited financial capacities of low-income families who can hardly keep pace with higher energy costs. The government also reiterated that all public institutions must achieve internal efficiency objectives, such as reducing administration costs. Finally, the government re-stated its priority for energy efficiency with a focus on best practices.

Ground source heat pumps in Québec and Canada are not only seen as a renewable energy source but also as an energy efficiency technology for buildings. By asking the Board to consider energy efficiency, the government is coherent with the deployment of its financial assistance programs for increased energy efficiency as well as oil replacement measures.

For more information, see
www.energy.gov.on.ca.

Source: GeoExpress Newsletter No. 14 / GeoExpress Bulletin No. 14

Policy

US and China commit to enhanced co-operation on HFC phase-down

President Obama and President Xi have committed to step up their co-operation on phasing down HFCs as part of their agreement on climate change targets, announced recently in China. They have announced climate change targets which will see the two countries make significant carbon reduction commitments, including China's first public statement of a date for 'peak' emissions. The two presidents committed to 'enhance bilateral cooperation' to begin phasing-down HFCs and to work together in a multilateral context as agreed at their meeting in St. Petersburg on 6 September 2013.

The United States intends to achieve an economy-wide target of reducing its emissions by 26 per cent to 28 per cent below its 2005 level in 2025 and to make best efforts to reduce its emissions by 28 per cent.

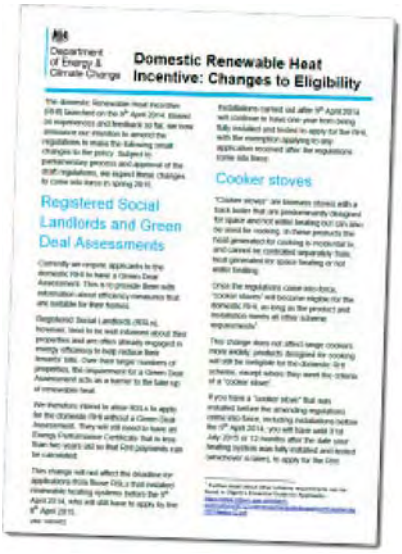
China intends to achieve the peaking of CO_2 emissions around 2030 and to make best efforts to peak early and intends to increase the share of non-fossil fuels in primary energy consumption to around 20 per cent by 2030. Both sides intend to continue to work to increase ambition over time.

In a joint statement, the US State Department said: "The United States of America and the People's Republic of China have a critical role to play in combating global climate change, one of the greatest threats facing humanity. The seriousness of the challenge calls upon the two sides to work constructively together for the common good."

"The United States and China hope that by announcing these targets now, they can inject momentum into the global climate negotiations and inspire other countries to join in coming forward with ambitious actions as soon as possible, preferably by the "first quarter of 2015".

Source: www.racplus.com

DECC proposes inclusion of high-temperature heat pumps to Domestic RHI


Department of Energy & Climate Change

Domestic Renewable Heat Incentive: Changes to Eligibility

Registered Social Landlords and Green Deal Assessments

Cooker stoves

The UK Department of Energy and Climate Change (DECC) has announced its *intention to amend the Domestic renewable heat incentive (dRHI) regulations to include high temperature air source heat pumps*. High-temperature heat pumps are a development of existing air source heat pump technology that can operate at temperatures as high as 80 °C, , suitable for use in properties where it is not appropriate to change radiators or use under-floor heating.

Source: www.ejarn.com and *www.gov.uk*

Working **fluids**

Company launches chiller that uses water as refrigerant

A German company has launched what it says is the world's first production-scale chiller using water as refrigerant. Efficient Energy announced its eChiller at the Chillventa exhibition in Germany recently. The company says that the key to using water as refrigerant is a micro-turbined centrifugal compressor that runs at up to 90 000 rpm to increase pressure and temperature in the water leaving the evaporator. The resulting steam is condensed directly into the remaining cool water flow. Use of variable oil-free turbo compressors enables the eChiller's cooling capacity to be adjusted continuously from 10 % to 100 %, while an intelligently controlled refrigerant circuit enables efficient modulation in response to changes in the ambient temperature. Among other touted features is a free-cooling function that doesn't require an additional free-cooling circuit.

Source: The HVAC&R Industry, Oct 16, 2014 and www.racplus.com

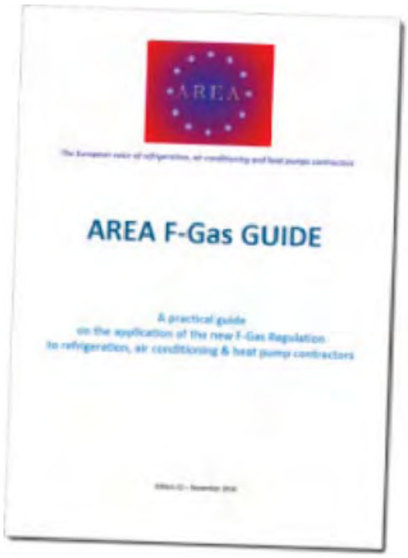
US industry announces $5 billion refrigerant research plan

The US Air-Conditioning, Heating, and Refrigeration Institute (AHRI) president and CEO Stephen Yurek has announced that the American HVACR industry will invest $5 billion in research and development funds over the next decade for new-generation refrigerants and equipment. The HVACR industry has been proactive in developing refrigerants with lower GWP. “Close to $2 billion has been spent by the industry since 2009 researching energy-efficient equipment and the utilisation of low-GWP refrigerants,” Yurek stated, “and over the next ten years, the industry will invest an additional $5 billion for R&D and capital expenditure to develop and commercialize low-GWP technologies.” He said that AHRI and its member companies launched the Low-GWP Alternative Refrigerants Evaluation Program in 2011, the first phase of which was completed at the end of 2013. The second phase of the program is currently underway. The aim of the program is to evaluate different refrigerants in several applications.

“The industry is committed to working with the international community in transitioning away from high-GWP refrigerants in a managed and orderly process, and this research is a tangible part of that commitment,” he said.

The Alliance for Responsible Atmospheric Policy, an industry coalition representing more than 95 per cent of U.S. HFC production and a significant majority of the user industries, announced actions that support a Montreal Protocol amendment to phase down the production and consumption of HFCs. The Alliance also announced that it commits to take actions and support policies with a goal to reduce global HFC greenhouse gas contribution by 80 per cent by 2050 relative to current emissions. This will be accomplished by advancing technologies; improving servicing practices; increasing recovery, reclamation, and reuse; and conducting technology assessments and workshops.
Source: www.racplus.com

AREA publishes guide to the new F-gas regulation



AREA, the European organisation of air conditioning, refrigeration and heat pump contractors, has issued a *guide on the new F-gas regulation*.

The new F-gas regulation includes many changes that will come into force from 1 January 2015. Refrigeration, air conditioning and heat pump contractors will be faced with a set of new requirements that will substantially affect the content and performance of their activities. With this guide, AREA aims to provide a tool that explains and clarifies the new rules, their impact and their practical application.

President of AREA Per Jonasson said: “This guide is the result of a work that started immediately after the EU Institutions found a compromise on the final text. Having closely followed the revision since the very beginning, we were well-placed to anticipate the questions the new Regulation would trigger. Yet addressing them into a comprehensive yet clear and practical guide required great efforts and commitment from the entire AREA membership over more than 6 months.”

The European Commission also co-operated to ensure common understanding and consistency.

The guide comprehensively addresses all aspects relevant to contractors. For each theme, it clearly highlights what the impact is on contractors and it makes suggestions as to what they should do. It also includes useful tools, such as a conversion table weight-CO_2-eq for the most commonly used refrigerants, a flowchart on leak check frequency and a table summarising certification requirements.
Source: www.acr-news.com

EPEE F-gas pledge to be catalyst for global action

Following the UN Climate Summit, the European Partnership for Energy and the Environment (EPEE) issued a *pledge* in which the European refrigeration, air-conditioning and heat pump industry commits to taking concrete, proactive action to facilitate the implementation of the EU F-gas Regulation.

“EPEE members are determined to make the EU’s new F-gas rules work and are ready to help the EU and its Member States reach their targets and ensure that Europe remains a leader in reducing greenhouse gas emissions,” said EPEE Director General Andrea Voigt.

The commitments focus on two main areas – (1) education, in terms of raising awareness and promoting the rules, and (2) supporting a smooth and viable transition towards lower-emission technologies.

EPEE's education commitments include organising regular and tailored events to raise awareness of the new rules, developing specific communication tools to enhance industry's understanding of its new obligations, and increasing cooperation with installers, owners, and schools to ensure all parts of the chain understand their responsibilities.

It will also set up a Low Emissions Task Force to examine how to overcome barriers which prevent the uptake of low emissions technologies, as well as monitor the effectiveness of the EU F-gas Regulation and raise global awareness.
Source: www.ejarn.com and JARN, October 25, 2014

Technology

Net-Zero-Energy 'Skin' Keeps Buildings Cool

Photo: M. P. Gutierrez (BIOM)

Researchers at the University of California-Berkeley have designed a new membrane that wraps around a building and is filled with microscale valves and lenses that open and close as they sense light, heat, and humidity. The facade reportedly works with no power at all—not even solar panels—and keeps the temperature comfortable, and light bright, inside. "It began with the aim of being a skin that can breathe, similar to our skin, that can open and close its pores, to regulate the temperature, humidity, and light conditions," says architect Maria-Paz Gutierrez, part of a research team collaborating on the new material, called SABER. The researchers are working to make the material an ultra-low-cost solution for developing countries, where energy use is increasing rapidly.
"When you augment natural ventilation, you augment human comfort without decreasing temperature," said Gutierrez. "If you augment and enable evaporation, you can with the same temperature have a much higher threshold for comfort."
Source: www.fastcoexist.com

Markets

EHPA hosts event for sustainable heating & cooling solutions

Heat pumps and district heating offer efficient, low-carbon and affordable solutions for heating, cooling and sanitary hot water for European cities. It is not a lack of technology, but a lack of planned concerted action, that prevents cities from making better use of available energy. Ambitious and consistent decisions are needed though. This is the concluding message of the event "*Sustainable and Affordable Heating and Cooling: The Case of European Cities*" hosted by the European Heat Pump Association (EHPA) in October. The event was well attended by various members of the renewable heating and cooling sector and representatives of local authorities from several European cities.

Keynote speaker Mrs Linda Gillham (UK/EA), member of the CoR, Commissions for the "Environment, Climate Change and Energy (ENVE)" and "Natural Resources (NAT)", introduced the event by insisting on the importance of minimizing the overall energy demand in the urban area and improving the use of affordable renewable energy and efficient technology in European cities.

Secretary general of EHPA, Thomas Nowak, mentioned the key role of cities in the climate and energy debate and the need to make the right decisions. "Instead of focusing on the search for new gas sources, decision makers should turn to available technologies that can heat, cool and provide sanitary water and are indigenous. Heat pumps are a perfect option to bridge the differences between supply and demand for heating and cooling in cities".

Source: www.ejarn.com, JARN, October 25, 2014 and www.ehpa.org

China: air source heat pump technical committee established

The 2014 Annual Conference of China Air Source Heat Pump Alliances (CHPA), organised by the Heat Pump Alliance of China Energy Conservation Association (CECA), was held in Beijing recently. More than 220 participants, including experts from the National Development and Reform Commission, China Energy Conservation Association, China Academy of Building Research, China National Institute of Standardization, the European Heat Pump Association (EHPA), and the International Energy Agency Heat Pump Centre and company representatives attended the conference.

Source: www.ejarn.com and JARN, September 25, 2014

European refrigerants market

The economic slowdown over the period from 2007 to 2009 adversely affected the European refrigerants market, and rising fuel prices affected the overall growth rate of the market. A *report from MicroMarketMonitor* estimates the size of the market in terms of volume and value, and reviews the main market drivers, challenges, and key issues.

Source: www.ejarn.com and *JARN, October 25, 2014*

China closes five HCFC production lines

China announced the shutdown of five HCFC production lines at the International Day for the Preservation of the Ozone Layer ceremony in mid-September. This should remove 59 000 tons of HCFC output, phase out 88 000 tons of HCFC production capacity and cut annual CO_2 emissions by more than 93 million tonnes.

Source: www.ejarn.com and *JARN, October 25, 2014*

ATW market in France

France is Europe's largest Air-To-Water heat pump market, recording an annual demand of more than 100 000 units at its peak in 2008. Oil prices in France surged in 2007-2008, and the government offered incentives to install heat pump heating and hot water supply systems to encourage the switch to electricity. Following this peak, however, the European economic crisis hit, leading to a downturn in the construction market and cuts to incentives. The ATW market then began to contract. The French economy continues to struggle, but is finally showing signs of gradual recovery. The French ATW market rose to 54 500 units in 2013, an increase of 9 % over 2012.

Source: www.ejarn.com

AN OVERVIEW OF RECENT HEAT PUMP TREND IN ASIA/OCEANIA REGION

Takeshi Hikawa, Director, Heat Pump & Thermal Storage Technology Center of Japan, TOKYO, JAPAN

Abstract:
This report introduces heat pump market trends in Asia and Oceania Region of which consists different levels of economies in a widespread area with variety of climates. It also picks out some installation cases of development or application of heat pumps, fitted in each policy or economy such as heat pump water heater, industrial heat pump application and so on. In addition, activity toward environmentally friendly society, such as introduction of standards for heat pump performance, development of heat pump unit designed to low GWP refrigerant, are also other features in this report.

Key words: ***Asia and Pacific, Heat Pump technologies, energy saving,***

1 INTRODUCTION

Firstly we would like to focus on how large primary energy consumes during its continuous economic expansion in Asia. As Figure1, the estimation of IEEJ (The Institute of Energy Economics Japan) based on IEA WORLD ENERGY OUTLOOK 2008 indicates, Asia is a larger energy consumer than any other regions and its consumption is expected to be surging up outstandingly. The consumption level in 2030 is predicted to be double compared to the 2010 level.

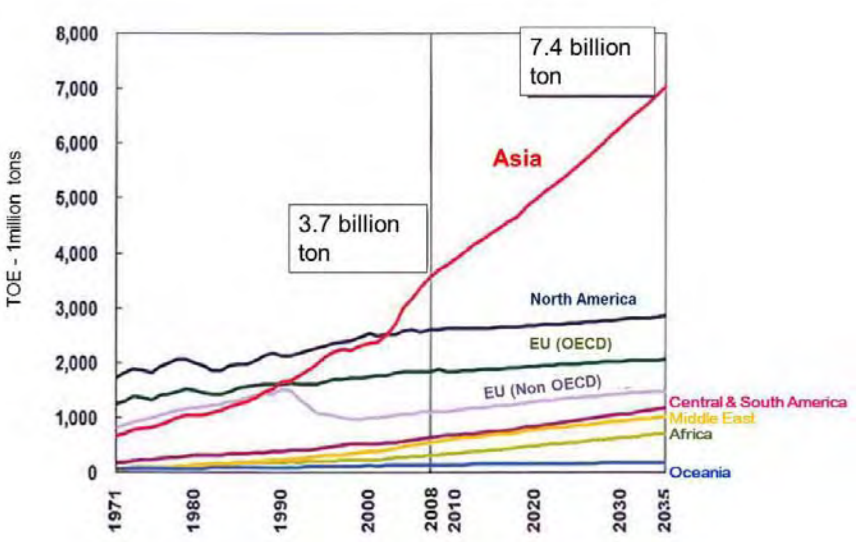


Figure1: Primary Energy Consumption by Region (IEEJ“ Asia/World Energy Outlook 2011”)

Amid the increase in energy consumption, heat pump demand is on the upward trend. Consequently, Asia and Oceania Region confronts twin increase in energy consumption and greenhouse gas emission.
Developed economies such as the EU, North America and Japan have been implementing policies for reduction in energy consumption and greenhouse gas emission. For instance, the

EU has been implementing the EU's climate package, known as "20-20-20". The main objective of this package is to achieve, by 2020 a 20 % reduction in greenhouse gas emission, a 20 % improvement in energy efficiency and a 20 % share for renewables in the EU energy mix. North America has also been conducting various phase-out policies along with R&D support for energy efficient devices.

Now let's take a look at the situation in Asia and Oceania Region. The key is how to achieve economic development and decrease in energy consumption simultaneously in Asia which contains many developing countries. As for heat pumps, it would be critical to disseminate energy efficient and CO2-reduction-effective equipment immediately to meet increasing demand in developing countries. Followings are features of markets, energy conservation policies, R&Ds for developing energy efficient equipment and issues surrounding refrigeration in Asia and Oceania Region.

2 FEATURES OF MARKETS IN ASIA AND OCEANIA REGION

2.1 Space Conditioning (RACs & PACs)

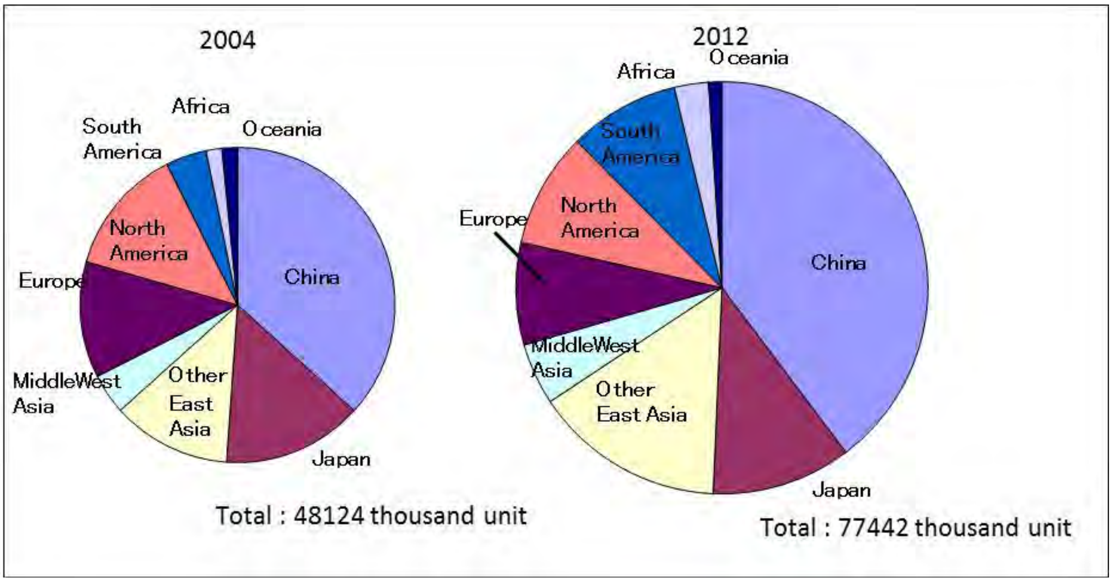


Figure 2: Room air-conditioner shipment (JRAIA 2013)

Room air-conditioners in Figure2 are defined as package unit type air-conditioners used mainly in households both for cooling-only units and heat pump reversible units. Both Window air-conditioners and small size split air-conditioners are in this category. The total number was nearly 50 million units in 2004 and increased up to around 80 million in 2012.

Asia and Oceania Region accounts for near 70 % with more than 60 million units in 2012. Growth in demand in China is particularly remarkable by achieving 30 million in 2012 which accounts for 60 % in East Asia. Japan has kept steady demand around 8 million a year recently and 100 % of them deploy inverter technology. Shares are small but India achieved 3 million, Korea 2 million followed by Taiwan, Malaysia and Thailand which show sharp increase up to nearly 1 million.

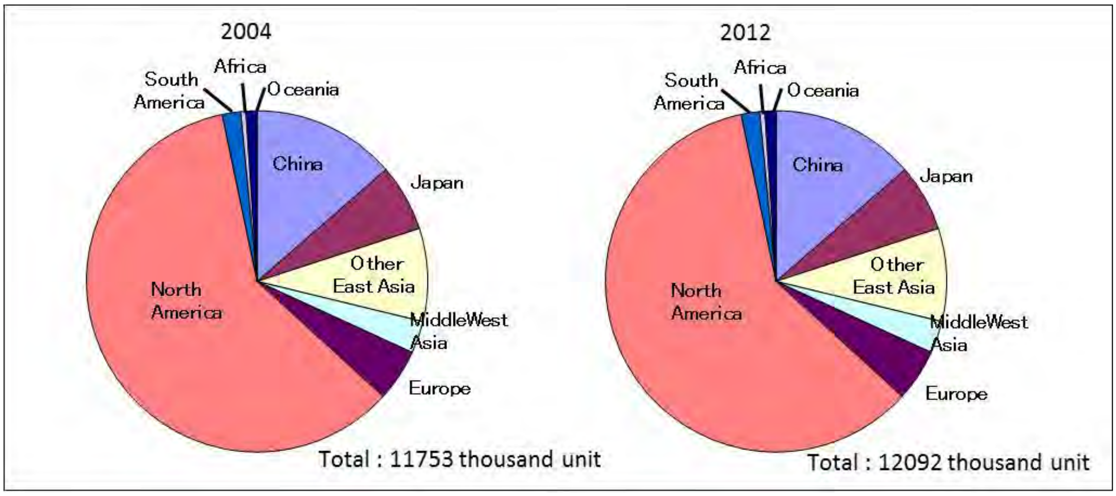


Figure 3: Package air-conditioner shipment (JRAIA 2013)

Other major category of air-conditioners, larger capacity than room air-conditioners consist of unitary type ones majorly used in the US and split type multi air conditioners for commercial use applied to medium size buildings. As Figure3 indicates unitary type in the US occupies a large share, while Asia accounts for 30 %. The demand in this region has been steady worldwide and it has also been stable around 4 million in entire Asia. A trend in Asia is that China and Japan occupy most of the demand.

2.2 Electric chillers and heat pumps

Electric chiller, classified into larger capacity category than package air-conditioners, is generally applied to central reversible air-conditioning system for large scale buildings and for industrial use as well. Compressor type is varied from reciprocating, scroll, screw to centrifugal depending on each capacity level.

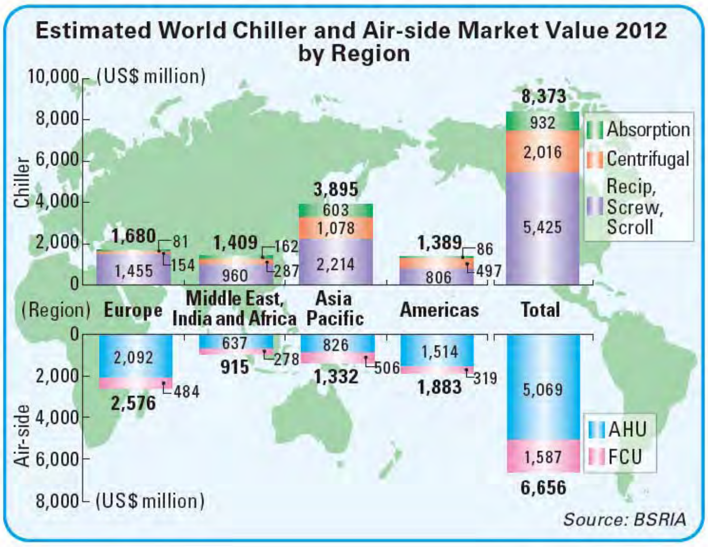


Figure 4: World Chiller Market by Region (JARN November edition 2013)

Their demand is heavily influenced by an economic situation. As Figure 4 shows, Asian market is the largest in the world, which accounts for nearly 50 %. An absorption chiller marks large growth thanks to lots of large-scale building development in China. Majority of demand in Japan, on the other hand is air-cooled electric chillers. Their trend is to be modularized for easy update, adjustability for system control and cost effectiveness. Besides, research and development for higher efficient technology has brought about innovation regarding heat exchangers and inverter technology for part load operation. Performance has been improved substantially, in fact some chillers in the market exceed IPLV value 11.

3. NEW TECHNOLOGY FEATURES IN ASIA

Some countries have made a unique development for energy saving, CO_2 reduction and introduction of renewables in accordance with its circumstance in Asia. This section picks out examples from Japan; CO_2 heat pump water heater and industrial heat pump application, China; Ground source heat pump and heat pump water heater and Korea: Smart city project.

3.1 Heat pump water heater

Japanese government introduced a grant program from 2002 to disseminate energy efficient Eco Cute, a nickname of domestic water heater with CO_2 refrigerant, which can reduce energy consumption in water heating. In addition, manufactures have been making every effort to hike a COP value by improving compressor and heat exchanger performance, heat insulation performance, system control technology and so on. As a result, Eco Cute products achieved an Annual Performance Factor (APF) value, which will be described in detail in latter section, over 3.8, equivalent to primary COP 1.5. Not to mention, these are more environmentally-friendly and emit far less CO_2 than direct combustion type of water heating equipment. The cost reduction and improvement of performance have led Eco Cute to annual shipment of 0.5 million and achieve accumulated shipment number to be over 3 million units.

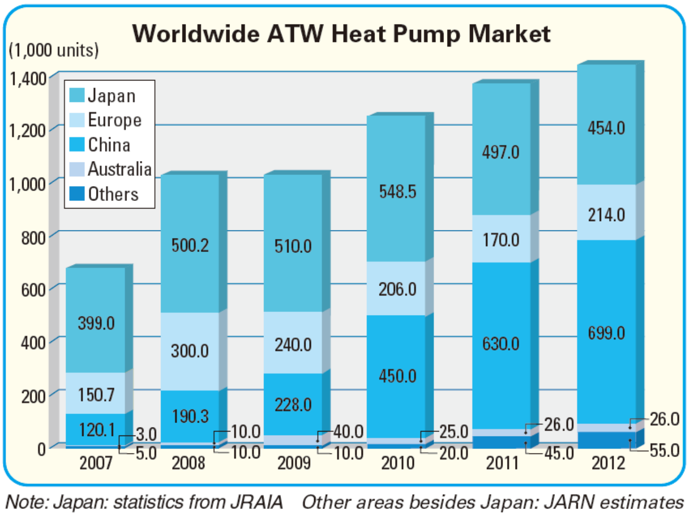


Figure 5: Worldwide ATW heat pump market trend (JARN August edition 2013)

Meanwhile an increase in deployment of heat pump water heaters in China is remarkable owing to its firm housing demand. Another reason for boosting Chinese demand can be found in the stable electricity rate, around 0.6 Yuan/kWh owing to majority in inexpensive coal fired power generation in China. As is shown in Figure 5, its shipment number has shown significant growth as the years go on. The number was over 0.6 million in 2011 and is expected to be over 1 million near future. China has exceeded Japan regarding an annual shipment and become the world's largest market.

Feature of the heat pump water heater is applying R134a as refrigerant and its heat exchanger is designed to be laterally-contacted on a tank. They are less expensive but the hot water temperature in the tank is relatively low of about 50°C.

3.2 Ground Source heat pumps in China

Ground source heat pump has been supported by the Chinese government as an eco-friendly technology because coal combustion type of heating equipment still dominates in China at present. The government support has spurred the expansion of ground source heat pump in China. Figure6 shows the market trend of ground source heat pump. It is predicted that the shipment number of ground source heat pump in China will become the largest in the world before too long. The feature of Chinese market is that there have been many introduction cases in large scale buildings such as commercial buildings, public facilities and multi dwellings.

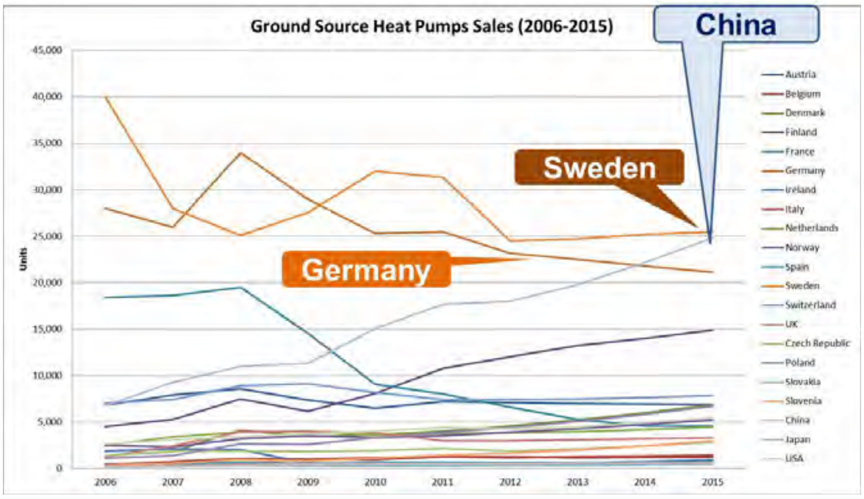


Figure 6: Ground Source Heat pump Sales Prediction (Oliver Peng 2013)

3.3 Heat pump Application for Industries

Boiler combustion has been used for various heating demand in industrial processes conventionally. Hence energy conservation utilizing alternative energy in industry sector has been considered seriously and consequently heat pump has been paid high attention. Actually, heat pump has been already used for cooling and heating in mild temperature range in industry process as well as for air-conditioning and refrigeration. However, for further energy conservation, waste heat can be used as a heat source for heat pump to produce higher temperature heat effectively.

On top of that, when both heating and cooling demand exist and utilized as heat source respectively at the same time, heat pump can save great deal of energy. These kinds of heat pump applications in industry, such as food processing factories and chemical processes for example, have been gradually increasing in Japan.
Figure 7 is an example of high temperature heat pump apparatus in practical use which can

produce 80 °C to 120 °C heat and are in practical use in wide range of processes from drying process of plastics to painting process of automobiles. Heat pump that can produce 160 °C heat has become available in the market and is expected to be used in variety of industrial applications.

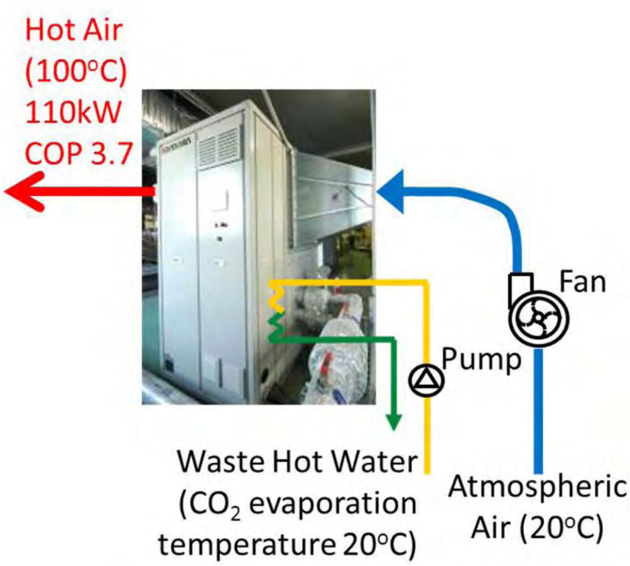


Figure 7: A case set up of Industrial heat pump (Watanabe 2012)

3.4 Korean Smart City Project

On expectation to realize a smart city in foreseeable future, the world's largest, cutting-edge smart grid test-bed was established on Jeju island, South Korea, in 2009. The test bed includes about 6,000 homes on the north-eastern part of the island. It is intended to lay the foundations for the early commercialization and creation of globally competitive export opportunities for smart grid technologies through demonstrations of smart grid technology as the key infrastructure element of green growth. (Park 2013)

4. POLICY & REGULATORY MEASURES

Asian nations have made an effort to conduct energy conservation policies by requiring manufacturers to meet minimum energy performance standards (MEPS) on the rated COP basis to remove low-performance appliances from the market or, by introducing a labeling system to motivate consumers to purchase higher efficient appliances.

Japan has conducted energy conservation policies since the 1970's after experiencing the 1st oil crisis. In addition, the Top Runner Program has enhanced the policies since its commencement in 1999. China, India and Taiwan have already introduced the labelling system and MEPS. Their MEPS value has improved up to the value used as a Japanese standard around 2000. Thailand has introduced the labelling system for air-conditioners since 1999 and MEPS since 2011. Vietnam has started the labelling system and MEPS will be introduced as early as 2014. Indonesia seems to be following these trends and we can say that energy conservation policies have been surely expanding throughout in East Asia.

Table 1: Recent MEPS Status in Asia

	COP	Period	Method	Remarks
Japan	**3.0→5.0	1999～2004	Top Runner	
	**4.9→5.7	2006～2010	Top Runner	APF
China	3.0	2010～	MEPS	Inverter(SEER)
	3.2	2010～	MEPS	Non Inverter
Taiwan	3.45	2011～	MEPS	
Thailand	2.82	2011～	MEPS	
India	2.7	2014～	MEPS	
	2.5	2012～14	MEPS	

**Figures represent an average COP value in a starting year and one required to be achieved in a targeted year respectively.

Japan established the Annual Performance Factor (APF) standard in order to accurately evaluate the performance under the usual operating condition. APF evaluates the energy efficiency of air conditioners in accordance with the actual conditions by taking into consideration the load conditions of buildings, intended use of air conditioners, load hours during heating/cooling periods in relation to outdoor temperatures and changes of the efficiency along with the capacity changes. APF indexes are widely used in Japan, including labeling as well as the Top runner program implemented by the Japanese government to promote energy conservation efforts.

In addition, the Japanese government has implemented the Top Runner Program to increase the average performance levels of products in the market.
The Top Runner Program is one of the most stringent programs, which introduces mandatory fleet-average energy efficiency requirements set at the level of the most efficient product on the Japanese Market at the time the requirement was formulated. Consequently, the energy efficiencies of Japanese products were successfully able to rise remarkably as shown in Figure 8.

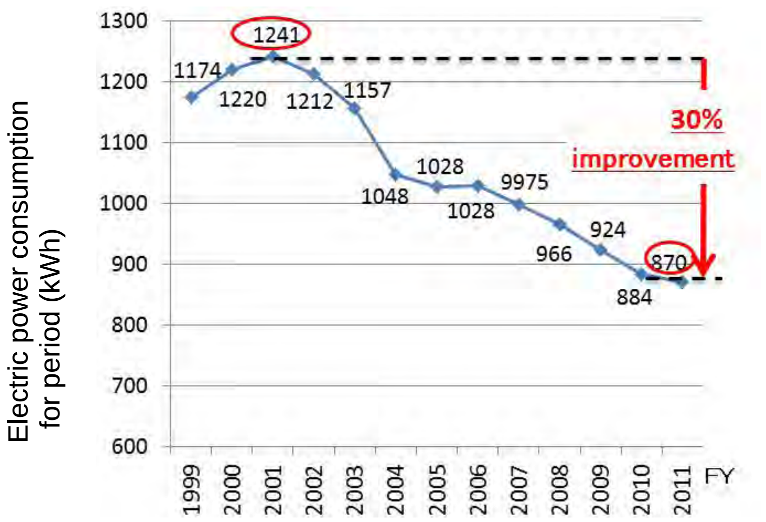
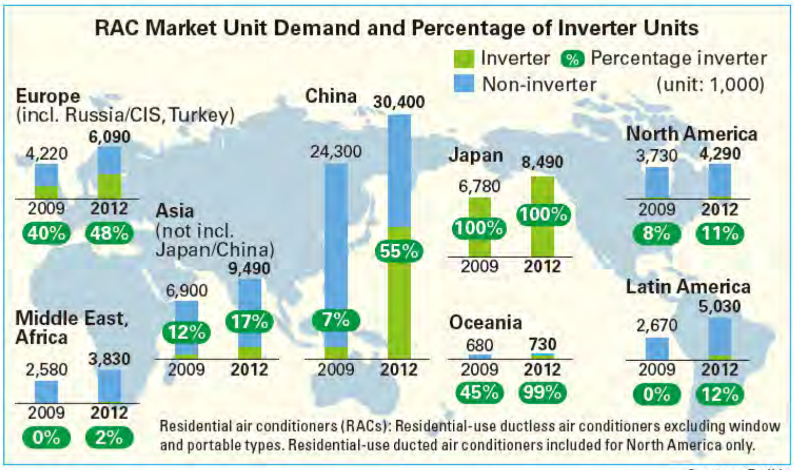


Figure 8: Saving energy trend of wall mounted air conditioner in Japan (Fukuda 2013)

(Note) Wall mounted cooling and heating units with cooling capacity of 2.8kW-class model; simple average values for a representative model of energy conserving-type products.

Furthermore, deployment of inverter technology has been accelerated in the Region because of its remarkable contribution to improve performance. As shown in Figure 9, 100 % of RAC in Japan deploy inverter technology thanks to the Top Runner program and the situation is similar in Oceania where nearly 100 % deployment has achieved. Labelling system and minimum energy efficiency standard requirement (MEPS) have boosted inverter dissemination in China also, achieving beyond 50 % share. The inverter trend can be observed in South East Asia with the shares of 17 % also. This rapid growth of inverter technology has been made within recent several years in Asia and Oceania Region and its effect of saving energy must be enormous.



Source: Daikin

Figure 9: RAC market and Percentage of Inverter Units (JARN July edition 2013)

4.1 Activities for heat pump performance validation

In order to make labelling systems and MEPS work effectively, it is important to have an independent third party which has capability to check an accuracy of declared performance values of products. When it comes to measuring heat pump performance, it is a must to equip precision measuring devices in a temperature-controlled testing room as well as to acquire its measuring know-how to get accurate data.

From this point of view, some of Asian nations have been working on establishing testing laboratories for accurate evaluation of air-conditioners' performance to promote energy efficient products. Japan has provided technological support for precise data acquisition method, or proper operation of testing facility.
Figure 10 shows technical exchanges between Japan Air Conditioning and Refrigeration Testing Laboratory (JATL) and Asia Oceania regions.
JATL which was established as a third-party institute for the purposes of offering appropriate information on air conditioning product performance has been networking with Korea Refrigeration and Air-conditioning Assessment Center (KRAAC) in Korea, Guang zhou Vkan Certification & Testing Institute (CVC) in China, Electrical and Electronics Institute (EEI) in Thailand, and so on. Furthermore Japan has provided some Asian countries with support of seasonal performance evaluation technology which requires accuracy through wide experiment range of capacity to evaluate real operation condition.

Figure 10: Technical exchanges among Asian countries testing laboratory (JATL 2013)

4.2 Japanese energy policy after the Great Eastern Japan Earthquake

Japanese energy policy has remarkably changed between pre- and post-great eastern Japan earthquake. All nuclear power stations in Japan have been stopped their operation and the ratio of thermal electric power generation has been increased after the incident. This has resulted in increase in CO_2 emission. In such a challenging situation, Japanese government has topped a great deal of incentives for promotion of renewable energy, energy conservation technology, unused energy usage or cogeneration technology in order to realize a low carbon society.
Energy Conservation Law has become effective since 1979 aiming at promotion of energy conservation policy in Japan. The present version after this earthquake focuses on supplying sides and promotes technology to level off the imbalance of electricity use in summer to meet severe supply demand tightness. Specifically, it encourages use of storage system of cold heat energy in the building, storage battery, building energy management system (BEMS), home energy management system (HEMS) or off-grid power system during peak hours of electricity consumption in a day.

5. REFRIGERANT DEPLOYMENT AS A MEASURE AGAINST GLOBAL WARMING

As refrigerant R22, R410A and R134a which currently used widely, have high GWP values of 1810, 2090 and 1430 respectively, heat pump designed for alternative refrigerants with relatively low GWP value, such as HFO, HFC(R32), HC and the mixture have been investigated dependent on application.

F-gas regulation has been under way in European and North American countries with targeting around 80% reduction. Meanwhile Japanese government has revised and tightened

the Act regarding Fluorocarbons. Japanese Cabinet approved the Bill for the Act for Partial Revision of the Act on Recovery and Destruction of Fluorocarbons on June 2013,
The revised Act is renamed to be the Act for Rationalized Use and Proper Management of Fluorocarbons, which concerns the complete life cycle of fluorocarbons. The following entities should be paid particular attention to the new Act as additional or stricter obligations have been imposed accordingly:

1. Manufacturers; encouraged usage of non-fluorocarbons or low GWP refrigerants.
2. Gas suppliers; requested to phase down of fluorocarbons family practically by adopting their alternatives and introducing more renewables to carry out planned reduction of their import volume.
3. Users of commercial products containing fluorocarbons (e.g. distribution industry); required to conduct a periodic check of products and to submit and publicize an annual report on quantity of fluorocarbons leakage.

5.1 Development of heat pump unit designed to low GWP refrigerant

When it comes to major refrigerant in Asia and Oceania Region, HFC(R410A) occupies a large share in Japan and Australia etc while HCFC(R22) does in rest of Asia.
Amid this trend, alternative refrigerants have been examined for developing of appliances considered from the point of views of energy conservation and eco-friendliness in Japan. Consequently, R32 has been paid attention as the most promising candidate alternative of R410A because its GWP is one third of R410A and it shows somewhat better characteristics than R410A.
As Table 2 shows, R32 has lower flammability of A2L classification which was a crucial risk must to be considered when developing products.

Table 2: Typical refrigerant's property (Matsuda 2011)

Refrigerant		Refrigerant Property					
		GWP	Relative Efficiency	ODP	Flammability	Toxicity	Condensing Pressure(MPa)
HCFC	R22	1810	100	0.06	A1	low	1.73
HFC	R407C	1770	99	0	A1	low	1.86
	R410A	2090	92	0	A1	low	2.72
	R32	675	97	0	A2L	low	2.80
	R1234yf	4	90	0	A2L	low	1.16
Other	R717(NH3)	0	106	0	A2L	high	1.78
	R290(C3H8)	3>	98	0	A3	low	1.53
	R744(Co2)	1	41	0	A1	low	(10)

The evaluation of flammability carried out in Japan has proved that the fire risk of wall hanging room air-conditioners in practical use is less than 10^{-10} incident of ignition / (year*unit), which is within accepted values in household electronics appliances in Japan. Consequently, products with R32 have been launched in a market since autumn 2012 and the number of shipment has already achieved over one million units.

On the other hand, HCFC（R22）is still widely used in China or other Asian and Oceanian countries and regions. Chinese government has been aimed to introduce propane for household air-conditioners while R32 and R410A for commercial ones.

In some south east Asia countries, shift from R22 to R32 has been considered recently, since the movement of shift to R32 in Japan has started. Thailand and Indonesia are planning to shift R22 to R32. In Australia, the Carbon Tax and HFC Levy has been introduced since July 2012, imposing around $40 per kilogram tax, which is another feature of measure regarding refrigerant. As aforementioned cases suggest, measures against global warming are steadily making progress in Asia and Oceania region.

6. CONCLUSIONS

This Asia and Oceania regional report introduces energy saving policies and counter measure against global warming surrounding heat pump industry in the region where heat pumps are spreading rapidly.
Especially, environmental friendly activities such as the rapid spread of inverter technology which enable saving energy a lot, the active dissemination movement of air conditioning verification system which makes more effective labeling system and MEPS, and the spread of air conditioner which is applied low GWP refrigerant are focused.
These movements of heat pump technology allow economic development while considering environmental preservation, and have been conducted throughout the region.

Heat pump is the remarkable breakthrough technology to realize economic development concurrently with low carbon society and stop global warming.

7. REFERENCES

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Oliver Peng (June 2013) “Global market and growth of Heating and Heat Pump” 2nd ASHP Asia Air-Source Heat Pump Development Forum China 2013

Seong-Ryong Park IEA HPC NEWSLETTER Volume 31 2013

Watanabe (October 2012) “Trends in industrial heat pump technology in Japan” IEA HPP Symposium

Ongoing Annexes

IEA HPP Annex 36 Quality Installation / Quality Maintenance Sensitivity Studies

Annex 36 evaluated how installation and/or maintenance deficiencies cause heat pumps to perform inefficiently (i.e., decreased efficiency and/or capacity). The investigations showed that operational deviations (or faults) are significant; that multiple deviations – when combined – can have an additive effect on heat pump performance degradation; and that some deviations (among various country-specific equipment types and locations) have greater performance impact than others. The focus and work undertaken by each participating country is presented in the table below.

The intended audience for the resultant Annex 36 information is:

- HVAC practitioners responsible for designing, selecting, installing, and maintaining heat pump systems in varied applications.
- Building owners/operators interested in achieving improved comfort conditioning and efficiency performance from their HVAC equipment.
- Entities charged with minimizing energy utilization in varied heat pump applications and geographic conditions (i.e. utilities, utility commissions, energy agencies, legislative bodies, etc.).

The three-year effort concluded in mid-2014 with results presented at the Annex 36 workshop held in conjunction with the 11th IEA Heat Pump Conference (Montreal, Quebec, Canada; 12 – 16 May 2014).

The Annex 36 Final Report has been submitted to the IEA HPC for review and is sent out to the IEA Executive Committee for electronic approval.

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Annex 36 Participants	Focus Area	Work Emphasis
France	EdF – Space heating and water heating applications.	**Field:** Customer feedback survey on HP system installations, maintenance, and after-sales service. **Lab:** Water heating performance tests on sensitivity parameters and analysis.
Sweden	SP – Large heat pumps for multi-family and commercial buildings KTH/SVEP – Geothermal heat pumps	**Field:** SP – Literature review of operation and maintenance for larger heat pumps. Interviews with real estate companies owning heat pumps. KTH/SVEP – Investigations and statistical analysis of ~ 68 000 heat pump failures. **Modeling/Lab:** Determination of failure modes and analysis of found failures (SP) and failure statistics (KTH/SVEP).
United Kingdom	DECC – Home heating with ground-to-water, water-to-water, air-to-water, and air-to-air systems.	**Field:** Monitor 83 domestic heat pumps and made modifications to improve performance. **Lab:** Investigate the impact of thermostatic radiator valves on heat pump system performance.
United States (Operating Agent)	NIST – Air-to-air residential heat pumps installed in residential applications (cooling and heating).	**Lab:** Cooling and heating tests, with imposed faults, to develop correlations for heat pump performance degradations due to those faults. **Modeling:** Seasonal analyses modeling to evaluate the effect of installation faults on heat pump annual energy consumption. Includes effect of different building type (slab vs. basement foundation) and climates in the assessment of impact on fault-imposed heat pump performance.

ACCA → Air Conditioning Contractors of America
DECC → Department of Energy and Climate Change (UK)
EdF → Electricité de France
KTH → Royal Institute of Technology (Sweden)
NIST → National Institute of Standards and Technology (US)
ORNL → Oak Ridge National Laboratory (US)
SP → SP Technical Research Institute of Sweden
SVEP → Swedish Heat Pump Association

IEA HPP Annex 40 Heat pump concepts for Nearly Zero Energy Buildings

5th IEA HPP Annex 40 working meeting held in Nagoya, Japan

The objective of IEA HPP Annex 40's work is to investigate and improve heat pump systems applied in Nearly or Net Zero Energy Buildings (nZEB). Currently, nine countries - CA, CH, DE, FI, JP, NL, NO, SE and US - are collaborating in Annex 40.

The 5th Annex 40 working meeting was held in Nagoya, Japan, on 10/11 Nov. 2014. In the frame of the meeting, interim results of the national contributions were presented.

Task 2 has seen performance of different simulation studies. Results have shown that heat pumps are among the most efficient and cost-effective systems solutions for application in nZEB. A related Japanese case study will investigate nZEB office buildings, and will be carried out for both Japanese and European boundary conditions. In addition, Norway is developing a design tool for heat pumps in nZEB, and the USA is developing a simulation tool for the comfort evaluation of radiant emission systems.

Task 3 is dedicated to technology developments and field monitoring. As far as development of new technologies is concerned, the work is investigating variants of integrated heat pump solutions for different building functions, as well as the combination of heat pumps with other heat sources, such as solar components and CHP. In field monitoring, different field tests have recently started in Norway in the first nZEB in Scandinavian climate conditions. Results will be presented at the next Annex 40 meeting. In Sweden, 30 monitoring projects are planned to start in 2015. The Netherlands has also an ongoing field test with different heating systems, which will also be investigated by system simulations. Germany will contribute a long-term monitoring project of low-energy office buildings. Many buildings are equipped with thermally-activated building systems, which also have the capability of short-term storage and load management. In Switzerland, two nZEBs with electric vehicles (EVs) as an electrical storage option are the objects of field monitoring.

The meeting was concluded with a half-day workshop for the exchange of Annex 40 interim results with nZEB activities of Japanese manufacturers and stakeholders.

All workshop presentations are available for download on the Annex 40 website at http://www.annex40.net.

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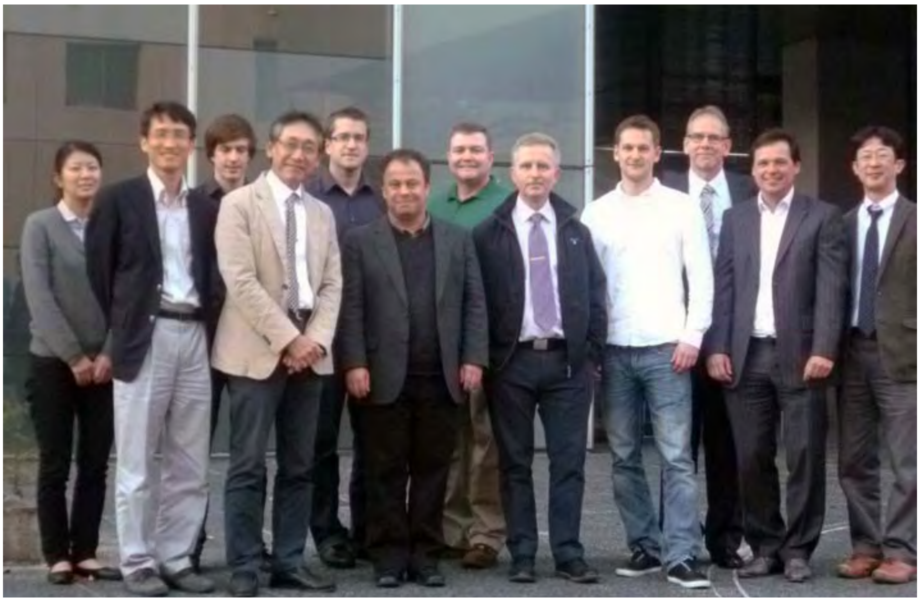

Annex 40: Working meeting held in Nagoya, Japan

IEA HPP Annex 41 Cold Climate Heat Pumps

Annex 41 began in July 2012 to revisit research and development work in different countries to examine technology improvements leading to successful heat pump experience in cold regions. The primary focus is on electrically driven air-source heat pumps (ASHP) with air (air-to-air HP) or hydronic (air-to-water HP) heating systems, since these products suffer severe loss of heating capacity and efficiency at lower outdoor temperatures. The main outcome of this Annex is expected to be information-sharing on viable means to improve ASHP performance under cold (≤ -7 °C) ambient temperatures.

In the past quarter the co-Operating Agents completed a draft summary interim report for the Annex (including Task 1 and 2 works through about August 2014). The draft interim report has been distributed to the Annex Participants for review. It will be revised based on those reviews and posted to the Annex web site in December 2014.

As reported in 2014 Newsletter issue 3, the Participants agreed at the Montréal workshop and business meeting (May 2014) to an extension of the Annex work period to allow all Participants time to complete their planned contributions. A proposal was submitted by the OAs to extend the Annex through July 2016 and the proposal was approved by the Executive Committee at its Nov. 2-5 meeting in Freiburg, Germany.

Planning is underway for the 3rd working meeting to be held May 2015 in Vienna at the Austrian Institute of Technology. The 2nd workshop is to be held August 2015 in Yokohama, Japan during the 2015 International Congress of Refrigeration – planning of the workshop program has started.

The Annex web site is http://web.ornl.gov/sci/ees/etsd/btric/usnt/QiQmAnnex/indexAnnex41.shtml

It has been updated to include all reports submitted to date as well as the presentations from the Montreal workshop.

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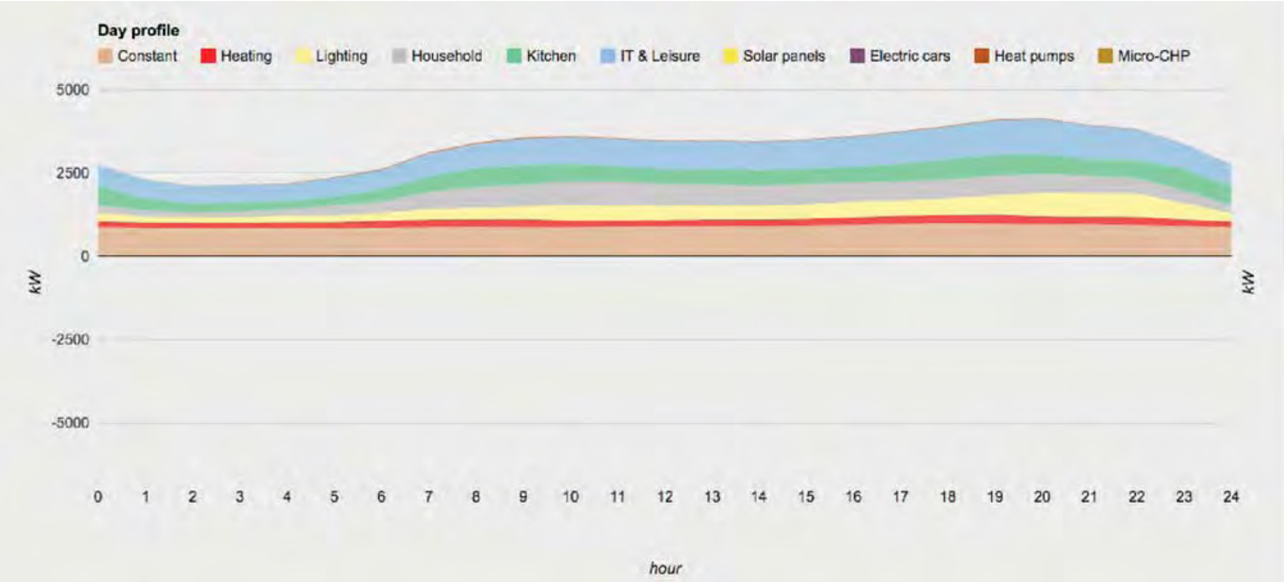
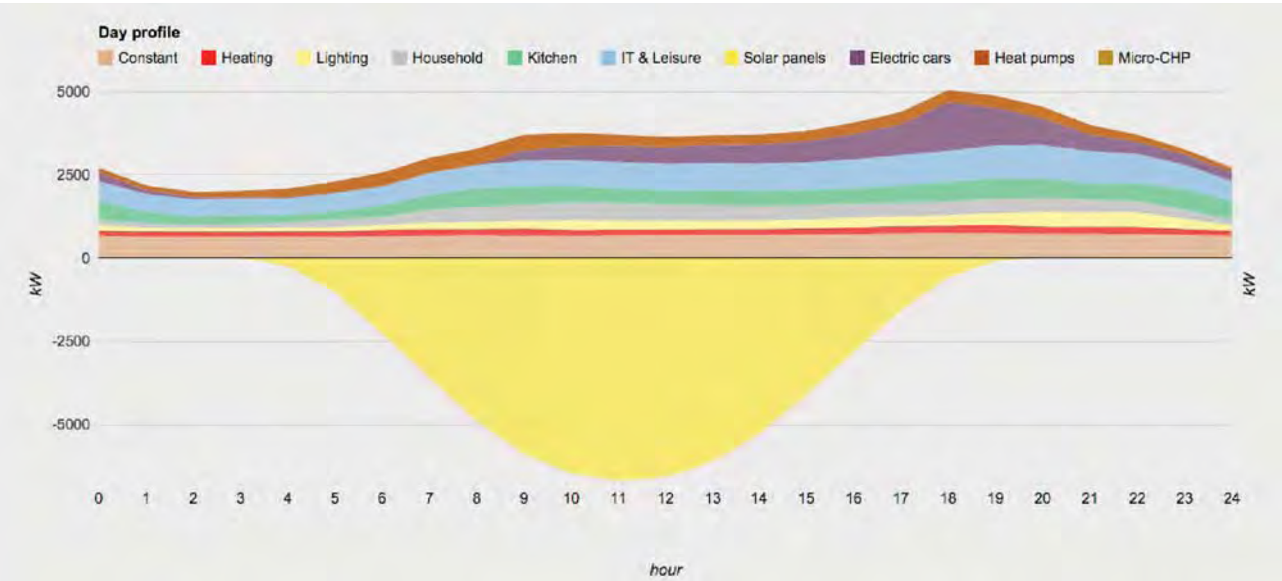
Annex 41: Montréal Annex 41 workshop





Annex 42: Typical power consumption on a bright summer day, in 2013 and 2030, for a typical urban area

IEA HPP Annex 42 Heat Pumps in Smart Grids

Work on this annex has effectively come up to speed during the course of 2014. During the tri-annual heat pump conference in Montreal in May we successfully organised a regular Annex 42 project meeting, an Annex 42 workshop open for all visitors to the congress, as well as a presentation accompanied by a scientific paper in the main program.

As Operating Agent, we also organised a workshop for a future Annex on hybrid heat pumps, which will start in Q2 of 2015.

Flexibility and storage are key elements related to heat pumps in smart grids, and consequently were the main topics discussed during the Annex 42 project meeting in October 2014 at Fraunhofer-Freiburg.
An indicator, known as a 'flexibility index', in the form of a figure to rate the degree of flexibility of a dwelling, is recognised as of potential interest for further investigation.

A visualisation of the necessity of smart grids and flexibility is shown in the diagrams above, which show the power demand of a typical urban area during a bright summer day in 2013 and 2030.

Task leadership fulfilment, task description, expected output, planning and a table of contents of the task reports were updated during further discussions. A firm action list was compiled and agreed upon. Task #1, Country Report, has been finalised.

Confirmed participants are United Kingdom, The Netherlands, South Korea, USA, Switzerland, Denmark, France, Germany and Austria. Sweden is in the process of arranging the process to become a full participant in Q4-2014/Q1-2015.

The national team meeting for the Netherlands after the summer break, on 16th September 2014, was also attended by Berenschot Energy & Sustainability (BES). BES can contribute specific knowledge about power production and pricing signal mechanisms in smart grids. As Operating Agent, we participated in the two-day HPC National Teams' meeting in Göteborg, Sweden, October 2014, and the ExCo meeting in Freiburg, November 2014.

The next Annex 42 meeting will be on 23rd January 2015, in Utrecht – The Netherlands.

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IEA HPP Annex 43 Fuel-driven sorption heat pumps

During the period while work was in progress on Annex 34 "Thermally Driven Heat Pumps for Heating and Cooling", there was a growing interest in the area of fuel-driven sorption heat pumps, with more and more products approaching market release. A new Annex, "Fuel-driven sorption heat pumps", was therefore proposed to the ExCo in March 2012. After an Annex definition meeting, a legal text was compiled and accepted as a draft by the ExCo. The new annex, Annex 43, started officially in July 2013, with a planned duration of four years. So far, seven countries have confirmed joining the annex (AT, DE, FR, IT, KR, UK, US); some more have expressed their interest (Poland, China), but of course more participants are welcome.

Objectives

The scope of the work under this Annex will be the usage of fuel driven sorption heat pumps in domestic and small commercial or industrial buildings or applications. If applicable, the additional possibility of supplying cold will also be considered. The main goal is to widen the use of fuel-driven heat pumps by accelerating technical development and market readiness of the technology, as well as to identify market barriers and supporting measures.

The Annex structure

The tasks are further specified as follows.

Task A: Generic Systems and System Classification

- Available sources and heating systems
- Existing market and regulatory boundary conditions

Task B: Technology Transfer

- Link research to industrial development for faster market penetration of new technologies
- Novel materials (e.g. MOFs for adsorption heat pumps)
- Novel components (integrated evaporators/condensers, compact heat exchangers)
- System designs (e.g. façade collector as heat source)

Task C: Field test and performance evaluation

- Measurement/monitoring procedure standardisation (e.g. how to cope with different fuel quality, system boundaries, auxiliary energy, etc.)
- Extend standards to seasonal performance factors at the system level

Task D: Market potential study and technology roadmap

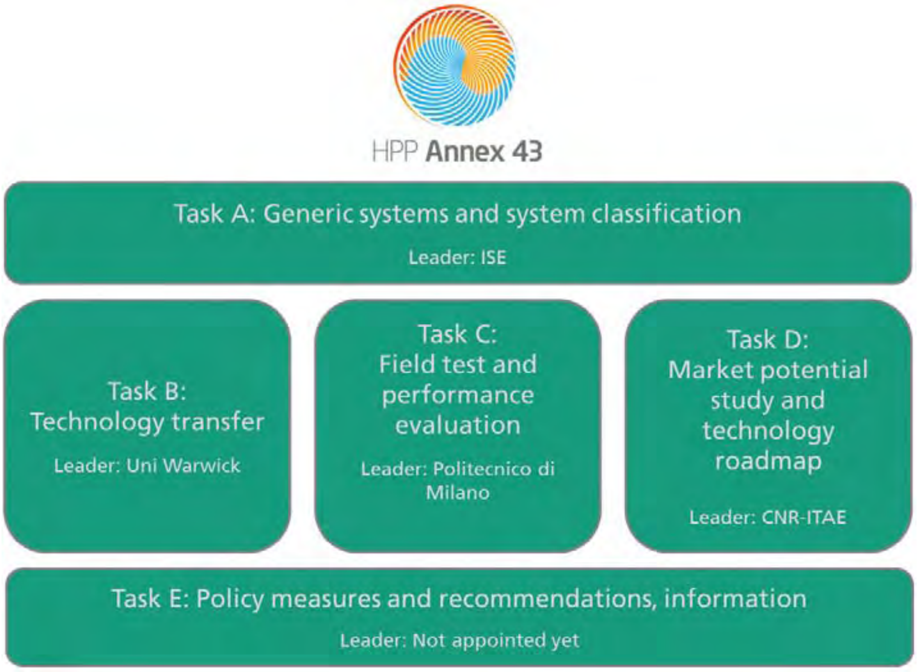
- Simulation study to evaluate different technologies in different climate zones, different building types and building standards
- Combine with market data and actual building stock for technology roadmap

Task E: Policy measures and recommendations, information

- Dissemination
- Workshops for planners, installers and decision makers
- Develop recommendations for policies, e.g. building codes and funding schemes

Within Task A, a template for the country report was prepared by ISE and sent out to the participants, it will be completed in spring 2015.

A presentation on the Annex was given at the Heat Pump Summit 2013 in October 2013 in Nuremberg, Germany, and at several more local events.



Annex 43: Annex structure

Annex 43: Group picture of the third Annex meeting held in Freiburg, Germany, November 2014.

This summer Korea joined the Annex as the seventh country.

The third meeting was held on November 6-7 in Freiburg with about 24 participants from 8 countries (including observers from the Netherlands and Russia). One of the major outcomes of this meeting was the common interest to start a large field test (> 1000 systems) on fuel driven sorption heat pumps to prove the efficiency of this technology, gather more information of ideal system layouts, and increase awareness. This idea will be discussed with interested parties from gas industry and heating manufacturers. If there is enough interest, possibilities for additional public funding will be evaluated.

More information about the annex can be found at:
https://www.annex43.org/

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IEA HPP Annex 44 Performance indicators for energy efficient supermarket buildings

Work in Annex 44 "Performance Indicators for Energy-Efficient Supermarket Buildings" is now focusing on collecting energy performance data from supermarkets in Sweden and The Netherlands. Two different approaches are being followed for this purpose: in Sweden, supermarkets have been asked to supply their data by means of a web questionnaire posted on the Swedish Annex website (*www.Annex44.com*), whereas in The Netherlands data for 2013 has been obtained from a refrigeration company that services a number of supermarket chains.

Data from The Netherlands relates to 150 supermarkets from a single supermarket chain. It presents the yearly energy consumption and yearly gas consumption in relation to the supermarket (sales) area, the opening hours and 65 energy-saving options that are or are not present in the individual supermarket stores.

A first analysis shows that the electrical energy consumption per m^2 is around 400 kWh/m^2,year, and is fairly independent of the supermarket size. This is somewhat surprising, as earlier work in Annex 31 showed a decreasing value for larger supermarkets (but the range of studied supermarket sizes was much larger in Annex 31). What is also obvious from the data from The Netherlands, is that there are large differences in overall electrical energy consumption per unit sales area, ranging from 100 to 600 kWh/m^2,year.

This could indicate that there is still room for considerable improvement of energy efficiency. The analysis will continue, to include the influence of the various energy-saving options on energy consumption.

Annex 44 is now nearing the time slot set for inclusion of new partners. There are still active discussions with partners from Australia/USA and Denmark concerning possible participation in this annex.

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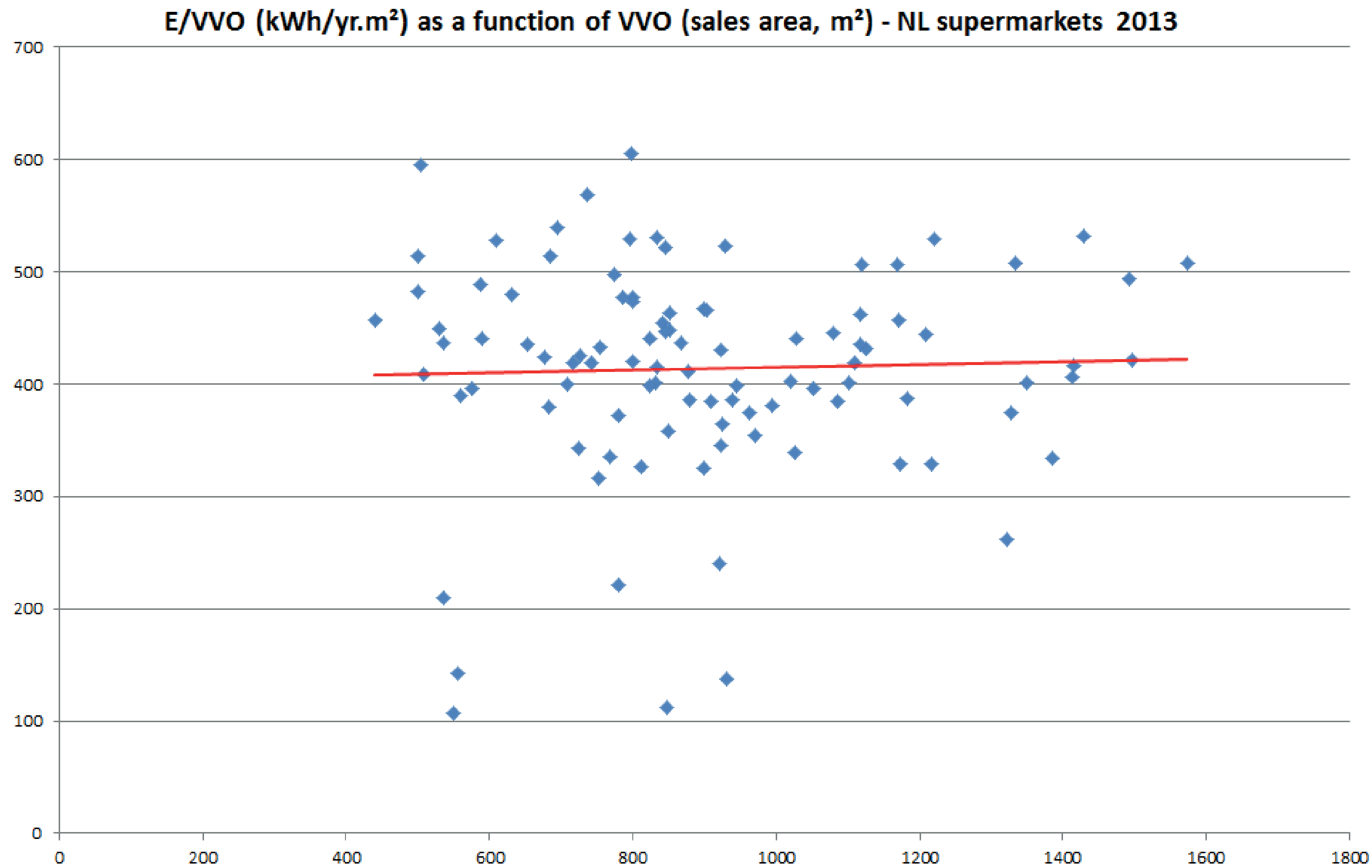

Annex 44: The annual energy consumption and annual gas consumption in relation to the supermarket (sales) area, the opening hours and 65 energy-saving options that are or are not present in the individual supermarket stores.

Ongoing Annexes

Bold text indicates Operating Agent.

Annex 36 Quality Installation/Quality Maintenance Sensitivity Studies	36	FR, SE, UK, US
Annex 37 Demonstration of Field measurements of Heat Pump Systems in Buildings – Good examples with modern technology	37	CH, NO, SE, UK
Annex 39 A common method for testing and rating of residential HP and AC annual/seasonal performance	39	AT, CH, DE, FI, FR, JP, KR, NL, SE, US
Annex 40 Heat Pump Concepts for Nearly Zero-Energy Buildings	40	CA, CH, DE, FI, JP, NL, NO, SE, US
Annex 41 Cold Climate Heat Pumps (Improving Low Ambient Temperature Performance of Air-Source Heat Pumps)	41	AT, CA, JP, US
Annex 42 Heat Pump in Smart Grids	42	CH, DK, FR, KR, NL, UK, US
Annex 43 Fuel Driven Sorption Heat Pumps	43	AT, DE, FR, IT, UK, US
Annex 44 Performance Indicators for Energy Efficient Supermarket Buildings	44	NL, SE

IEA Heat Pump Programme participating countries: Austria (AT), Canada (CA), Denmark (DK), Finland (FI), France (FR), Germany (DE), Italy (IT), Japan (JP), the Netherlands (NL), Norway (NO), South Korea (KR), Sweden (SE), Switzerland (CH), the United Kingdom (UK), and the United States (US).
All countries are members of the IEA Heat Pump Centre (HPC). Sweden is the host country for the Heat Pump Centre.

Development of a High Efficient Heat Pump System using Seawater Heat Source and Exhaust Energy with the Automatic Decontamination Device

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Abstract: The heat pump system using seawater heat source and exhaust energy without fouling of heat exchangers is developed and its performance characteristics of heating and cooling operation are presented. The heat pump system is made of a waste heat recovery system and a vapour compression refrigeration system and the automatic decontamination device. The working fluid is R-22. The heat pump system COPs are measured during heating and cooling operation modes for an indoor culture system, and the resultant COPs are 12.5 and 11, respectively, which are higher than those of the heat pump itself. Therefore, the performance of the heat pump system using exhaust energy and seawater heat source with a decontamination device is excellent compared with that of a general heat pump. The experimental data can be effectively used for the design of the high efficient heat pump using seawater heat source and exhaust energy.

Key Words: Waste heat recovery, Seawater heat source, automatic decontamination device, Vapor compression cycle, COP

1 INTRODUCTION

The heat pump is an energy-efficient and environment-friendly apparatus for heating and cooling of built environment. Since the 1950s, researches have been developed heat pump system structure, thermodynamics, working fluids, operation controlling, numerical simulation and economical analysis (Arif et al. 2009).

From 1985 (January) to 1986 (February), The Fisheries Research Center of the Japan Kinki University used heat pump system to breed fish (sea breams) fry in land based aquaculture system, and COP of system was about 2.8 (Michiyasu and Tadao Tsuji 1992). This study showed heat pump system can be used economically, energy saving in breeding fish in land based aquaculture system (Kim IB 1993). Recently, Oh et al. have been developed a high efficient heat pump system using exhaust water heat source that can be used to breed flatfishes in land based aquaculture system. This study showed that the COP is about 6, respectively, and energy saving is eleven times of the maximums and five times of the average than an oil boiler system (Oh et al. 2000).

In this study, for the purpose of fouling seawater treatment in plate heat exchangers, the ceramic ball are installed to ball collector and ball separator as the automatic decontamination device, here it is refined the fouling seawater. The seawater is compressed and boosted by water pump and ejector to clean when the heat exchanger plates is fouled. Fouling seawater is separated by the cyclone. Where the ball ceramic is dropped into a collector and separator and a pure seawater is delivered to breed fish in the farm.

The automatic decontamination device (ADD) is developed in effort to refined fouling seawater, the ADD is retrofitted for heat pump system, and specially it is designed to enable a quick cleaning of the fouling on the plate's surface. A comparison of the COP between the heat pump system with ADD and itself was also presented.

2 EXPERIMENTAL APPARATUS AND METHOD

2.1 General remarks

In this study the experimental facility shown in Figure 1, is specially constructed for the purpose. The major components of the experimental apparatus are; heat recovery, compressor, condenser and evaporator. The heat pump is one-stage refrigeration cycle, and was typically classification as a water-to-water heat pump. Two PHEs that made from copper were used for condensing and evaporating. The heat recovery system is used to heat exchange of exhaust water and new seawater that is made of stainless steel.

The temperature of coastal seawater and fresh water in Korea vary from 4^0C to 30^0C every year. The seawater from sediment cistern at first was pumped to the first PHE where it was preheated (in winter) or precooled (in summer) by water of indoor culture system. The sediment water at the outlet then delivered to condenser to increase the temperature and flow into the indoor culture system.

The fouling water from the other outlet of the first PHE was continue pumped to the second PHE. After exchanging heat, fouling water was delivered to the cyclone where it is separated to ball and pure water. The balls then were accumulated in ball collector while the pure water go back to the sediment cistern. When the ejector was used, the ball from collector will be mixed with the water from indoor culture system and go to the first PHE for the new cycle.

The automatic decontamination device includes ball separator and ball collector. The ejector and water pumps are installed in heat pump system for the purpose of fouling treatment in plate heat exchanger surfaces.

2.2 Working refrigerant loop

The schematic of the working refrigerant loop is also illustrated in Figure 1. A high temperature and pressure gas of ①compressor enter in the ②condenser. The water is heated by heat exchange in the condenser. After condensing in the condenser the refrigerant liquid enter a ⑤receiver. The refrigerant liquid of receiver is passed through the ⑥core - dryer, ⑦sight glass, ⑧solenoid valve, ⑨shutdown valve and ⑩expansion valve. From there the refrigerant liquid is changed by the low temperature and pressure refrigerant. This vapour-liquid refrigerant flow into ③evaporator. The water is cooled by heat exchange in the evaporator, and the refrigerant is evaporated. After evaporating in the evaporator, the refrigerant vapour and any unevaporated liquid are separated in the ④accumulator. The refrigerant vapour is compressed in the compressor.

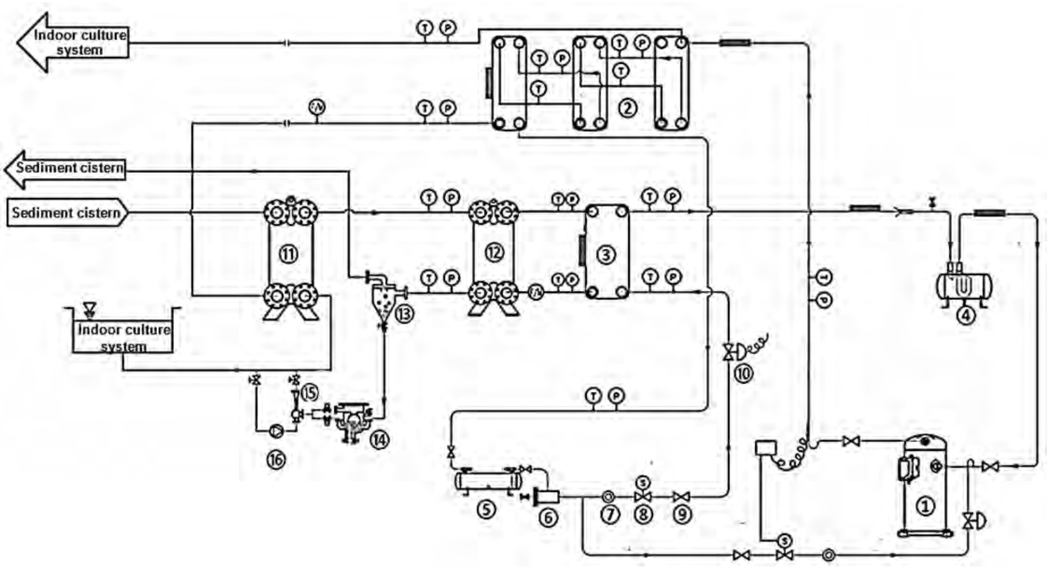
2.3 Cleaning plate heat exchangers loop

Fouling occurs when impurities cling on the heat exchanger surfaces, and it can decrease heat transfer effectiveness significantly. For the fouling treatment in heat recovery systems, as show in Figure 1, the waste water is compressed and boosted by ⑯water pump and

⑮ejector, and it is passed to ⑪the first and ⑫the second heat recovery. It is mixed with impurities cling to clean on the plate heat exchanger surfaces. The waste water is passed through ⑬ball separator, here, the ceramic ball is separated it by the cyclone. The new water from ball separator is passed to ⑭ball collector for refining and mixed waste water circulation for cleaning plate heat exchanger surfaces.

2.4 Test procedure

The heat pump is installed the automatic decontamination device, and is operated to determine COP. The refrigerant flow rate in heat pump was controlled by electric expansion valve. All data were taken after the operating conditions reached a steady state that all temperature and refrigerant flow rate has not changed.



① Compressor	⑤ Receiver	⑨ Shutdown valve	⑬ Ball separator
② Condenser	⑥ Core - dryer	⑩ Expansion valve	⑭ Ball collector
③ Evaporator	⑦ Sight glass	⑪ 1st heat recovery	⑮ Ejector
④ Accumulator	⑧ Solenoid valve	⑫ 2nd heat recovery	⑯ Pump

Figure 1: Schematic diagram of heat pump system

Table 1: Experimental conditions

Parameter		**Dimension**
Ceramic ball	$m^2.h.^0C/kcal$	0 – 0.000496
Compressor	Type	Copeland - ZR380KC
Condenser (KAORI)	Temperature (^0C)	13 ~ 28
	Heat transfer area (m^2)	7.176
	Number of plates	40
Evaporator (KAORI)	Temperature (^0C)	-10 ~ 7
	Heat transfer area (m^2)	10.77
	Number of plates	100
Heat recovery (Model DX-17DLH-1p-95)	Number of plates	95
	Heat transfer area (m^2)	15.81
Refrigerant	R22	

2.5 Plate heat exchanger area of heat recovery system

This heat pump system used the PHE (plate heat exchanger) for the heat recovery system, condenser and evaporator. Fundamental performance model is to be simplified for the calculation of heat transfer and PHE area. For analysis of heat transfer characteristic of PHE, Cooper A. and Usher J.D's correlation is used to measure turbulent flow in PHE chevron angle (β) 120°.

$$\mathrm{Nu} = 0.2\,\mathrm{Re}^{0.67}\,\mathrm{Pr}^{0.4}(\eta/\eta_W)^{0.1} \tag{1}$$

In the equation (1), the overall heat transfer coefficient with a calculated heat transfer coefficient can be calculated as follows.

$$1/U = 1/h_{\mathrm{h}} + 1/\lambda_{\mathrm{p}} + 1/h_{\mathrm{c}} + \mathrm{R}_{\mathrm{f}} \tag{2}$$

where λ_p and R_f are thermal conductivity and fouling factor of the PHE, respectively.

The characteristic analysis of the heat exchanger can be carried out using ε-NTU method (Lee et al. 2000). The NTU method has gained greatest acceptance in connection with design of compact heat exchangers where a large surface area per unit volume exists.
Heat exchanger efficiency was mentioned in the previous section as

$$\varepsilon = \text{Actual heat transfer rate / Maximum possible heat transfer rate} \tag{3}$$

The actual heat transfer rate is given by

$$\dot{q} = C_h(t_{hi} - t_{ho}) = C_c(t_{co} - t_{ci}) \tag{4}$$

The maximum possible heat transfer rate is expressed by

$$\dot{q}_{\max} = C_{\min}(t_{hi} - t_{ci}) \tag{5}$$

This is true because the maximum heat transfer would occur if one of the fluids were to undergo a temperature change equal to the maximum in the heat exchanger, (t_{hi}-t_{ci}). The fluid experiencing the maximum temperature change must be the one with the minimum value of C to satisfy the energy balance.

The fluid with the minimum value of C may be the hot or the cold fluid. For $C_h = C_{min}$, using Eqs. (4) and (5)

$$\varepsilon = \dot{q}/\dot{q}_{\max} = C_h(t_{hi} - t_{ho})/C_{\min}(t_{hi} - t_{ci}) = (t_{hi} - t_{ho})/(t_{hi} - t_{ci}) \tag{6}$$

For $C_c = C_{min}$

$$\varepsilon = \dot{q}/\dot{q}_{\max} = C_c(t_{co} - t_{ci})/C_{\min}(t_{hi} - t_{ci}) = (t_{co} - t_{ci})/(t_{hi} - t_{ci}) \tag{7}$$

It is, therefore, necessary to have two expressions for the efficiency (Eq. (6) and (7)). When efficiency is known, the outlet temperature may be easily computed.
For example, when $C_h < C_c$

$$t_{co} = \varepsilon(t_{ci} - t_{hi}) + t_{hi} \tag{8}$$

Also

$$t_{co} = \left(\dot{q}/C_c\right) + t_{ci} = \left[C_h/C_c(t_{hi} - t_{ho})\right] + t_{ci} \tag{9}$$

$$t_{co} = \left[\left(C_h / C_c \right) . \varepsilon . (t_{hi} - t_{ci}) \right] + t_{ci} \quad (10)$$

The NTU parameter is defined as UA/C_{min} and may be thought of as a heat transfer size factor. That may also be observed that flow configuration is unimportant when $C_{min}/C_{max} = 0$. This corresponds to the situation of one fluid undergoing a phase change where c_p may be thought of as being infinite. Evaporating or condensing refrigerant as well as condensing water vapor are examples where $C_{min}/C_{max} = 0$. The NTU method has gained greatest acceptance in connection with design of compact heat exchangers where a larges surface area per unit volume exists.

2.6 COP of Heat Pump System

The COP_{HP} of heat pump is then equal to the heat output divided by the work input:

$$COP_{HP} = Q_c / AW_{comp} = (Q_e + AW_{comp}) / AW_{comp} = 1 + Q_e / AW_{comp} \quad (11)$$

The heating COP_{hs} of heat pump system are different with the COP_{HP} of heat pump because the total input energy is more than the compressor power. The heating COP_{hs} of heat pump system is defined as

$$COP_{hs} = Q_{tc} / AW_e \quad (12)$$

Where, the Q_{tc} is the sum of the condensation heat energy and the rejected heat energy from a motor, the AW_e is the sum of the compressor work (AW_{comp}) and the electricity energy of a condensation fan, etc.

And the COP_{cs} calculation of cooling is the same as the heating calculation method in heat pump system. The cooling COP_{cs} of heat pump is defined as

$$COP_{cs} = Q_{te} / AW_e \quad (13)$$

Where, the Q_{te} is the sum of the evaporation heat energy and the heat flow energy of the surrounding.

The compressor and expansion valve are as follows: Assumption for the characteristic analysis of heat pump system.
(1) The compressor is a company manufacture.
(2) The pressure variation in heat exchanger has nil.
(3) The expansion valve has a constant enthalpy expansion.

3 RESULT AND DISSCUSION

3.1 Simulation in Heat Pump System

3.1.1 Heating Operation

Figure 2 shows the heating capacity of a heat pump system along with various refrigerant evaporation temperatures. As shown in Figure 2, the heating capacity increased with an increase of evaporation temperature, and decrease of condensation temperature. Figure 3 shows the COP_{hs}(heating) of heat pump system along with various refrigerant evaporation temperatures. The COP_{hs} increased with an increase of evaporation temperature, and decrease of condensation temperature. It is similar to the trends as heating capacity.

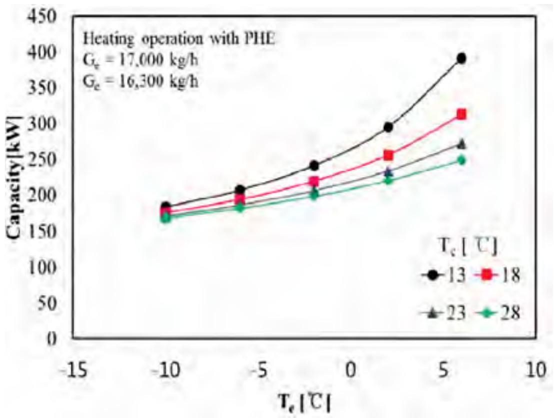


Figure 2: Heating capacity of heat pump system, G_e= 17,000 kg/h; G_c = 16,300 kg/h

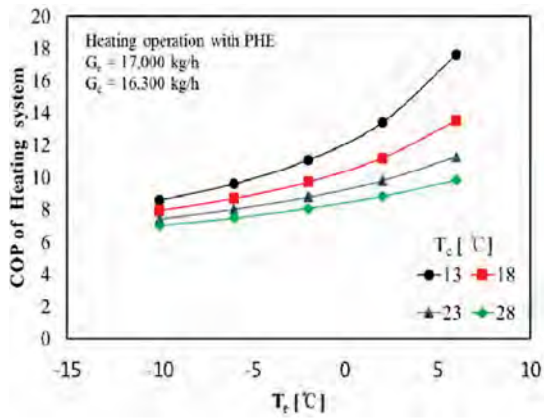


Figure 3: COP_{hs} (heating) of heat pump system, G_e= 17,000 kg/h; G_c = 16,300 kg/h

3.1.2 Cooling Operation

Figure 4 shows the cooling capacity of a heat pump system along with various refrigerant evaporation temperatures. As shown in Figure 4, the cooling capacity increased with increase of evaporation temperature. Figure 5 shows the COP_{cs} (cooling) of heat pump system along with various refrigerant evaporation temperatures. This is similar to the trends as shown in Figure 3. That is, the COP_{cs} increase with a decrease of condensation temperature when the evaporation temperature is constant.

3.2 Comparison of simulation and experimental data in Heat Pump

The COP of simulation result and experimental data is compared with changing evaporation and condensation temperatures simultaneously.

3.2.1 Heat pump COP with new heat exchangers

Figure 6 and Figure 7 show the COP_{hs} (heating) and COP_{cs} (cooling) of heat pump with using new heat exchangers, respectively. As shown in figures, The COP of simulation is higher about 20% than that of experimental data. And Figure 6 and Figure 7 has a bigger error with increasing evaporation and condensation temperature when compare simulation value with experimental data.

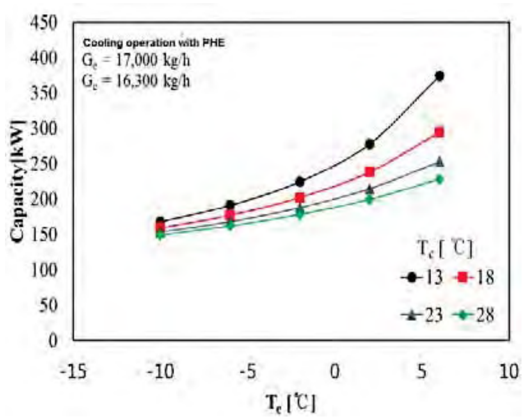


Figure 4: Cooling capacity of heat pump system, G_e= 17,000 kg/h; G_c = 16,300 kg/h

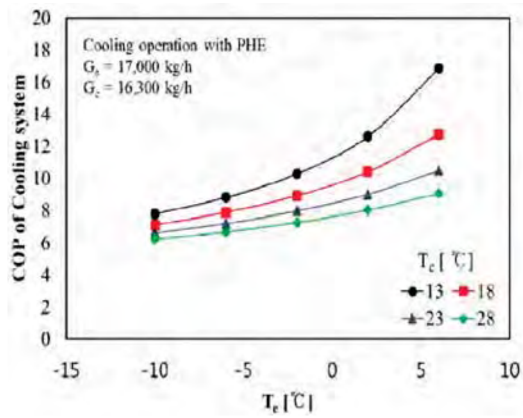


Figure 5: COP_{cs} (cooling) of heat pump system, G_e= 17,000 kg/h; G_c = 16,300 kg/h

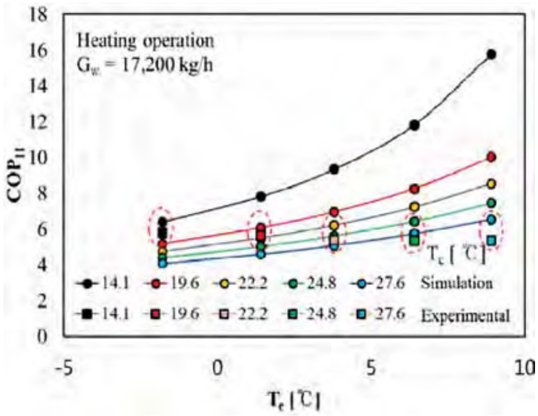


Figure 6: Comparison of COP of experimental data and simulation result in heating operation with new heat exchangers

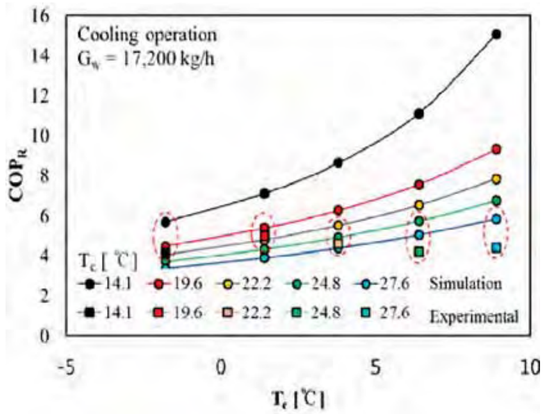


Figure 7: Comparison of COP of experimental data and simulation result in cooling operation with new heat exchangers

3.2.2 Heat pump COP with fouling heat exchangers

Figure 8 and Figure 9 show the COP_{hs} (heating) and COP_{cs} (cooling) of heat pump with using fouling heat exchangers. As shown in figures, The COP of simulation is higher about 20% than that of experimental data. Namely, it is similar to the trends as Figure 6 and Figure 7, respectively.

3.2.3 Heat pump COP with cleaning heat exchangers

Figure 10 and Figure 11 show the COP_{hs} and COP_{cs} of heat pump with using cleaning heat exchangers. As shown in figures, The COP of simulation is higher about 13% than that of experimental data.

Therefore, The COP of heat pump with using new heat exchangers is highest of heat pump with using cleaning and fouling.

3.3 COP of heat pump system in indoor culture system

3.3.1 Heating Operation

Figure 12 shows the COP along water temperature of sediment cistern when the heat pump

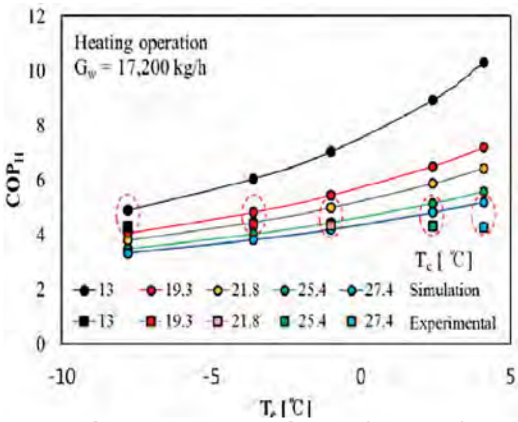


Figure 8: Comparison of COP of experimental data and simulation result in heating operation with fouling heat exchangers

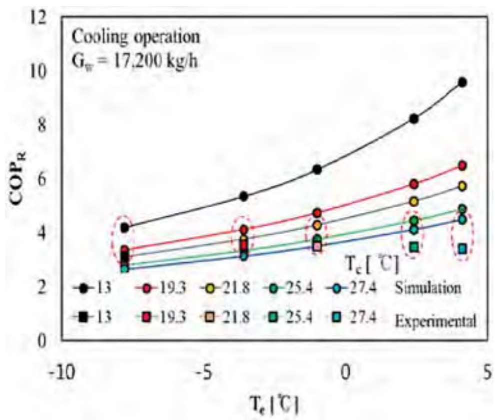


Figure 9: Comparison of COP of experimental data and simulation result in cooling operation with fouling heat exchangers

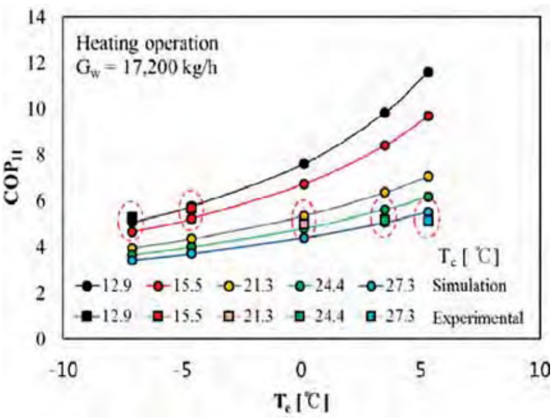


Figure 10: Comparison of COP of experimental data and simulation result in heating operation with cleaning heat exchangers

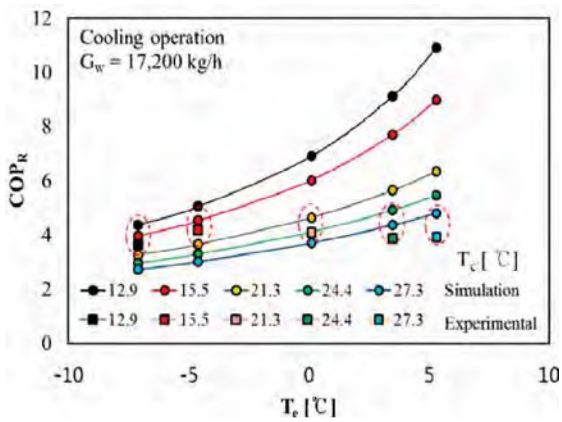


Figure 11: Comparison of COP of experimental data and simulation results in cooling operation with cleaning heat exchangers

system did operate heating. As can be seen in the figure, at a constant sediment cistern water temperature, 4^{0}C, the supply water temperature is heated to 18.7^{0}C. And the COP of heat pump system is about 18.

The region A in Figure 12 represents a ON/OFF operation of a compressor for a supply water temperature control. The region B shows a heating operation for a supply water temperature variation at a constant sediment cistern temperature. As shown in the figure, the supply water temperature in the indoor culture system remained constant of 18.7^{0}C under a constant sediment cistern water temperature, 5^{0}C. In the tests of experimental apparatus, the supply water temperature is 18.7^{0}C because a heat loss is caused by the surroundings of indoor culture system. The COP of heat pump system is 18 when a run was conducted as indicated region B of Figure 12. These results suggest that the heat pump system has operated at a steady state.

3.3.2 Cooling Operation

Figure 13 shows the COP along with water temperature of sediment cistern when the heat pump system operated under the cooling condition. This is similar to the trends as shown in Figure 12. The supplied water temperature entered into indoor culture system is controlled at a constant, 20.1^{0}C. In Figure 13, the region A is the beginning of a cooling operation, the region B is a cooling operation process for a supply water temperature variation at a constant

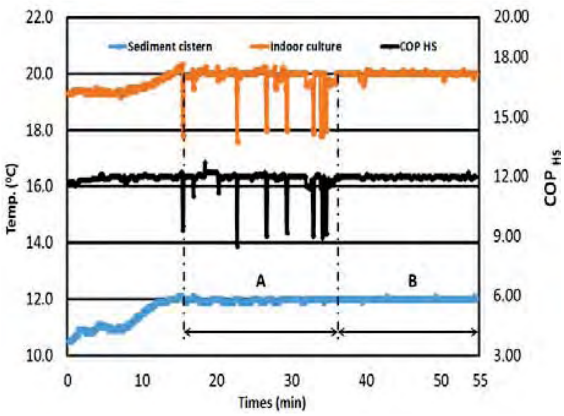


Figure 12: COP_{HS} of heating operation of heat pump system, T_{tank}=20.1^{0}C, G_w=6000kg/h

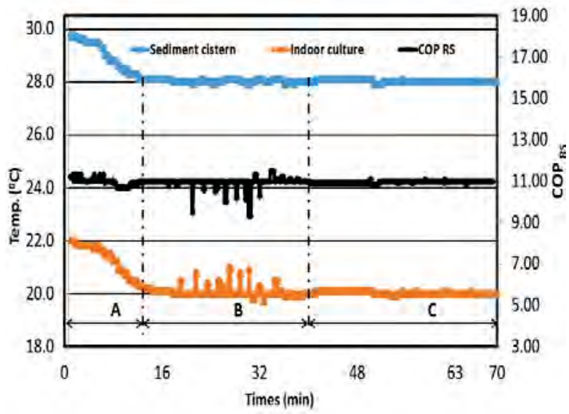


Figure 13: COP_{RS} of cooling operation of heat pump system, T_{tank}=20^{0}C, G_w=6000kg/h.

sediment cistern water temperature, 28^{0}C. The COP of heat pump system is from 10.8 to 11 when it was operated in region B and C. These results proved consistent in several tests.

4 CONCLUSIONS

The result of experimental involving a high efficient heat pump system using seawater heat source and exhaust energy with the automatic decontamination device may be summarized as follows:

1. The COP of simulation is higher about 20% than that of experimental data in heat pump with using plate type heat exchanger.
2. The fouling factor is effect on performance of this system. The COP decrease with its increase.
3. The COP of heat pump with automatic decontamination device is higher than a heat pump itself on a heating and cooling operations.
4. The heating and cooling COP of heat pump system is 18 and 12.5 when the heat pump system has operated with a steady state in indoor culture system, respectively.

5 ACKNOWLEDGMENTS

This work was supported by the Korea Institute of Energy Technology Evaluation and Planning (KETEP) Grant no. (2011T100100464) of the Energy Technology Development.

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OPTIMIZATION OF CONTROL STRATEGIES – SWITCHING BETWEEN PASSIVE COOLING AND REVERSIBLE HEAT PUMP

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Abstract: So far analyzed geothermal systems in office buildings differ in their operation, so that a generalization of the results is restricted. Within the scope of the R&D project geo:build ground coupled supply systems for heating and cooling in the buildings are being analyzed both in theory and practice. The focus is to study an adjustment of the cooling modes and switching between passive cooling and chiller mode. Furthermore, a development of energetic and economical sensible combinations of these technologies is foreseen. In the project, four office buildings will be measured and analyzed. The planned targets for efficient combination of passive cooling and chiller mode are not yet sufficiently implemented, despite of implemented improvements.

Key Words: ground coupled reversible heat pump, energetic and economic efficiency, optimization, control strategies, monitoring

1 INTRODUCTION

In theory, it is planned to use primarily an efficient passive cooling for cooling the office buildings in the summer. However, in practice, it is determined, that buildings with integrated chiller use continuously mechanical cooling. This is partly due to an insufficiently coordinated control of passive cooling (PC) and chiller mode (CM) operation and, on the other hand, due to a raised soil temperature. As a result, a balance between heat injection and extraction can be disturbed and the possibility for using the passive cooling is often not guaranteed.

Within the scope of the ongoing R&D-project "Optimization of ground coupled heating and cooling supply systems in office buildings - reversible heat pump and passive cooling", control strategies of switching between passive and active cooling (rev. heat pump) are being analyzed both in theory and practice. The project is conducted by the IGS - Institute of Building Services and Energy Design at the Technical University of Braunschweig in cooperation with a scientific partner and the two industrial partners.

2 METHODOLOGY OF THE RESEARCH AND DEVELOPMENT PROJECT

At the beginning of the research project, a closer look is taken at coordination and switching between passive cooling and chiller mode. Subsequently, possible energy efficient and economically sensible combination of this technology will be developed.
The focus is on the monitoring of four office buildings (Table 1) with a reversible heat pump and passive cooling mode as well as the analysis of their operation strategies and operating system functions. Additionally building and plant simulations as well as simulations of heat extraction and heat injection into the soil and its thermal behavior will be carried out. The knowledge acquired from the simulations will be implemented and tested (see Figure 1).

The aim of the R&D-project is to optimize the implementation of geothermal heat and cold storages as well as to develop and test more efficient use of these storages. To achieve an improved storage efficiency ratio, optimized operation strategies and application-oriented storage concepts should be further developed and evaluated in particular with ground coupled chiller and the operation of switching between chiller and passive cooling mode.

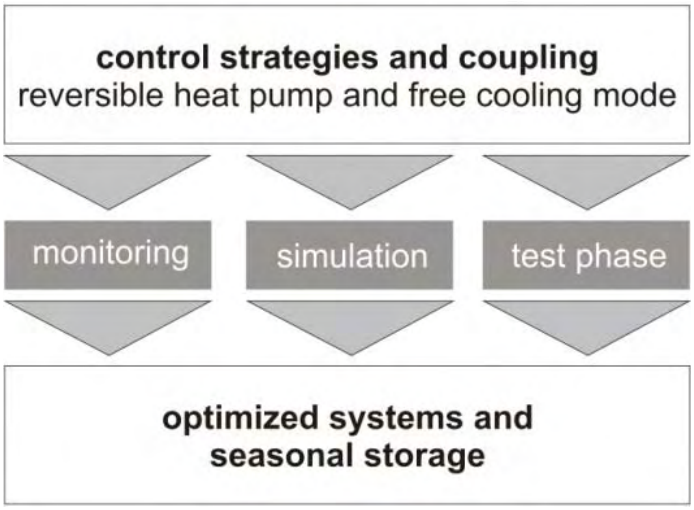

Figure 1: Proceedings

Approaches to design and control in terms of

- operation and control concepts enabling an immediate switching between chiller and passive cooling mode,
- potentials of an intermittent operation of the probe field,
- the regeneration phase between chillers operation and passive cooling mode,
- a constant thermal conditions in the soil as well as a realizable heat extraction and heat injection, the available load to the building and the correlation between heat extraction and heat injection

should be processed.

Table 1: Data of building, geothermal systems and heating-/cooling concepts (NFA – net floor area)

Gelsenwasser AG, Gelsenkirchen		
	building data	office building NFA 6 189 m² year of construction 2004
	geothermal system	36 borehole heat exchanger à 150 m
	design heating load	total building 207 kW / 43.6 W/m^2_{NFA} heat pump 326 kW
	design cooling load	total building 305 kW / 9.3 W/m^2_{NFA} passive cooling 200 kW rev. heat pump 320 kW
VGH Regionaldirektion, Lüneburg		
	building data	office building NFA 3 957 m² year of construction 2002
	geothermal system	101 energy piles à 20 m
	design heating load	total building 350 kW / 88.5 W/m^2_{NFA} heat pump 85 kW
	design cooling load	total building 120 kW/ 30.3 W/m^2_{NFA} passive cooling 80 kW rev. heat pump 89 kW
Office Freundlieb am See, Dortmund		
	building data	office building NFA 2 930 m² year of construction 2010 / 2011
	geothermal system	12 borehole heat exchanger à 144 m
	design heating load	total building 125 kW / 42.7 W/m^2_{NFA} heat pump 87.6 kW
	design cooling load	total building 95 kW / 32,5 W/m^2_{NFA} passive cooling 60 kW rev. heat pump 68.4 kW
Lecture Hall, Salzgitter		
	building data	lecture hall building NFA 3 296 m² year of construction 2012 / 2013
	geothermal system	15 borehole heat exchanger à 95 m
	design heating load	total building 92 kW / 28 W/m^2_{NFA} heat pump 60 kW
	design cooling load	total building 152 W / 46 W/m^2_{NFA} passive cooling 60 kW rev. heat pump 45 kW

3 CONTROL STRATEGIES FOR SWITCHING BETWEEN THE MODES

Already at the beginning of the project, it could be determined how large and variable implementation and integration of passive and active cooling in a geothermal plant can be. Moreover how important it is to implement and coordinate this combination in the right way to operate buildings efficiently and to protect the ground.
A literature review as well as a survey of the heat pump manufacturers show that the previous and common rules for the combination of passive cooling and chiller is based on simple comparisons. According to it, four main control strategies can be defined:

1. set-point regulation: as soon as a defined temperature is exceeded or undercut, the operation mode changes from PC to CM and vice versa. Parameters such as supply temperature or return temperature (distributor, CCA, HVAC, etc.), outlet or inlet temperature from / into the soil, room temperature or ambient air temperature are considered.

2. difference-regulation: change between the two modes is done in response to a predefined temperature difference. To form the differences, the parameters supply and return temperature at the distributor, the inlet and outlet temperatures of soil, primary and secondary side or extraction and undisturbed soil temperature are being used.

3. duration scheme: this strategy is based on, for example a certain time program which defines switching between the PC and CM or it is based on running in intervals

4. algorithms: lately, the algorithms are being implemented to draw up efficient control strategy including a holistic attitude to building. The factors considered: balanced soil temperature, energy / primary energy consumption, energy cost, comfort, weather forecasts, etc.

In addition, all combinations of the presented control strategies are possible and implemented.
Already on basis of the examined buildings in the project, it becomes clear how different the implementation of the scheme of passive cooling and chiller mode can be. It can be seen that primarily the focus is laid to the set-point regulation – due to diversity of variants - based on the primary or secondary side of the system.

The currently defined control strategies and switching parameters between passive cooling and chiller operation provide that:

- Building GEW: approval for chiller operation, if the outlet temperature from the soil exceeds 18 C.
- Building VGH: passive cooling ends as soon as the supply temperature to the emission systems exceeds the set value.
- Building FAS: approval for the chiller operation, if the term

  $(T_{\text{outlet BHE}} + 2K) > (T_{\text{set emission system}} + 1K)$ is satisfied.
- Building HSS: passive cooling in operation until the supply temperature to the distributor exceeds the 16 °C.

4 FIRST MONITORING RESULTS - FUNDAMENTALS

The collected experiences from the preceding project "WKSP - heat and cold storage in the foundation area of office buildings" (BMWi, FKZ 0327364A) are the basis for carried out monitoring and further optimizations in the research project geo:build.
First optimizations and bug fixes are implemented in the buildings VGH and Gelsenwasser AG as well as in the Lecture Hall regarding the performance of the systems, heat injection and heat extraction to achieve an even balance in the soil as well as the implementation of the designed operation.
The design goals concerning more efficient use of the passive cooling mode compared to the chiller operation was still not yet been adequately implemented. This is where geo:build starts.

4.1 VGH Regionaldirektion, Lüneburg

Based on the heat injection and heat extraction (Figure 2) it becomes clear that until 2007 no planned heating or cooling operation of the geothermal system has been scheduled.
To increase the extraction of heat from the soil and to establish a systematic heating and cooling mode, the following measures and optimizations have been implemented as part of the first monitoring:

- Elimination of design errors in the concrete core activation system (e.g. improperly installed valve)
- Elimination of control errors in the building management system (BMS)
    - Modification of the calculation of the mean ambient temperature
    - Adaption of heating and cooling boundary set point temperatures
    - Adjustment of the heating and cooling curves of concrete core activation and ventilation systems
- Localization of errors in the internal controller of the geothermal heat system.
- Implementation of the heating operation in spring 2007.
- Partial realization of the cooling operation in summer 2009.

Through the carried out optimization measures and the implementation of the designed cooling operation the ratio of cooling supply by chiller to passive cooling in the year 2007 was 89% to 11% and will be reduced in 2009 to 55% to 45%. There is no information on the designed proportion of compression chiller/ passive cooling.
Due to the high heat extraction and in result disturbed energy balance in the soil (Figure 2), the temperature in the soil is very low. Thus, a further reduction of the chiller operation and implementation of the efficient passive cooling mode should be possible.

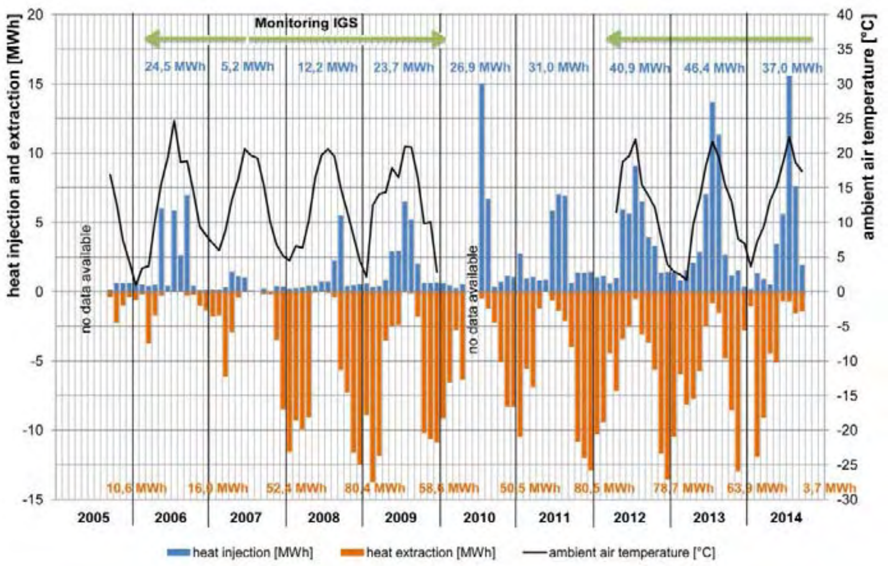

Figure 2: Monthly heat injection and heat extraction (2005 – 2013), VGH

4.2 Gelsenwasser AG, Gelsenkirchen

During 2006 to 2009 significantly more heat was injected into the soil than extracted (about two to three times). The heat came e.g. from the building itself (combined operation mode) and the high fraction of the chiller. The result is a warming of the soil to an unfavorable temperature level for passive cooling mode, so that during the cooling mode mainly the chiller was operated.
According to the planning documents for the Gelsenwasser AG, a ratio for cooling supply by the passive cooling mode to chiller was designed 68% to 32%. However, to date, it was only possible to achieve a ratio of 49% to 51% (see Figure 3).
As a part of the existing monitoring, measures and optimization were carried out to minimize the heat injection, in particular the combined heating and cooling mode and to reduce the high fraction of chiller operation.

Measures and optimization:

- Optimized ventilation strategy:
  - No cold supply during office hours at low outside temperatures.
- Use of self cooling through the building envelope and the supply air flaps
  - No space cooling during the night when the outside temperatures are less than the room temperature.
- Priority for the free night cooling.
- Changing the control strategy of the geothermal system:
  - Increasing the temperature limit (outlet temperature from the ground heat storage) for approval of operating chillers.

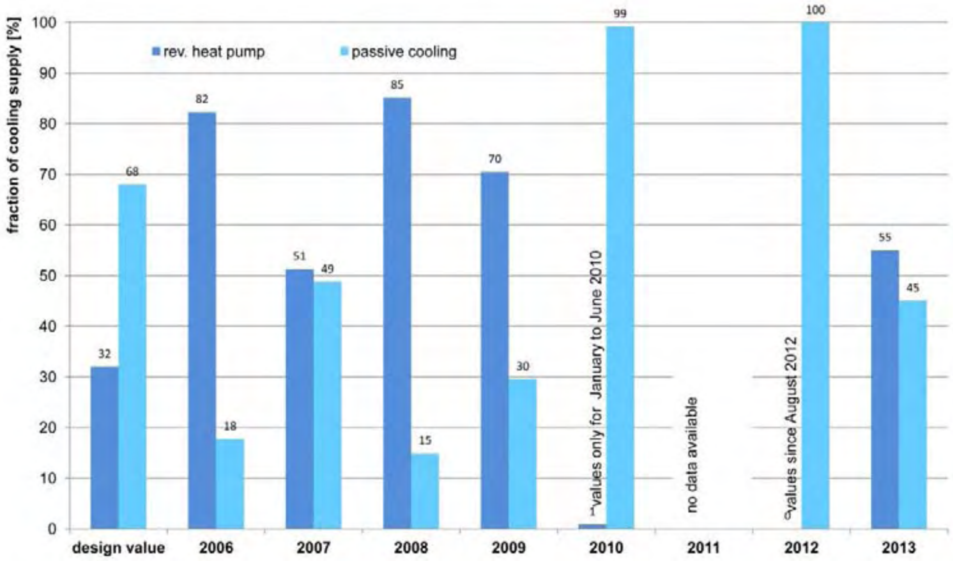

Figure 3: Percentage distribution of refrigeration by passive cooling and chiller operation (GEW), 2006 - 2013

4.3 Lecture Hall, Salzgitter

Since March 2013 the full monitoring of the Lecture Hall building in Salzgitter has been added with the start of using. In planning, a ratio of 54% operation of the reversible heat pump to 10% passive cooling was defined based on thermal simulations. In order to cover peak loads, two additional compression chillers are integrated to cover the remaining 36% of the total cooling energy demand.
Figure 4 shows the first ratio for the period from August until December 2013, after the commissioning phase of the operation of building and measurement equipment. In essence, the cold is provided by the heat pump in the chilling mode. The potential of the passive cooling is generally limited at the end of the cooling period due to rising temperature in the soil. An optimization is necessary to raise the proportion of passive cooling at the end of the cooling period 2014.

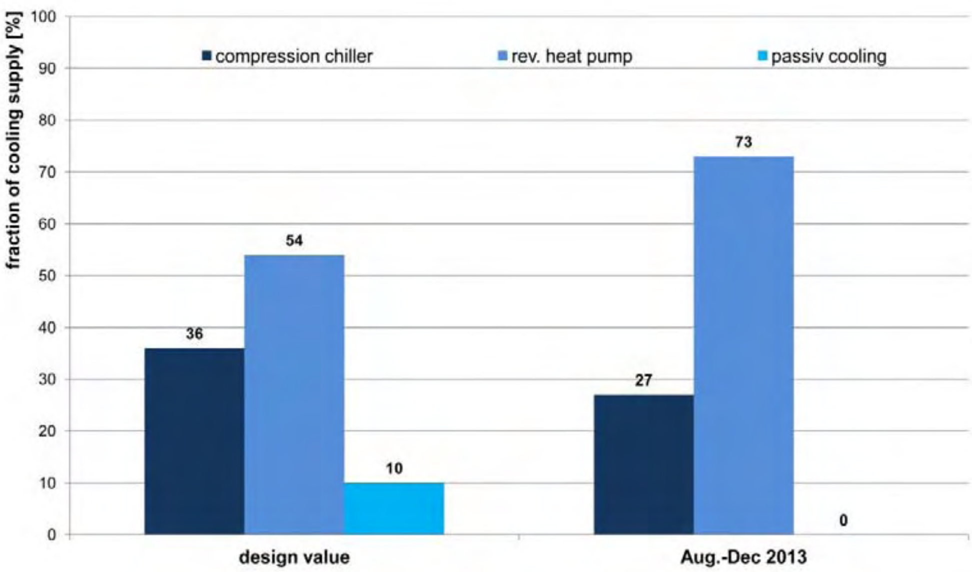

Figure 4: Percentage distribution of refrigeration by passive cooling and chiller operation (HSS), 2013

5 TRNSYS SIMULATION

Using the simulation software TRNSYS, the lecture hall in Salzgitter and the VGH Lüneburg are created in a holistic building and plant model. The model is imaged with all building-specific configurations. The rooms in the buildings will combined in zones according to their thermal boundary conditions (Figure 5 and Figure 6) and their physical properties. Important boundaries to heating, cooling, ventilation and internal loads, etc. are also determined for each zone.

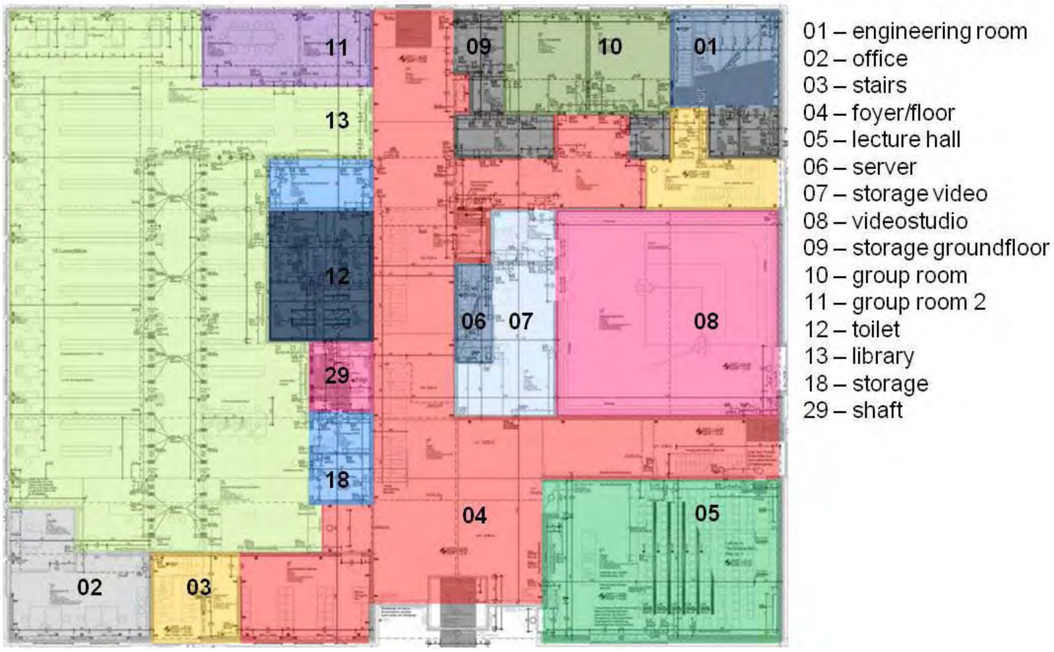

Figure 5: Zoning lecture hall Salzgitter, ground floor (total 29 zone)

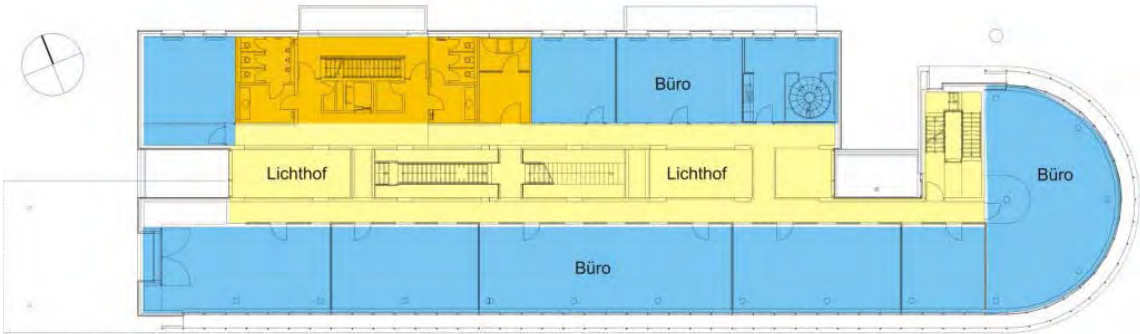

Figure 6: Zoning VGH, 1. floor (blue: office, separated in north and south; orange: toilet and kitchen, yellow: floor (areaway))

A universal base simulation model is created for the buildings, including the plant technology. The model used general data interfaces for input parameters, such as weather data, and data outputs such as load curves (Figure 7). The building models, created for each project, are linked with pre-defined transfer interfaces on the load profiles for heating and cooling with the plant technology. Due to the general structure of the simulation model, the effort will be reduced to simulate the various buildings and equipment designs. So only the different performing systems for each building will be supplemented and adjusted in the standard deck. The simulation data are stored in output files and can be used for further processing, graphic processing and for direct comparison with measured data from the real operation.

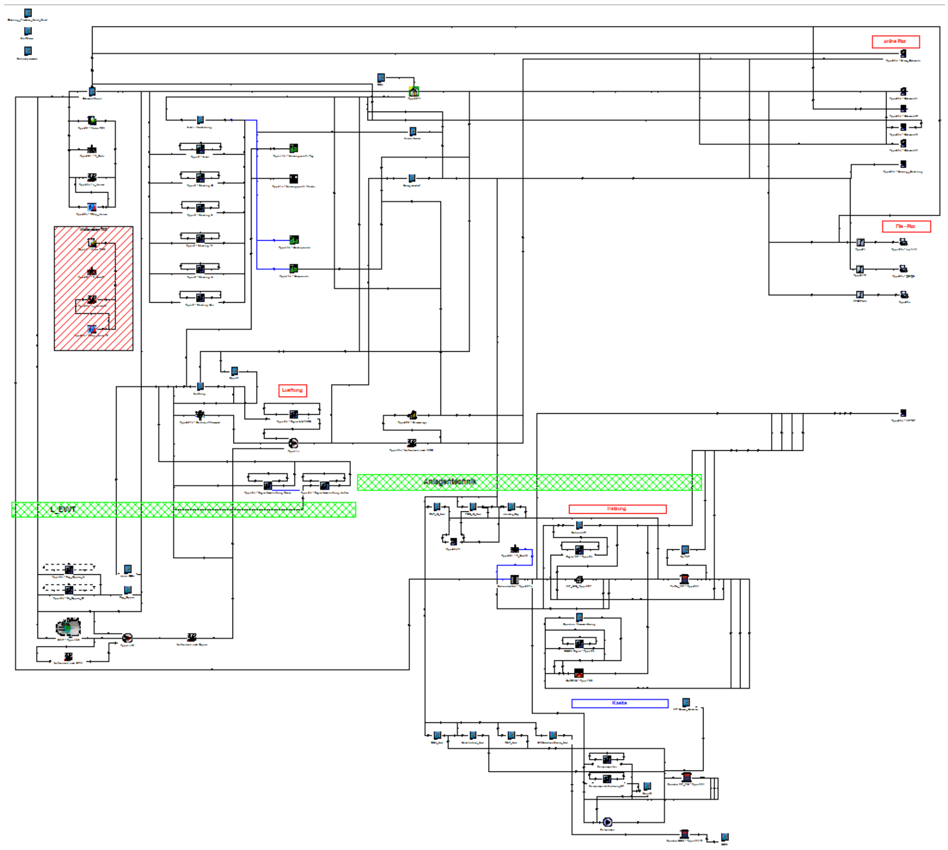

Figure 7: current „universal deck“

The validation of the model with existing data is currently being proceeded, so that the building and the control strategies can be tested.
The validation can achieve a very good result. For example the borehole heat exchanger (used Type 557): according to Figure 16 it can be seen that the deviation between simulation and measured values amount to ~ 7% (Figure 8). All settings and parameters from the planning data and the TRT (thermal response test) can be applied to lecture hall Salzgitter.

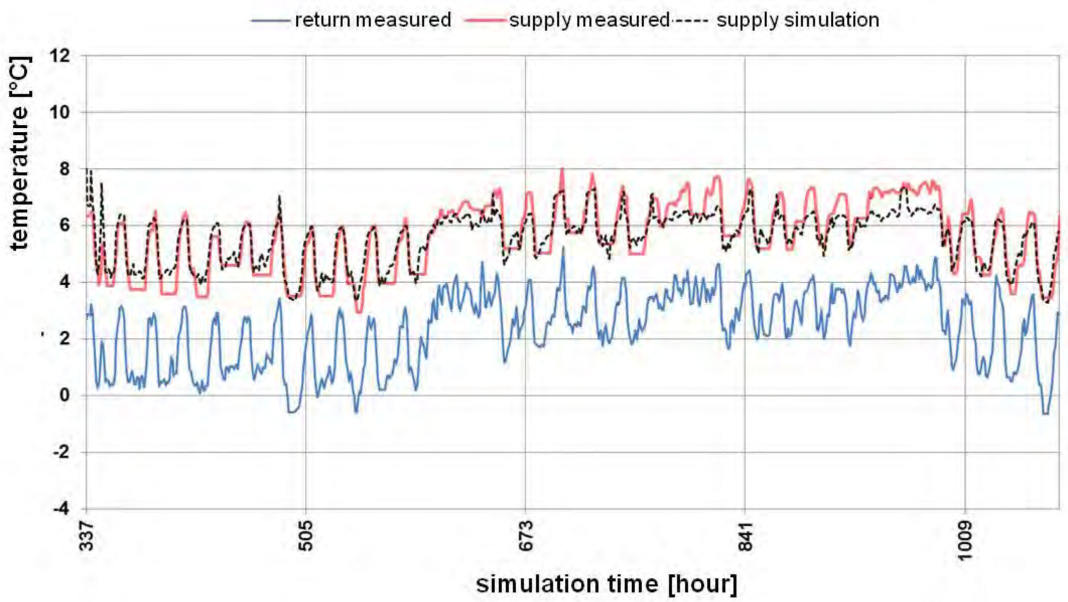

Figure 8: Result of validation of the borehole heat exchanger

Due to the comparison of the results from simulation, design and measured values from monitoring, it is possible to analyze optimization approaches prior to field studies.

6 FIRST ESTIMATIONS

Based on the measured cooling consumption and the soil temperatures (both 15-minute values) of the VGH, a first rough estimation of the different control strategies is made. The assessment does not consider the actual response of the soil or the building. It is only compared to the duration of passive cooling vs. mechanical cooling with reference to the existing measured data and the varied control strategies. To select and preliminary analyze the strategies, the ambient air temperature, supply and return temperature in the building as well as the inlet and outlet temperatures of the energy piles were used.

Strategies:

Actual / current strategy: Passive cooling ends as soon as the supply temperature to the emission systems exceeds the set value of the cooling curve.

A – set value soil: Passive cooling ends as soon as the outlet temperature from the energy piles (ground) exceeds 17.5 °C or the ambient air temperature exceeds 24.5°C.

B – temperature difference energy piles: Passive cooling starts when the temperature difference exceeds 2K. The chiller operation works between 0 – 2K.

C – temperature difference emission system: If the difference is smaller than 0.6 K, the passive cooling starts.

D – program + set value: passive cooling between 8 pm and 2 am as well as between 6 am and 10.30 am and if set value for emission system is > 20°C. Otherwise active cooling.

E – program + set value 2: passive cooling between 8 pm und 6 am and if set value for emission system is > 20°C. Otherwise active cooling.

Table 2 shows the comparison of the tested control strategies in terms of the potential of passive cooling. Based on the results, an initial assessment can be done, which control strategies are suitable for increasing the passive cooling operation and which cannot.

Using the first rough results it should be highlighted that it is a great potential in the change of control strategies, to use the more efficient passive cooling and thus to conserve the soil. The potentials also show that it is crucial to look not only to the predetermined cooling curve of the building by creating a control strategy. The strategies should be implemented considering mutual interactions between the building and the soil.

Table 2: Results of first estimations concerning variation of strategies

Strategy	Operation time passive cooling simulation	Operation passive cooling reality	Potential
A – set value soil	1 252 h	305 h	947 h
B – temperature difference energy piles	1 099 h		794 h
C - temperature difference emission system	405 h		100 h
D - program + set value	1 257 h		952 h
E - program + set value 2	1 133 h		828 h

7 CONCLUSION

The monitoring results show that the post commissioning monitoring of operation is important in order to identify and resolve problems at an early stage.
With the completion and validation of the shown models of buildings and plants, the foundation is laid, to identify further optimization potential and to develop valuable control strategies for planning. Before an implementation in field installations such a simulation allows the testing of new control strategies for building cooling with geothermal systems in theory.

8 ACKNOWLEDGEMENTS

The authors gratefully acknowledge the Federal Ministry of Economics and Technology through funding the research project geo:build (BMWi, FKZ 03ET1024A).

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Techno-economic evaluation of combining heat pump and mechanical steam compression for the production of low pressure steam from waste heat

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Editor´s note: All costs stated in this article are in US dollars.

Abstract: Most of the industrial use of energy is for thermal applications such as drying and heating. A large percentage of the energy used is lost or dissipated into the environment as heat through warm liquid effluents and gaseous effluents. We evaluated an approach relying on combined use of a mechanical closed-cycle heat pump and mechanical steam compression. The heat pump produces steam at moderate vacuum conditions which is afterward compressed above atmospheric pressure. This sequence is to be used for upgrading of waste heat at temperatures of 60 °C, with the subsequent production of low pressure steam (110 °C or above) as a hot utility. A techno-economic evaluation of such a sequence is presented in the context of heat recovery from wet air of a paper drying process. A COP higher than three is achievable. Although such a sequence is currently hardly economically justifiable in a context of low natural gas prices, it could eventually be beneficial, especially where renewable energy is available and where the price of electricity compares to fuel prices.

Key Words: heat pump, mechanical steam compression, waste heat, wet air, heat recovery

1 INTRODUCTION

Most of the industrial use of energy is for thermal applications such as drying and heating of reactors. In the Canadian industry, about 70 % of the energy used is simply lost or dissipated into the environment as heat (Stricker & Associates Inc. 2006). Namely, heat is lost though warm liquid effluents (40 °C or above) which have to be cooled with a cooling tower or other heat dissipating device. Heat is also lost through gaseous effluents such as flue gas from combustion processes and wet air from air drying processes. In many cases, the quality of lost heat is poor because its temperature is too low for practical use or economic recovery by means of conventional heat recovery devices such as heat exchangers. There is a need for a flexible, efficient and economic approach, specifically for applications requiring heat at temperature above 100 °C.

The use of closed-cycle heat pumps (HP) for heat valuation is well known but there are limitations related to the achievable final temperature (below 120 °C) and the economically achievable temperature lift (typically below 50 °C). Furthermore, the performance of a heat pump is strongly related to the composition and conditions of each of the streams serving as heat source and heat sink, respectively. The idea of using HP for heat recovery in air drying process is not new and there are many commercial applications relying on this technology (Prasertsan and Saen-saby 1998, Morris 1993, Colak and Hepbasli 2009a and 2009b). Nevertheless, the use of HP for the production of steam (as a hot utility) is less considered, although this application of HP had been evaluated in the past (Binet et Frote 1987, Abrahamsson et al. 1997). The use of two stages HP or a combination of HP with mechanical steam compressors or thermo compression device such as ejectors had also received some attention. Mechanical steam compressors alone have also been examined (Mujumdar 2006).

In this paper, we briefly present an approach relying on combined use of mechanical heat pump and mechanical steam compression. This sequence is to be used for valorisation of heat from waste heat down to temperatures of 50 °C with the production of low pressure steam (117 °C or above) as a hot utility. It is considered that such a utility can fulfill a large part of the energy requirements, namely in the pulp and paper industry and the agro-food industry. A basic techno-economic evaluation of such a sequence is presented, in the context of heat recovery from wet air from drying process.

2 SYSTEM PROPOSED

2.1 Description of the System

We present here the techno-economic evaluation of a heat recovery system for the valorisation of heat from a typical paper air drying process. Characteristics of the waste heat effluent (wet air from paper drying process) that we consider for the purpose of techno-economic evaluation are shown in Table 1. The heat valorisation system is to recover about 60 % of heat in a wet air stream having a 60 °C dew point. Recovered heat is to be upgraded as low pressure steam at 130 kPa and at a temperature of 117 °C. Water at 40 °C is used for the production of this utility steam.

Table 1 : Parameters considered for heat valorisation system

Wet air (waste heat)		**Outlet air (saturated)**		**Vapour produced (utility)**	
Flow	25 000 Nm^3/h	Temperature	47 °C	Pressure	130 kPa
Temperature	85 °C	Humidity	0.07 $kg_{water}/kg_{dry\ air}$	Superheat	10 °C
Pressure	101.3 kPa			Temperature	117.1 °C
Dew point	60 °C (0.15 $kg_{water}/kg_{dry\ air}$)				

Figure 1 shows the sequence proposed.
a) Direct-contact heat exchanger device for the recovery of heat from wet air
b) HP system comprising evaporator, compressor, and condenser and throttling valve
c) Steam compressor.

Heat from wet air is recovered as hot water in a direct-contact heat exchanger instead of feeding it directly to the evaporator of the HP system. The use of such auxiliary equipment allows a more versatile use of HP system as well as a better control. This type of exchanger allows good heat transfer coefficients and is already used in drying hoods for the production of hot water from hot wet air (Thermal energy international Inc, Newswire 2010). In such a system, outlet air is at a temperature closed to the temperature of the inlet water (temperature difference of 5 °C or below) (Sofame Technologies inc. 2009). As shown in the figure, hot water exiting the direct heat exchanger is directed to the evaporator (hot side) of the HP system. Excess water coming from condensation of the wet air stream is evacuated and can be reuse for another application.

In the evaporator, heat is transferred to an evaporating HP fluid or refrigerant (cold side of the evaporator). At the exit of the evaporator, vaporised HP fluid is compressed before being condensed in the condenser (hot side of the condenser). Liquid water maintained at moderate vacuum condition (50 kPa) is fed in the cold side of the condenser for the purpose of producing steam. The produced steam is then compressed at the desired pressure (130 kPa), and some water is added to the steam at the exit of the steam compressor, for adjustment of the superheating of the steam.

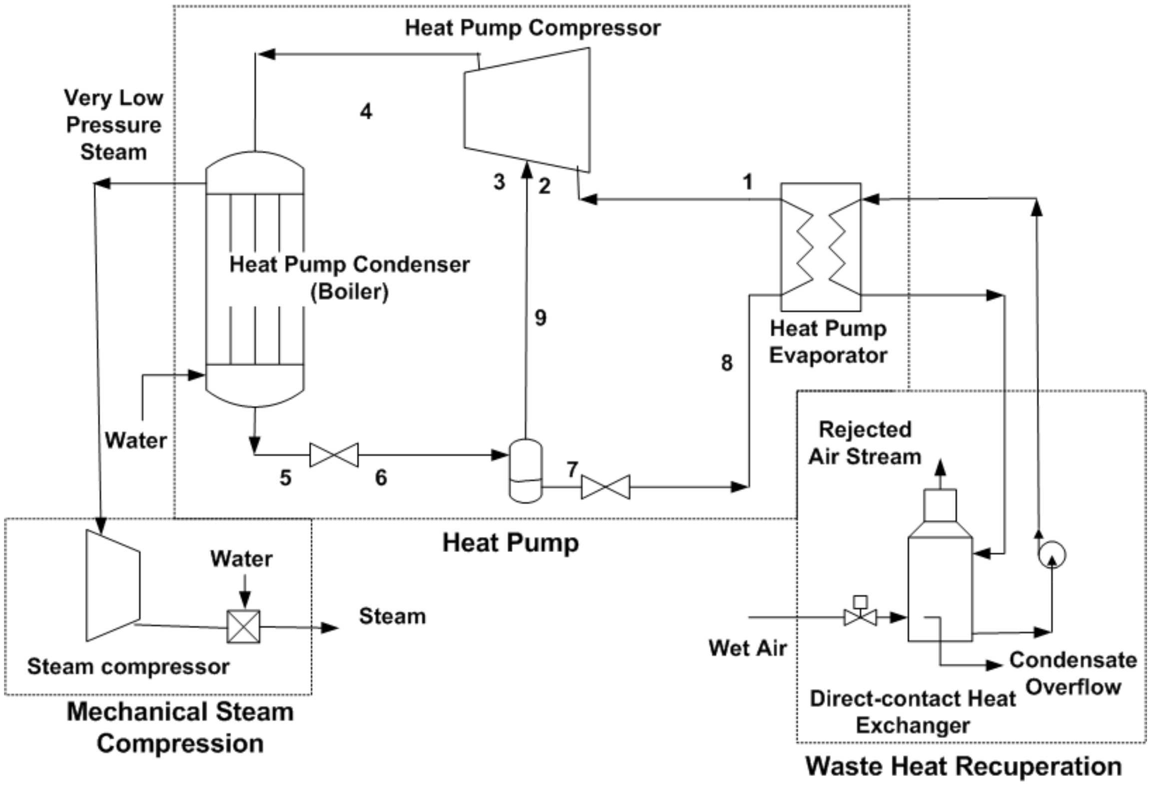


Figure 1: Schematic of system for the valorisation of heat from wet air from a drying process

In this study, heat pump fluids R-245fa and R-134a are considered. The fluid R-245fa, with a critical temperature of 154 °C presents interesting characteristics for high temperature applications although slightly toxic (ASHRAE Std 34 safety group B1, cannot be used for human comfort). The fluid R-134a, with a critical temperature of 101 °C, is a HFC traditionally used for medium temperature HP (generally with condensation below 80 °C). More recently developed fluids with low Global Warming Potential (GWP) index , R-1234ze and R-1234yf, were also modeled. In principle, HP using one of these fluids could easily be used to recover heat from a heat source of 40 °C or below for generation of heat at 80 °C.

One of the critical parts of HP is the compressor. Screw compressors appear to be the most appropriate for the application considered, because of their versatility regarding the span of flow rates and compression ratios. Furthermore, the presence of some liquid can be tolerated. Compressors without oil injection, used for steam compression, can support elevated temperature (225 °C) but are limited to compression ratio below 7.

2.2 Mass and Energy Balance

EES (Engineering Equation Solver from F-Chart Software) was used for a complete calculation of the mass and energy balance of such a system. Thermo physical properties required for the calculations are available and can be recovered by the software solver (properties for steam, R-245fa, for R-134a, for R-1234yf and for R-1234ze).

2.2.1 Waste Heat Recuperation

Direct-contact heat exchanger is simulated as a process involving direct contact of hot water and hot wet air with counter current flow. Condensed water is evacuated and the balance of the water outlet from the direct heat exchanger goes to the evaporator of the HP system. The temperature difference between dew point of the inlet air and outlet water temperature is specified as 5 °C (which is the inlet water of the hot side of the evaporator). Similarly, the temperature difference between the exit saturated air and inlet water temperature is specified as 5 °C (which is the outlet water of the hot side of the evaporator). The temperature of the outlet saturated air is specified to be 47 °C for the recovery of 60 % of the heat stream (reference at 25 °C). The flow rate of the water fed into the direct exchanger is obtained from the energy balance.

2.2.2 Heat Pump Evaporation and Condensation

The heat pump evaporator is assumed to be a plate-and-frame heat exchanger. The evaporation temperature is 5 °C less then the outlet water at the hot side of the evaporator. Flow rate of the HP fluid is calculated according to the heat transfer requirement for waste heat recuperation (1 736 kW) and a 10 °C of superheating.

The heat pump condenser is assumed to be a shell-and-tube heat exchanger. The cold side is fed with liquid water at 40 °C and at 50 kPa absolute pressure (which is vapour pressure at a saturation temperature of 81 °C) and its flow is adjusted for a temperature pinch of 7 °C.

The heat exchange surface, used to estimate heat exchangers cost, is calculated from the global heat transfer coefficient (U, in kW/m²°C) and the heat duty. The global heat transfer coefficient (U), respectively for the evaporator and the condenser, is calculated from:

$$\frac{1}{U_{Evap}} = \frac{1}{h_{fluid,evap}} + R''_{wall} + \frac{1}{h_{water}}, \tag{1}$$

$$\frac{1}{U_{Cond}} = \frac{1}{h_{fluid,cond}} + R''_{wall} + \frac{1}{h_{water\ boiling}}, \tag{2}$$

Where the heat transfer coefficients (h, in kW/m² °C) are obtained from correlation for plate and shell-and-tube heat exchangers. The calculations procedure is not detailed here but the correlation can be found in the literature (for fluid boiling in a plate heat exchanger ($h_{fluid,evap}$) : Ayub 2003; for water circulation in a plate heat exchanger (h_{Water}) : Kays and London 1984; for fluid condensation in an horizontal tube ($h_{fluid,cond}$): Dobson and Chato 1998 ; and for water boiling (nucleate) (h_{Water}): Rohsenow 1952). To take into account wall and surface fouling, the term $1/R''_{wall}$ is assumed to be approximately 7.5 kW/m² °C.

2.2.3 Heat Pump Compression

Heat pump compressor is used to compress fluid vapour up to a pressure corresponding to a saturation temperature close to 90 °C. The isentropic efficiency ($\eta_{isen,HP,}$) of the compression process is assumed to be 75 % and about 20% of the enthalpy increase is assumed to be dissipated to the environment through oil cooling and various heat losses ($f_{Heat\ Loss}$).The electrical efficiency ($\eta_{el,HP}$) is assumed to be 90%. As illustrated in Figure 1, 2 and 3, the heat pump uses an economizer cycle. The economizer port in the screw compressor is assumed be located where the volume ratio is 1.5. The point 2 is located just before the economizer port and the point 3 is located just after the port. The pressure drop of the stream passing through the economizer port is assumed to represent about 30% of the difference between pressure at point 6 and at point 2. Equations 3 to 8 were used to calculate the compressor power. The enthalpy (h, in kJ/kg) in the first step of the compression process is calculated using:

$$h_{2,no\ heat\ loss} = h_1 + \frac{(h_{2s} - h_1)}{\eta_{isen,HP}}, \tag{3}$$

$$h_2 = h_1 + (1 - f_{Heat\ Loss,HP})(h_{2,\ noheat\ loss} - h_1), \tag{4}$$

where h_{2s} is the enthalpy obtained from an isentropic compression process and $h_{2,no\ heat\ loss}$ is the enthalpy at point 2 without considering the heat losses. The fluid state after the economizer port (point 3) is calculated using:

$$u_3 \dot{m}_{fluid,3-4} = u_2 * \dot{m}_{fluid,1-2} + h_9 * (\dot{m}_{fluid,3-4} - \dot{m}_{fluid,1-2}), \tag{5}$$

$$P_3 = P_2 + (P_9 - P_2) * 0.7, \tag{6}$$

$$\dot{m}_{fluid,3-4} / \rho_3 = \dot{m}_{fluid,1-2} / \rho_2, \tag{7}$$

where u is the internal energy (in kJ/kg), $\dot{m}_{fluid,1-2}$ is the fluid mass flow rate (in kg/s) before the compressor port, $\dot{m}_{fluid,3-4}$ is the fluid mass flow rate after the port, ρ is the density (in kg/m^3) and P is the pressure. The enthalpy (h) in the second step of the compression process is obtained from equations analogous to equations 3 and 4. Finally, the compressor power is calculated using:

$$\dot{W}_{HP} = \frac{\dot{m}_{fluid,1-2}(h_{2,no\ heat\ loss} - h_1) + \dot{m}_{fluid,3-4}(h_{4,no\ heat\ loss} - h_3)}{\eta_{el,HP}}, \tag{8}$$

Expansion processes (5 to 6 and 7 to 8) are assumed to be isenthalpic expansion processes.

2.2.4 Steam compression

Steam at the outlet of the HP condenser is compressed from 50 to 130 kPa. The isentropic efficiency of the compression process is assumed to be 75 % and about 10% of the enthalpy increase is assumed to be dissipated to the environment. The electrical efficiency is assumed to be 90%. Liquid water at 40 °C is mixed with the pressurized water vapour in order to achieve 10 °C of superheat.

2.3 Economic evaluation

Economic evaluation of the system is essentially based on calculating the cost of the compressors and of the heat exchangers. For each of these equipments, the delivered equipment cost (C) is calculated and adjusted using:

$$C = C_{Base} f_{mat} \frac{CE_{Study}}{CE_{Equation}} f_{delivery}, \tag{9}$$

where C_{Base} is a calculated equipment cost and f_{mat} is a material factor. CE_{Study} is the Chemical Engineering (CE) index used for this study (CE_{Study} value is set to 600, approximately the CE index in 2012 and 2013) and $CE_{Equation}$ is CE index related to the empirical equation used for C_{Base}. The value of $f_{delivery}$, a factor used to obtain the delivered cost from the FOB cost, is assumed to be 1.1 (Silla 2003).

For the heat pump screw compressor, the following expression was used for C_{Base} (Walas 1988, p. 667):

$$C_{Base,HPCompressor} = 1490 * W_{HPComp}^{0.71}, \tag{10}$$

where $W_{HP\ Comp}$ is the heat pump compressor power (in HP). The CE index is 325. For the steam screw compressor (functioning in vacuum), the following expression was used for C_{Base} (Seider 2009, p. 595):

$$C_{Base,SteamComp} = 7840.5 * Flow_{Steam\ Comp}^{\ 0.38}, \tag{11}$$

where $Flow_{Steam\ Comp}$ is the actual flow at suction (in m^3/h) . The CE index is 500 and f_{mat} is 1. For the heat pump evaporator, a plate-and-frame heat exchanger, the following expression was used (Seider 2009, p. 592):

$$C_{Base,Evap} = 24\ 090 * A_{Evap}^{\ 0,42}, \tag{12}$$

where A_{Evap} is the heat-transfer area (in m^2). The CE index is 500 and f_{mat} is 1(Stainless steel).   For the heat pump condenser, a kettle shell-and-tube heat exchanger, the following expressions were used (Seider 2009, p. 571):

$$C_{Base,Cond} = \exp\left\{9.89757 - 0.8709(\ln(A_{Cond})) + 0.09005(\ln(10.7634 * A_{Cond}))^2\right\}, \tag{13}$$

$$f_{mat} = a + 1.18(A_{Cond}/100)^b, \tag{14}$$

where A_{Cond} is the heat-transfer area (in m^2). For Cr-Mo steel both sides, a is 1.7 and b is 0.07.The CE index is 500.

The cost for direct-contact heat exchanger was assumed to be 175 000 $ for the capacity of the system under study (Chabot 2012). When the wet air stream flow rate is modified, the following equation was used:

$$C = 175\ 000 * \left(\frac{Flow_{wet\ air}}{25\ 000}\right)^{0.7}, \tag{15}$$

where the flow rate of wet air is in normal cubic meters per hour.

The capital cost of the project is obtained using:

$$Capital\ Cost = \sum f_{DC}\, f_{IC}\, f_{CCF}\, C, \tag{16}$$

where f_{DC} is equipment direct installation cost factor (assumed to be 1.2 for compressors, 2 for the condenser, 1.5 for the evaporator and 1.5 for the direct-HX), f_{IC} is the indirect cost factor (assumed to be 1.34) and f_{CCF} is the construction and contractor fees factor (assumed to be 1.18) (Silla 2003).

Amortization cost was calculated for a period of 5 years. Income tax and interest were not considered in the economic calculations. The annual operation and maintenance cost is calculated assuming 5 % of the total capital cost. The electricity cost is calculated assuming a price of 0.05 $/kWh. In the next section, Table 2 and 3 summarize the economic evaluations procedure and the results for the case studied.

3 RESULTS AND DISCUSSION

3.1 Base Case Results

Figures 2 and 3 show detailed results from simulation calculations for a heat source with a dew point of 60 °C and the production of steam at 130 kPa and 117 °C (parameters in Table 1) respectively using HP fluid R-245fa and HP fluid R-134a.

In the first case, the system allows the production of 2.3 MW of heat as steam at 130 kPa and 117 °C (flow rate of 0.90 kg/s), with an electrical consumption of 734 kW for compressors. Global coefficient of performance (COP), the ratio of heat produces on electricity consumed, is 3.13. It is approximately 50 % of the ideal Carnot COP based on the saturation temperature of the steam produced and the temperature of the air exiting the direct-contact heat exchanger (47 °C). The heat pump itself presents a COP of 4.13 or 60 % of ideal Carnot COP (based on HP fluid evaporation and condensation temperatures). The heat pump compressor power is 513 kW for a pressure ratio of 4.45. Flow rate of the HP fluid R-245fa at the exit of the compressor is 14.5 kg/s, requiring a compressor capacity of about 3600 m^3/h (assuming a volumetric efficiency 80 %). Discharge temperature at the exit of the fluid compressor is 91 °C which is a moderate temperature. Energy consumption of the steam compressor (working at vacuum conditions at the inlet) is 221 kW for a compression ratio of 2.6. The steam compressor volumetric capacity is high, about 12 500 m^3/h. Globally, we can expect a good performance for such a system.

In the second case (Figure 3) where the HP fluid is R-134a, global coefficient of performance (COP) is 2.84 which is 9% lower then calculated global COP for the system using R-245fa. HP itself presents a COP of 3.6 or 52 % of ideal Carnot COP. Flow rate of the HP fluid is R-134a at the exit of the HP compressor is 19.9 kg/s, requiring a compressor capacity of about 1300 m^3/h (assuming a volumetric efficiency 80 %). The R-134a condensation pressure of 3200 kPa is high, but compressors that withstand this pressure are available.

Table 2 and 3 present the details of the economic evaluation. The cost of the heat produced is 0.053 \$ and 0.055 \$ per kWh_{th} for, respectively, R-245fa and R-134a (14.7 and 15.3 \$/GJ). This is equivalent to 10.3 and 10.7 \$/GJ of combustible (ex: natural gas) used at 70 % efficiency (0.39 and 0.41 \$/m^3). However, after the 5 year simple amortization period considered in this study, the price would drop by more than 50 % and be of 0.023 and 0.025 \$/$kWh_{th.}$ Considering the context of low natural gas prices, such a system is currently hardly economically justifiable for the base case presented in this study.

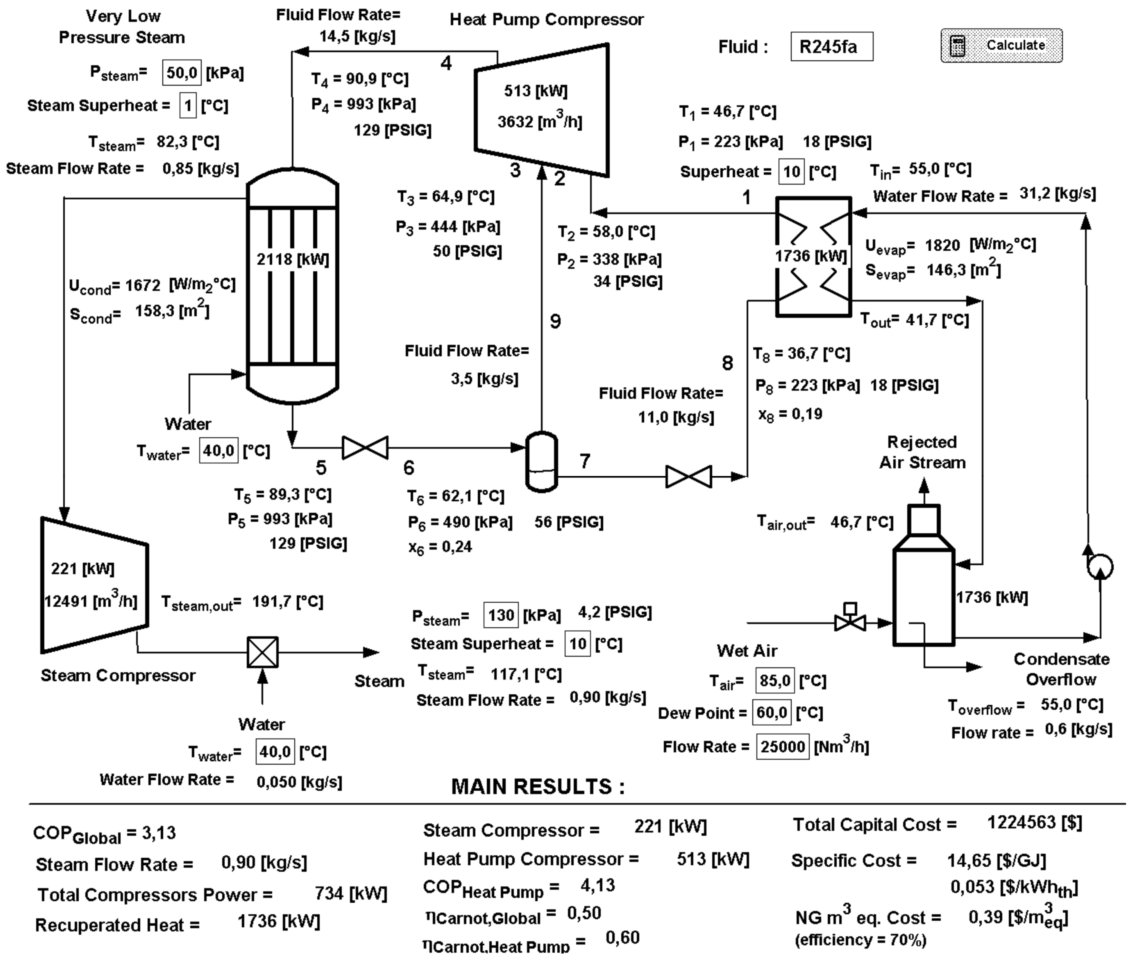


Figure 2: Schematic of system based on HP and mechanical steam compression with R-245fa for production of 2.3 MW of heat (steam at 130 kPa and 117 °C)

Table 2: Economic evaluation for a system based on HP and mechanical steam compression with R-245fa for production of 2.3 MW of heat (steam at 130 kPa and 117 °C)

	HP Comp.	Steam Comp.	Direct-HX	Evapora-tor	Conden-sor	Total
Key Parameter	513 kW	221 kW	1736 kW	146 m²	158 m²	
Delivered Equipment Cost ($)	312703	342 606	175 000	258 103	136 161	1 224 573
Installation Factor	1,2	1,2	1,5	1,5	2	
Indirect Cost Factor	1.34	1.34	1.34	1.34	1.34	
Contingency and Contractor Fees Factor	1.18	1.18	1.18	1.18	1.18	
Capital Cost ($)	593 335	650 075	415 065	612 168	430 595	2 701238
Operation Cost Factor	0.05	0.05	0.05	0.05	0.05	
Amortization Period (y)	5	5	5	5	5	
Utilisation (h/y)	8000	8000	8000	8000	8000	
Annual Amortization Cost ($/y)	118 667	130 015	83 013	122 434	86 119	540 248
Annual Electricity Cost ($/y)	205 031	88 409				293 440
Annual Operation Cost ($/y)	29 667	32 504	20 753	30 608	21 530	135 062
Total Annual Cost ($/y)	353 365	250 927	103 766	153 042	107 649	968 750
Specific.Cost ($/kWh$_{th}$)	0.0192	0.0137	0.0056	0.0083	0.0059	0.053
Spec. Cost ($/GJ)	5.34	3.79	1.57	2.31	1.63	14.65
Spec. Cost (GJ eff. 70%)						10.25
Spec. Cost ($/NG m³ eff. 70%)						0.39

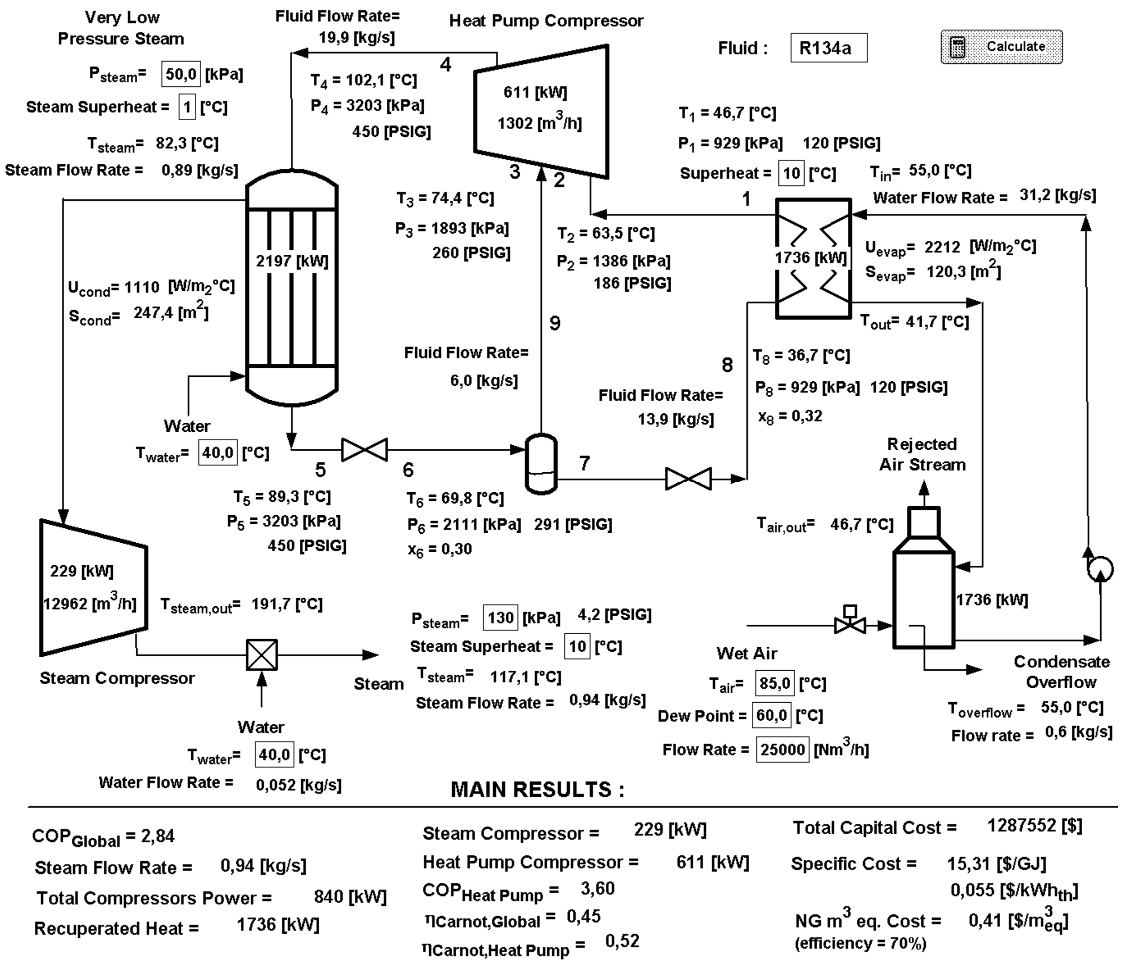


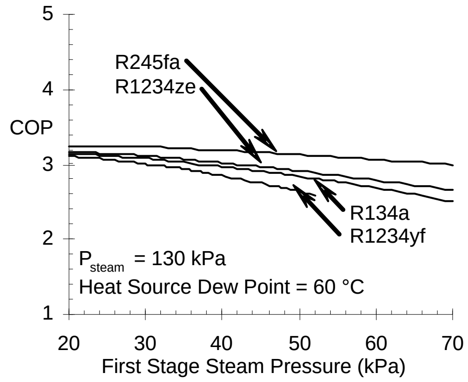
Figure 3: Schematic of system based on HP and mechanical steam compression with R-134a for production of 2.4 MW of heat (steam at 130 kPa and 117 °C)

Table 3: Economic evaluation for a system based on HP and mechanical steam compression with R-134a for production of 2.4 MW of heat (steam at 130 kPa and 117 °C)

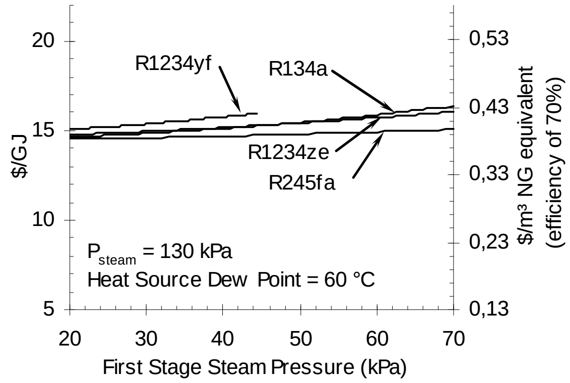
	HP Comp.	Steam Comp.	Direct-HX	Evaporator	Condensor	Total
Key Parameter	611 kW	229 kW	1736 kW	120 m²	247 m²	
Delivered Equipment Cost ($)	354 132	347 461	175 000	237 783	173 177	1 287 553
Installation Factor	1.2	1.2	1.5	1.5	2.0	
Indirect Cost Factor	1.34	1.34	1.34	1.34	1.34	
Contingency and Contractor Fees Factor	1.18	1.18	1.18	1.18	1.18	
Capital Cost ($)	671 944	659 286	415 065	563 974	547 655	2 857 925
Operation Cost Factor	0.05	0.05	0.05	0.05	0.05	
Amortization Period (y)	5	5	5	5	5	
Utilisation (h/y)	8000	8000	8000	8000	8000	
Annual Amortization Cost ($/y)	134 389	131 857	83 013	112 795	109 531	571 585
Annual Electricity Cost ($/y)	244 300	91 744				336 044
Annual Operation Cost ($/y)	33 597	32 964	20 753	28 199	27 383	142 896
Total Annual Cost ($/y)	412 286	256 565	103 766	140 993	136 914	1 050 525
Specific.Cost ($/kWh$_{th}$)	0.0216	0.0135	0.0054	0.0074	0.0072	0.055
Spec. Cost ($/GJ)	6.01	3.74	1.51	2.05	2.00	15.31
Spec. Cost (GJ eff. 70%)						10.71
Spec. Cost ($/NG m³ eff. 70%)						0.41

3.2 Other parameters

The fist stage steam pressure of the system presented above was set to 50 kPa (82 °C). Figure 4 shows the impact of this parameter on the system efficiency and economics for various HP fluids. The repartition of the load on either the HP or the steam compressor does not affect significantly the efficiency or the cost of the system for R-245fa. Technical considerations on the available equipments (size, pressure ratio, max. pressure, resistance to vacuum, etc.) should determine the final choice. R-245fa shows the best performances, but the other fluids (R-134a or low GWP index fluids such as R-1234ze and R-1234yf) constitute valid options.



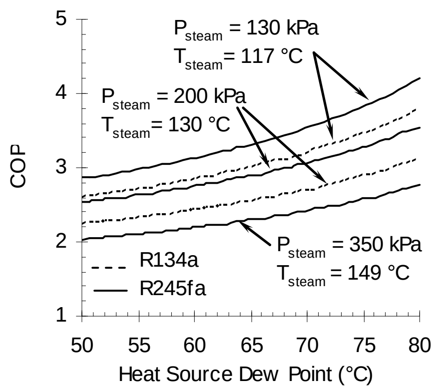
a.



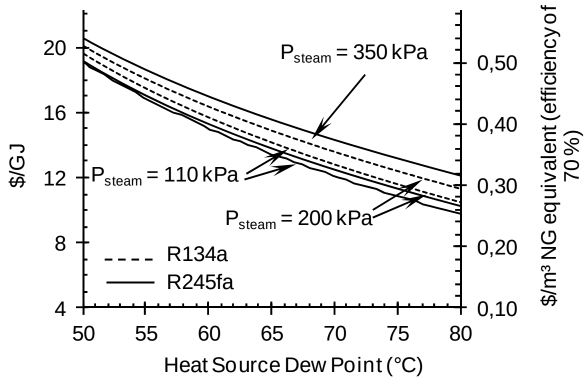
b.

Figure 4: Impact of First Stage Steam Pressure on Efficiency (a) and Economics (b)

Figure 5 shows the impact of heat source dew point on the production of various steam pressures. The specific cost of the 350 kPa steam is only about 15 % higher than for 130 kPa steam. However, the wet air stream dew points have a significant impact on the system efficiency and steam production cost. A higher dew point does not only provide a higher temperature, but a significantly higher humidity ratio (i.e. energy content). The specific cost of 130 kPa steam decreases by 28 % with a heat source dew point at 75°C instead of 60°C (10.6 $/GJ instead of 14.7 $/GJ) and the COP rises from 3.1 to 3.8. If hot water is used directly as the heat source, the cost would decrease by 10% and it should be considered 5 °C higher to fit with Figure 5.



a.



b.

Figure 5: Impact of Heat Source Dew Point on Efficiency (a) and Economics (b)

As shown in Figure 6a, the capacity and the utilisation have a significant impact on the specific cost of the steam produced. The wet air stream flow rate should not be much lower than 25 000 Nm³/h, the flow rate used in this study. Ideally, the steam production process should also be continuous (8000 hours per year). Figure 6b presents the impact of electricity rates and amortization period on the specific cost of 130 kPa steam. Steam specific cost is not much affected by electricity rate fluctuations. When the amortization period is 10 years instead of 5 and the air stream dew point is 75 °C, the system is more economically interesting with a specific cost of less than 8 $/GJ (0.21 $/m³, natural gas at 70 % efficiency).

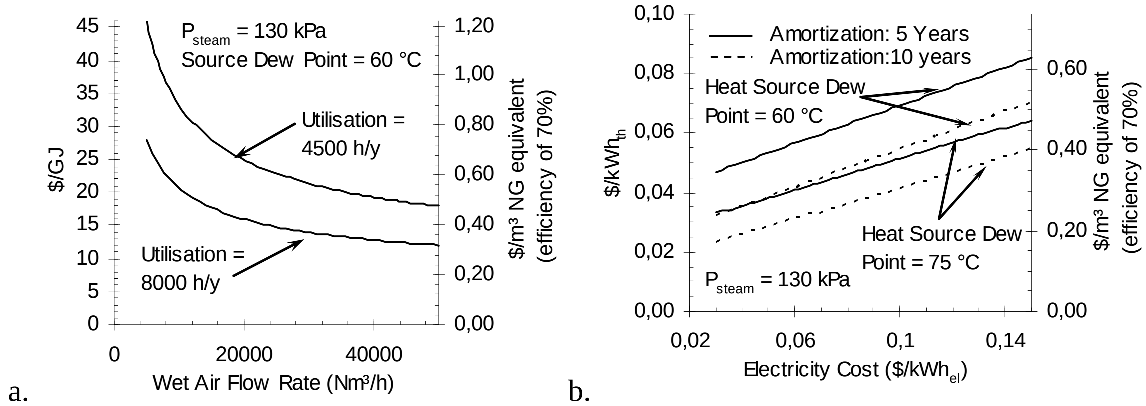


Figure 6: Impact of Wet Air Flow Rate and Utilisation (a) and Electricity Cost (b)

4 CONCLUSIONS

This preliminary techno-economic evaluation suggests that 130 kPa steam (117°C) can be produced from a wet air stream (dew point 60°C) by the combined use of a mechanical heat pump and mechanical steam compression with a COP higher than three and at a cost of 15 $/GJ. This is equivalent to 10.5 $/GJ of combustible (ex: natural gas) used at 70 % efficiency (0.40 $/m^3). After the 5 year simple amortization period, the price would decrease by more than 50 %.The study suggests that the steam pressure could be raised to 350 kPa without much affecting the steam specific cost (15% raise). The waste wet air stream dew point has a significant impact on the system efficiency and economics: at 75°C instead of 60 °C , the COP rises to 3.8 and the specific cost decreases to 10.5 $/GJ. When an amortization period of 10 years instead of 5 is used, the specific cost decreases to 8 $/GJ (equivalent to 0.21 $/m^3 natural gas at 70 % efficiency). Steam cost is not much affected by electricity rate fluctuations. The capacity and the utilisation have a significant impact on steam cost: the wet air stream flow rate should ideally be higher than 20 000 Nm^3/h and the steam production process should be continuous (8000 hours per year). The authors would like to underline that, given the lack of precision of such techno-economic evaluation, the data presented here should only be interpreted as rough approximations and tendencies. For a better cost estimate, a study with actual cases and actual costs for specific project is necessary.

Although a sequence combining a HP and mechanical steam compression for the production of low pressure steam as a hot utility from waste wet air stream is currently hardly economically justifiable in a context of low natural gas prices, it could eventually be beneficial, especially where renewable energy is available and where the price of electricity compares to fuel prices.

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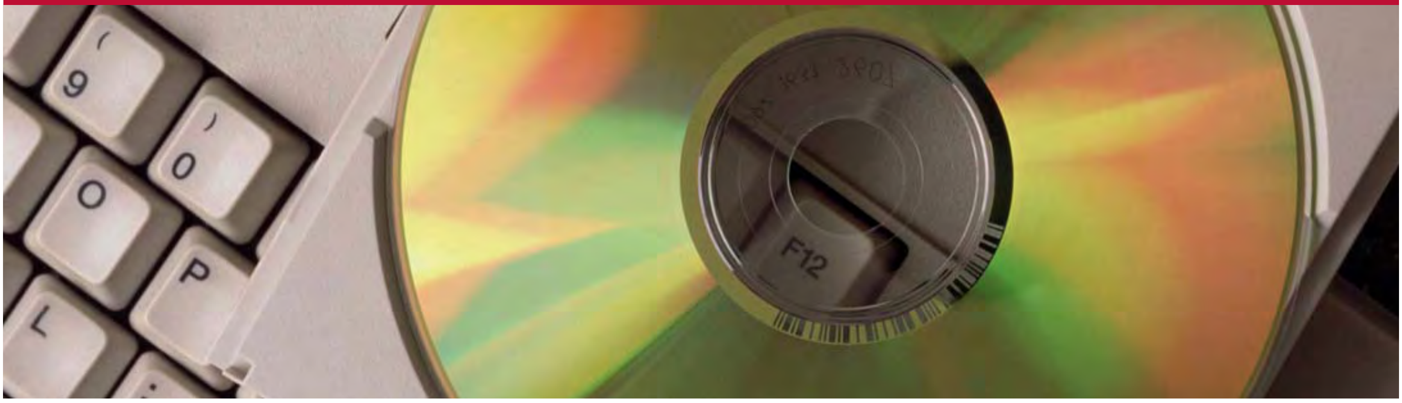
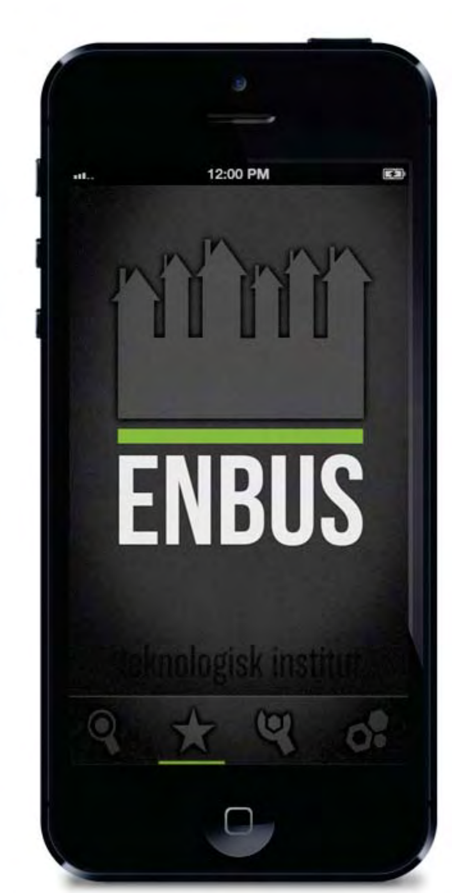


Reduce energy consumption in the building sector – with an app

Construction, maintenance and operation of buildings are some of Europe's most energy-consuming activities since about 40 % of the total energy use corresponds to buildings. The new ENBUS App makes it easier for users to find products that can reduce energy consumption in the construction and building sector.



The ENBUS App makes it possible to search through products in different product groups, for example windows, insulation, ventilation and heating. It gives a good and fast overview of products in a specific group and it is also possible to narrow the search within the product group. A normal Internet search will only give links to different webpages from distributors and it will take a lot of time to make a comparable list. This App will give you the list on the fly and include search parameters.

The ENBUS App also gives a detailed guideline on what to consider when you are thinking of renovating your house or flat. There is furthermore a thorough explanation of the different search parameters under each product category.

The App calculates the potential energy saving if changing for example the windows of the house. It is possible to choose between three different "standardized" house types (detached house, row house and flat) and two locations (Copenhagen in Denmark and Munich in Germany). The energy savings are made with the calculation programs ASEPI and Energy Plus.

This App has been made as part of the EU project "ENBUS – Energising the building sector". It has been made as an example on how to spread awareness, motivation and information among house owners that have thought about renovating their house. The database in this App is not so large to begin with, but can be increased to thousands of products in the future.

The ENBUS App can be downloaded for free at App Store or by the QR-code below. If you don't have an iPhone, you can instead use the interactive website here: *http://enbus.eu/gmp*

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2015

12 – 14 January
5th Jordanian IIR Conference on Refrigeration and Air Conditiong
Aqaba, Jordan
http://jnciirconf.ju.edu.jo/Home.aspx

24 – 28 January
ASHRAE Winter Conference
Chicago, USA
http://ashraem.confex.com/ashraem/w15/cfp.cgi

3 – 5 February
ATMOsphere Asia 2015
Tokyo, Japan
http://www.atmo.org/events.details.php?eventid=27

24 – 27 February
Climatizacion
Madrid, Spain
http://www.ifema.es/ferias/climatizacion/default_i.html

26 – 28 February
ACREX 2015
Bangalore, India
http://www.acrex.in/

10 – 14 March
ISH
Frankfurt, Germany
http://ish.messefrankfurt.com/frankfurt/en/aussteller/willkommen.html

16 – 17 March
ATMOsphere Europe 2015
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16 – 18 April
6th IIR Ammonia and CO_2 Refrigeration Conference
Ohrid, Republic of Macedonia
http://www.r744.com/events/view/615

19 – 24 April
World Geothermal Congress
Melbourne, Australia
http://wgc2015.com.au/index.php

6 – 9 May
Advanced HVAC and Natural Gas Technologies 2015
Riga, Latvia
http://www.hvacriga2015.eu/

19 – 21 May
13th IEA Energy Conservation through Energy Storage Greenstock Conference 2015
Beijing, China
http://iea-eces.org/eces/event-calendar.html , scroll to May 2015

1 June – 6 June
eceee 2015 Summer Study on energy efficiency
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25 June – 26 June
ATMOsphere America 2015
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27 June – 1 July
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16 – 22 August
ICR 2015 – The 24th IIR International Congress of Refrigeration
Yokohama, Japan
http://www.icr2015.org/

20 – 21 October
Heat Pump Summit
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http://www.hp-summit.de/en/

20 – 23 October
8th International Conference on Cold Climate-Heating, Ventilation and Air-Conditioning (Cold Climate HVAC 2015)
Dalian, China
http://www.coldclimate2015.org/

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23 – 27 January
ASHRAE Winter Conference
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7 – 9 April
4th IIR Conference on Sustainability and the Cold Chain
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http://www.iifiir.org/medias/medias.aspx?INSTANCE=exploitation&PORTAL_ID=general_portal.xml&SETLANGUAGE=EN

21 – 24 August
Gustav Lorentzen Natural Working Fluids Conference
Edinburgh, Scotland
http://www.ior.org.uk/GL2016

11 – 13 October
Chillventa
Nuremberg, Germany
http://www.chillventa.de/en/

In the next Issue

Quality installation / Quality maintenance
Volume 33 - No. 1/2015

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster co-operation among its participating countries, to increase energy security through energy conservation, development of alternative energy sources, new energy technology and research and development.

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The Programme strives to achieve widespread deployment of appropriate high quality heat pumping technologies to obtain energy conservation and environmental benefits from these technologies. It serves policy makers, national and international energy and environmental agencies, utilities, manufacturers, designers and researchers.

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