heat pump centre

IEA HEAT PUMP CENTRE

NEWSLETTER
VOL. 29
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Heat Pumps - The Solution for a Low-Carbon World

10th IEA Heat Pump Conference 2011 - a website conference

Selected articles from the Heat Pump Conference

Rittinger Award

Market report: South Korea

IEA **Heat Pump** CENTRE
NEWSLETTER

Volume 29 - No. 3/2011

In this issue

Heat Pump Centre Newsletter, 3/2011

This issue is dedicated to the recently held 10th IEA Heat Pump Website Conference. Each of the three regional coordinators has selected two papers for publication in the HPC Newsletter; we gratefully acknowledge the permission to reproduce them here. They are also available on the 2011 Heat Pump Conference Proceedings (available via the HPC website), as are a large number of other interesting articles.

Further in this issue, the 2011 Rittinger awardees are presented.
In addition, we get a presentation of the heat pump market in South Korea.

Enjoy your reading!

Johan Berg
Editor

Heat pump news

Regulars

Topical articles

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Published by IEA Heat Pump Centre
Box 857, SE-501 15 Borås,Sweden
Phone: +46 10 516 55 12
Fax: +46 33 13 19 79

Publisher:
IEA Heat Pump Centre
PO Box 857, S-501 15 BORAS
SWEDEN
Tel: +46-10-516 50 00, Fax: +46-33-13 19 79
E-mail: hpc@heatpumpcentre.org
Internet: http://www.heatpumpcentre.org

Editor in chief: Monica Axell
Technical editors: Johan Berg, Roger Nordman, Ulf Mårtensson - IEA Heat Pump Centre
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IEA OECD heat pump centre

Foreword

The 10th International IEA Heat Pump conference 2011

Thomas Kopp
Chair, International Organizing Committee

Dear heat pump friends,
The 10th International IEA Heat Pump conference 2011 in Tokyo, Japan, was closed at the end of August 2011. We had the pleasure of following a very interesting conference in an innovative website format. This was something of an experiment. Yet we all know that the organizing committees IOC and NOC Japan were forced to choose this solution because of the tragic consequences from the tsunami of March 11.

The honorable effort of NOC Japan resulted in very professional and excellent Conference Proceedings and a widely visited web conference. 191 papers, 14 keynote presentations, 7 address speeches and 7 reports on ongoing annexes were presented. From the opening of the website conference until closing at the end of August, 377 participants visited the website 1665 times and studied 18108 pages. As initially planned, the conference consisted of 9 sessions. Each was to offer a keynote presentation, several oral and many poster presentations. Unfortunately, only some of the oral presentation authors took the chance to post a video-speech on the conference website. But all initially planned oral presentations can be studied in the form of papers in the very interesting conference proceedings and were presented in power point format in the website conference. Due to many abstract proposals after launching the first announcement at the end of 2009, both the «System and Components» and «Applications» subjects had to be presented in two sessions:

1. Opening session with welcome addresses from IOC, NOC, NEDO, IEA, AHRI, EHPA, JSRAE and reports from the Regions North and South America, Europe and Africa and Asia and Oceania as well as a report of the Japanese platinum network
2. Heat Pumps for a sustainable society (policy and market), 3 keynotes, 16 papers
3. Systems and Components I, 1 keynote, 39 papers
4. Ground Source Heat Pumps, 1 keynote, 30 papers
5. Applications I, 1 keynote, 30 papers
6. Systems and Components II, 1 keynote, 30 papers
7. Air Source Heat Pumps, 1 keynote, 22 papers
8. Applications II, 1 keynote, 24 papers
9. Heat Pump Programme – Reports from ongoing Activities, 1 keynote and 7 papers

Based on all these contributions, I am certain that also this 10th International Heat Pump Conference in Tokyo was a great success and the important role of heat pumps could undoubtedly be proven. Also in terms of technical improvement, many possibilities and new equipment were presented and new ideas show even further steps to reach higher efficiency and lower cost. I'm

sure that you will read all related articles in this newsletter even if you didn't have the chance to attend the website conference.
Naturally, I would have preferred to close the conference being physically present at the initially planned Venue 'Chinzan-so' in Tokyo and saying 'Good bye' to all attendees. But unfortunately this was not possible this time. Of course, we could think of increasing the number of web-conferences. This is really a very environment-friendly way of exchanging information. There are definitely no CO_2 emissions by travel and no expenses for accommodation and food. However, and this is the sad aspect, neither are there any meetings, any personal discussions, any interesting excursions and lovely conference dinners. We will have to weigh up both sides of the argument, but when I think back to all my impressions from different conferences, I honestly think that meeting and discussing together is more fascinating!
I would like to finish this foreword with a great thank you to all individuals and bodies who contributed to the 10th IEA Heat Pump conference with their tremendous work. First I wish to present my deepest thanks to the National Organizing Committee of Japan with its chairman Momoki Katakura for the logistical preparation under these difficult conditions. I would also like to thank all partner organizations and sponsors as well as the international Country Sponsors of the IEA Heat Pump Programme. I also extend my thanks to my colleagues of the International Organizing Committee who were responsible for the conference programme.
I do hope that we can personally meet each other in one of the next IEA Heat Pump Conferences, as the planning for the next conference in this series will soon be under way.

Report from the 10th IEA Heat Pump Website Conference

Makoto Tono
Secretary, National Organising Committee

On December 22, 2008 when Christmas was just around the corner, we received a notification from the HPP Chairwoman saying "Japan has been selected as the next host country for the 10th IEA Heat Pump Conference 2011." That was literally a Christmas present for us. This is the second time that Japan hosts the IEA Heat Pump Conference, since hosting the 3rd Conference in 1990 for the first time. Both international and national organising committees for the conference have been established accordingly. We held the 1st International Organising Committee (IOC) meeting at Amersfoort, The Netherlands in May, 2009, since when a total of five IOC meetings have been held. The IOC is mainly responsible for conference programmes, while the National Organising Committee (NOC) is responsible for all on-site arrangements. The first NOC meeting was held in September 2009, and a total of 24 NOC meetings has been held.

Year 2009
The NOC selected Chinzan-so as the conference venue in September and opened the conference website and started the "call for papers" through the conference website in December, 2009. "New refrigerants" was picked out as one of the abstracts' topics from which authors chose. 7000 copies of first fliers were distributed all over the world for promotion of call for papers. Three Regional Coordinators (RCs) - Mr. Gerald Groff for North and South America region, Ms. Monica Axell for the European and African region, and Mr. Makoto Tono for the Asia and Oceania region - were responsible for abstracts. Abstracts were collected online through the conference website, which was the first time that this had been done for the IEA heat pump conference series. In addition, we established an online abstract management system, which enabled us to share on-going information about abstracts.
The subtitle of this conference is "The Solution for a Low-Carbon World", which reflects our expectation that heat pumps can contribute a great deal to global decarbonisation.

Year 2010
The deadline for abstract submission was July 31, 2010. Thanks to dedicated promotional activities of the three RCs, we collected 302 abstracts from 27 countries or regions. Online participation and full paper registrations started on October 1, 2010. Early-bird rate was set at 50 000 yen per person, expecting many participants during the worldwide recession. Second fliers were sent out to promote participation in December.

Year 2011
The year of the conference finally arrived. We had obtained industrial sponsors by then. A programme committee was in the final stage of deciding the conference programme structure. " Air source heat pumps" was adopted as one of ten session titles, and a committee completed paper classification into oral or poster presentations af-

Column continued

ter intense discussion by email exchanges. All keynote and address speakers were selected too.

When preparation of the 10th IEA Heat Pump Conference was at full speed, the Great Eastern Japan Earthquake of massive M9 force, and its devastating tsunami, struck Japan on March 11, 2011.

Web Conference

The organising committees of the conference had no choice but to give up arranging for participants to meet at the venue, Chinzan-so, Tokyo, Japan. Instead, the organisers decided to hold a virtual website conference from June 27 to August 31, 2011 in order to publish many collected papers. The registration period was extended in accordance with the postponement of the conference dates, and we finally collected 212 papers from 22 countries or regions and 377 participants from 26 countries or regions. This form of conference, via a website, was now used for the first time for the IEA Heat Pump Conferences. For authors, paper submission was obligatory, with power point presentations, voice and video messages on a voluntary basis. Participants were able to get access to the website conference with their ID and password at any time, and wherever they were, during the conference period as long as they were online. The conference website used a Q & A function, as an alternative to Q & A sessions after each oral presentation or during poster sessions at the real conferences. Total visit numbers to the website exceeded 1600, so we believe that many participants took part in the website conference. Although the form of the conference was forced to change to a virtual one, we were able to mark the 10th conference, in the same way as the past nine “real” conferences, and we are happy about it. The venue is not yet decided, but we are looking forward to meeting you at the next, 11th, IEA Heat Pump Conference in 2014. No doubt we will see further dissemination and deployment of heat pumps.

Acknowledgements

Last but not least, the conference organisers would like to extend their sincere appreciation to both the IOC and NOC chairmen, the three RCs, country and industrial sponsors, paper authors, participants and all the other individual and corporate contributions. At the same time, the NOC would like to express its deep gratitude for your words of sympathy, support and encouragement for the devastating casualties and damage caused by the twin disasters that Japan experienced on March 11.

General

Residential power use declining

U.S. homes are more cluttered than ever with electricity-consuming devices, particularly consumer and mobile electronics. However, electricity demand is leveling off. From 1980 to 2000, residential power demand grew by about 2.5% per year. From 2000 to 2010, the growth rate slowed to 2%. Over the next 10 years, demand is expected to decline by about 0.5% a year, according to the Electric Power Research Institute, a nonprofit group funded by the utility industry. The group attributes the decline to new homes that are being built to use less electricity, and government subsidies for home energy savings programs that are helping older homes use less power.

http://serbiagbc.org/power-use-declines-as-numbers-of-gadgets-rise/?lang=en
Source: The HVAC&R Industry for September 22, 2011

Changes to fan efficiency requirements proposed for ASHRAE, IES energy standard

A proposed change to ASHRAE and IES's energy standard would encourage advancement of efficient fan design.

Proposed addendum *u* to ASNI/ASHRAE/IES Standard 90.1-2010, Energy Standard for Buildings Except Low-Rise Residential Buildings, would add a reference to the Air Movement and Control Association (AMCA) International's Standard 205-10, Energy Efficiency Classification for Fans. This would require fan efficiency requirements to be classified based on fan efficiency grades.

http://www.ashrae.org/pressroom/detail/changes-to-fan-efficiency-requirements

Do clothes make the man hotter or cooler? Role of fashion in thermal comfort studied by ASHRAE

The role of international fashions in determining how cool or hot we are is being studied by ASHRAE. It's not the impact of Gucci or Chanel on our style but rather how non-western wear, such as burqas or saris, affects our thermal comfort.

Comprehensive data exists on western clothing insulation values but little research exists on non-western. Having information on attire like saris could influence the design of ventilation and air-conditioning systems to provide the best thermal comfort for occupants.

http://www.ashrae.org/pressroom/detail/do-clothes-make-the-man-hotter-or-cooler/

FAA funds geothermal system at Portland Airport

The Portland International Jetport will include a $3 million geothermal heating and cooling system as part of a major expansion project. The geothermal system is expected to save the 12 700 m^2 terminal more than 200 000 litres of oil every year. That equates to savings of more than $200,000 per year and more than $8 million over the system's life. The system is largely funded by a $2.5 million grant from the FAA's Voluntary Airport Low Emission program (VALE). The program has funded 42 low-emission projects at 22 airports over the last five years.

http://www.pressherald.com/business/new-energy-at-jetport_2011-04-30.html

Policy

Department of Energy releases 2011 Strategic Plan

The U.S. Department of Energy (DOE) has released its 2011 Strategic Plan, a comprehensive blueprint to guide the agency's core mission of ensuring America's security and prosperity by addressing its energy, environmental, and nuclear challenges through transformative science and technology solutions.

The DOE Strategic Plan is organized into four distinct categories, representing the broad cross-cutting and collaborative efforts taking place across the department's headquarters, site offices and national laboratories. It also covers four objectives.

1. Enforce the standards [the DOE has] in place.
2. Review minimum appliance efficiency standards at least every five years.
3. Develop standards and processes to address the entire spectrum of energy intensities for a given product class, not just the least (or most) efficient limits.
4. Leverage precompetitive research and development to understand the potential and limits for new technologies to improve building, appliance, vehicle, and industrial efficiency while providing economic benefits.

http://www.achrnews.com/articles/department-of-energy-releases-2011-strategic-plan

Finance for energy efficient equipment

The UK Carbon Trust and Siemens Financial Services have launched a new scheme to enable UK manufacturers to invest in a range of energy efficient technologies including air conditioning and refrigeration equipment.

Research carried out by the Carbon Trust and Siemens has shown that the estimated market for energy-efficient equipment finance over the next three years in the manufacturing sector is £4.6 billion.

The new scheme is designed to

"boost green growth and unlock business investment in the low carbon economy.
http://www.acr-news.com/news/news.asp?id=2531&title=Finance+for+energy+efficient+equipment

ASHRAE urges congress to continue funding for important building data survey

The recent announcements regarding the U.S. Energy Information Administration's (EIA) decision to not release the results of the 2007 Commercial Buildings Energy Consumption Survey (CBECS), and to halt work on the 2011 edition of the Survey, have prompted ASHRAE to request action.
EIA has opted not to release the 2007 CBECS results— a national sample survey that collects information on the stock of U.S. commercial buildings, their energy-related characteristics, energy consumption and expenditures—and has suspended work on a 2011 Survey due to statistical issues and funding cuts, respectively.
ASHRAE has issued a letter strongly urging Congress to include funding for CBECS in the Fiscal Year 2012 appropriations bills to allow work on the 2011 edition of the Survey to continue. This is particularly important in light of the 2007 CBECS data discrepancies.
http://www.ashrae.org/pressroom/detail/ashrae-urges-congress-to-continue-funding/

CECED: The Energy Efficiency Directive - a missed opportunity

The European Commission's proposal for an Energy Efficiency Directive represents a serious setback to the EU's stated commitments to achieve energy savings according to CECED.
The Coalition for Energy Savings calls on the European Parliament and Member States not to give up on the 20% energy savings commitment by 2020. The draft Directive must be substantially strengthened if Europe is to realise the huge potential of energy efficiency for the climate, the economy, and society. The lack of a clear process for the planned review in 2014 is also worrying to the Coalition, since it may mean that the decision on the introduction of binding targets could be deferred until the next European Commission.
http://www.ceced.org

Working Fluids

The Environmental Investigation Agency calls on the UN to end HFC-23 credits

The Environmental Investigation Agency (EIA) is calling on the UN to end emission offset credits for destroying the high GWP gas HFC-23, a by-product in the manufacture of the refrigerant R22
On the eve of an expected decision by the UN's Clean Development Mechanism (CDM) on HFC-23 methodology, EIA is calling for the methodology to be retired and current contracts for HFC-23 destruction not renewed.
Earlier this year Europe voted to ban the credits after it was widely recognised that the system perversely encouraged developing countries to over-produce R22 in order to cash in on credits available for destroying the high GWP by-product HFC-23.
Since 2005, Kyoto signatories are said to have spent several billion euros to obtain about 260 million carbon credits for offsets resulting from HFC-23 destruction projects.
http://www.acr-news.com/news/news.asp?id=2486&title=EIA+calls+on+the+UN+to+end+HFC23+credits

Over 50% of companies are trading illegally

F-Gas Support, the body set up in UK by Defra to provide guidance for companies and individuals in achieving F-gas certification, has confirmed reports that over 50% of UK air conditioning and refrigeration companies are now trading illegally.
AREA president and ACR News blogger Graeme Fox first highlighted the possibility that on the eve of the July 4 deadline only 50% of companies had upgraded to the mandatory full certification leaving around 2,500 companies trading illegally.
F-Gas support today confirmed that this was indeed the case and that it would now be working with its certification bodies to identify and follow up organisations that previously held an interim certificate and should now hold a full certificate.
http://www.acr-news.com/news/news.asp?id=2525&title=Over+50%25+of+companies+are+trading+illegally

Australia to set a carbon price for HFCs

The Australian Government has announced plans for tackling climate change which include a carbon price being fixed to HFC refrigerants which they say will be implemented through existing legislation on synthetic greenhouse gases.
The Australian Prime Minister Julia Gillard, Deputy Prime Minister and Treasurer Wayne Swan, and Minister for Climate Change and Energy Efficiency Greg Combet announced the Government's 'Securing a clean energy future' plan on 10 July.
The detailed plan explains how the country will aim to cut 159 million tonnes a year of carbon pollution from the atmosphere by 2020, which incorporates a carbon pricing mechanism which it is hoped will encourage investment in renewable energy like wind and solar power, and the use of cleaner fuels.
Under the plan, carbon will be priced at $23 AUD (about €16.9) per tonne from 1 July 2012, and will rise by 2.5 per cent each year during a three-year fixed price period until 1 July 2015. The price mechanism will then transition to an emissions trading scheme, where the price will be determined by the market.
http://www.acr-news.com/news/news.asp?id=2536&title=Australia+to+set+a+carbon+price+for+HFCs

AHRI to evaluate low GWP refrigerants

AHRI, the US Air-Conditioning, Heating, and Refrigeration Institute, has launched a research programme to identify and evaluate the

application of low GWP alternative refrigerants.
The programme aims to assess the industry's response to environmental challenges raised by the use of high GWP refrigerants and to assist industry research to avoid a duplication of work.
According to Karim Amrane, AHRI vice president of regulatory affairs and research: "The intent of the programme is to help industry select the most promising refrigerants, understand technical challenges, and identify the research needed to use these refrigerants. The programme will not prioritise these alternatives; rather, it will identify potential refrigerants replacements for high GWP HFCs, and present the performance of these replacements in a consistent and standard manner."
http://www.acr-news.com/news/news.asp?id=2424&title=AHRI+to+evaluate+low+GWP+refrigerants

Conference calls for more natural refriger-ants training

A major European conference on renewable energy has repeated calls for more engineers to be trained in the use of CO_2 in refrigeration systems.
Marco Buoni, vice president of the Air Conditioning and Refrigeration European Association (AREA), speaking at the 14th European Conference on Renewable Energy and Heating in Milan called for more training of technicians and other staff handling natural refrigerants, not just HFCs.
http://www.acr-news.com/news/news.asp?id=2500&title=Conference+calls+for+more+natural+refrigerants+training

Honeywell to invest $33M in production of HFO-1234ze

Honeywell has announced plans to invest $33M in the production of HFO-1234ze at its Baton Rouge, Louisiana, manufacturing facility.
The new low-GWP gas is already finding applications in foam blowing and aerosols but is also being considered to replace HFC-134a for large stationary refrigeration and air conditioning applications. Production is scheduled to begin in late 2013.
http://www.acr-news.com/news/news.asp?id=2527&title=Honeywell+to+invest+%2433m+in+production+of+HFO%2D1234ze

US looks set to follow EU ban on R134a in car AC

The US looks set to follow the EU and ban the use of R134a in car air conditioning systems. The US EPA has agreed to grant a petition filed by a trio of environmental groups to withdraw the agency's approval for the use of the gas which will be followed by a formal "notice and comment" rulemaking to set the phase-out schedule. This could take six months or more.
The decision comes after the new HFO refrigerant 1234yf - the preferred replacement option in Europe - was granted final US approval at the end of last month.
The Natural Resources Defence Council, a US-based environmental action group, took the lead on the original petition, and was joined by the Institute for Governance & Sustainable Development and the Environmental Investigation Agency.
http://www.acr-news.com/news/news.asp?id=2423&title=US+looks+set+to+follow+EU+ban+on+R134a+in+car+ac

Markets

Absorption sales to reach over $900 million by 2017

A new report from Global Industry Analysts (GIA) suggests the worldwide market for absorption chillers is forecast to reach $924.2 million by the year 2017, prompted by rising environmental concerns, requirement for low cost, high efficiency cooling systems and need for cutbacks in spiraling electricity charges.
The report highlights the need for comfort cooling, uninterrupted power supply and alternatives for fluorocarbon based chillers which are expected to spur growth in the absorption chillers market, by increasing usage in world markets such as Asia-Pacific, Europe and US which will further propel robust expansion in the long term.
In contrast to the European and US markets where centrifugal and positive displacement chillers occupy a dominant position, absorption chillers drive demand in the Asian chiller markets, particularly in Japan, China and Korea, which account for a significant 75% of the global market.
http://www.acr-news.com/news/news.asp?id=2496&title=Absorption+sales+to+reach+%24924%2E2+million+by+2017

Recovery to continue this year

Last year's recovery in the HVACR industry in Europe looks set to continue in 2011, according to Eurovent Market Intelligence (EMI).
The latest report, which covers Europe, the Middle East and Africa (EMEA), reveals that the chiller market rose to 16,000,000 kW in 2010, a 10% increase over 2009. Forecasts for 2011 range from 11% and 17% for Eastern Europe, 11% for Africa and between 5% and 12% for the rest of Europe.
The air handling unit market amounted to more than 1.5 bn Euros in 2010. Main markets are north east Europe (a quarter for Germany and 8% for Russia). By itself, this area accounts for 58% of sales.
Overall growth was witnessed in northern Europe and in some southern countries such as Portugal but there were decreases in most of the Eastern European countries such as the Czech Republic, where sales fell by more than 10%. The forecasts for 2011 are optimistic, with increases between 5% and 7% predicted, and even exceeding 20% for Ukraine and the former Soviet Union countries.
http://www.acr-news.com/news/news.asp?id=2487&title=Recovery+to+continue+this+year

Geothermal heat

pump shipments to double by 2017

Geothermal heat pump sales will experience strong growth in the next several years, with annual unit shipments in the United States increasing from fewer than 150,000 in 2011 to more than 326,000 units by 2017. According to a report by Pike Research, the total worldwide capacity for geothermal direct use applications will reach 179% of current levels during the same period. "The potential for geothermal heat pumps is high, but installations currently represent just 1% of the heating and cooling market," says industry analyst Mackinnon Lawrence. "However, growing electricity demand, rising energy prices, and increasing regulation around emissions and efficiency are all expected to push demand higher."

http://www.pikeresearch.com/newsroom/geothermal-heat-pump-shipments-to-double-in-volume-to-326000-units-annually-in-the-united-states-by-2017

Cost of Solar PV systems falling

The installed cost of solar photovoltaic (PV) power systems in the United States fell by roughly 17% in 2010, and by an additional 11% in the first six months of 2011. A report from Lawrence Berkeley National Laboratory says the recent cost reductions are attributable, in part, to "precipitous" decreases in the price of PV modules. The report also says that nonmodule costs, such as installation labor, also fell for residential and commercial PV systems in 2010.

http://newscenter.lbl.gov/news-releases/2011/09/15/tracking-the-sun-iv/

Source: The HVAC&R Industry for September 22, 2011

Heat pumps on the rise in New Zealand

Heat pumps are now found in one in four homes in New Zealand, and they are installed in half of all new homes, according to data compiled by the Energy Efficiency and Conservation Authority (EECA).

The sales data also shows that two-thirds of heat pump buyers are opting for ones carrying the blue Energy Star mark, which is an independent, international mark of energy efficiency.

http://www.acr-news.com/news/news.asp?id=2416

GEA given go-ahead to complete Bock takeover

GEA says it will retain the Bock name and make Bock's Frickenhausen manufacturing site its "competence centre" for small compressors.

The German industrial group made the announcement after being given the legal go-ahead to complete the acquisition of the German refrigeration compressor company.

The announcement also secures the future of over 340 staff at Frickenhausen and those at Bock's foreign plants.

GEA said it will continue to market the smaller compressors under the Bock name, the medium and large compressors will be further branded Grasso. Thies Hachfeld and Klaus Stojentin from GEA Grasso in Berlin will join Udo Klaussner from Bock Kältemaschinen as the new management trio at Frickenhausen.

GEA announced its intetions to buy the family-owned German refrigeration compressor manufacturer at the end of last year. The move, which included additional manufacturing sites in the Czech Republic, India and China, extends GEA's compressor offering in the lower and medium output ranges.

http://www.acr-news.com/news/news.asp?id=2431&title=GEA+given+go%2Dahead+to+complete+Bock+takeover

Dow in Chinese perchloroethylene joint venture

US chemical company Dow is to produce perchloroethylene, a key component in the production of HFC refrigerants, in a joint venture with the Chinese Befar Group.

Perchloroethylene (PCE or perc), is a key raw material in the manufacture of R134a and HFC 125. Demand in China is said to be growing rapidly but, to date, local supply has been limited.

The new manufacturing facility in Binzhou, Shandong Province, China, will have an initial target capacity of 40kt per year, with the ability to double production to 80kt soon after. Production could begin in 2014.

http://www.acr-news.com/news/news.asp?id=2425&title=Dow+in+Chinese+perchloroethylene+joint+venture+

Hydro-Québec plans to increase its financial assistance for residential geothermal heat pump installations

When Natural Resources Canada will end its ecoENERGY Retrofit - Homes Program in March 2012, Hydro-Québec suggests increasing its financial assistance from \$2000 to \$6375 for retrofits. In the new home market segment, Hydro-Québec suggests increasing the financial assistance from \$2800 to \$4000.

In addition, Hydro-Québec also says it would like to offer some form of financial assistance in the upstream to stimulate the supply side of the construction industry by signing agreements in priority with home builders in new developments. This financial assistance would target the builder's commitment and would be modulated according to its participation. The contribution could be as high as \$8000 per home.

Finally, and with the goal of ensuring the quality of geothermal heat pump installations, Hydro-Québec wishes to increase its support to the Canadian GeoExchange Coalition (CGC) to help the Coalition in its customer education efforts and communicate the advantages of CGC system certification. This component of the rate case is in perfect line with other measures and actions taken by the utility since 2002 to support the CGC in its geothermal heat pump market transformation initiative.

Source: CGC Newsletter

Heat pump market and technology in Korea

Jun-Young Choi, Korea Testing Laboratory, KOREA

Abstract

This paper describes the Korean market situation for heat pumps and advances in their technology. Heat pumps are seen as an alternative to current fossil-fuel-based means of heating and air conditioning. The paper also describes the potential of the heat pump in Korea in mitigating future CO_2 emissions.

Introduction

The International Energy Agency (IEA) has recently started promoting research to estimate national performance indicators for heat pumps in terms of quantifying their CO_2 reduction potential. Based on this, there is a move to quantify the amount of energy savings. In addition, the International Organization for Standardization (ISO) is working for better understanding by governments and consumers of the value of Annual Performance Factors (APF) in defining annual energy consumptions.

Residential, commercial and public buildings account for approximately 24 % of total energy consumption in Korea, and so energy-efficient buildings are indispensable to achieving a green developed country. Air conditioners, which account for the largest energy consumption in buildings, use 28 % of the country's thermal energy and 13% of its electricity. As the thermal energy is consumed mainly in the housing and building sector (housing sector, 90 %; business sector, 8 %; public sector 2 %), improvements in the sector's efficiency of energy use will have a corresponding significant effect on national energy consumption andCO_2 emissions.

The heat pumps used for high-efficiency cooling and heating in buildings are recognized as an alternative to existing primary heat resource device. The need for technology development and improved marketing of heat pumps are growing as the major means of response to the Framework Convention on Climate Change. This paper summarizes the current Korean market situation and technology developments of heat pumps.

Market

Expansion of the worldwide market for heat pumps is accelerating. As the heat pump becomes an more widely recognised alternative to fossil fuel, take-up is growing rapidly, with Europe as the centre. The heat pump market, which was as large as the shipbuilding market with a 61.5 billion dollar valuation in 2008, is expected to increase rapidly to 170 billion dollars in 2012. In particular, the global residential heating market for heat pumps has grown by more than 53 % per year.

Responding to the change in the market, the EU passed legislation to recognise aerothermal, geothermal or hydrothermal energy captured by heat pumps as renewable energy. Major European countries, such as France, Belgium and the Netherlands, encourage the take-up of heat pumps by means of various subsidies. Japan boasts the world's best technical skills in the heat pump field, and with COPs of 6 or more the use of water-heating heat pumps using natural refrigerant CO_2 is already commercialised. Japan has taken a lead with the introduction of a different product family: the ECO CUTE residential heat pump for water heating and low-temperature space heating in the form of floor heating. This system, and its technological development, has been actively supported by the Japanese government.

As a newcomer, South Korea is making various efforts to expand the heat pump market. Although the heat pump market has expanded recently, the domestic market is still at a poor level.

As of 2008, sales of air conditioners in Korea (the fourth largest producer of air conditioners in the world) amounted to 1.26 million units (residential air-to-air cooling-only units), whereas the share of heat pumps was around 5 %, which is a low value (see Figure 1), and even this is concentrated on the commercial VRF market (see Figure 2). The main reason is that as

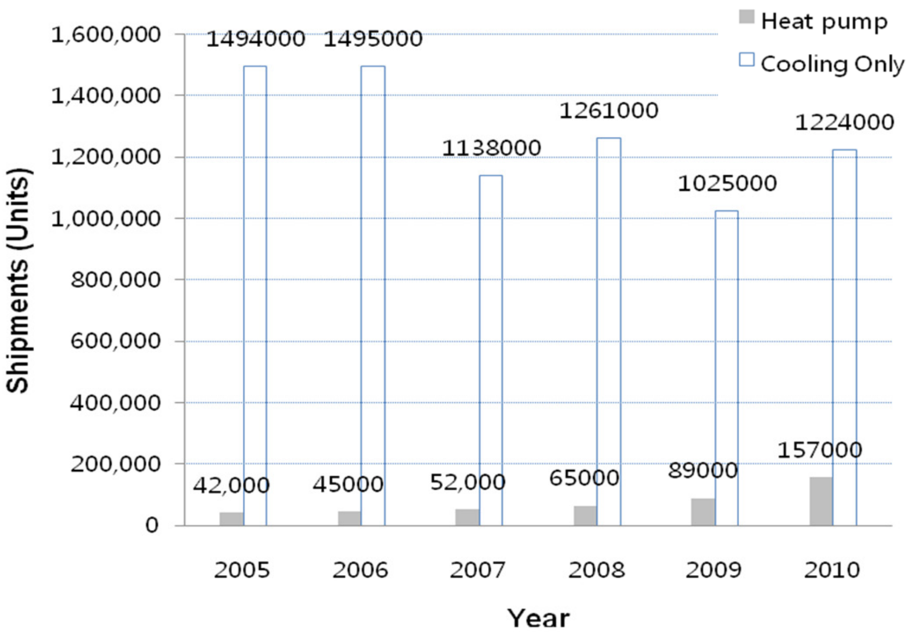

Figure 1 Shipments of residential air-to-air heat pump and cooling-only air conditioners below 23 kW cooling capacity
**Source: KEMCO(Korea Energy Management Corporation) Annual Report 2011*

most of the population lives in high-rise apartments and condominiums in urban areas, room air conditioners are preferred for reasons of space. The present technology level of heat pumps does not satisfy consumers' demands, and so their is a psychological barrier to their acceptance. This differs from Japan and China. In addition, the traditional Korean Ondol culture favours floor heating systems that circulate hot water from a fossil-fuelled boiler.

Traditionally, gas or oil boilers have accounted for 60 % of heating supplies in the residential space heating market. However, the market for heat pump water heaters is growing, creating a US$ 62 million market in 2008. But air-to-air heat pumps are dominant in the Korean market, with a limited demand for air-to-water, water-to-water and water-to-brine heat pumps for the commercial and industrial sector. Unfortunately, lack of data means that these markets are not easy to analyse. With subsidies for the use of renewables, geothermal heat pump numbers have grown steadily every year (see Figure 3) .Water-to-water heat pumps dominate the geothermal heat pump market as they are generally chosen by schools, public and commercial buildings.

The price of electricity in Korea has increased only by 25 % in 20 years, but the price of LNG and kerosene has soared sharply. Heat pumps are already competitive in the commercial and industrial sectors, while residential heat pumps are expected to be competitive when reliable electricity supplies can be assured through the use of nuclear energy in the future.

In addition, it is expected that the market share of heat pumps will increase dramatically as energy sources are diversified and technology evolves. Other factors supporting an increase in take-up include greater environmental awareness, increases in fuel price and the need to use renewable energy sources. The market for heat pumps for domestic application is expected to grow continuously in the sector of skyscrapers and commercial buildings (restaurants etc.), as well as replacing fossil-fuelled boilers in the residential sector.

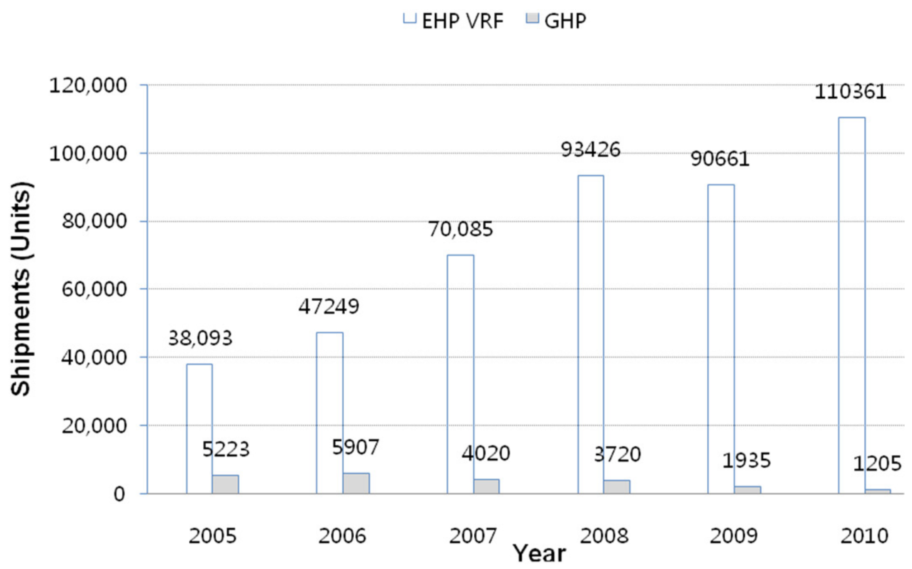

Figure 2 Shipments of EHP VRF and GHP
Key : EHP VRF is Variable Refrigerant Flow system by Electric Heat Pump GHP is Gas Driven Heat Pump
**Source: KRAIA(Korea Refrigeration and Air conditioning Industry Association), Statistical Data 2011*

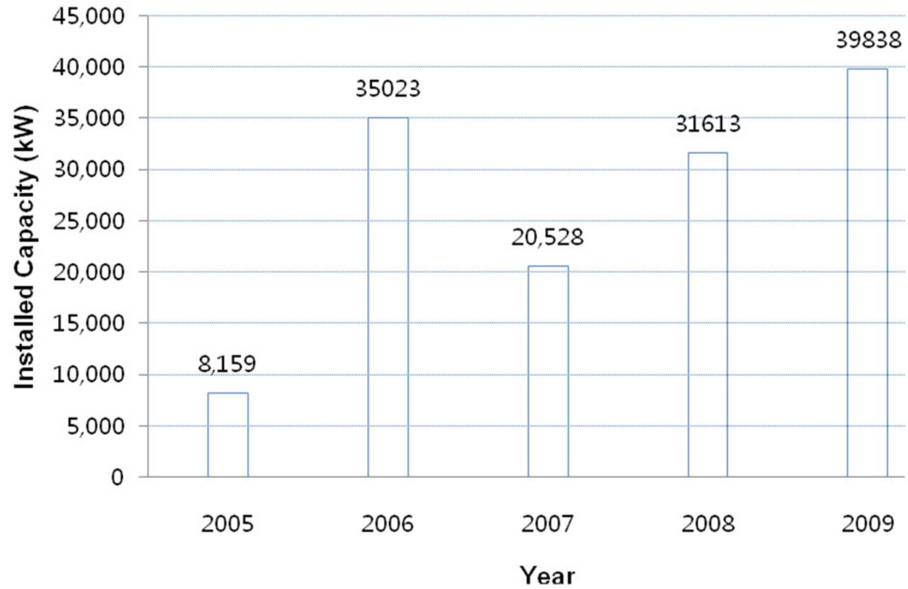

Figure 3 Annual installed capacity of geothermal heat pumps
Note: 1RT is 3.5 kW
**Source: REC(Renewable Energy Center), 2009 Report*

Technology

Although improvements in efficiency and reduction of environmental impact have progressed steadily, it is now compressor technology, improvement of product structure and development of heat exchangers and the use of microprocessors that are the main areas of active work. Investment in, and support for advanced technology for, heat pump water heaters are expanding, as is support for systematic production and R&D by industry, universities and research institutes.

In the Government's 2011 green energy strategic road map, the heat pump sector is expected to see ".... expansion of the domestic market and achievement of international competitiveness through the development of new concept heat pumps". The goal is to increase market penetration and improve the technology level, from 90% and 85% respectively until last year, to

100% by 2030. Similarly, the country's global market share is expected to increase over the same period from 8 % to 20 %. The Government recognises the value of heat pumps in reducing CO_2 emissions through replacement of existing primary heat source systems, and recognises the need for technology development as a main means of meeting the country's commitments in response to climate change.

For promoting the heat pump industry, the road map divides R&D programmes into commercialisation and original technology development. One area in focus is that of commercialisation of 'Refrigeration, Air Conditioning and Freezing' in the form of a medium- capacity air-to-water (ATW) heat pump system. The project aims to promote the domestic market and to create an international market by developing 'Refrigeration, Air Conditioning and Freezing' together in a heat pump system with high marketing opportunities based on VRF heat pump technology. This strategy should kill three birds with one stone in that it a) encourages a product family aimed at the export market, but based on domestic supply, b) improves energy efficiency and c) reduces CO_2 emissions.

A medium-capacity ATW product family(See Figure 4), should expand the market and develop a technology for the future, while reducing environmental impact protection and increasing the use of renewable energy. This plan also includes commercialisation of ATW heat pumps to replace existing heating and cooling systems using boilers, as well as chillers that use primary energy. At the same time, it concentrates on original technology development for 'Medium-Capacity, Hot Water Multistage Compression Heat Pump Systems' and 'Heat Pump System Using Latent Heat Storage', which are suitable for waste heat recovery and use in industrial processes. These systems can be expected to become increasingly important in the heat pump market in the long term.

In particular, the roadmap has proposed measures to boost the heat pump industry. These include renewable energy facilities and classification of high-efficiency appliances, incentives for high-efficiency heat pump products, etc.

Conclusions

Although the Korean heat pump market is relatively small, the Government is trying to deal with climate change and energy problems by expanding the heat pump market. Heat pumps are seen as a product presenting a dynamic force for new growth to promote the green energy industry. In the future, the heat pump market will grow as a result of boiler replacement, and is expected to have a great market potential for industrial use as well as for residential use. The Government is trying to support technology development and to systemise certification of performance, standard, quality and testing of heat pump devices and system by selecting the heat pump as one of 15 green energy sectors of the green energy strategic road map. If this effort is successful, it can achieve a very large effect nationally as the major means of reducing climate gas emissions, while at the same time supporting an internationally competitive export-intensive industry.

Author contact information
Dr. Jun-Young Choi
Chief Researcher, Team Leader
Korea Testing Laboratory
Liya67@ktl.re.kr

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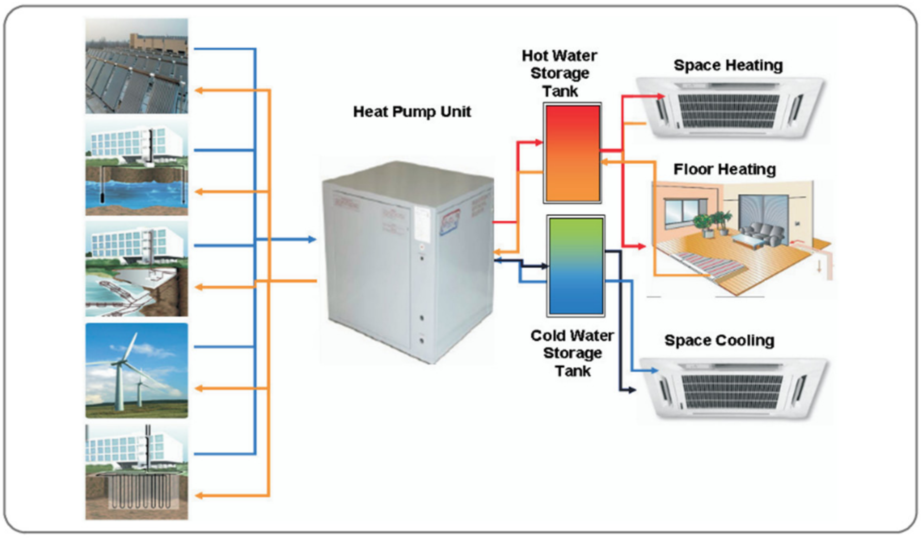


Figure 4 High-efficiency heat pump water heater with hybrid source
** Source: KETEP, "Green Energy Strategic Roadmap for Heat pumps", 2009 Report*

Annexes, ongoing

IEA Annex 34: Thermally Driven Heat Pumps for Heating and Cooling

The objective of this Annex is to reduce the environmental impact of heating and cooling by the use of thermally driven heat pumps. It is based on the results of Annex 24, "Absorption Machines for Heating and Cooling in Future Energy Systems", and cooperates with Task 38, "Solar Air-Conditioning and Refrigeration" of the IEA "Solar Heating and Cooling" (SHC) Implementing Agreement. Annex 34 is concerned with the development of performance eval-uation standards and the further development of higher efficient thermally driven heat pumps.

Following up the last newsletters, this newsletter focuses on the development of highly effi-cient evaporators.

The evaporator is the heat exchanger that connects the chilled water circuit to machines part where 'cold' is effectively produced through the evaporation of water vapour. As the adsorber generates the required temperature gradient between the chilled water (high pressure) and the water inside the evaporation pool (low pressure) – separated via a solid tube wall as it is typi-cally applied - the evaporator represents the thermal resistance for the heat flow. This heat resistance (R) consists of five individual resistances as they are expressed through the second formula in Figure 1.

Improving the heat transfer alongside the cross-sectional area of a tube - as regarded in Figure 1 - each participating resistance is to lower. For the internal heat transfer well known options are increasing the volume flow inside the tube or inserting turbulent flow generators. To lower the resistance of the tube wall materials with high thermal conductivities as well as huge con-tact areas on both sides are

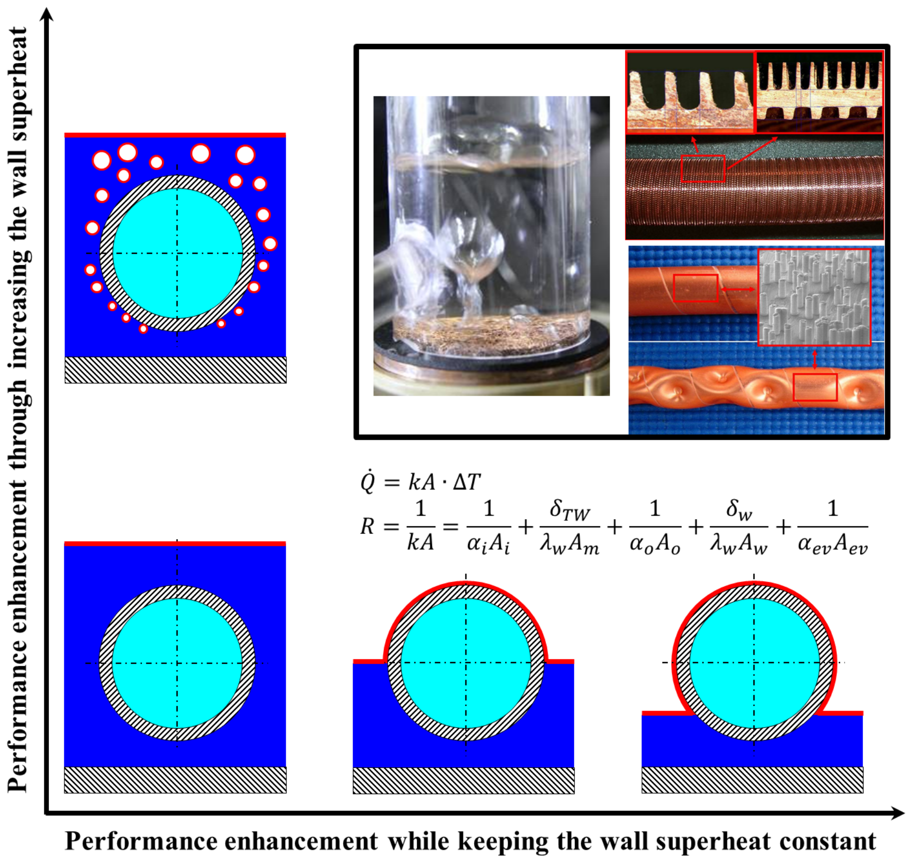


Figure 1: Enhancement technics to increase the outer heat transfer of an immersed tube and a cut-out with pictures of nucleate pool boiling at a 3D-metal structure manufactured by Fraunhofer Institute for Manufacturing Technology and Advanced Materials (IFAM) on the left hand as well as high performance evaporation tubes from the manufacturers Wieland Werke AG (top right) and La Mont Kessel GmbH & Co. KG with a surface treatment by MiCryon Technik GmbH (bottom right).

to apply. Observing the resistances outside the tube appropriate options are visualised in Figure 1. Thus, it is either possible through reaching the boiling re-gime of nucleate boiling (cf. top left) or to lower the thickness of the water layer. As cooling application require low temperature differences between chilled water stream and the water inside the evaporation pool high performance surfaces are required to reach the region of nu-cleate boiling at already little wall superheats. Left hand side of the cut-out in Figure 1 shows such a high performance 3D-metal structure where the initiation of nucleate boiling has been observed at wall superheats of already 8 K at a pressure of 10 mbar. Lowering the thickness of the water layer (cf. horizontal axis) – which simultaneously increases the area of free surface evaporation (cf. red layers) - is beneficial as it has a direct impact on heat transfer already at low wall superheats. Tube wetting is reached through capillary surface structuring where water moves upwards automatically. The cut-out in Figure 1 shows high performance evaporation tubes able to make use of this capillary-assisted-evaporation where 'art exists' in ensuring a continuous tube wetting during the operation. Additionally, these pictures show turbulent flow generators realised through internal fins (top) and a tube deformation (bottom). Both combinations result in a high tube performance while higher pressure drops inside the cold water stream require an additional pump energy input and are to respect while designing the evaporator.

Applying falling film evaporation which is as well accompanied by thin water layers and ad-ditionally uses the benefit of forced convection is also well-known to increase the heat trans-fer. Nevertheless, moving parts and the electrical energy input to run the refrigerant pump are to consider. Thus, the question is whether the energy conservation through the use of a more compact and therefore less 'pressure drop consuming' evaporator exceeds the electrical energy input to run the refrigerant pump or not.

The following Task description summarises the current progress:

Task A: First results of the collected country reports were presented on the 10th IEA Heat Pump Conference. The final report of Task A will be completed within the next few weeks which will then serve as basis for the handbook. A summary will be handed in to the journal "Renewable and Sustainable Energy Reviews". Unfortunately, the second attempt for an EU-funding, aiming to ensure the cooperation of the participating institutions beyond the project duration, was also rejected.

Task B: The data base review of already existing standards was completed. A proposal for the determination of performance indicators has been prepared under the label prEN 14599 and was presented at the 8th Annex 34 meeting at the International Sorption Heat Pump Confer-ence (ISHPC11). The proposal is currently being assessed by several groups and compared to the results of other proposals (e.g. VDI). The German VDI-Directive for Gas Heat Pumps was published. A working group of the European Committee for Standardization (CEN) started their work on the CEN 12309 regarding gas-powered heat pumps where some members of this Annex are represented.

Task C: The round robin tests for the determination of balance and conductivity were contin-ued and extended by other commercially available materials. At the same time even more test laboratories are participating and listed on the website. Parallel to that, the data base of sorbent materials is growing continuously (internal web pages). A proposal for a directive to determine the properties of sorption materials and a comparison of two reference materials were also presented at the ISHPC after the meeting. In addition, work is underway on the measurement of the cycle stability of adsorption materials. New tests to evaluate the mechani-cal stability started and more results of the test setup to check absorption cycles are gained. Work on the development of evaporators, new adsorber based heat exchangers, materials and coating techniques are being carried forward.

Task D: There was a large feedback on the template for data acquisition on existing plants. According to this, the task participants were informed about the best case studies which are currently being prepared for the website. The nomenclature was changed and adapted to that from the IEA Task 38. A description of standardized test procedures has started. Different procedures of individual laboratories are being collected and compared regarding the ad-vantages and disadvantages of each. The list for the use of potential simulation tools has been completed and has been published on the website.

Task E: The website (www.annex34.org) was complemented with a literature database, links to other relevant pages and was also moved to a faster server. In parallel, the access list of the internal pages has been updated. Several papers were presented at the International Sorption Heat Pump Conference and the 10th IEA Heat Pump Conference. A special edition of the HPC newsletter: "Thermally driven heat pumps" was published earlier this year.

IEA HPP /IETS Annex 35 / 13 Application of Industrial Heat Pumps

The Annex 35 / 13, a joint venture of the IEA Implementing Agreements "Industrial Energy-Related Technologies and Systems" (IETS) and "Heat Pump Programme" (HPP) organized its annex meeting on June 16, 2011 in connection with
the 6th Annual Conference on Industrial Energy Efficiency "From Research to Low CO_2 Plants" at EDF R&D, Les Renardières (France). The meeting was attended by 18 participants from 9 member countries.

The discussions about the programme of work started with the approval of the
Task 1 report, presenting market overviews of the energy situation, the energy use of the industrial sectors and the barriers for heat pump application in all participanting countries. These findings should provide the basis to further work for the wider application of industrial heat pumping technologies. The report will be available by the end of the year.

The major bottleneck of the programme of work is still Task 2 "Modeling calculation and economic models". Two options have been discussed: to update the old screening program of the Annex 21 "Global Environmental Benefits of Industrial Heat Pumps (1992 -1996)" or to develop a new programme, also based on the pinch analysis. So far no final decision has been taken.

The next annex meeting will take place on the September 27, 2011 in connection with the next "European Heat Pump Summit" at the Congress Center Nürnberg/Germany on the September 28-29, 2011.
It was also agreed to.to organize a Annex 35/13 workshop on "Practical applications of industrial heat pumps" as part of the Heat Pump Summit on the September 28, 2011. The workshop is planned as an introduction of a workshop in connection with a major industrial conference, such as Achema 2012, Frankfurt or Chisa 2012, Prague.
H.J. Laue

Status Update on IEA Annex 36: Quality Installation / Quality Maintenance Sensitivity Studies

A working meeting of the Annex 36 Participants was held on 14 – 15 June 2011 in Stockholm, Sweden. The meeting reviewed the focus and work to be undertaken (see table below), progress by each participating country, formulated the report format structure, and established the schedule for future meetings. In conjunction with the working meeting, three heat pump facilities in the Stockholm area were toured: (1) a central city district heating and cooling plant, (2) a single-family home with a ground geothermal HP system, and (3) a river-source geothermal system installed in a moderate sized hotel.

The Annex is scheduled to run through November 2013 with additional working meetings planned for September 2012 (in the U.S.) and the fall 2013 (in France).

Annex 36 Background: It is widely recognized that residential and commercial heat pump equipment suffer significant performance loss (i.e., capacity and efficiency) depending on how the components are sized, matched, installed, and subsequently field-maintained. Annex 36 is evaluating how installation and/or maintenance deficiencies cause heat pumps to perform inefficiently and waste energy. Specifically investigated is the extent that operational deviations are significant, whether the deviations (when combined) have an additive effect on heat pump performance, and whether some deviations (among various country-specific equipment types and locations) have larger impacts than others.

Annex 36 Participants	Focus Area	Work to be Undertaken
France	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. KTH/SVEP - investigations and statistical analysis of 22000 heat pump failures. Modeling/Lab: Determination of failure modes and analysis of found failures (SP) and failure statistics (KTH/SVEP).
United Kingdom	Home heating with ground-to-water, water-to-water, air-to-water, and air-to-air systems.	Field: Replace and monitor five geothermal heating systems Lab: Investigate the impact of thermostatic radiator valves on heat pump system performance.
United States (Operating Agent)	Air-to-air residential heat pumps installed in residential applications (cooling and heating).	Modeling: Examine previous work and laboratory tests to assess the impact of ranges of selected faults covered augmented by seasonal analyses modeling to include effects of different building types (slab vs. basement foundations, etc.) and climates in the assessment of various faults on heat pump performance. Lab: Cooling and heating tests with imposed faults to correlate performance to the modeling results.

IEA HPP Annex 37

The aim of this project is to demonstrate and disseminate the economic, energy, and environmental potential with heat pumping technology. The focus will be on modern technology and results from already performed field measurements will be used to calculate energy savings and CO_2 reduction. It should be possible to predict the most suitable heat source and heat pump system for a certain application in a certain geographic region. In order to draw the right conclusions it is of most importance that the quality of the measurements is guaranteed. The criteria for good and secured quality will be defined in the project. An additional goal is to establish a data base, connected to the Heat Pump Centre website, where data from field measurements are presented.

In order to achieve the objectives of the Annex, the activities have the following structure:

Task 1 (Taskleader SP, Sweden)
Make a common template of what should be communicated. The focus is on the template content. Cosmetics are not considered in this task.

Task 2 (Taskleader Plainair SA, Switzerland)
Define criteria for good quality of field measurements (e.g. boundaries of the measured systems, number of and placement of measuring points, measurement uncertainty, time increments etc.) and decide what pa-

rameters are important for assured quality.

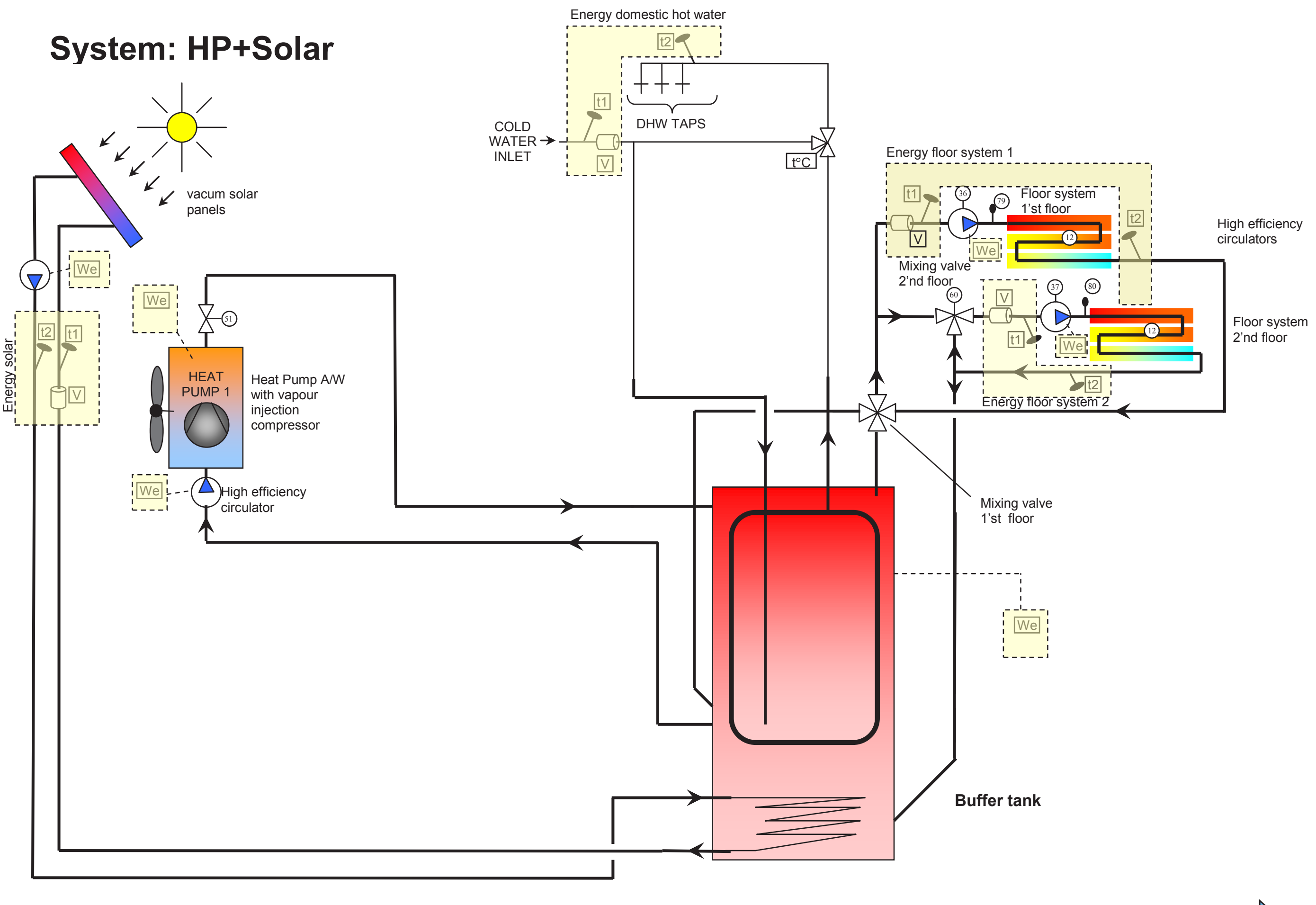
Task 3 (Taskleader AIT, Austria)
Collection and evaluation of current and concluded field measurements on heat pump systems. The focus is on the best available technology.

Task 4 (Taskleader not decided)
Agree on how to recalculate the selected annual performance measures, such as seasonal performance factor, energy savings and carbon footprints. Calculation of SPF, electricity consumption, energy savings and CO_2 reductions from the collected measurements. These parameters will be compared with those for other heating systems. For thermally driven heat pumps, Annex 34´s definition of SPF will be applied. Regarding systems with heat pumps combined with solar thermal, the results from the work with combining the solar fraction or solar savings fraction with an SPF, that is ongoing in Task 44 of the IEA Solar Heating and Cooling Programme/Annex 38 of the IEA Heat Pump Programme, will be considered. For an example, see the figure.

Task 5 (Taskleader SP, Sweden)
Establish a database connected to the HPC website, based on data from field measurements and the common template; the best examples will be documented.

Task 6 (Taskleader DECC, United Kingdom)
Information dissemination
Information to installer and manufacturers shall contain good examples, but it could also contain bad examples, with mistakes that are often made and should be avoided. This information should support further development of training documentation (e.g. EU Certified Heat Pump Installer) and also installation manuals and regulations supplied by the manufacturers.
The work in Task 1 is completed. Good examples will be published in the database. The following information should be given:

- Information about the building (year of construction, year of installation, type of building, application area, geographical location, climate, heated area etc). It is not decided how to describe the climate. It is desirable to supply degree days and to make an agreement about how to calculate degree days.
- Photo of the building.
- Information about the heat pump (heating capacity, type, purpose, type of heat source, type of heat sink, heat source system, distribution system, alternative / complementary heating system etc).
- System scheme.
- Key parameters of the control (e.g. heating curve).
- Information about measurements (number of measuring points, location of measuring, sampling interval, system boundaries etc). (The declared seasonal efficiency and savings depends both on system boundaries and on where in the system the heat meters are located.



- Results (curves and key figures). Savings should be expressed in both energy and terms of carbon savings/carbon footprint. Discussions about how to express carbon savings are ongoing.

The work in Task 2 is ongoing and is expected to be completed early September.
A working meeting is planned on September 27th in Nürnberg in connection with the Heat Pump Summit. A workshop is planned on September 29th. Other countries are invited to join Annex 37.

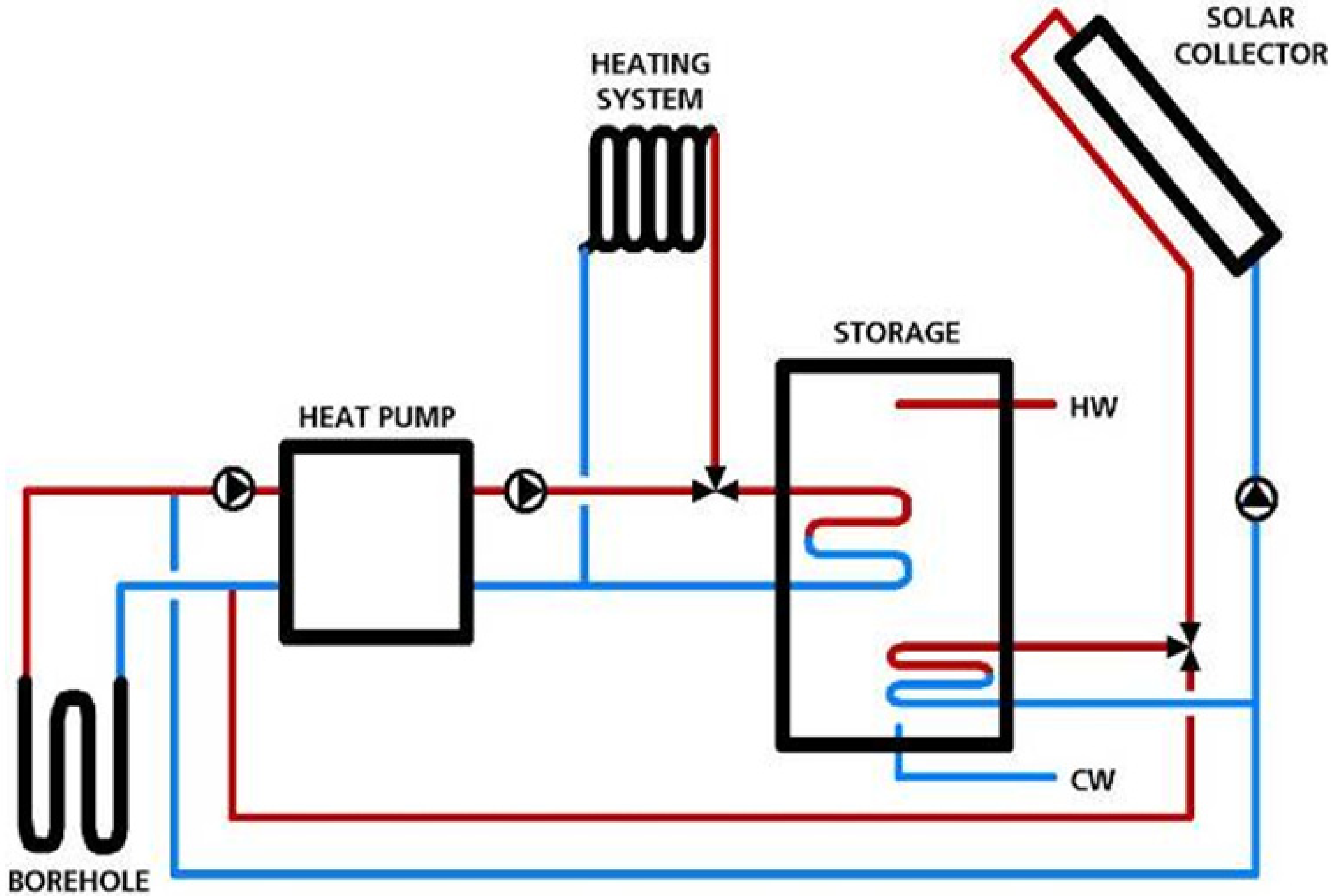
Contact:
Pia Tiljander
E-mail: pia.tiljander@sp.se

IEA HPP Annex 38: Solar and Heat Pump Systems

Heat pumps, promoted by electric utilities, have become a popular heating system, and when combined with solar energy the energy savings only increase. Manufacturers see a bright future for this technology and are working to optimize their systems. This is where the IEA HPP Programme together with the SHC Programme comes in.
The goal of Annex 38, a joint effort with SHC Task 44 (T44A38), is to optimize the combination of solar thermal energy and heat pumps, primarily for single-family houses using electrically driven heat pumps and glazed or unglazed solar collectors. This combination is the main share of the current expanding market of S+HP solutions.

Companion Technologies?

Solar and heat pump technologies share some common traits, such as the use of some electricity to make "free" energy available for hot water and space heating, the variation of the quality of the source over a day or season, the dependency upon the temperature of the source, and the need for storage.
The optimisation of one technology will often help the other one, and it makes much sense to look at both with a consistent and global view.
To combine these two "mature" technologies is not easy and solar companies and research institutes are currently busy working on increasing the solar fraction for heating and hot water while decreasing the cost.Heat pump companies and research institutes are mainly focused on increasing the annual COP or SPF (above 5 is the target) and solar can help !



Example of System Type 2: Active Regeneration

Moving From Non-Integrated to Fully Integrated Systems

There are basically two kinds of systems that can be designed when working with two heat producers:

A non-integrated solution: the heat pump system provides the heating and serves as the back up for the domestic hot water.

A fully integrated system: the heart of the system is the heat pump but solar provides energy to the evaporator side of the heat pump, either through a storage tank or directly, and when possible to the DHW tank and/or to the heating distribution system.

In T44A38, more than 20 monitored projects are under investigation. Serval basic configurations have been classified as "generic", i.e., as representative of all others.

The main feature of this system is the active use of the solar energy for support of the ground heat source regeneration. It is not yet clear if this regeneration brings a definite advantage. Annex participants are working on the theoretical principle.

Comparisons

This is main the goal of Task 44/Annex38.

To facillitate this process, Annex participants have created the 'square view'. This is a fast way to understand any configuration.

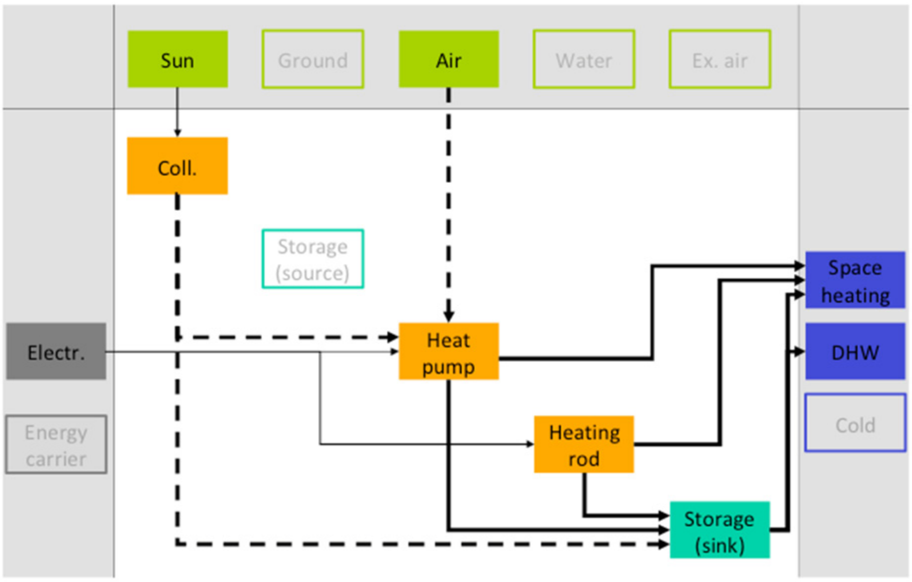
Participating Countries

Austria, Belgium, Canada, Denmark, Finland, France, Germany, Italy, Spain, Sweden, Switzerland, UK, USA

Conclusions: Annex expectations

- The IEA framework provides a unique opportunity to meet and share with the experts from universities and industries working on one side on thermal solar and on the other side on heat pumps to exchange new ideas and knowledge and to test them in conjunction.

Article contributed by Jean-Christophe Hadorn, Operating Agent of SHC Task 44/HPP Annex 38, jchadorn@baseconsultants.com
www.iea-shc.org/Task44



Example of a square view , a new way to describe any

Ongoing Annexes

Bold text indicates Operating Agent. * Participation not finally confirmed, ** Participant of IEA IETS or IEA SHC

Annex 34 Thermally Driven Heat Pumps for Heating and Cooling	34	AT, CA, CH, **DE**, FR, IT, NL, NO, US
Annex 35 Application of Industrial Heat Pumps (together with Task XIII of "Industrial Energy-Related Technologies and Systems" (IEA IETS))	35	AT, CA, DK**, FR, **DE**, JP, NL, KR, SE, CH*
Annex 36 Quality installation and maintenance	36	CA*, CH*, DE*, FR, JP*, KR*, SE, UK**, US**
Annex 37 Demonstration of field measurements of heat pump systems in buildings – Good examples with modern technology	37	AT, CH, **SE**, UK
Annex 38 Systems using solar thermal energy in combination with heat pumps	38	AT**, BE**, CA**, **CH**, DE, DK**, ES**, FI, FR**, IT**, 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, NL, KR, **SE**, US

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

2011 Ritter-von-Rittinger Awards IEA Heat Pump Programme

In conjunction with the 10th International Heat Pump Conference, the International Energy Agency Heat Pump Programme has given the prestigious Ritter von Rittinger Medal to three awardees.

The award is presented triennially in conjunction with the International Heat Pump Conference to individuals and teams who have distinguished themselves through international achievements in advancing heat pumping technology, markets or applications that result in improved energy efficiency and environmental benefits. This award is named for Peter Ritter von Rittinger, an Austrian engineer credited with the design and installation of the first practical heat pump system at a salt works in Upper Austria in 1856. The award was presented for the first time at the IEA International Heat Pump Conference in 2005 in Las Vegas USA, and is awarded once every three years at the International Heat Pump Conference.

The 2011 Rittinger awardees are Professor Per-Erling Frivik of Trondheim, Norway, Professor-Doctor Hermann Halozan of Graz, Austria and Mr. John D. Ryan of Bethesda, Maryland (U.S.A.). As, due to the earthquake in Japan, the conference was held as a virtual event, these individuals received the awards in Paris on May 16, 2011. Dr. Sophie Hosatte, Chairman of the IEA Heat Pump Programme Executive Committee, presented the awards.

Professor Hermann Halozan and Mr. John D. Ryan. Unfortunately, professor Frivik could not attend the ceremony.

Professor Per-Erling Frivik

Professor Per-Erling Frivik received the award in the “Markets” category. Throughout his professional life he was affiliated with SINTEF, and NTH/NTNU in Trondheim, Norway. His credo was ‘technological progress through international cooperation’. As the Research Director at the SINTEF Refrigeration Engineering Division from 1986 through 1997, Professor Frivik built up a large group in heat pumps and refrigeration engineering. It is within this group that Professor Gustav Lorentzen developed the idea of using CO_2 as a working fluid. Professor Frivik was instrumental in raising the funds necessary for the research and technology transfer with major international equipment manufacturers at an early stage. This is evidenced by the success of CO2 heat pumps. He also initiated an effective cooperation between his research group at SINTEF and the NTH/NTNU’s Institute for Refrigeration Engineering which served internationally as a model (the GEMINI model) of co-existence between academic and applied research.

He was strongly involved at the International Institute of Refrigeration, as a member and President of Commission E2 from 1995 to 1999, and as a member of the Scientific Council during the same period. He served on the special Committee responsible for development of the IIR’s first Strategy Plan, adopted in 1999. He was instrumental in the process that led to the IIR launch of the IIR Gustav Lorentzen Conference on Natural Working Fluids. In addition, he raised funding from international friends so that IIR could establish the IIR/IIF Gustav Lorentzen Medal.

Professor Frivik was also a member of the IEA Heat Pump Programme Executive Committee between 1985 and 1997, and the Chairman for several years, shepherding the Programme through a major reconstruction beginning in 1989. He established the Programme Advisory Board and was instrumental in recruiting high level individuals for this Board. He also established the Norwegian National Team, which he headed for several years. He participated in many IEA Heat Pump Conferences, serving as Chairman of the

International Organizing Committee and was known for his passionate presentations advocating heat pump technologies. At the 2002 Conference in Beijing Professor Frivik presented a retrospective talk entitled "A Half Century of Heat Pump Applications" in which he recalled many of his experiences during the latter half of the 20th Century.

Professor Doctor Herman Halozan
Professor Halozan received the award in "Technology, Markets, and Applications" category. Professor Halozan has been providing his vast expertise in heat pumps to a vast number of international associations and organizations for more than 20 years. He has strongly influenced the European heat pump industry by supporting the development and dissemination of quality standards for heat pumps at European and national levels, promoting international collaborations for developing knowledge, systems and practices in heat pumps through research, development, demonstration and deployment. He was instrumental in promoting heat pumps in national and international political agendas and has greatly contributed to their diffusion in the European heat pump market.

We can emphasize his long-serving and valuable role in Austria and Europe, for instance at the Association of the Austrian Heat Pump Industry and the European Heat Pump Association. He has also been an active member of the International Institute of Refrigeration, as a private member since 1984, and since 1999, as the President of Commission E2, Heat Pumps and Heat Recovery, and a member of the Management Committee.
He plays a significant role at the IEA:

- Since 1994, he is the Austrian Delegate at the IEA Working Party on End-Use Energy Technology, and the chairman since 2008;
- Since 1992, he is the Austrian delegate at the IEA Heat Pump Programme Implementing Agreement, and chairman for the period 1998-2001;
- Since 1994, he leads the Austrian National Team at the IEA Heat Pump Programme.

Professor Halozan is now a consultant for industry and government organizations.

Mr. John Ryan
Mr. John Ryan received the award in the "Administration/Organization" category. Mr. Ryan was one of the initiators of the IEA Heat Pump Programme shortly after he joined the U.S. Department of Energy in 1978 and served as the US delegate to the Executive Committee until his retirement in 2007. For nearly three decades, John devoted the bulk of his DOE career to establishing and guiding the Department's Research Development & Demonstration/Deployment efforts to develop and promote adoption of advanced heat pump technologies. He worked tirelessly to integrate the IEA HPP with his DOE program, being directly involved in the very first HPP project (Annex 1, Common Study of Advanced Heat Pumps).

Mr. Ryan has always been very active in the organization of the work of the programme. In the early years of the HPP he was a member of the Heat Pump Centre (HPC) Steering Committee and served as its chair in 1989 when the HPC underwent a major reorganization. He was a founding member of the HPP Executive Committee and served as its chairman from 1995-1997. He has been a staunch supporter of the HPP within the USA, and led efforts to establish and maintain a U.S. National Team comprised of representatives from U.S. industry and research organizations. As a result, the U.S. has participated or will participate in most of the Annexes and other activities over the duration of the programme. He was a leader in establishing Annex 10 (Technical and Market Study of Advanced Heat Pumps) as well as Annexes 12, 18, 21, and 26, all highly successful projects.

Furthermore, Mr. Ryan was chairman of the International Organizing Committee (IOC) for the 1993 and 2002 International Heat Pump Conferences (held in Maastricht and Beijing, respectively). He also played key roles in organization and promotion of the 1987 and 2005 conferences that were held in the U.S.

Sophie Hosatte
Chair, Heat Pump Centre

EXAMINATION REGARDING AIR-CONDITIONERS AND HEAT PUMPS, USING THE NEXT GENERATION REFRIGERANTS

Shigeharu TAIRA, Takeshi YAMAKAWA, Akinori NAKAI, Ryuuzaburou YAJIMA
DAIKIN INDUSTRIES,LTD., 1000-2 Aza-Ohtani, Okamoto-cho, Kusatsu, Shiga 525-8526 Japan

Abstract: In recent years, to meet the Montreal Protocol requirements, reduction of the fluorocarbon refrigerants designated by the Protocol has almost been completed in Japan by replacing them with HFCs that have zero-Ozone Depletion Potential (ODP).However, although HFCs have zero-ODP, some HFCs known as alternatives to HCFCs have high Global Warming Potential (GWP). Now it has become imperative to find refrigerants that could satisfy the requirements of both the Montreal Protocol and the Kyoto Protocol at the same time. When choosing refrigerants, we should not focus on GWP only since the degree of global warming impact is greatly affected by various factors such as capacity of equipments in use, safety, economy, and Life Cycle Climate Performance (LCCP).When used in heat pump systems, those factors vary from country to country according to the climate of regions, their operating conditions, the way they are used, electricity rate system, and applicable laws and regulations. In this paper, we propose a notion of diversity of refrigerant choice, in which we suggest the most suitable refrigerants for various applications in various regions by employing the best available technology. Our proposal includes not only developed countries but also developing countries whose contribution to global warming has been becoming more significant than before due to their rapid economic growth.

Key Words: Refrigerant, GWP, LCCP

1 INTRODUCTION

In recent years, the refrigeration and air conditioning industry and academic researchers have been studying the possibility of adopting new refrigerants to save energy and reduce the environmental impact to help mitigate global warming. When selecting a new refrigerant, environmental consciousness, safety, performance, economy and other aspects, as shown in Figure 1, under given application and operating conditions must be considered to make a scientific and rational choice to have a substantial impact on global warming. At present, possible new refrigerants have been identified for car air conditioners, which present relatively few technological problems to be overcome. On the other hand, more studies are needed on new refrigerants for stationary air conditioners (Taira and Nakai 2010). Figure 2 shows trends in adoption of next-generation refrigerants for use in air conditioning and hot water supply. In Figure 2, we see that since the Kyoto Protocol came into effect, Europe has led the way in adoption of next-generation refrigerants, prioritizing leakage prevention and implementation of substance control regulations. These substance control regulations have first been applied to automotive air conditioners, which feature relatively few technical hurdles and easily replaceable refrigerants. Regulation of stationary air conditioners is the next on the agenda.

In fact, in highly eco-conscious Europe, a great amount of research is already underway on new low-GWP (Global Warming Potential) refrigerant candidates such as ammonia, hydrocarbons (HC), CO2 and others, as well as on air conditioning, refrigeration and hot water supply systems employing these natural low-GWP refrigerants. In terms of synthetic

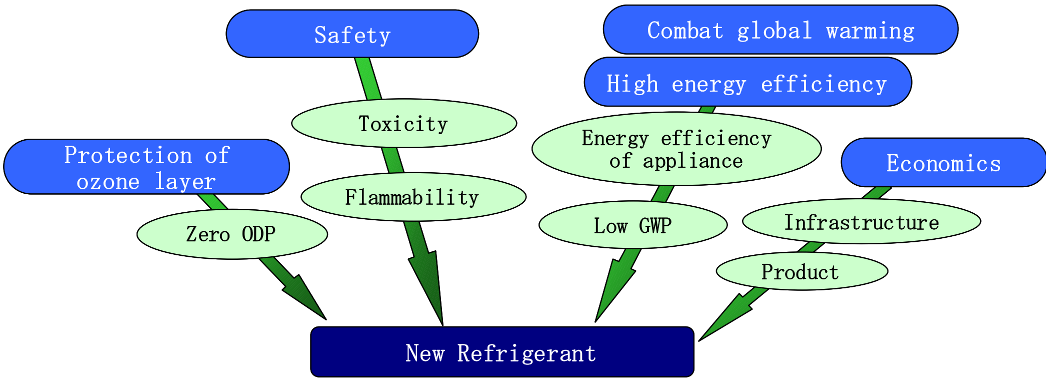


Figure 1: Examination Item for selection of New Refrigerant

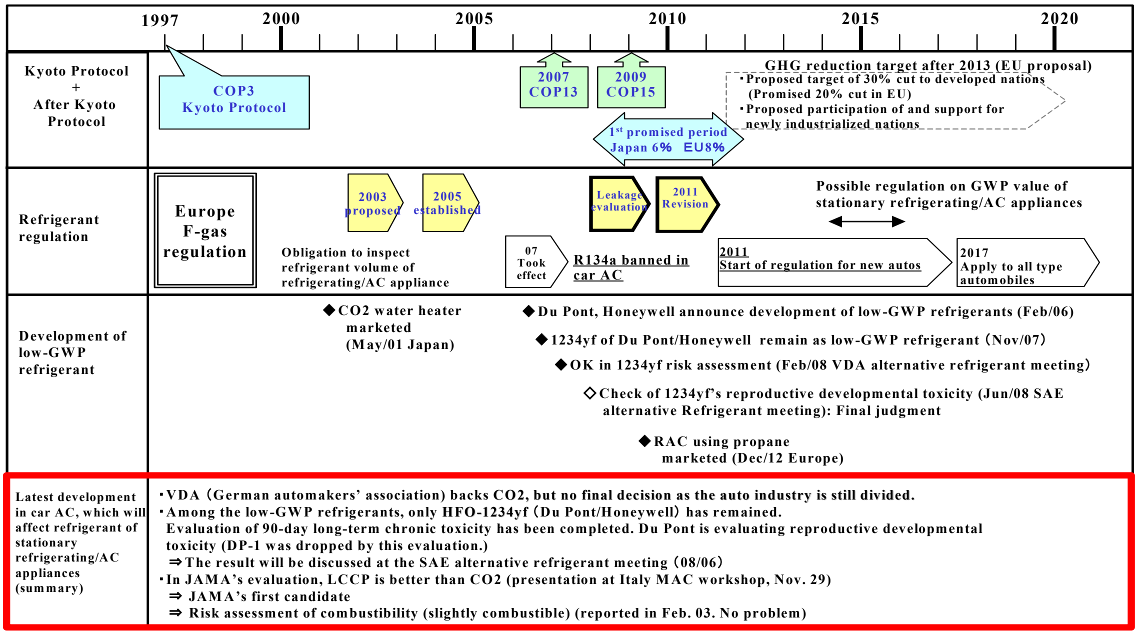


Figure 2: New refrigerants for air-conditioning and hot water supply

Table 1: Properties of refrigerants compared to R410A

		Cond. Press Mpa*1 @45℃	Volumetric Capacity*1 kJ/m3	COP	GWP (IPCC4)	Flamm -ability	Toxicity
HFC	R410A	2.73	100%	100%	2088	No	Low
	R32	2.80	100-110%	100-103%	675	Low*2	Low
	HFO-1234yf	1.15	40-45%	100-106%	4	Low*2	Low
	R32/yf Mix *3 (50% : 50%)	2.26	80-85%	100-103%	340	(Low)	(Low)
HC	R290 (Propane)	1.53	55-60%	100-106%	<3	High	Low

*1 : Calculated by NIST Refprop Ver.8.

low-GWP refrigerants, R32 has already been developed. Some major refrigerant manufacturers have developed HFO-1234yf, intended as a new low-GWP refrigerant option, which is now undergoing wide-ranging assessments. Meanwhile, in Japan, 2001 saw the introduction of heat-pump water heaters employing CO2, a natural refrigerant and one of the next-generation refrigerant candidates. Around the same time, air conditioners using propane went on the market, albeit in negligible quantities. However, national and regional variations in operating conditions, usage configurations, electric power networks, and laws and regulations mean that natural refrigerants are not a panacea for prevention of global warming, nor always the best possible option.

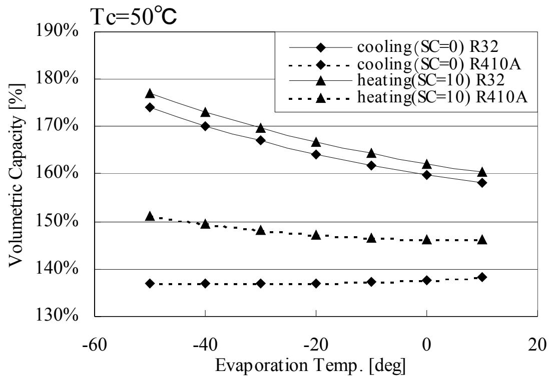


Figure 3: Volumetric Capacity (compared with R22)

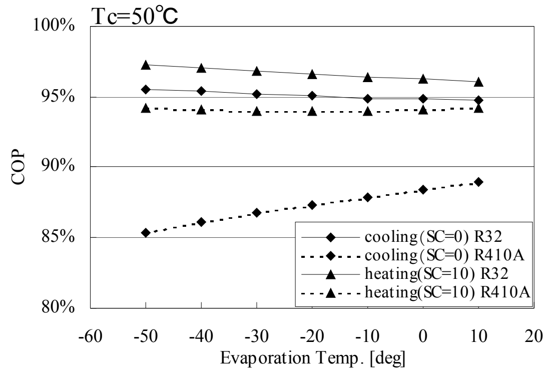


Figure 4: COP (compare with R22)

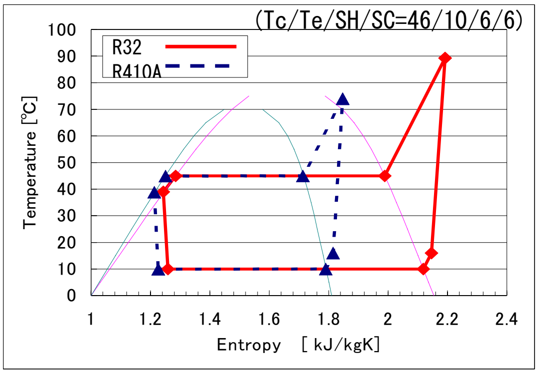


Figure 5: Characteristics of R32 on T-S diagram

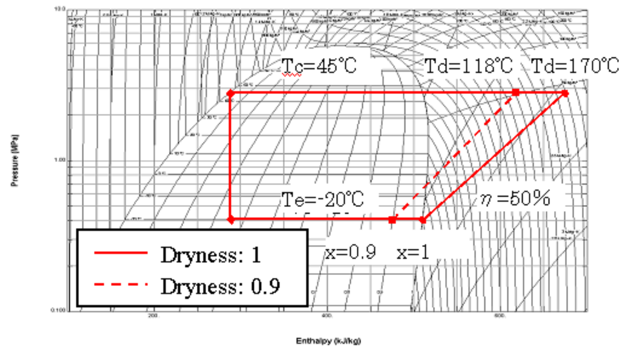


Figure 6: To reduce discharge temp of R32

Table 1 shows properties of new refrigerants. Each has a problem in either flammability, efficiency or GWP. No refrigerant is perfect – which means that the most suitable refrigerant must be chosen depending on the application and the volume of refrigerant to be charged.

For residential and package air conditioners, possible new refrigerants under consideration include fluorocarbon refrigerants with very low flammability - HFO-1234yf and R32 - and their mixture, as well as propane, a hydrocarbon refrigerant with superior performance and low GWP but high flammability. For VRV, which requires a large volume of refrigerant, CO_2 is also a possibility. Technological development is underway to put them to practical use. Against this background, a new category “lower flammable refrigerants with a maximum burning velocity of <= 10cm/s” (2L) has been created in safety standards for HFO-1234yf, R32 and other refrigerants whose flammability is so low that they may be able to be used as virtually nonflammable in practical use when treated properly. This move is intended to promote the use of these refrigerants, which offer advantages in combating global warming.

Among the numerous new refrigerants, this paper focuses on R32, which makes a substantial contribution to energy conservation and requires minimal modification of equipment configurations. This paper discusses various considerations regarding the benefits and challenges of practical adoption of R32.

2 COMPARISON OF REFRIGERANT CHARACTERISTICS

Figure 3 and Figure 4 show a comparison of theoretical performance of R410A and R32. Considering the possibility of switching from R22 equipment, they are presented as in comparison to R22 (Yajima et al. 2010)

Compared with R410A, R32’s volumetric capacity ratio and COP tend to increase as the evaporation temperature drops. R32 is advantageous in cold areas because the lower the evaporation temperature, the higher the volumetric capacity ratio.

As shown in Figure 5, a disadvantage of R32 is that its discharge temperature is higher than that of R410A. A refrigerant with larger volumetric capacity ratio tends to have a higher discharge temperature due to an increase in the specific heat ratio.

3 MAIN SOLUTION OF PROBLEM POSED BY R32

There are three main issues with R32. One is the rise in discharge temperature, another is selection of refrigerant oil, and the third is the protocol for handling of slightly flammable refrigerants.

First, let us consider the high discharge temperature. Figure 6 shows discharge temperature considerations for a cycle employing R32. In Figure 6, compressor efficiency is set at 50%, condensation temperature at 45 °C, and evaporating temperature at -20 °C, and an example of discharge temperature change when compressor intake dryness changes are given. At compressor intake dryness 1, the discharge temperature is 170 °C, not a viable value when taking into consideration compressor and refrigerant oil reliability. It is possible to bring the discharge temperature from 170°C to 120 °C by lowering the compressor intake dryness to 0.9 from 1.

However, excessive lowering of the intake dryness runs the risk of liquid compression occurring inside the compressor, or obstruction of compressor lubrication due to a decline in refrigerant oil viscosity inside the compressor. Accordingly, it is necessary to pinpoint the dryness limit in accordance with the characteristics of the compressor and refrigerant oil, and to take this limit into consideration.

Table 2: Frequency of low temperature conditions

Mean Temp.	Days of Mean Temperature (Jan.–Dec. 2009)						
	Beijing	Cicago	N.Y.	Munich	Helsinki	Moscow	Sapporo
< −10℃	1	10	0	1	12	24	1
< −15℃	0	2	0	0	2	7	0
< −20℃	0	2	0	0	0	3	0
< −25℃	0	0	0	0	0	0	0

Min. Temp. (day)	Days of Minimum Temperature (Jan.–Dec. 2009)						
	Beijing	Cicago	N.Y.	Munich	Helsinki	Moscow	Sapporo
< −10℃	18	34	4	12	39	42	5
< −15℃	1	9	0	5	10	24	0
< −20℃	0	2	0	0	1	7	0
< −25℃	0	1	0	0	0	2	0

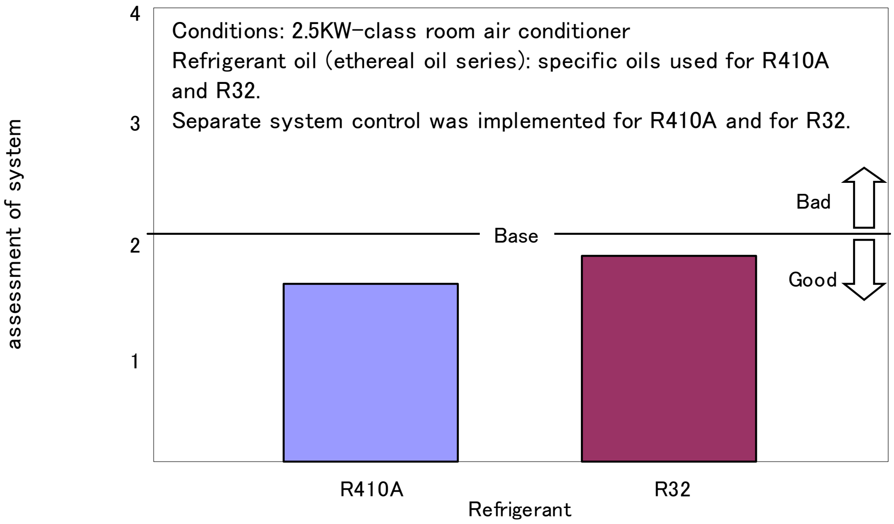


Figure 7: assessment of system

Part of the evaporation capacity of the refrigeration cycle will be consumed by cooling of the discharge gas, leading to a decline in COP. However, as evidenced by Table 2, under the winter temperature conditions in major cities of the world, operating conditions such as these occur infrequently, and year-round impact on COP is negligible.

When the evaporating temperature reaches -30 °C, it is necessary to decrease dryness in order to further lower the discharge temperature. When -30 °C conditions constitute a steady state, it is likely to have a major impact on compressor reliability and performance, necessitating the consideration of measures such as use of different compressor material or refrigerant oil, or use of a liquid or gas injection cycle.

Next, Figure 7 shows the results of a system reliability evaluation after 4,000 hours of operation, using R410A and R32. The evaluation test was conducted under the operating conditions of a 2.5KW-class room air conditioner. In accordance with refrigerant characteristics, operation was carried out with different system control software for R410A and for R32. Ether-series refrigerant oil, (however, detailed formula was selected to be appropriate to the unique properties of each refrigerant) was used in the compressor for both R410A and R32. The results showed that product reliability after 4,000 hours when R32 was used with the next-generation refrigerant oil X was well within the same scope as, and nearly equal to currently employed R410A.

Regarding the flammability of R32, a new category, class 2L (lower flammable refrigerants with a maximum burning velocity of <= 10cm/s), has been created in addition to the existing flammability classification, as shown in Table 3, ASHRAE34/ISO817 safety classification. Organizations in various countries are in the process of establishing a practical safety standard. R32 and HFO-1234yf belong to the class A2L, and a realistic safety standard different from that for A2 will facilitate its practical application. As shown in Figure 8, the charge limit for R32 and HFO-1234yf in relation to room area is higher than that for R290 and other flammable refrigerants, so their practical application is foreseeable in the near future.

Table 3: ASHRAE34/ISO817 safety classification

A1 (No toxicity, noncombustible)	A(B)2L (A: No toxicity, slightly combustible, B: Toxic, slightly combustible)	A2 (No toxicity, combustibility: low)	A3 (No toxicity, combustibility: high)
R744(CO_2) R410A R22	R717(ammonia, B2L) **HFO-1234** **R32**	R152a	**R290 (propane)**

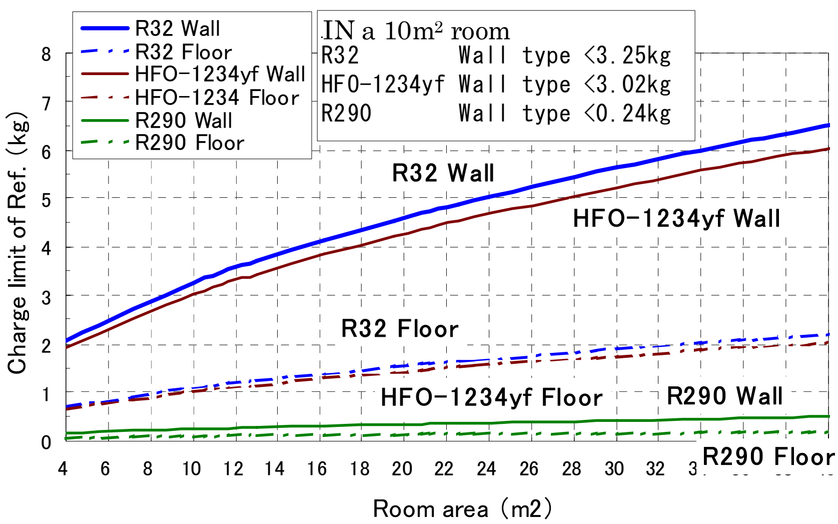


Figure 8: Charge Limit or Refrigeration

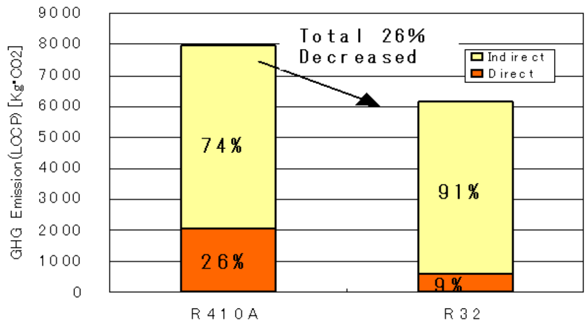


Figure 9: RAC 4kW LCCP(Amount of GHG Emission)

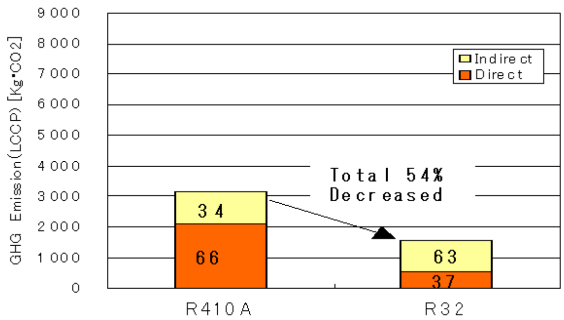


Figure 10: RAC 4kW LCCP(Amount to GHG Emission) Indirect 20%

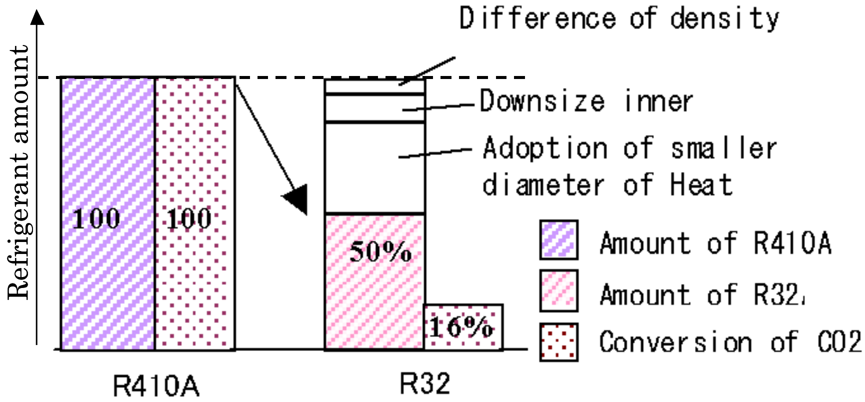


Figure 11: Reduction of Refrigerant

4 POTENTIAL TO MITIGATE GLOBAL WARMING

The impact of an air conditioner on global warming throughout its lifetime is evaluated based on life cycle climate performance (LCCP), an index covering both the direct impact of its refrigerant and the indirect impact of consumption of energy. The direct impact is calculated by multiplying the amount of refrigerant emitted into the atmosphere by its GWP. As far as direct impact is concerned, the lower the GWP, the better.

Figure 9 shows R32's effect to reduce LCCP based on APF of JISC9612. R32's GWP is about one third of that of R410A, so the direct impact can be reduced to one third. As mentioned earlier, R32 has higher efficiency than R410A, so it can reduce indirect impact by cutting annual and instantaneous power consumption. By switching from R410A to R32 refrigerant, LCCP can be reduced by 26%.

According to some reports (Tahara and Takada 2010), (Takada and Tahara 2010), actual operating time of air conditioners in total is only about 20% of the figure used by JISC9612. Figure 10 shows calculations based on operating time of 20% of that JIS9612 describes. LCCP is reduced to 54% of that of R410A..

Switching to R32 whose the GWP is one third of R410A will change the ratio of direct and indirect impacts. Direct impact by the refrigerant will not be a dominant factor in emission of greenhouse gasses anymore. Continued efforts to reduce indirect impact are important.

Reducing the amount of refrigerant charge is another way to reduce direct impact (Yajima et al. 2000). As R32 has larger volumetric capacity than R410A, the amount can be reduced to 50% of that of R410A for equal performance, as shown in Figure 11, by design optimization such as using thinner heat exchanger tubes, and downsizing the compressor cylinder as well as by the difference of density.

Table 4: New refrigerants compartmentalization

Application	Existing refrigerant	Possible new refrigerant		Remarks
MAC	R134a	HFO-1234yf CO_2		- HFO-1234yf is waiting for result of toxicity evaluation to be released in June 2008.German manufacturers support CO_2.
Direct expansion RAC	R410A	High outdoor temperature, warm area	R32 other	- HFO-1234yf is a low-pressure refrigerant and not suitable for direct expansion system due to pressure loss. Measures must be taken to reduce pressure loss. - CO_2 has low cooling COP, which is about 65% that of HFC, so it is not suitable for application in warm areas. Its heating COP is about 85% that of HFC and capacity drop under low outdoor temperature is small, which makes CO_2 a possible candidate in cold areas. - R32 can replace R410A because property of R32 is similar to R410A's.
		Cold area	R32 CO_2	
Positive displacement chiller	R134a	Large size	HFO-1234yf	-R134a can be simply replaced with HFO-1234yf.
	R407C R410A	Medium to small size	R32 Other	
Centrifugal water chiller	R134a	HFO-1234yf		-R134a can be simply replaced with HFO-1234yf.
Water heater, hot water heating	R134a	Hot water heating	HFO-1234yf	- HFO-1234yf and its mixed refrigerant for heating water to about 65℃ - CO_2 has advantage in applications to heat low-temperature water to high temperatures, such as Eco Cute - CO_2 is not suitable to applications to circulate hot water, because its COP drops sharply when the temperature of returning water is high.
	R407C R410A	Water heater & hot water heating	R32 Other	
	CO_2	Hot water supply only	CO_2	

5 CONCLUSION

Our studies so far have revealed the following:

- R32 has higher discharge temperatures than R410A, but by lowering the compressor inlet dryness, it can be used under climate conditions found in many cities.
- The problem of flammability is being solved as standards are in preparation to treat it as having very low flammability.
- R32's effect to mitigate global warming is significant when air conditioner operating time is assumed to be shorter than that assumed by JIS.
- Because R32 has higher volumetric capacity, the charge amount can be reduced.
- Product reliability after 4,000 hours when R32 is used with the next-generation refrigerant oil X is well within the same scope as, and nearly equal to currently employed R410A.

All in all, it can be concluded that R32 is a refrigerant enabling quick action against global warming, since it has fewer hurdles to its practical application and early launch of products can be expected.

However, R32 is not almighty. Other refrigerants can be suitable depending on environment and intended purpose. Table 4 shows a "refrigerant map" listing the various applications of the next-generation refrigerants. It is evident that, at present, rather than narrowing down refrigerants excessively, the most effective way of combating global warming is to use the most efficient refrigerant in each field and selecting the most appropriate refrigerant for each application from among a wide range of options.

Along with development of an innovative new refrigerant, which would require extensive studies before commercialization, introducing a quick, practical measure is another way to contribute to the fight against global warming.

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OPTIMIZING HEAT PUMP DESIGN FOR REFRIGERANTS R407C AND R410A

Eric Winandy, Director Application Engineering,
Enrico Fraccari, Product Manager Residential Comfort
Guy Hundy, Technical Consultant,
Emerson Climate Technologies, Pascalstrasse 65,52076 Aachen, Germany

Abstract: This theoretical study based on actual compressor performance and backed up by experimental evidence shows how condensing temperature can be minimised and evaporating temperature maximised by circuit design and control. The effect of temperature glide in the heat exchangers can be used to good effect with R407C and use of a suction/liquid line heat exchanger can improve evaporator utilisation. With R410A, temperature differences are limited by other factors and compressor performance is better. Seasonal analysis for space heating both underfloor and radiator, using the latest vapour injection scroll technology indicates where R410A shows advantages over R407C

Key Words: heat pumps, simulation, efficiency

1 INTRODUCTION

The technique for using compressor data for simulation of heat pump performance was reported in previous papers, Winandy and Hundy, 2008, 2007. The study reported in the IEA paper also demonstrated very good correlation between the simulated data for several rating points and the test data obtained from WPZ tests. The test points were chosen to correspond to the WPZ data at EN14511 conditions. The simulation technique has now been used to make a detailed comparison between R407C and R410A in air and ground source heat pumps. In each case the minimum practical temperature differences in the heat exchangers have been applied so that this represents a best case scenario for each refrigerant. The benefits of each refrigerant in these applications is investigated.

2 TEMPERATURE DIFFERENCES IN CONDENSERS

A visualisation of the heat exchanger temperature profiles in plate heat exchanger (PHX) condensers is given in Figures 1 and 2. For R407C the limiting condition is seen to be at the water inlet. This “pinch point” is governed by the need to maintain a positive temperature difference (2K) at this point. Any subcooling at that point will increase the temperature difference and hence increase the condensing temperature. The negative effect on COP of raising the condensing temperature is greater than the positive effect of increasing the enthalpy difference in the heat exchanger. A liquid receiver is therefore beneficial and in practice with R407C it tends to result in a small amount of subcooling due to stratification, typically 1.5K. The effect of refrigerant pressure drop is small and has been neglected.

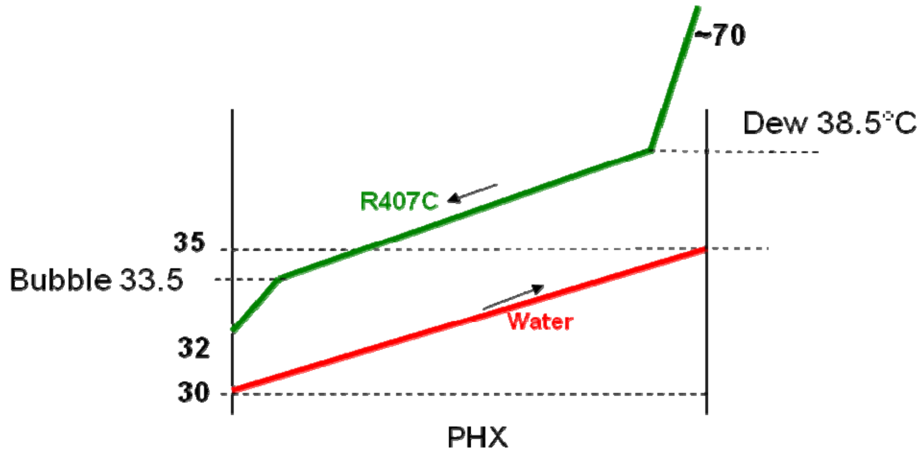

Figure 1: Condenser Temperature Profiles with R407C

On the contrary, for R410A the pinch point is at the water outlet and there is also a larger average Temperature difference through the condensation process. At the water outlet itself the high temperature gas is de-superheated, and so, with reference to the water outlet temperature, the condensation temperature difference can be as low as 1.5K. In that case, subcooling is beneficial as it increases the heating capacity without affecting directly the condensing temperature, and 3.5K subcooling is possible without liquid receiver.

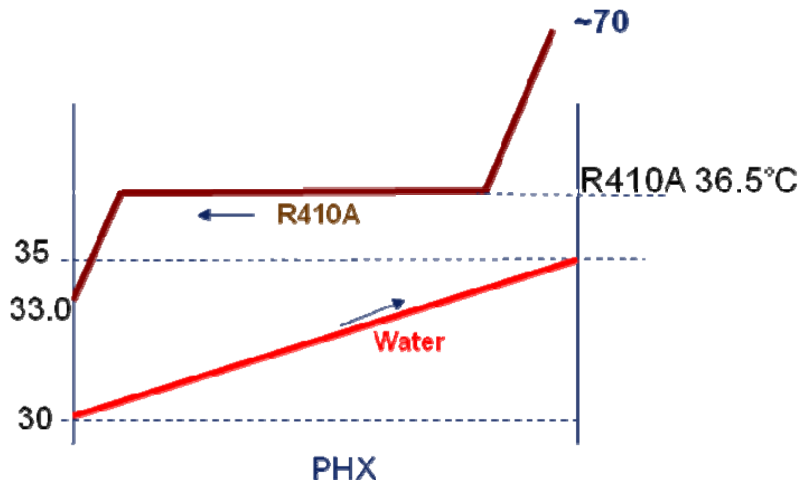

Figure 2: Condenser Temperature Profiles with R410A

3 TEMPERATURE DIFFERENCES IN PLATE HEATEXCHANGER EVAPORATORS

The temperature profiles are shown in Figures 3 and 4 for R407C and R410A respectively. For both refrigerants, with a standard expansion device requiring about 5K, the maximum evaporating temperature is limited by the superheat required (bold lines). But the inherent glide of R407C during its evaporation, a suction/liquid heat exchanger (Figure 5) allows the evaporation temperature to be increased by taking the superheated region out of the evaporator without any other constraint. In that case, an increase of 3K is possible in practice (Figure 3, dashed lines). Note that due to difficulties in keeping a stable control loop at the level of the expansion device, it is better to retain some superheat at the evaporator outlet. By using a suction/liquid heat exchanger with R407C in this way the practical maximum evaporating temperature is -3°C.

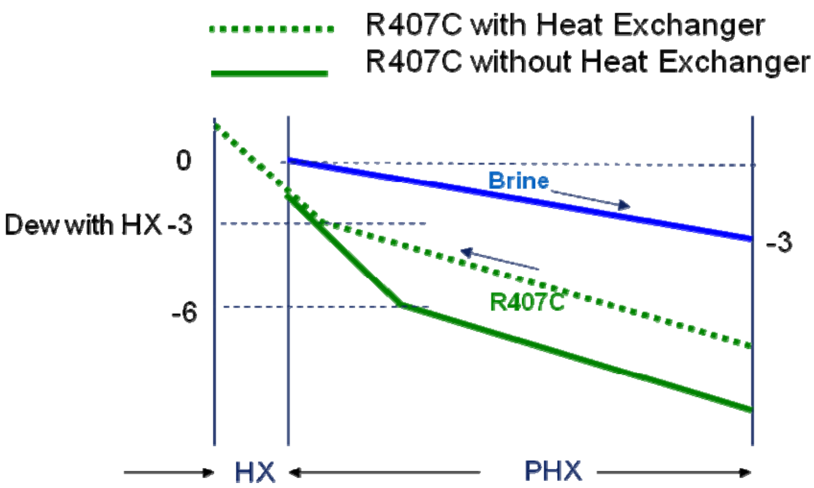

Figure 3: PHX Evaporator Temperature Profiles with R407C

In the case of R410A, a suction/liquid heat exchanger will not be as beneficial since a raise of evaporation temperature will be limited by another pinch point appearing at the brine outlet zone for the Plate Heat Exchanger due to the absence of glide with R410A. In that case, the best that can be done is to reduce the superheat to a minimum value with the help of an electronic valve for example. An evaporation of -4°C at rating conditions is possible then (Figure 4, dashed lines). So minimum temperature difference can be achieved with R410A if superheat controlled at 2K. Beyond this point the TD at the brine outlet starts to impose a limit.

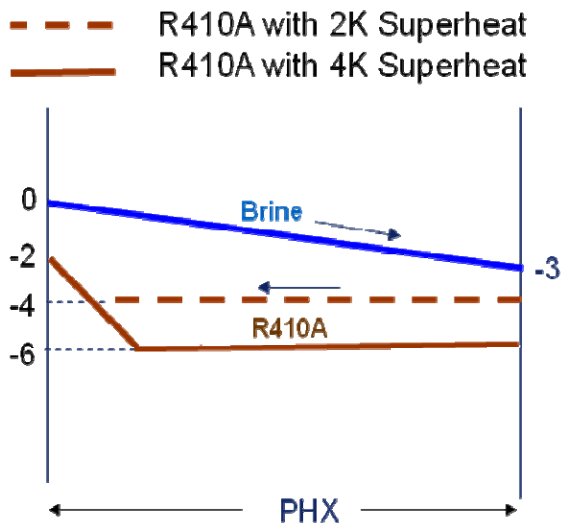

Figure 4: PHX Evaporator Temperature Profiles with R410A showing a superheat of 4K for conventional TEV and 2K for an electronic valve

The diagrams in Figure 5 show the location and effect of the suction/liquid line heat exchanger. This device would not be necessary for R410A.

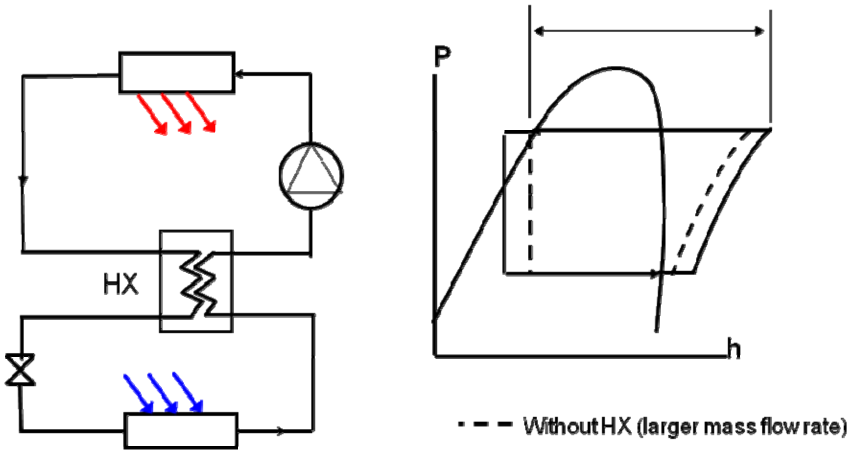

Figure 5: Suction/Liquid Line Heat Exchanger

4 TEMPERATURE DIFFERENCES IN AIR COIL EVAPORATORS

For air coil evaporators a minimum temperature difference of 8K has been chosen because it corresponds to state-of-the-art coils for both refrigerants. A slightly lower temperature difference should be achievable with R410A because of its inherently better heat transfer properties compared to R407C. Having said that, there are other considerations such as cost, dimensions and defrost capability that will influence the coil performance.

5 SUMMARY OF BOUNDARY CONDITIONS

Table 1: Ground Source Boundary Conditions

	R407C	R410A
Pump power, % compr power*	5%	5%
Superheat	4K	2K
Delta T, (Dew – Water Out)	3.5K	1.5K
Delta T, (Dew – Brine Out)	3K	4K
Subcooling	1.5	3.5

*Fixed value, based on design condition

Table 2: Air Source Boundary Conditions

	R407C	R410A
Fan power , % compr power*	10%	10%
Superheat	4K	4K
Delta T, (Dew – Water Out)	3.5K	1.5K
Delta T, (Airside)	8K	8K
Subcooling	1.5	3.5

*Fixed value, based on design condition

6 SIMULATION RESULTS FOR RATING POINT

For ground source B/W heat pumps four compressor technologies have been investigated, and are shown in Figure 6:
ZP36 Air conditioning R410A Scroll Compressor
ZH09K1P Heating Optimized R410A Scroll Compressor
ZHI11K1P Heating Optimized R410A Scroll Compressor with vapour Injection
ZH21 Heating Optimized R407C Scroll Compressor
ZH13 Heating Optimized R407C Scroll Compressor with vapour Injection
The Heating Optimized type has a higher compression ratio scroll set together with a dynamic discharge valve. This gives a flatter isentropic efficiency curve with better isentropic efficiency at the higher pressure ratios required for heating applications.

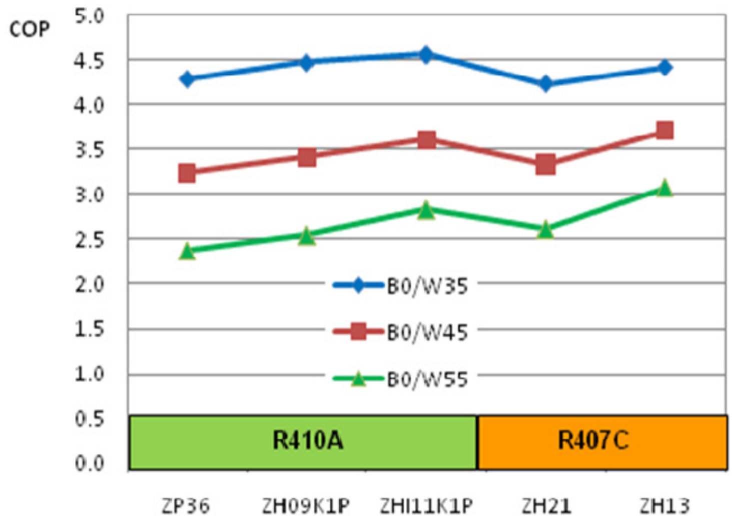

Figure 6: Ground Source HP COPs with water temperatures 35, 45, 55°C

- R410A gives better performance when all the boundary conditions are optimized.
- For underfloor heating at 35°C the benefit of vapour injection is mainly in heating capacity. The vapour injection model has approximately 18% more capacity than the standard heating model. This can be easily found from Compressor Selection Software (2010) The COP advantage is small as can be seen from Figure 6.
- There is better COP advantage for vapour injection with higher water temperatures, 45 and 55°C, Figure 6. There is also more heating capacity enhancement.

For Air Source, examples of the same scroll technologies have been analysed, with separate charts for 35°C water and 55°C water, Figures 7 and 8:
ZP42 Air conditioning R410A Scroll Compressor
ZH12K1P Heating Optimized R410A Scroll Compressor
ZHI11 K1P Heating Optimized R410A Scroll Compressor with vapour Injection
ZH38 Heating Optimized R407C Scroll Compressor
ZH13 Heating Optimized R407CScroll Compressor with vapour Injection

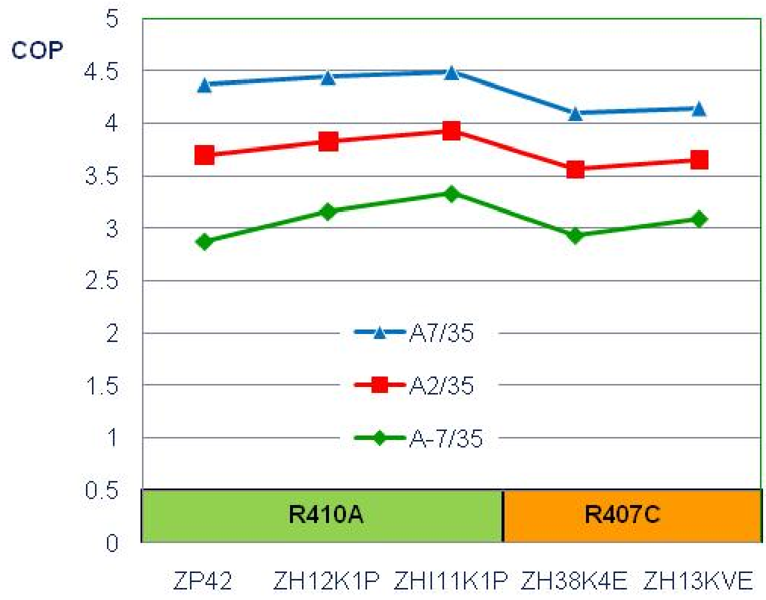

Figure 7: Air Source HP COPs with water temperature 35°C

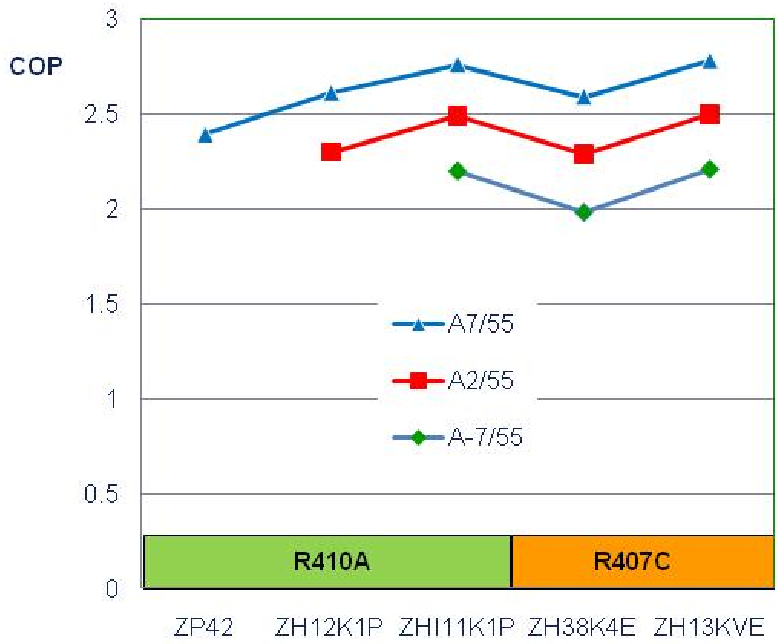

Figure 8: Air Source HP COPs with water temperature 55°C

- R410A gives distinctly better performance at the lower water temperature, and is equivalent to R407C at high water temperature where its lower critical temperature starts to gain influence.
- Vapour injection shows a distinct COP advantage, particularly at lower air temperatures.
- Operation at low air temperatures for non-vapour injected models is limited by the discharge temperature with both refrigerants. The absence of simulated COP points in Figure 8 shows the influence of this limitation. The operating envelope diagrams for all models can be found in the Compressor Selection Software (2010).

7 SEASONAL PERFORMANCE

Seasonal performance has been investigated for air source models using software based on the prEN14825 method. The term SCOP, Seasonal Coefficient of Performance, is defined as the ratio of heating annual heat delivered to annual electrical energy absorbed, including pump/fan (tables 1,2). Figure 9 shows a typical screen with the cooler climate profile illustrated at the top left. The graph visualizes the load and the HP capacity, showing the bivalent point. In this example the cooler climate with 55°C water outlet is chosen.

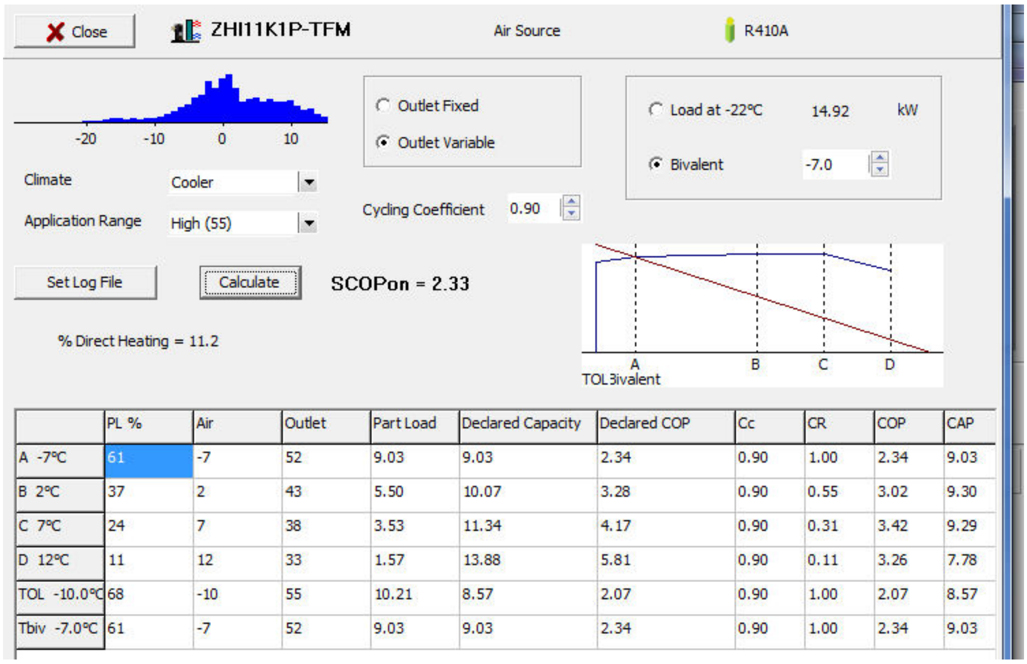

Figure 9: SCOP calculation screen in Selection Software

Figures 10 - 13 are example results of SCOP analysis using the above method for Air/Water heat pumps. This tool enables rapid comparisons between various compressor technologies when applied in different climates and with different distribution systems. In Figures 10 and 12 the distribution is to underfloor heating at 35°C and in Figures 11 and 13 the distribution is via radiators at 55°C. SCOPs have been translated in annual running cost terms for the model sizes used in the analysis. An average electricity cost of 0.16 Euro per kWh has been applied.

The bivalent point for cold climate is -7°C, and for the average climate -1°C. Building load is 19kW at -22°C for the cold climate and 13kW at -10°C for the average climate (same building envelope). For air temperatures below the bivalent point, back up direct electric heat is added as necessary to fulfill the load requirement. The back up heater annual energy input is between 1.7 and 3% of the total annual load, except in the case of the air conditioning scroll when applied to air source, where it is 8%. This is because in the lowest ambient temperature bands this scroll type is unable to operate due to discharge temperature limits. The vapour injected type will deliver a higher proportion of the heat heat demand below the bivalent point, and hence minimize the back up heat requirement.

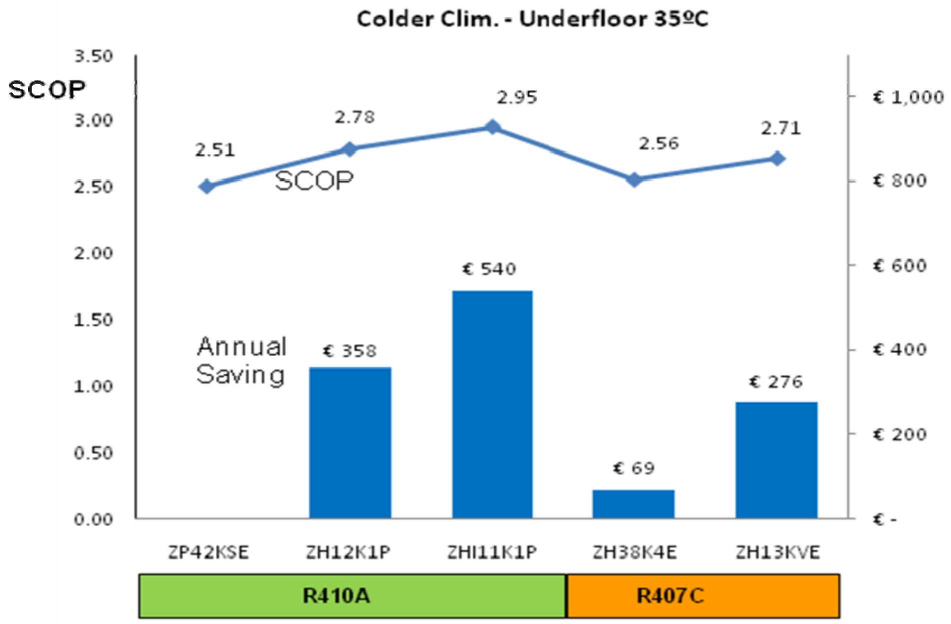

Figure 10: SCOP values and corresponding annual savings in the colder climate. The cost baseline is ZP42KSE

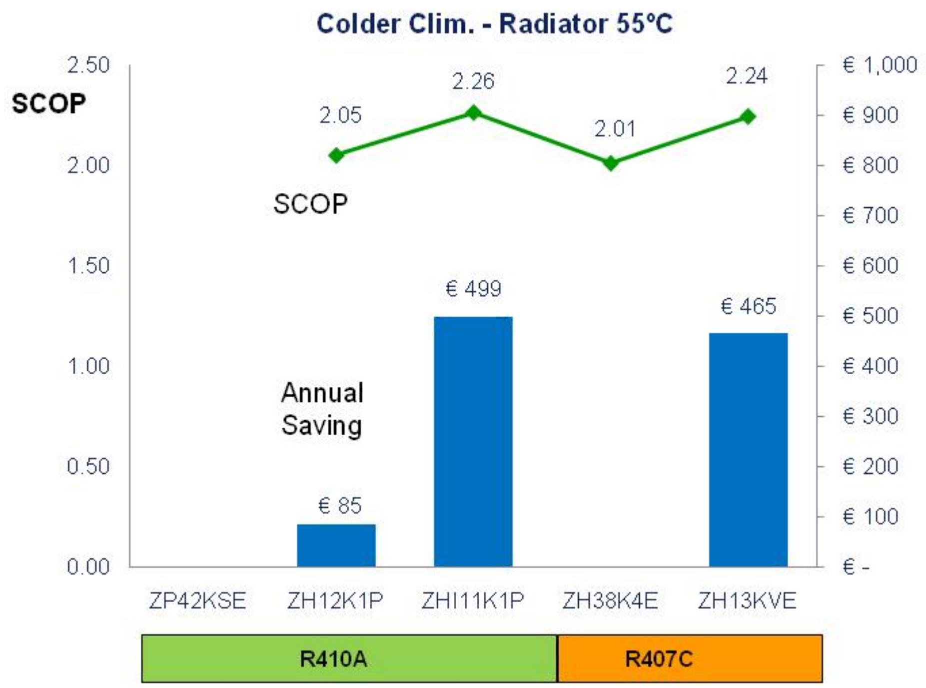

Figure 11: SCOP values and corresponding annual savings in the colder climate. The cost baseline is ZH38K4E

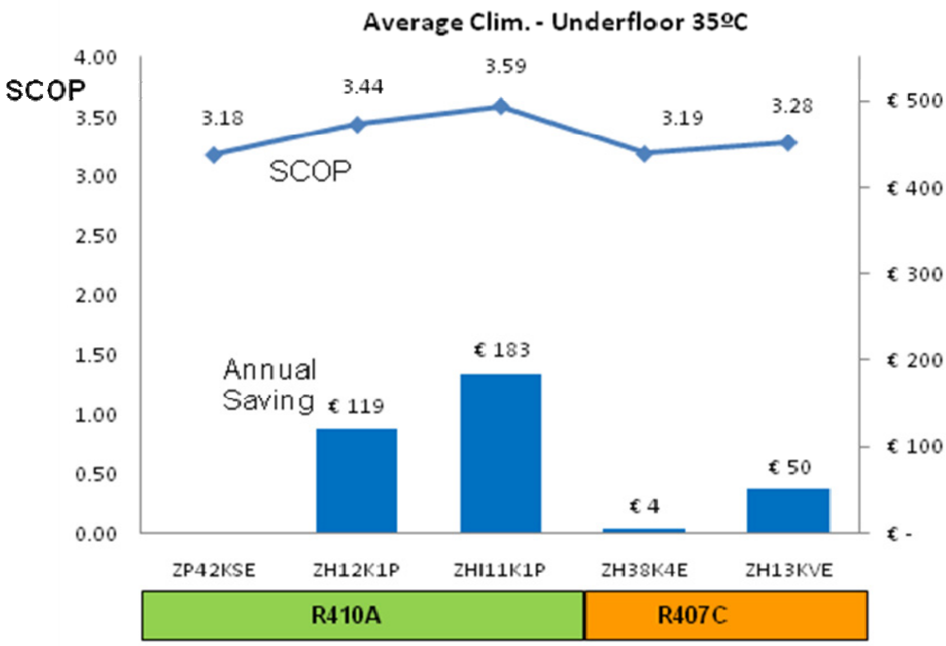

Figure 12: SCOP values and corresponding annual savings in the average climate. The cost baseline is ZP42KSE

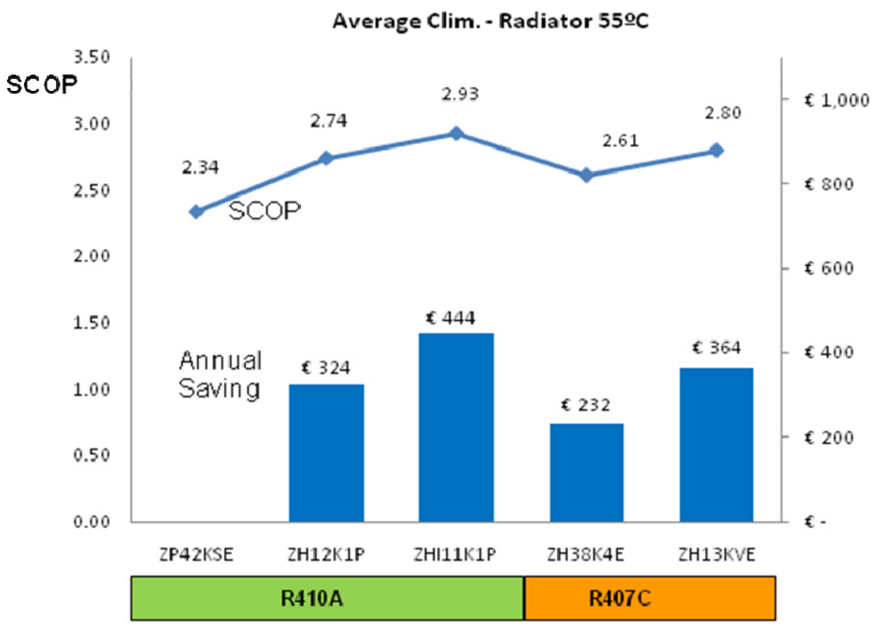

Figure 13: SCOP values and corresponding annual savings in the average climate. The cost baseline is ZP42KSE

8 CONCLUSIONS

A direct refrigerant to refrigerant properties comparison is insufficient to determine the best refrigerant for a given heat pump application. The compressor influence as well as the system influence will not vary always in the same direction and a detailed analysis is needed. In this paper, it has been shown how system design choices can be directly influenced by the inherent properties of the refrigerant and maximize their beneficial effect.

As a result, it has been shown that R410A has the potential to outperform R407C in most applications when using optimized scroll technology and when considering the most favourable boundary conditions in each case.

- Vapour injection give a good SCOP advantage at highest lift and therefore gives most benefit with air source/radiator systems
.
- When applying standard air conditioning scrolls with air source systems, direct heating is required at low air temperatures because of envelope limits, and this reduces SCOP.

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AN EXPERIMENTAL STUDY ON HEAT PUMP CYCLE USING ZEOTROPIC BINARY REFRIGERANT OF HFO-1234ze(E) AND HFC-32

Shigeru Koyama, Professor, Faculty of Engineering Sciences, Kyushu University, Kasuga, Fukuoka 816-8580, Japan
Nobuo Takata, Senior Technical, Interdisciplinary Graduate School of Engineering Sciences, Kyushu University, Kasuga, Fukuoka, 816-8580, Japan
Sho Fukuda, Graduate Student, Interdisciplinary Graduate School of Engineering Sciences, Kyushu University, Kasuga, Fukuoka, 816-8580, Japan

Abstract: In the present study the performance tests on heat pump cycle have been carried out for a near-azeotropic refrigerant R410A,a pure refrigerant HFO-1234ze (E) and zeotropic binary refrigerant mixtures of HFO-1234ze (E) and HFC-32 at heating mode, using a compressor developed for R410A. It is confirmed that the COP value and the heating capacity of pure HFO-1234ze(E) are the lowest among the refrigerants tested in the present study due to its low vapor density and latent heat. Also, due to the same reason, the pressure drops in condenser and evaporator of HFO-1234ze(E) are the highest among the tested refrigerants. It is also found that adding HFC-32 into HFO-1234ze(E) dramatically improves not only the COP value but also heating capacities; the COP values of 20mass%HFO-1234ze/80mass%HFC-32 mixture are almost the same as those of R410A at the same heating load. As a result, the present cycle performance tests prove that mixtures of HFO-1234ze(E) and HFC-32 are strong candidates for replacing R410A in domestic heat pump systems.

Key Words: Experiments, Cycle performance, Binary refrigerant mixture, HFO-1234ze(E), HFC-32, R410A

1 INTRODUCTION

Since Midgeley synthesized chlorofluorocarbons (CFCs) successfully in 1928, CFCs and hydro-chlorofluorocarbons (HCFCs) had been widely used as working fluids (refrigerants) in air-conditioning and refrigeration industry for many years. However, in 1974, It was pointed out that CFCs could cause the depletion of the stratospheric ozone layer (Monica and Roland 1974), and then a trend to control the emission of CFCs and HCFCs internationally had reached a peak. As result, in 1987, the Montréal protocol for the phase–out of ozone depleting substances such as CFCs and HCFCs was concluded. In that situation, the development of new alternative refrigerants without the ozone depletion potential (ODP) have been proceeded, and hydro-fluorocarbons (HFCs) such as R134a, R410A and R407C were developed as alternatives of CFCs and HCFCs in 1990's. However, at the 1997 Kyoto Conference (COP3), it was determined that the product and use of HFCs should be regulated due to their high global warming potential (GWP). This requirement has forced us to develop environmentally acceptable alternative refrigerants with zero ODP and relatively low GWP. In the last decade, natural refrigerants such as i-C_4H_{10}, water, CO_2 and NH_3, have been attracting a great deal of attention all over the world because of their zero ODP and extremely low GWP. Some heat pump and refrigeration systems such as domestic refrigerators using i-C_4H_{10}, heat pump water heaters using CO_2 and refrigerated storehouses using NH_3/CO_2 have been put in practical use. However, appropriate alternatives for domestic air-conditioning systems still can not be found up to now.

In the above mentioned circumstance surrounding the air-conditioning and refrigeration industry, we have noticed from early stage on HFO-1234ze(E) (*Trans-1,3,3,3-Tetrafluoro-propene*) that was developed as a cover gas for casting process of Magnesium alloy because its zero-ODP and extremely low GWP. In the present study, we have carried out the cycle performance test in order to investigate the possibility to introduce HFO-1234ze(E) and/or its mixtures with R32 as low-GWP alternatives for domestic heat pump systems. The tested refrigerants were a near-azeotropic refrigerant R410A, a pure refrigerant HFO-1234ze(E) and binary zeotropic refrigerant mixtures of HFO-1234ze(E) and HFC-32.

2 EXPERIMENTAL APPARATUS AND METHOD

Figure 1 shows the schematic view of an experimental apparatus, which was used for the present experiments on the cycle performance of a domestic heat pump system. The experimental apparatus consists of a refrigerant loop, a sink water loop and a heat source water loop. The refrigerant loop is composed an inverter controlled hermetic type compressor, an oil separator, a double-tube type condenser, a liquid receiver, an electric expansion valve and a double-tube type evaporator. Using constant-temperature baths, the heat sink water and heat source water are supplied to the condenser and the evaporator, respectively. Four mixing chambers are installed between components in the refrigerant loop to measure the refrigerant pressure and temperature. The other four mixing chambers are installed in heat sink and heat source water loops to measure water temperatures at the inlet and outlet of the condenser and the evaporator. The specifications of the condenser and the evaporator are listed in Table 1. Both of the condenser and the evaporator are double-tube type and coiled, where the refrigerant flows inside the inner tube, while heat sink/heat source water flows in the annulus surrounding the inner tube.

Table 2 shows the experimental conditions at heating mode. The refrigerants tested in the present experiments are R410A, pure HFO-1234ze(E) and mixtures of HFO-1234ze(E)/HFC-

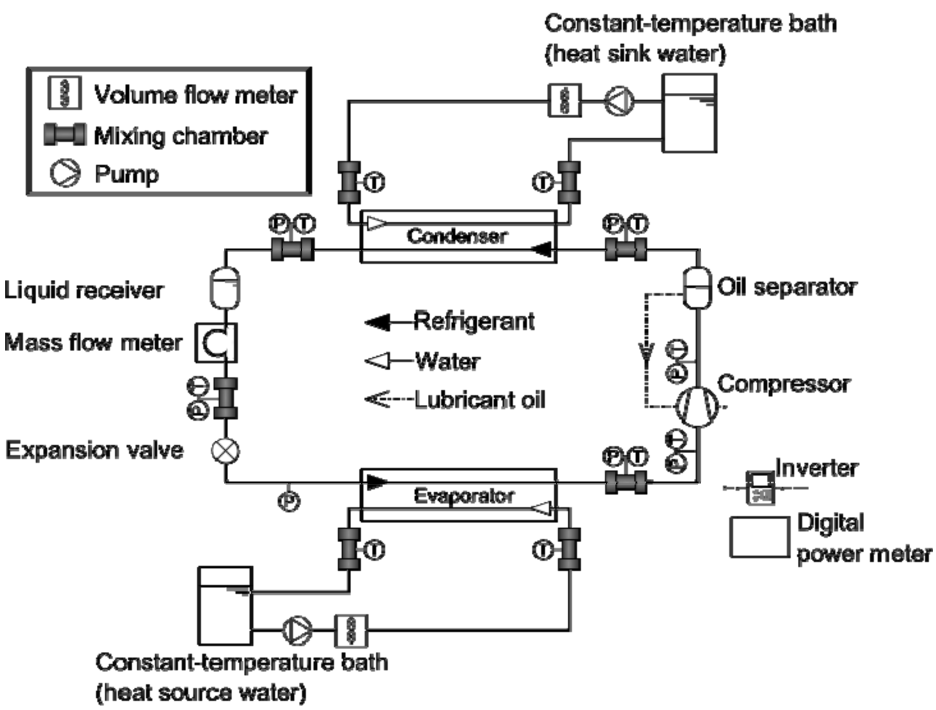


Figure 1: Schematic view of experimental apparatus

Table 1: Specifications of double-tube type heat exchangers

		OD (mm)	ID (mm)	Length (mm)	Type of tube
Condenser	Outer tube	12.7	10.7	5000	smooth tube
	Inner tube	9.52	7.53	5000	micro-fin tube
Evaporator	Outer tube	12.7	10.7	4500	smooth tube
	Inner tube	9.52	7.53	4500	smooth tube

Table 2: Experimental conditions at heating mode

	Refrigerant				
	R410A	HFO-1234ze(E)/HFC-32			
		20/80 mass%	50/50 mass%	80/20 mass%	100/0 mass%
GWP	675	542	342	142	9
Condenser: Heat sink water temperature [°C]	20 → 45				
Evaporator: Heat source water temperature [°C]	15 → 9				
Heating capacity [kW]	1.8~2.8	1.6~2.8	1.8~2.8	1.4~2.8	1.0~1.8
Degree of superheat [K]	3				

-32. The mass fractions of HFO-1234ze(E) in the mixtures are 20, 50 and 80 %. For all experiments the degree of superheat at evaporator outlet is fixed at 3 K. Temperatures of the heat sink water and heat source water are also fixed for all experiments as follows: heat sink water temperatures at inlet and outlet of condenser are kept at 20 °C and 45 °C, respectively, and heat source water temperatures at inlet and outlet of evaporator are kept at 15 °C and 9 °C, respectively, and the range of the heating capacity changes from 1 kW to 2.8 kW depending on kinds of refrigerants. The refrigerant charge amount is also changed as one of experimental parameters in the discharge-suction pressure range less than 6.

In the experiments, the heat pump system is operated to achieve a specified experimental condition by adjusting the compressor frequency and the opening of the electric expansion valve. As the system reaches a steady state as satisfying an experimental condition, the following physical quantities are measured directly using a data acquisition system:
(1) electric power inputs to the inverter and the compressor,
(2) refrigerant temperature and pressure in every mixing chambers and refrigerant temperature at the discharge port of the compressor,
(3) mass flow rate of refrigerant and volumetric flow rates of heat sink water and heat source water,
(4) temperatures of heat sink water and heat source water at the inlet and outlet of condenser and evaporator.
Then, heat transfer rates of condenser and evaporator are calculated from the water-side energy balance equations using the measured volumetric flow rate and temperatures of heat sink water and heat source water, and then the coefficient of performance (COP) at heating mode is obtained from the electric power input to the inverter and the heat transfer rate of condenser. Thermodynamic properties of R410A are calculated using the program package REFPROP Ver.8 (Lemmon et al. 2007), while those of pure HFO-1234ze(E) and mixtures of HFO-1234ze(E)/HFC-32 are calculated using the program package REFPROP Ver.8 combined with the HFO-1234ze(E) program package (Akasaka 2010). Figure 2 shows the

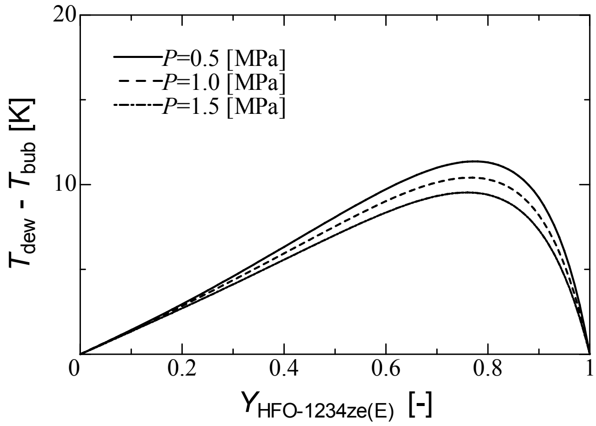


Figure 2: Temperature difference between dew and bubble points of HFO-1234ze(E)/HFC-32 mixture

calculated temperature difference between dew and bubble points of HFO-1234ze(E)/HFC-32 mixture for reference.

3 EXPERIMENTAL RESULTS AND DISCUSSION

Figure 3 shows the experimental results of R410A at heating mode, where the ordinates in Figs. 3(a) and (b) express COP and degree of sub-cooling at condenser outlet, respectively, while the abscissa in all figures represents the heat transfer rate of condenser (heating load). Symbols are employed to distinguish the refrigerant charge amount. In each case of refrigerant charge amount, the value of COP increases with increase of heating load, and it reaches a maximum and then decreases. In case of a constant heating load, the value of COP has a maximum at a certain refrigerant charge amount. The degree of sub-cooling increases with increase of refrigerant charge amount and it almost maintains a constant value when the heating load increases at a constant refrigerant discharge amount.

Figure 4 shows the experimental results of 100mass% HFO-1234ze(E) at heating mode, where the ordinates and abscissa in Figs.4(a) and (b) are the same as in Fig. 3., respectively, and symbols are also employed to distinguish the refrigerant charge amount. In some cases of refrigerant charge amount, the value of COP increases with increase of heating load and then decreases, while in other cases of refrigerant charge amount it decreases with increase of heating load. On the other hand, the degree of sub-cooling increases with increase of heat load in all cases of refrigerant charge amount. It is noted that the COP value has a maximum at a certain refrigerant charge amount. It was also found through the present drop-in experiments that the heating capacity of HFO-1234ze(E) was considerably lower than that of R410A

Figure 5 shows the experimental results of the mixture of 50mass% HFO-1234ze(E)/50 mass% HFC-32 at heating mode, where the ordinates and the abscissa in Figs. 5(a) and (b) are also the same as in Figs. 3(a) and (b), respectively. The relation between COP and heating load of this mixture is almost the same as in case of 100mass% HFO-1234ze(E), but the COP value of this mixture is higher than that of 100mass% HFO-1234ze(E) especially as the heating load increases. This mean that the addition of HFC-32 into HFO-1234-ze(E) improves the cycle performance effectively. In each case of refrigerant charge amount, the degree of sub-cooling increases moderately with the increase of heating load.

Figure 6 shows the experimental results to investigate the effects of the refrigerant charge amount on the cycle performance at the heating mode. The results of R410A, 100mass%

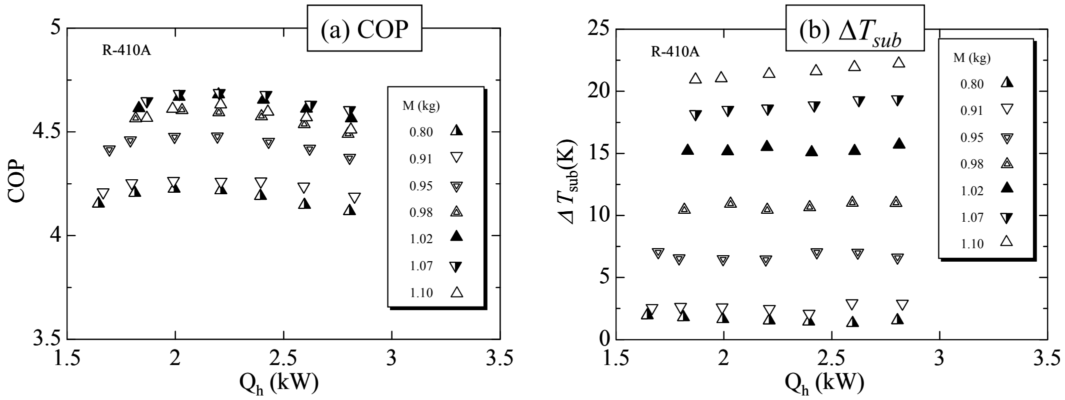


Figure 3: Experimental results of R410A

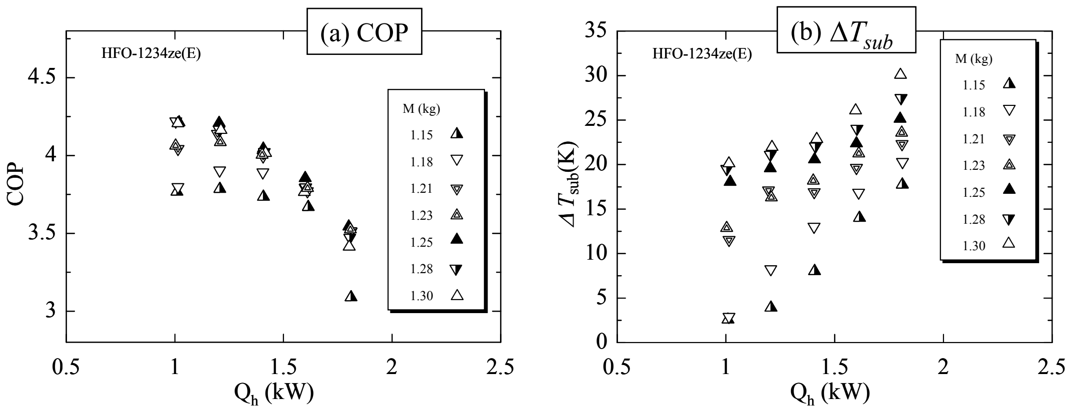


Figure 4: Experimental results of 100mass% HFO-1234ze(E)

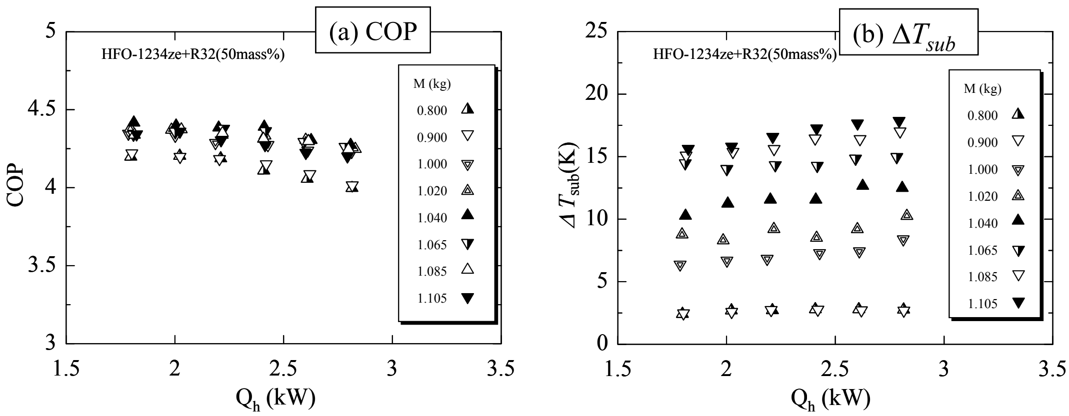
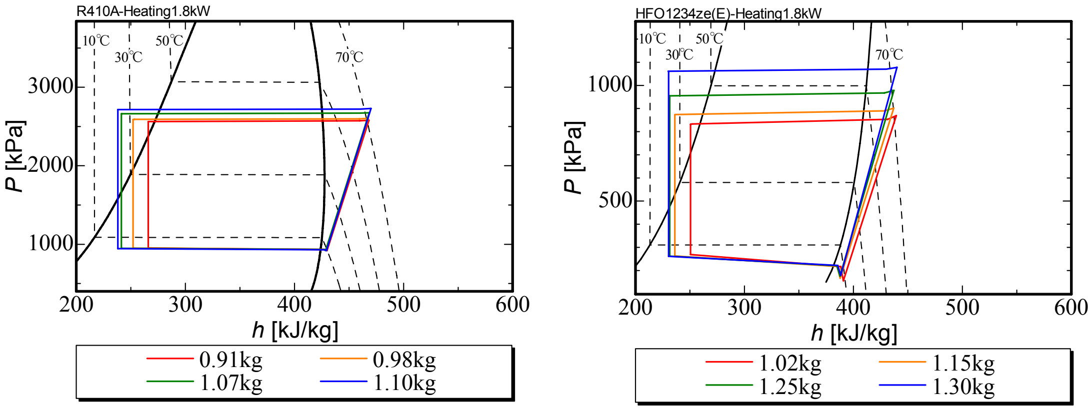


Figure 5: Experimental results of 50mass% HFO-1234ze(E)

HFO-1234ze(E), the mixture of 80mass% HFO1234ze(E) and the mixture of 20mass% HFO-1234ze(E) are plotted on the pressure–specific enthalpy diagrams shown in Figs. 6(a), (b), (c) and (d), respectively, where the difference of refrigerant charge amount is distinguished by colors of lines. In all cases of refrigerants, the change of specific enthalpy in the compressor, the condenser and the evaporator increases with the increase of refrigerant charge amount. It is also found that the pressure in condenser increases with increase of refrigerant charge amount, while the pressure in evaporator is little affected by the refrigerant charge amount. It is also found that the pressure drop of 100mass% HFO-1234ze(E) in evaporator is the largest among the test refrigerants.

Figure 7 shows the relation between COP and degree of sub-cooling at condenser outlet in case of a heating load (1.8 kW), where symbols of circle, square, triangle, inversed triangle and diamond represent the results of R410A, 20mass% HFO1234ze(E), 50mass% HFO-1234ze(E), 80mass% HFO-1234ze(E) and 100mass% HFO-1234ze(E), respectively. In all cases of test refrigerants, the COP value once increases with increase in the degree of sub-cooling and reaches a maximum. Then, it decreases with increase in degree of sub-cooling.



(a) R410A

(b) 100mass% HFO-1234ze(E)

(c) 80mass% HFO-1234ze(E)

(d) 20mass% HFO-1234ze(E)

Figure 6: Heat pump cycle plotted on P-h diagram

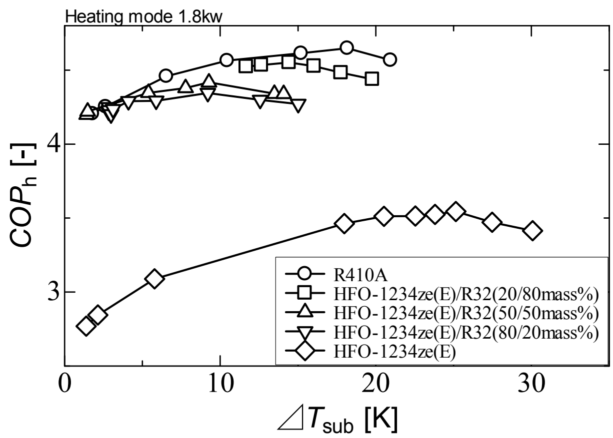
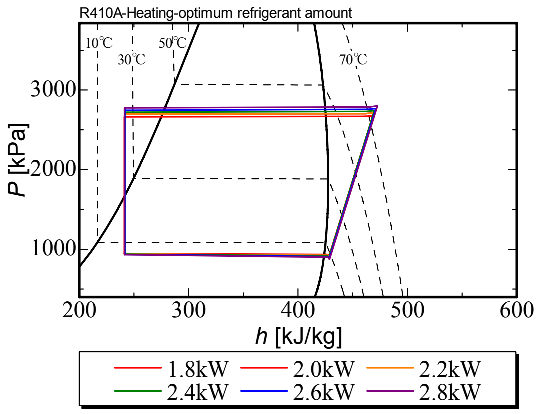


Figure 7: Relation between COP and degree of sub-cooling

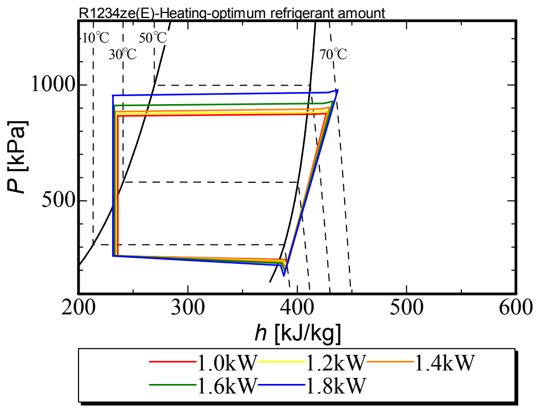
The optimum value of refrigerant charge amount is determined as the refrigerant charge amount at a maximum COP value. Compared with the maximum COP values of R410A, those of 100, 80, 50 and 20 mass% HFO-1234ze(E) are about 24 %, 7%, 5% and 1% lower, respectively. This means that 100 mass% HFO-1234ze(E) is not suitable for an alternative of R410A, but mixtures of HFO-1234ze(E) and HFC-32 are available for alternatives of R410A.

Figure 8 shows the heat pump cycles of test refrigerants, plotted on the pressure–specific enthalpy diagrams, where Figs. 8(a), (b), (c) and (d) are results of R410A, 100mass% HFO-1234ze(E), the mixture of 80mass% HFO1234ze(E) and the mixture of 20%mass% HFO-1234ze(E) at optimum refrigerant charge amounts, respectively. In all cases of test refrigerants, the condenser pressure and enthalpy change in condenser and compressor increase with increase of heating load. This trend is remarkable in the cases of 100 and 80 mass% HFO-1234ze(E).

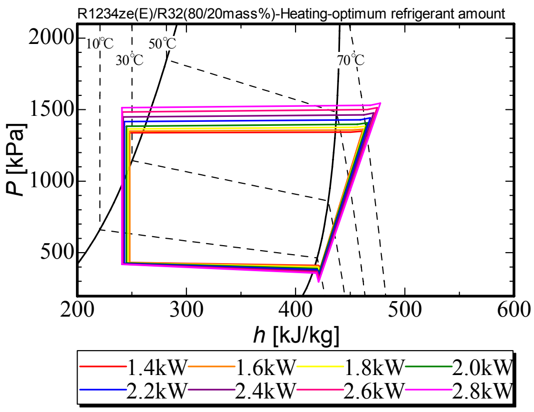
Figure 9 shows the relation between COP and heating load at optimum refrigerant charge amount, where symbols of circle, square, triangle, inversed triangle and diamond represent the results of R410A, 20mass% HFO1234ze(E), 50mass% HFO-1234ze(E), 80mass% HFO-1234ze(E) and 100mass% HFO-1234ze(E), respectively. The COP value of R410A is the highest among test refrigerants and is a little affected by heating load, while that of 100 mass% HFO-1234ze(E) is the lowest among test refrigerants and decreases considerably with increase of heating load. This also proves that 100 mass% HFO-1234ze(E) is not suitable for an alternative of R410A, but addition of HFC-32 into HFO-1234ze(E) is one of



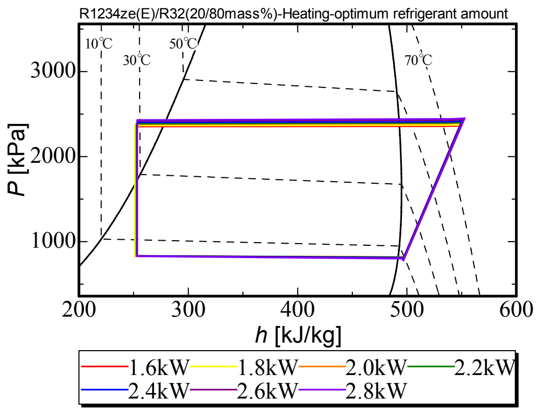
(a) R410A



(b) 100mass% HFO-1234ze(E)



(c) 80mass% HFO-1234ze(E)



(d) 20mass% HFO-1234ze(E)

Figure 8: Heat pump cycle potted on the P-h diagram

promising methods to introduce a Low GWP refrigerant as an alternative of R410A; the COP value of the 20mass% HFO-1234ze(E) mixture is only 1% lower than that of R410A.

Figure 10 shows the relation between refrigerant pressure drop in heat exchangers and refrigerant flow rate, where Figs.10(a) and (b) are the results of condenser and evaporator, respectively. In each figure symbols of circle, square, triangle, inversed triangle and diamond represent the results of R410A, 20mass% HFO1234ze(E), 50mass% HFO-1234ze(E), 80mass% HFO-1234ze(E) and 100mass% HFO-1234ze(E), respectively, and among of them the symbols red-circled are the results at heating load 1.8 kW. As well known, the refrigerant pressure drop in heat exchangers increases with increase of refrigerant flow rate in all cases of test refrigerants; the increase of refrigerant flow rate corresponds to the increase of heating load. At the same heating load (1.8 kW), the pressure drop of R410A is the smallest among test refrigerants, while that of 100 mass% HFO-1234ze(E) is the largest . The addition of HFC-32 into HFO-1234ze(E) reduces the pressure drop in heat exchangers.

Figure 11 shows the relation between refrigerant temperature and heat transfer rate in heat exchangers at heating load 1.8 kW, where Figs.11(a) and (b) are the results of condenser and evaporator, respectively. In each figure symbols of circle, square, triangle, inversed triangle and diamond represent the results of R410A, 20mass% HFO1234ze(E), 50mass% HFO-1234ze(E), 80mass% HFO-1234ze(E) and 100mass% HFO-1234ze(E), respectively, and the chain line denotes water temperature. In evaporator, the temperature difference

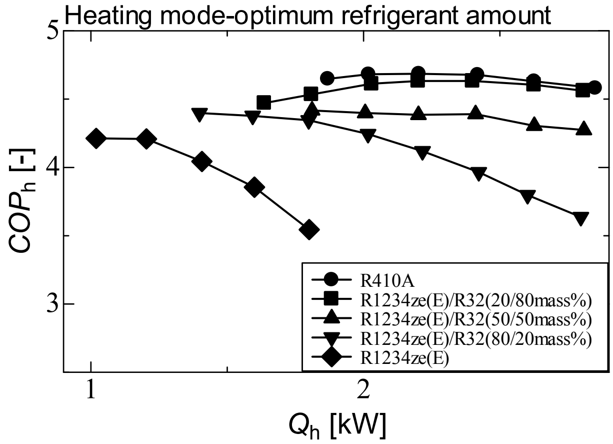


Figure 9: Relation between COP and heating load

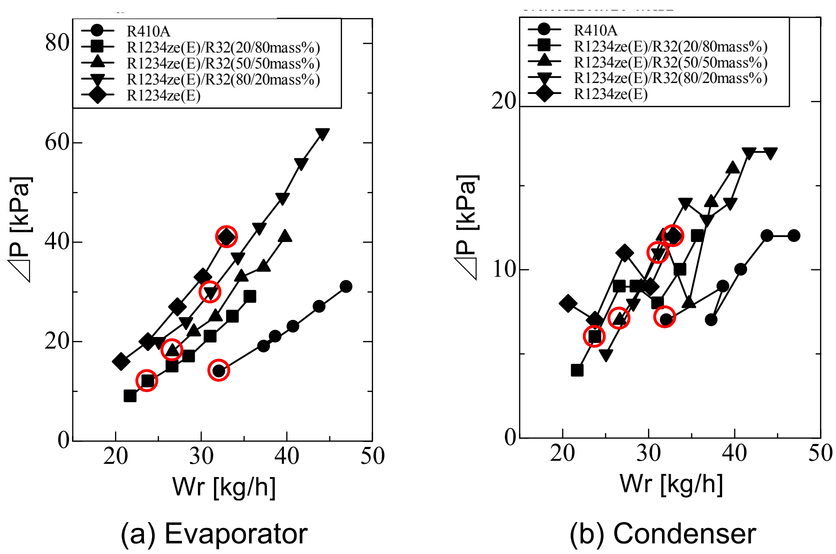
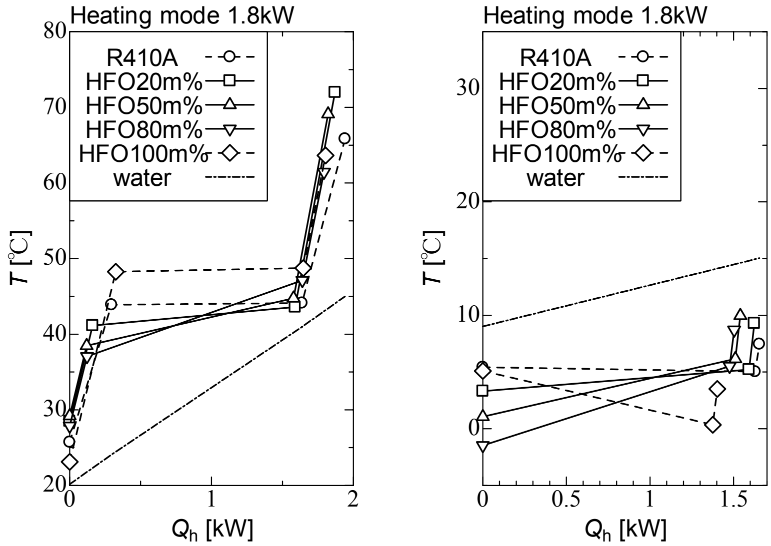


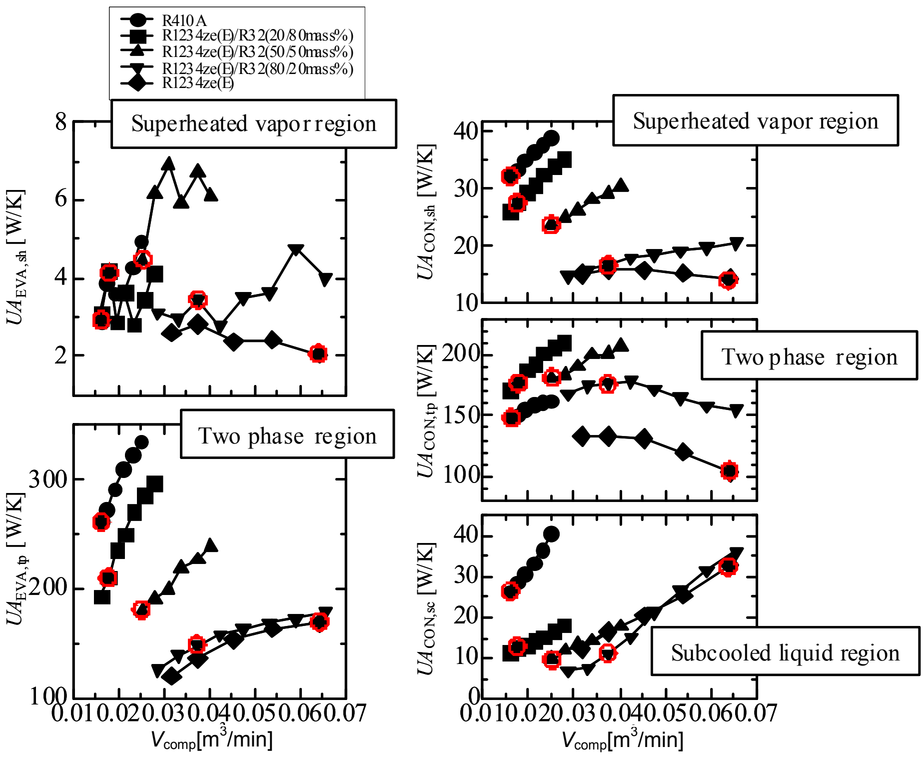
Figure 10: Relation between refrigerant pressure drop and mass flow rate

between refrigerant and water of R410A is the smallest among test refrigerants, and the pressure drop of 100mass% HFO-1234ze(E) is the largest. The refrigerant temperature of test mixtures rises in refrigerant flow direction overcoming the saturation temperature drop due to pressure drop increase. In condenser, the temperature difference between refrigerant and water of 100mass% HFO-1234ze(E) is the largest among test refrigerants, and the refrigerant temperature of test mixtures drops in refrigerant flow direction due to the temperature glide during phase change.



(a) Evaporator side (b) Condenser side

Figure 11: Relation between temperature and heat load in heat exchanger



(a) Evaporator side (b) Condenser side

Figure 12: Relation between UA and V_{comp}

Figures 12(a) and (b) show the heat exchange performance of test refrigerants in evaporator and condenser, respectively, where the ordinate denotes the overall heat transfer coefficient multiplied by the heat transfer area, and the abscissa denotes the volumetric refrigerant flow rate at the suction port of compressor. Symbols in both figures are the same as those in Figure 10, and red-circled symbols are the results at heating load 1.8 kW. Comparison among red-circled symbols suggests us the following heat transfer characteristics in two-phase region:

(1) Heat transfer performance in evaporator decreases, in order, R410A, 20mass%HFO-1234ze(E), 50mass% HFO-1234ze(E), 100mass% HFO-1234ze(E) and 80mass% HFO1234ze(E). It is inferred that the nuclear boiling of 100mass% HFO-1234ze(E) is suppressed by the forced convection effect, and those of HFO1234ze(E) mixtures are suppressed by the mass transfer resistance in the liquid surrounding vapor bubbles.
(2) Heat transfer performance in condenser decreases, in order, 20mass%HFO-1234ze(E), 50mass%HFO-1234ze(E), R410A, 80mass%HFO-1234ze(E), and 100mass% HFO1234ze(E). Heat transfer performances of mixtures are relatively higher than pure refrigerants. This is mainly due to temperature glide effect of mixtures.

4 CONCLUDINGS REMARKS

The main findings in the present study are summarized as follows:

(1) Pure HFO-1234ze(E) is not suitable for an alternative of R410A, but the addition of HFC-32 into HFO-1234ze(E) are one of promising methods to introduce a Low GWP refrigerant as an alternative of R410A; the COP value of the 20mass% HFO-1234ze(E) mixture is only 1% lower than that of R410A.
(2) The addition of HFC-32 into HFO-1234ze(E) is also effective for the reduction of the pressure drop in heat exchangers.
(3) The boiling heat transfer of 100mass% HFO-1234ze(E) may be the lowest among test refrigerant due to suppression of nuclear boiling by the forced convection effect, and the nuclear boiling of HFO1234ze(E) mixtures may be suppressed by the mass transfer resistance in the liquid surrounding vapor bubbles.
(4) Heat transfer performances of mixtures are relatively higher than pure refrigerants. This is mainly due to temperature glide effect of mixtures.

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A STUDY OF INDIRECT GHG EMISSIONS FOR NORTH AMERICAN RESIDENTIAL HEATING AND COOLING EQUIPMENT

Robert B. "Dutch" Uselton, Senior Principal Engineer, Lennox Industries Inc., Carrollton, TX, USA

Abstract: R&D that improves the energy efficiency of residential heating and cooling equipment will reduce site energy consumption, reduce the primary energy necessary to supply the site energy and also reduce the greenhouse gas (GHG) emissions attributable to heating and cooling of homes. This paper benchmarks the performance of common heating and cooling equipment sold in North America with respect to the equivalent CO2 emissions (CO2e) from their use. Operation in several North American cities is examined to see both the effects of local climate and the regional variation in the full-fuel cycle (FFC) power plant efficiency. It turns out that both are important to the understanding the likely GHG impact of heating and cooling equipment selection.

Key Words: efficiency, CO2, GHG, heat pump, furnace

1 INTRODUCTION

Heat pumps have the promise of reducing source energy requirements and CO2e emissions resulting from space conditioning of buildings. This study examines the net source energy, including so-called "precombustion" energy, for two typical residential class space conditioning systems:

- A natural gas furnace with an electric vapor compression air conditioner, and
- A vapor compression heat pump.

The direct and indirect CO2e emissions from annual operation of the equipment are also estimated. The climactic data for six U. S. cities are used and the regional variations in power system characteristics have been taken into account.

Two topics in particular are of current interest to the heat pump design community:

- How significant is the GWP of the refrigerant when estimating life-cycle CO2e emissions, and
- What might be a good target efficiency for a cold-climate heat pump for the northern US and Canada?

Information on these two questions is provided.

2 METHODOLOGY

Heating and cooling systems that combust fossil fuels have direct greenhouse gas emissions; CO2 is a natural byproduct of combustion. They have indirect emissions from the fossil fuel energy consumed in the extraction, processing and transport of the fuel. Moreover, virtually all residential heating and cooling systems consume electricity. There can be substantial indirect greenhouse gas emissions because of the fossil fuel energy used throughout the steps from fuel extraction, fuel transport and electrical generation at the power plant. The extraction process releases other gases, methane in particular, with non-zero global warming potential, GWP. The accounting of all these *equivalent CO2* emissions,

CO_2e, has become a manageable task as data has become available for the various processes that generate them. It is possible to work backward from the electrical and fossil fuel energy consumed on-site to make reasonable estimates of the CO_2e emissions related to that energy use.

2.1 Heating Equipment

In North America, there are a variety of residential space heating systems. Most common is the natural gas-fired warm air furnace. The next most popular system is the electric heat pump. Boiler systems are relatively uncommon except in the northeastern US. In the United States, the Department of Energy's Appliances and Commercial Equipment Standards Program develops test procedures and minimum efficiency standards for residential appliances.

2.1.1 Gas Furnaces

Residential warm air natural gas-fired furnaces are covered by the US Department of Energy's Uniform Test Method for Measuring the Energy Consumption of Furnaces and Boilers (US DOE, Furnaces 1997). The annual fuel utilization efficiency (AFUE) of these products is required to meet a minimum standard of 78%. This metric estimates the seasonal efficiency of natural gas utilization and is based upon the higher heating value of the fuel. It does not consider the electrical energy consumption of the furnace. It is expected that, in the near future, portions of the country with a large number of heating load hours (HLH) will have the minimum AFUE raised to 90%. Canada has a parallel test standard and also uses AFUE for furnace ratings.
The direct CO_2 emissions from warm air furnaces can be calculated just knowing the heating energy supplied to the home over the season, the efficiency (AFUE) of the furnace, and the rate of CO_2 production characteristic for burning natural gas in this type of system. There are also indirect emissions from the furnace's electrical consumption, and from the various emissions that occurred in delivering the natural gas and electricity to the home.

2.1.2 Heat Pumps

In the United States, residential heat pumps are covered by the US Department of Energy Test Procedure for Residential Central Air Conditioners and Heat Pumps (US DOE, Heat Pump 2005). The seasonal heating efficiency of a heat pump is described by the heating seasonal performance factor (HSPF). It is a descriptor with mixed units but, if divided by the number 3.412, gives the dimensionless heating coefficient of performance (HCOP). The HSPF is computed differently for different regions of the United States. The most commonly quoted HSPF is for Region IV, which corresponds to the middle latitudes of the country. The same set of test data is used to calculate the HSPF for the six different regions, the difference comes from how the weightings are changed between the low temperature and higher temperature tests. Note that the HSPF calculation has, built-in, an assumption that auxiliary resistance heat is used when the heat pump does not have sufficient heating capacity. What this means is that the same piece of equipment will have a lower HSPF rating in the colder regions and a higher HSPF in milder regions of the country. Sales of heat pumps in Canada rely on HSPF ratings. The most appropriate "region" designation for most of populous Canada is the US Region V.
Heat pumps can have direct greenhouse gas emissions only if the refrigerant used has a non-zero global warming potential (GWP) and some of the refrigerant leaks from the system. (In the United States, the Environmental Protection Agency requires that R-22 and R410A, the two most common refrigerants, be recovered during servicing and at the end-of-life of the equipment.) Indirect greenhouse gas emissions arise from the use of electricity and from many of the processes related to getting the electricity to the home.

2.2 Cooling Equipment

In the United States, residential air conditioners, including heat pumps operating in cooling mode, are tested and rated according to the same heat pump standard mentioned above. The efficiency descriptor for cooling is called the seasonal energy efficiency ratio (SEER). It is a mixed unit descriptor but, when divided by 3.412, is the dimensionless cooling coefficient of performance (CCOP). The US national minimum SEER is 13 however, for new cooling equipment installations in warm regions of the country it may be raised to 14.
As with heat pumps, direct greenhouse gas emissions can only come from inadvertent leaks of refrigerant. Indirect greenhouse gas emissions can be estimated by working backward from the site electrical energy used to meet the seasonal cooling load, taking into account all of the emissions factors related to getting that electrical energy to the site.

2.3 Climatological Data

Table 1 lists six US cities spanning a range of North American climates. The cities are arranged by US Department of Energy Heat Pump Region number. Heating and cooling design temperatures are listed to indicate the range of winter and summer conditions. The design temperatures are the 99% heating and 2% cooling design temperatures from the 2009 edition of the ASHRAE Handbook of Fundamentals. The assumed summer heat gain rate for the house is 0.607 kW/°C. The cooling design load is calculated by multiplying this heat gain rate times the difference between the design temperature and an 18.3°C (65°F) balance point temperature. The cooling design loads indicate the minimum-sized cooling system needed for the homes. The heating design loads for this study were calculated by following the method for estimating the design heat requirement (DHR) found in the US DOE Test Procedure for Residential Central Air Conditioners and Heat Pumps (US DOE, Heat Pump 2005). Two assumptions were made:

- The 8.3°C heat pump heating capacity is equal to the 35°C cooling capacity (As it turns out, this is frequently true, within +/- 5%.)
- The home's design heat requirement is midway between the Procedures' minimum and maximum DHR values (the calculation method generates a minimum and a maximum)

When these heating design loads are used to generate seasonal heating energy requirements, they give values that are reasonably consistent with home heating energy needs determined from analysis of actual home energy consumption. As might be expected, the variation in heating and cooling loads from home to home, within a region, can vary considerably. Fortunately, the conclusions drawn from this study are not significantly affected by this.

Table 1: Seven Cities Considered in the Analysis

City	State	DOE Region	Cooling Design Temp. (°C)	Heating Design Temp. (°C)	Cooling Design Load (kW)	Heating Design Load (kW)
Orlando	Florida	I	32.8	2.8	8.8	6.2
Ft. Worth	Texas	II	36.0	-2.8	10.7	10.0
Nashville	Tennessee	III	32.2	-8.3	8.4	10.6
Indianapolis	Indiana	IV	30.2	-15.0	7.2	13.2
Minneapolis	Minnesota	V	29.4	-23.3	6.7	16.9
Portland	Oregon	VI	28.6	-1.1	6.2	6.2

2.4 Heating and Cooling Load Estimation

The annual cooling and heating loads are determined using the US Department of Energy Test Procedure for Residential Central Air Conditioners and Heat Pumps (US DOE, Heat

Pump 2005). The procedure provides cooling and heating load hours for each of the defined regions. The annual cooling load is equal to the cooling load hours (CLH) times the cooling design load. The annual heating load, by this procedure, is equal the heating load hours (HLH) times the heating design load times a correction factor of 0.77. In the procedure, the factor is justified as an adjustment that "tends to improve the agreement between calculated and measured building loads."

Table 2:Annual Heating and Cooling Loads for a Typical Home in Seven Cities

City	DOE Region	Annual Cooling Load (kWh)	Annual Heating Load (kWh)
Orlando	I	21100	3600
Ft. Worth	II	19000	9600
Nashville	III	10500	14200
Indianapolis	IV	5600	22900
Minneapolis	V	2800	35700
Portland	VI	1200	13000

2.5 Site Energy Consumption

In developing estimates of annual greenhouse gas emissions for residential cooling and heating systems in North America, it is necessary to work backward from the site energy consumption. In this study, two different classes of equipment are being considered: electric cooling with warm-air natural gas furnace heating <u>and</u> electric heat pumps with auxiliary electrical resistance heat. The efficiency level of the equipment must be specified in order to calculate losses and determine the required input energy. The equipment selections reflect what might be the most likely equipment that would be chosen for installation.

2.5.1 Electric Cooling with Natural Gas Heating

With the SEER level of the air conditioner and the seasonal cooling load both known, it is straightforward to first figure the cooling COP and then the amount of electrical energy consumed to do the work. Table 3 summarizes this information. Note that Indianapolis, Minneapolis and Portland use 13 SEER air-conditioners while the other cities use 14 SEER equipment. The right-most column provides the seasonal cost of operation if the cost of a kWh is equal to the latest official nationally-averaged residential price published by the US Department of Energy (US DOE, Residential Energy Costs).

Table 3: Cooling Season Electrical Energy Use for Electric Air Conditioner

City	DOE Region	Mini-mum SEER	Corres-ponding Seasonal CCOP	Cooling Load (kWh)	Cooling Electrical Energy Use (kWh)	Cooling Season Operating Cost @ $0.115/kWh ($)
Orlando	I	14	4.10	21100	5143	591
Ft. Worth	II	14	4.10	19000	4629	532
Nashville	III	14	4.10	10500	2571	296
Indianapolis	IV	13	3.81	5600	1477	170
Minneapolis	V	13	3.81	2800	738	85
Portland	VI	13	3.81	1200	324	37

The calculations for heating season furnace operation are similar except that there are two energy sources that must be accounted for. Table 4 summarizes heating season results. Note that Indianapolis, Minneapolis and Portland use 90% efficient (90 AFUE) furnaces. The furnaces for homes in Orlando, Ft. Worth and Nashville are 80% efficient (80 AFUE) non-condensing type furnaces. The natural gas cost used is the nationally-averaged residential price published by the US Department of Energy (US DOE, Residential Energy Costs).

Table 4: Heating Season Electrical and Natural Gas Energy for Warm-Air Furnace

City	DOE Region	Min. AFUE	Furnace Electrical Power (kW)	Heating Load (kWh)	Natural Gas HHV Fuel Input (kWh thermal)	Heating Electrical Energy Use (kWh)	Heating Season Op. Cost @ $0.115/kWh & $0.04074/kWh (thermal)
Orlando	I	80	0.5	3600	4400	375	224
Ft. Worth	II	80	0.6	9600	12100	750	578
Nashville	III	80	0.5	14200	17800	875	825
Indianapolis	IV	90	0.4	22900	25400	900	1138
Minneapolis	V	90	0.6	35700	39700	1650	1808
Portland	VI	90	0.4	13000	14500	1100	716

2.5.2 Heat Pump

For the case of heat pumps, the efficiency combination of 14 SEER and 8.5 HSPF will be used for all cities. The corresponding cooling COP for 14 SEER level is 4.103. The cooling COP can be divided into the seasonal cooling load to give the cooling season electrical energy use. In Table 6 an estimate of the cooling season energy usage and operating cost are provided.

Table 5: Cooling Season Electrical Usage for Heat Pump

City	DOE Region	SEER	Cooling COP	Cooling Load (kWh)	Cooling Electrical Energy Use (kWh)	Cooling Season Operating Cost @ $0.115/kWh ($)
Orlando	I	14	4.103	21100	5143	591
Ft. Worth	II	14	4.103	19000	4629	532
Nashville	III	14	4.103	10500	2571	296
Indianapolis	VI	14	4.103	5600	1371	158
Minneapolis	V	14	4.103	2800	686	79
Portland	VI	14	4.103	1200	301	35

Table 6 examines heat pump energy usage. The Region IV 8.5 HSPF designation assumes the minimum design heat requirement in its calculation. For other design heat requirements and for other regions, the HSPF must be recalculated. The design heat requirement is important. It is customary to use a heat pump with oversized cooling capacity in northern climates because of the savings that comes about from higher heating season heat pump output. (This is because less auxiliary resistance heat is needed to meet the load.) Table 6 shows the adjusted HSPF values appropriate for each region, cooling oversize factor and assuming a mid-range design heat requirement. The heating electrical energy use and heating operating cost are also shown.

Table 6: Heating Season Electrical Usage for Heat Pump

City	DOE Region	Region IV (min DHR) HSPF	"Cooling Oversize Factor"	HSPF for mid-DHR, Region, Sizing Factor	Heating COP	Heating Load (kWh)	Heating Electrical Energy Use (kWh)	Heating Season Operating Cost @ $0.115/kWh ($)
Orlando	I	8.5	1	10.59	3.10	3600	1144	130
Ft. Worth	II	8.5	1	9.96	2.92	9600	3300	380
Nashville	III	8.5	1	9.27	2.72	14200	5200	600
Indianapolis	IV	8.5	1.25	8.21	2.41	22900	9500	1100

Minneapolis	V	8.5	1.5	6.73	1.97	35700	18000	2100
Portland	VI	8.5	1.14	10.27	3.01	13000	4300	500

2.6 Energy Extraction, Generation and Distribution Effects

Recently, considerable effort has gone into quantifying the upstream energy consumption and GHG emission effects of the North American energy delivery system. There are national metrics for quantities such as a pre-combustion CO2e factor linked to site-consumed natural gas and regional and national electric power transmission and distribution loss factors. Taking into account these upstream effects has been termed Full Fuel Cycle (FFC) analysis. A report summarizing many of these metrics that may be useful to building engineers is a publicly available publication from NREL (Deru and Torcellini 2007).

2.6.1 Electricity

The electrical distribution system for the continental US is subdivided into three large "interconnections". Alaska and Hawaii have their own systems. Mexico and Canada have separate electrical grids that are not significantly interconnected with the large US system. The "interconnections" are managed by the North American Electrical Reliability Council (NERC). The Western Interconnection encompasses the western US and part of western Canada. The ERCOT Interconnection covers most of Texas. The Eastern Interconnection takes in the remainder of the continental US, west of the Rocky Mountains, and part of Canada. The average electrical transmission loss factor for the aggregate NERC system is 9.9%. ERCOT has a loss factor of 16.1%, the Western, 8.4% and the Eastern, 9.6% (Deru and Torcellini 2007). The fuel/energy sources and types of generating equipment have different proportions in the different interconnects and this is reflected in the regional source energy factors used to estimate the full fuel cycle. The western interconnect uses higher fractions of hydroelectric and natural gas-fired generation and lower fractions of nuclear and coal-fired generation, compared to the other interconnects. Figure 1 shows the full fuel cycle source energy requirements to operate these two different representative cooling/heating systems in the six different cities.

In the process of generating electricity, CO2 is emitted by fossil fuel power plants. There are other greenhouse gas emissions, as well, and it is useful to lump these in with the CO2, on an "equivalent global warming potential" basis, to give an equivalent CO2, or CO2e. This macro-scale assessment looks at all CO2e emissions from the generating stock under consideration, i.e. the electrical energy output from nuclear and hydroelectric power plants is included even though these plants have low or no CO2 emissions. The aggregate NERC system produces 0.620 kg CO2e per kWh of generated electrical output. The corresponding emission factors for the three large interconnects are: ERCOT 0.624, Western 0.486, and Eastern 0.650 (Deru and Torcellini 2007).

There is another part of the electrical power system that has significant CO2e emissions although they are more difficult to aggregate since they come from many smaller sources. These emissions are the result of extracting the power plant fuel from the earth, processing it and then transporting the fuel to the generating facilities. These are termed "precombustion" emission factors. The aggregate NERC system has a precombustion emission factor of 0.0699 CO2e per kWh of generated electricity. That is to say, it is about 1/10 of the generation emission factor. The corresponding precombustion emission factors for the three large interconnects are: ERCOT 0.0947, Western 0.0625 and Eastern 0.0688 (Deru and Torcellini 2007). Finally, to convert emissions related to generation into emissions related to site consumption, the electrical transmission loss factors must be applied.

2.6.2 Natural Gas

Natural gas is composed mainly of methane. It has several other components and the mixture varies over time and geography in North America. In this study, a higher heating value of 37,631 kJ/cubic meter (1010 BTU/cubic foot) will be used. Coincidentally, as with the electrical distribution network, the energy expended to deliver the commodity to the site is around 9%.
CO2e emissions from combustion of natural gas vary according to the combustion system used. In this study, the warm air furnace is the device used for on-site combustion. The on-site combustion CO2e emission factor for a residential furnace burner is 1.93 kg per cubic meter of natural gas. (Less than 1% of this factor comes from non-CO2 emissions.)

All of the non-combustion CO2e emission effects for site-delivered natural gas have been summed and tabulated (Deru and Torcellini 2007) as a “precombustion” emission factor. This factor includes the emissions related to extraction, processing and delivering the fuel. Their estimate is a value of 0.446 kg CO2e per cubic meter of delivered fuel. This is approximately one fourth of the value of the combustion emissions from a warm air furnace.

2.7 Full Fuel Cycle Source Energy

Figure 1, below, shows the full fuel cycle source energy requirements given the previously calculated annual site energy consumption of these systems in the six cities. (The efficiency of delivery of natural gas is assumed to be the same for all cities.) The characteristics of the three NERC interconnects’ generation mix and fuel types have been accounted for in the calculation of source energy factors for the electricity used. Energy from renewable energy sources is not considered in the source energy accounting since minimization of non-renewable energy use is the objective for equipment efficiency goals. Note that if national average electrical source energy factors had been used instead, projected source energy requirements, especially for the heat pump, would be overestimated because the Western interconnect has considerable hydroelectric resources.

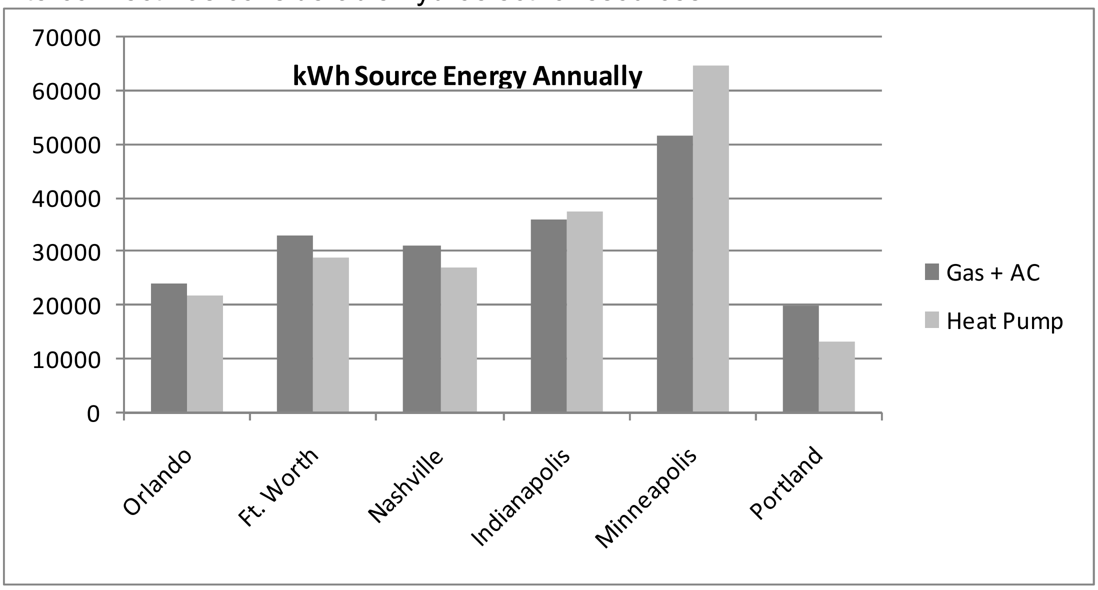

Figure 1: Source Energy Requirements for Two Different Heat/Cooling Systems

2.8 Equivalent CO2 Emissions

Using the identified CO2e emission factors, estimates of direct plus indirect greenhouse gas emissions can be determined for the candidate systems. The emissions include so-called precombustion emissions and account for electrical transmission losses. Table 7 shows the results when national average emission factors are used.

Table 7: Annual kilograms CO2e Emissions Using National Average Emission Factors

City	AC + Furnace Annual CO2e Emissions (kgs)	Heat Pump Annual CO2e Emissions (kgs)
Orlando	5192	4765
Ft. Worth	6817	6012
Nashville	6652	5915
Indianapolis	7575	8238
Minneapolis	10842	14257
Portland	4370	3508

These calculations are repeated in Table 8 but using the NERC interconnect CO2e emissions factors instead of national average emission factors.

Table 8: Annual Kilograms CO2e Emissions Using NERC Interconnect CO2e Emission Factors

City	AC + Furnace Annual CO2e Emissions (kgs)	Heat Pump Annual CO2e Emissions (kgs)
Orlando	5357	4954
Ft. Worth	7226	6615
Nashville	6756	6149
Indianapolis	7646	8564
Minneapolis	10913	14821
Portland	4217	3013

Table 8 CO2e emission estimates best represent reality. As stated in section 2.1.2, CO2e direct emissions from air conditioners and heat pumps only occur if the refrigerant has non-zero GWP and if some refrigerant escapes to the atmosphere. One can overlay the CO2e effects of refrigerant leakage during the life of this class of equipment if a few assumptions are made. For this study, the assumptions about refrigerant leakage have been taken from a 2002 study by Arthur D. Little Inc. (A. D. Little, 2002). They are:

- R410A with 2088 GWP
- 15 year equipment life
- 2% leak rate per year (fleet average)
- 15% loss of refrigerant at end of life (average over all systems).

Additionally, an assumption was made that 0.5 kg refrigerant is required per kW cooling capacity. Figure 2 shows the lifetime CO2e emissions for the systems from Table 8. The dark line at the top of each bar is the CO2e contribution from direct refrigerant emissions, which average 3.9% of total emissions. The conclusion from this is that a system using an alternative refrigerant with zero GWP needs to be at least 96% as efficient in order for there to be net CO2e emission reductions.

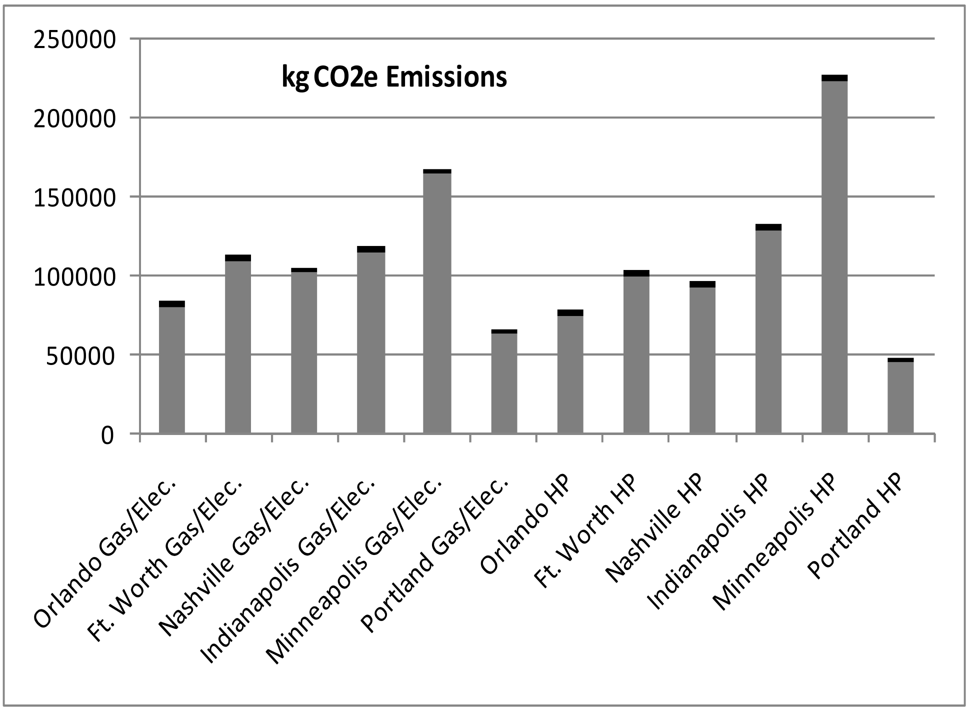

Figure 2: Chart showing 15 year lifecycle kg CO2e Emissions

3 Creating A New Benchmark for Heat Pumps in Cold Climates

Using national average energy costs, the annual operating cost of the heat pumps in this study are lower than the furnaces plus air conditioner operating costs for each city, aside from Minneapolis. (For some cities the operating costs are close to the same.) The CO2e emissions are lower for heat pumps in all cities except Indianapolis (it is close) and Minneapolis. These results suggest a higher efficiency heat pump is needed for cold climates.

A separate study was done looking at only Minneapolis, representative of the northern tier of the US and significant portions of Canada. In this study, the furnace used was a 98% efficient (98 AFUE) model and the cooling unit was kept at 3.81 COP (13 SEER). (While a higher efficiency model of air conditioner could have been used, the number of cooling hours is so low that it would not be a compelling choice.) The heat pump selected was a premium five ton two-stage model with nominal (Region IV) heating COP of 2.68 (HSPF 9+). This heat pump would be configured to use only the first cooling stage to handle the summer cooling load; the high cooling stage would be locked-out. By increasing the ultimate heat pump heating capacity, a considerable energy savings is achieved by displacing electrical resistance auxiliary heating. The results of this analysis are given in Table 9.

Table 9: Minneapolis Results for Best Available Gas/Electric AC and Heat Pump Equip.

Metric	Units	Gas Furnace + Elec. AC	Upsized 2-Stage Heat Pump
Cooling Efficiency	COP	3.81	5.10
Cooling Elec. Energy Used	kWh	706	528
Heating Efficiency	% Effic. / HCOP	98	2.47
Heating Gas Fuel Energy	kWh	35621	---
Heating Elec. Energy Used	kWh	1650	14146
Annual Operating Cost	$	1722	1687
Annual Source Energy	kWh	47012	50522
Annual CO2e Emissions	Kg CO2 equiv.	9956	11563

The best widely available heat pump technology still requires more source energy in cold climates. Annual operating costs are comparable. A heat pump with 10% higher efficiency would use less source energy than the premium furnace plus air conditioner system. A 16% heat pump performance improvement would yield CO2e emissions comparable to the premium furnace plus air conditioner. This suggests that a 20% heat pump heating performance improvement might be a good target for a cold climate heat pump R&D program.

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DEVELOPMENT OF A HIGH PERFORMANCE AIR SOURCE HEAT PUMP FOR THE US MARKET

Omar Abdelaziz, R&D Staff, Oak Ridge National Laboratory, Oak Ridge, TN, USA;[*]
Bo Shen, R&D Staff, Oak Ridge National Laboratory, Oak Ridge, TN, USA;
Zhiming Gao, R&D Staff, Oak Ridge National Laboratory, Oak Ridge, TN, USA;
Van Baxter, R&D Staff, Oak Ridge National Laboratory, Oak Ridge, TN, USA;
Ipseng Iu, R&D Engineer, Johnson Controls Inc., Wichita, KS, USA;

Abstract: Heat pumps present a significant advantage over conventional residential heating technologies due to higher energy efficiencies and less dependence on imported oil. The US development of heat pumps dates back to the 1930's with pilot units being commercially available in the 1950's. Reliable and cost competitive units were available in the US market by the 1960's. The 1973 oil embargo led to increased interest in heat pumps prompting significant research to improve performance, particularly for cold climate locations. Recent increasing concerns on building energy efficiency and environmental emissions have prompted a new wave of research in heat pump technology with special emphasis on reducing performance degradation at colder outdoor air temperatures. A summary of the advantages and limitations of several performance improvement options sought for the development of high performance air source heat pump systems for cold climate applications is the primary focus of this paper. Some recommendations for a high performance cold climate heat pump system design most suitable for the US market are presented.

Key Words: heat pumps, cold climate, multi-stage

1 INTRODUCTION

Air-source heat pumps (ASHPs) provide efficient heating by augmenting their energy consumption with heat collected from the ambient air and "pumped" to the required supply temperature. Reversed cycle air-conditioners were presented in the 1930's as a means to efficiently provide heating in buildings (Kerr Jr. *et al.* 1934, Neeson 1938, Brace and Crawford 1938, Labberton 1939). However, these systems where not introduced to the market before the 1950's and started to be reliable and economically feasible in the 1960's as described by Hiller 1976. This industry received an increasing interest following the 1973 oil embargo prompting significant research to improve performance, particularly for cold climate locations.

In late 1975 Carrier Corporation initiated an extensive heat pump research effort (Groff and Reedy 1978 and Groff *et al.* 1979). Four residential split-system ASHPs (based on the vapor-compression refrigeration cycle) located in Seattle, Minneapolis, Syracuse, and Boston were instrumented and monitored for a full year. The field tests illustrated that these heat pumps achieved significant energy saving compared to electric resistance heating systems. Additional studies pointed out the benefits of increasing heat pump capacity for colder

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climate locations despite negative impacts on cooling season performance (Groff *et al.* 1978 and Bullock *et al*. 1980). The average efficiency of residential heat pumps sold in USA increased 2.5% per year in 1980s (Calm 1987). In 1995, EPA introduced Energy Star specifications for residential heating and cooling products, including ASHPs. Today's Energy Star label is only awarded to ASHPs with a Seasonal Energy Efficiency Ratio (SEER) of 14-14.5 (cooling SPF 4.10-4.25) or higher and Heating Season Performance Factor (HSPF) of 8-8.2 (heating SPF 2.34-2.4) or higher[1]. The current most efficient air-source heat pumps have SEERs (cooling SPFs) of 20 (5.86) and higher while the SEER of a heat pump in 1979 was just 7. Thus the energy efficiency of modern air-source heat pumps is almost three times higher than those available 30 years ago. This great efficiency achievement has resulted from technical advances in vapor compression systems and components (e.g. compressors, heat exchangers, and flow control devices, etc.) as well as microprocessor-based control, variable-speed motors, etc., all achieved while making the switch from ozone depleting refrigerants to HFCs (Karen and Herold 1993). However, several issues still negatively impact ASHP heating performance under cold ambient conditions (Roth *et al.* 2009). First, ASHP heating capacity and COP significantly decrease as ambient temperature decreases. Second, ASHPs have the drawback of accumulating frost on outdoor coils, which deteriorates energy efficiency and lowers thermal comfort.

Over the last several decades, a number of technologies and design modifications have been proposed to improve the COP and heating capacity of cold climate heat pumps. Homes and buildings in cold climates usually require higher space-heating design loads than space-cooling design loads. In an effort to increase ASHP energy efficiency in cold weather, US manufacturers are gradually introducing new products specifically designed for better cold weather performance - Hallowell International (Acadia 2010) and Nyle Special Products (Hadely *et al.* 2006) are two examples. These new products use a combination of innovative technologies coordinated by the control systems to enhance their performance. For example, the Acadia™ cold climate heat pump uses a two-cylinder compressor to accomplish efficient multi-stage compression process.

The strategy of the multi-stage vapor injection compression cycle (with multiple compression stages) is becoming attractive to improve the COP and heating capacity of heat pumps at cold operating conditions. Theoretical and experimental results presented by US national laboratories and Universities reported that multi-stage vapor injection compression cycles achieved higher COP and capacity than single-stage cycles (Domanski 1996, Bertsh and Groll 2008, Mathison *et al.* 2011, and Wang *et al.* 2009) at cold ambient conditions. Domanski (1996) evaluated the thermodynamic performance for the ideal two-stage cycles, and conclude that the two-stage cycle improves the COP for every fluid, but the degree of COP improvement is larger for working fluids with large heat capacity. Bertsch and Groll (2008) simulated, designed and tested a two-stage heat pump using R410A at ambient temperature as low as -30°C and supply temperature of up to 50°C. Their heat pump was equipped with two compressors to operate low- and high-stage compression processes. The results show that a COP of 2.1 was achieved at -30°C ambient temperature with double the heating capacity of a conventional heat pump system running at the same conditions. To reduce the cost and system complexity, Wang *et al.* (2009) replaced the two compressors by a single multi-stage scroll compressor with refrigerant vapor injection ports. Vapor injected cycles showed 20% COP gain at -17.8°C ambient temperature versus that of the same cycle without injection. Mathison *et al.* (2011) from Purdue University theoretically analyzed the performance limit for multi-stage vapor compression cycle with continuous refrigerant injection. The results illustrated a COP increase varying from 18% to 51% depending on the refrigeration application, with larger temperature lift cycles benefiting most significantly. At least one residential air-source heat pump using multi-stage compression is already commercially available in US (Acadia 2010, Hadely *et al.* 2006, and Trane 2010).

[1] http://www.energystar.gov/index.cfm?c=airsrc_heat.pr_crit_as_heat_pumps

Use of CO_2 as refrigerant in a vapor compression cycle has also been investigated to improve the performance of cold climate heat pumps. A CO_2 refrigerant cycle can provide 35% greater capacity at low ambient temperature, which can decrease the use of electric resistance heating. CO_2 is considerably different from conventional refrigerants; its critical pressure and temperature are fairly low (7.38 MPa and 31.1°C respectively). Hence, CO_2 vapor compression cycles usually result in a transcritical cycle with subcritical low-side and supercritical high-side pressure (Kim *et al.* 2004). Prototype residential CO_2 heat pump systems have demonstrated higher capacity and comparable COP compared to R410A or R22 heat pump systems at lower outdoor temperatures (Kim *et al.* 2004). The higher capacity of the CO_2 system at lower outdoor temperatures has considerable impact when accounting for the overall seasonal system efficiency for an application, as the dependence on supplementary heating is reduced (Richter *et al.* 2003). Comparison between a R22 unit and a CO_2 prototype system show the CO_2 system achieves 20% better energy efficiency due to a lower need for supplementary heat (Richter *et al.* 2003 and Nekså 2002). The CO_2 technologies are still under development and have great potential for further improvements. A two-stage CO_2 compressor has shown a potential for 20% COP improvement (Kim *et al.* 2004), as was confirmed by Groll and Kim (2007). This new compressor technology is intended for lower-temperature refrigeration applications, but also is of interest to energy saving in air conditioning and heat pumping. Further CO_2 system developments include: using of microchannel heat exchanger, increasing the isentropic efficiency of the compressor, and using an ejector or expander for expansion work recovery. There has been extensive research in improved heat pump cycles and designs over the world; the above review was limited to the current efforts in the US.

2 AIR SOURCE HEAT PUMPS: THEORY AND ADVANCEMENT

The typical configuration of an ASHP consists of a compressor, indoor and outdoor coils (air-to-refrigerant heat exchangers), two expansion valves (one for cooling and one for heating), an accumulator, and a reversing valve. In cooling mode, the indoor coil is the evaporator and the outdoor coil is the condenser, and vice versa in heating mode. This kind of system is widely used in the southern part of the United States where the winter weather is mild. At the AHRI standard heating condition, i.e. 21°C dry bulb indoor and 8.3°C dry bulb/6.1°C wet bulb outdoor (AHRI 2008), a 7.7 HSPF (2.25 heating SPF) ASHP can operate at around 3.5 COP. For high efficiency systems, the COP can be increased to the 4.5 range if larger heat exchanger coils and electronically commutated motors (ECM) are used. Compared to the electric resistance heater, where the COP is always 1, the application of the heat pump has gained significant interest from end users.

However, the COP of ASHPs decreases quickly at low ambient conditions. At the AHRI low temperature heating condition, i.e. 21°C dry bulb indoor and -8.3°C dry bulb/-9.4°C wet bulb outdoor, the COP drops by 30 to 35%. The COP drop for high efficiency systems is less - about 25 to 30%. Even though heat pumps can still operate above 1 COP at low ambient conditions, the heating capacity provided is generally not enough for comfort. Therefore, in colder climate regions, ASHPs generally must use a secondary heat source (usually electric resistance heat, etc.). When the heating load demand is high and the heating capacity from the heat pump alone is not enough, the secondary heat source turns on to supplement the heat output and ensure a comfortable living space.

There are other system configurations that can extend the heat pump application range to lower ambient conditions. Here are few examples:

1. Booster system (Shaw 2007): In this kind of system, a booster compressor is connected in series with the primary compressor. The booster normally has less capacity than the primary compressor. Figure 1 compares the booster cycle to the conventional vapor compression cycle on a P-H diagram. The booster is off during

normal operation. When the ambient temperature is low enough and the primary compressor alone cannot satisfy the load demand, the booster turns on to provide extra heating capacity and boost up the overall system COP.

2. Vapor injection technology (Lifson 2005, Siddharth *et al.* 2004, and Wang *et al.* 2009, Heoa *et al.* 2010): The vapor injection technology creates a multi-stage compression. Figure 2 (a) shows a two-stage vapor injection cycle working principle on a P-H diagram. A phase separator such as flash tank is installed after the expansion valve. The liquid portion goes through a second expansion and circulates to the evaporator, while the vapor portion is injected back to the compressor and creates a second compression effect. The compression process used in this cycle can be accomplished with multiple single-stage compressors or one multi-stage compressor. Scroll compressors can be equipped with vapor injection ports resulting in a single multi-stage compressor. Using a single multi-stage compressor is more cost effective and results in simpler system configuration. The vapor injection cycle is a proven technology that can improve heating capacity and COP at low ambient conditions.
3. Ejector technology (Elbel and Hrnjak 2008): The ejector is used to recover some of the throttling loss at the expansion valve. An ejector takes the high pressure refrigerant from the condenser to be the motive fluid and the low pressure refrigerant from the evaporator to be the suction fluid. This is illustrated as the ejection line on Figure 2 (b). The ejector mixes both fluids and ejects the mixture to a phase separator. The liquid portion goes through an expansion valve and circulates to the evaporator, while the vapor portion goes to the compressor suction. Since the ejector mixed the high and low pressure refrigerants, the suction pressure to the compressor is increased. As a result the system performance is increased.

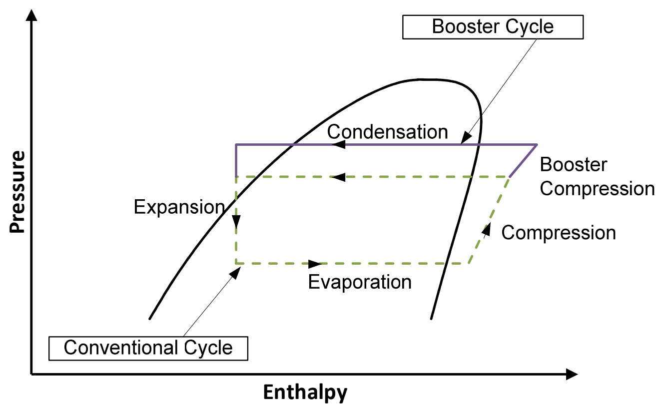

Figure 1: Comparison between conventional and booster cycles.

All of these technologies have a common effect that provides a temperature lift to the vapor compressor cycle; as a result, the heating performance can be improved. However, besides the booster system, the other two technologies have yet to be widely commercialized in the residential market. The recent development of these technologies creates an opportunity to develop a non-conventional, yet more efficient and wider application range ASHP.

Among these three technologies, the vapor injection technology is the most cost effective way to implement and can provide significant performance gain. In this paper we will focus on 3 multi-stage vapor injection cycle configurations: flash tank cycle, economizer heat exchanger (HX) cycle, and flash injection circuit cycle. We developed an in-house modeling tool and used that with an off-the-shelf multi-stage compressor map along with conventional indoor and outdoor heat exchanger designs to study the impact of operating and design parameters on the performance of these cycles. Following is a review on the different multi-stage cycles, simulation results, and discussion.

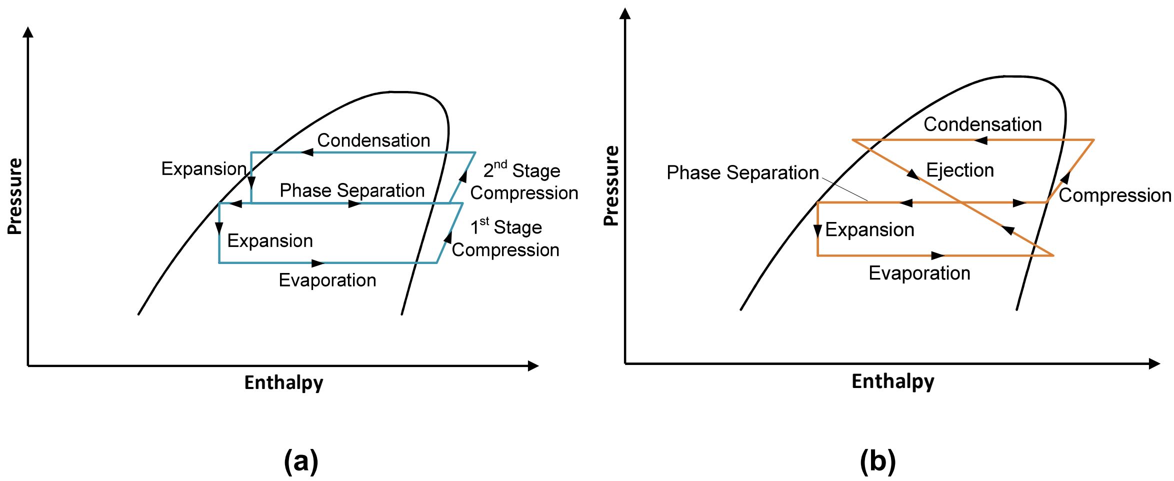

Figure 2: Advanced vapor compression cycles: (a) Vapor injection cycle, (b) Ejector cycle.

3 VARIATION OF VAPOR INJECTION CYCLES

Multi-stage vapor injection compression cycle can be classified into two fundamental configurations: (a) Flash tank cycle and (b) Economizing heat exchanger cycle. Figure 3 shows the schematics of a 2-stage cycle for each configuration.

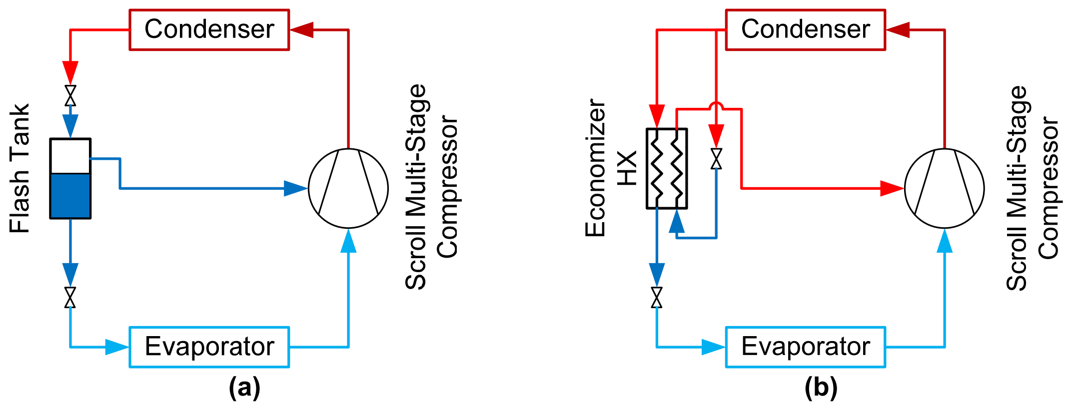

Figure 3: Schematics of two-stage cycles with flash tank (a) and two-stage cycle with economizing heat exchanger (b).

In a two-stage cycle with flash tank (Figure 3 (a)), the two-phase refrigerant after the first expansion is separated into saturated liquid and vapor by a flash tank. It has the advantage of feeding 100% of saturated vapor to the compressor injection port. However, the amount of refrigerant going to the injection port is difficult to control and is solely determined by the high side pressure.

The two-stage cycle with economizing heat exchanger (Figure 3 (b)) allows part of the liquid refrigerant at the condenser outlet to pass through an expansion valve before entering the economizer HX to further subcool the main-stream refrigerant coming from the condenser. The superheated intermediate pressure refrigerant leaving the economizer HX enters the intermediate compressor port. As a result, the separation with economizer HX is not always 100% as compared to the flash tank separation. In the mean time, the subcooled main-stream refrigerant is expanded by a second expansion valve, and then enters the evaporator.

Hence, refrigerant flow rate and pressure entering the intermediate compressor port can be easily controlled using thermostatic expansion valves. As such, this two-stage cycle has been widely investigated. Wang *et al.* (2009) demonstrated that two-stage cycle with economizer HX achieves performance improvement comparable to that of two-stage cycles with flash tank. The former has a wider operating range of injection pressure due to its freedom of setting the injection refrigerant superheat at the injection port. A few commercial heat pump products based on the concept of two-stage cycle with economizer HX have been available.

Besides the basic injection cycle configurations, Takahashi 2010 discussed the benefits of using a flash injection circuit cycle that comprises 3 electronic expansion valves. The cycle configuration of the proposed injection circuit is shown in Figure 4 below. In this design, the subcooled refrigerant leaving the condenser is first slightly expanded into a receiver containing a suction line HX (power receiver). The expanded refrigerant is then further subcooled in an economizer HX similar to that used in the cycle shown in Figure 3 (b). Using electronic expansion valves allows for the intermediate pressure refrigerant to leave the economizer HX at near saturated conditions before it enters the compressor. This results in improved heating performance. The system controller is devised such that the flash injection circuit maintains refrigerant circulation even at lower ambient conditions.

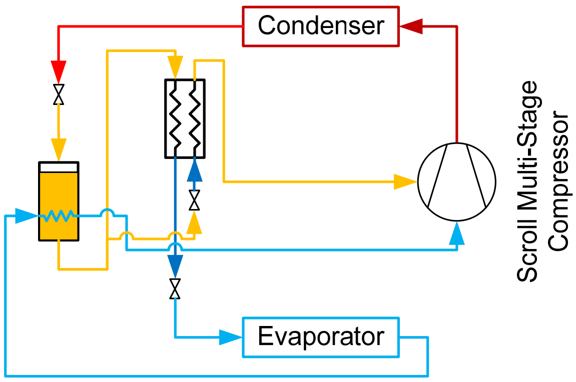

Figure 4: Two-stage cycle with flash injection circuit (Takahashi 2010)

4 RESULTS AND DISCUSSIONS

A system simulation tool was developed in-house. The tool is a component based simulation tool with a Newton-Raphson non-linear solver. Different component models are being used: segmented HX model, modified compressor map to model the multi-stage compressor, overall UA/effectiveness HX model for refrigerant-to-refrigerant economizer HX, and a flash tank model. Component connections are described in a system configuration file; hence any system configuration can be simulated. We have developed 3 system configuration files; one for the flash tank cycle described in Figure 4 (a), one for the economizer cycle described in Figure 4 (b), and the other for the flash injection circuit described in Figure 5. System components have been sized based on existing heat pumps that are available on the market. The compressor is a prototype 5 hp R410A scroll compressor with vapor injection ports.

A numerical experiment was designed to evaluate system performance under varying ambient conditions with different design parameters such as outdoor coil (OD) airflow, OD superheat (SH), indoor coil (ID) airflow, ID subcooling (SC), and economizer(s) effectiveness. The results of this numerical experiment are described in the following subsections.

4.1 Flash Tank Cycle

The flash tank cycle of Figure 3 (a) was modeled. In this cycle, the intermediate pressure was predetermined from the compressor map. A parametric study was constructed to vary

OD airflow (75% to 200% the design value), OD SH (0.56 to 11.11°C), ID airflow (75% to 200% the design value), ID SC (0.56 to 11.11°C). This study has shown that the only parameter that has noticeable impact on the performance at low ambient conditions is the superheat. The optimal OD SH is 2.78°C at design conditions and 0.56°C for the rest of the operating ambient conditions as shown in Figure 5. Furthermore, Figure 5 shows that the performance is almost constant below OD SH of 8°C for design ambient conditions (8.3°C) and below OD SH of 2.78°C for the lowest ambient conditions (-26°C).

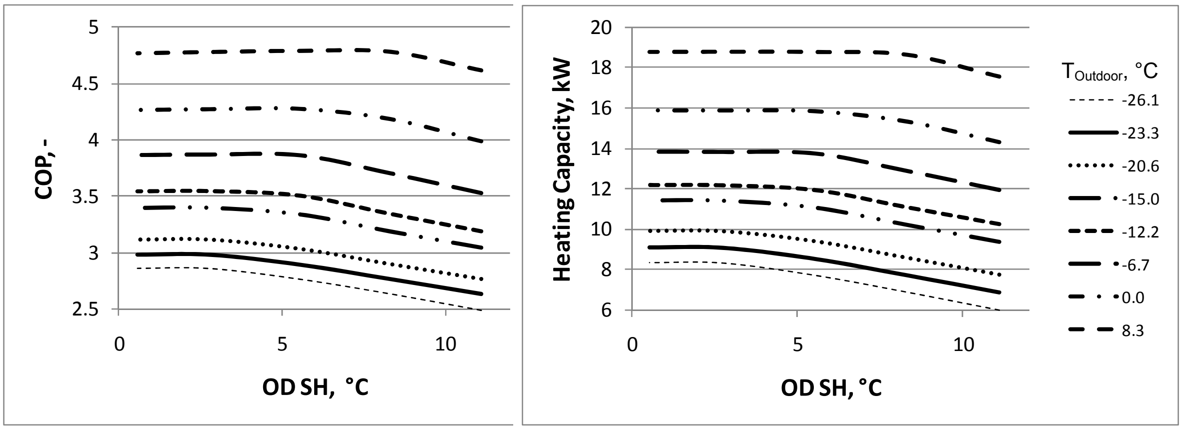

Figure 5: the impact of OD SH on the performance of the flash tank cycle.

The impacts of the other design parameters on the performance of the cold climate heat pump are summarized in Figure 6. In Figure 6, the x-axis represents the value of the different design parameters as a percentage of the design value. The cycle COP, excluding any fan power consumption, was plotted as the dependent value on the y-axis for the lowest ambient conditions of -26°C on the left and the design ambient conditions of 8.3°C on the right. The OD airflow rate had no impact on the performance at low ambient conditions while doubling the OD airflow resulted in only 3.4% performance improvement at the design conditions. Doubling the ID airflow rate resulted in 20% improvement in COP at the design conditions and only 4.5% improvement at low ambient conditions. Finally, the design ID SC of 5.6°C was found to be the optimum at design ambient conditions. At low ambient conditions, the optimal ID SC was found to be 0.6°C resulting in only 1.8% performance improvement.

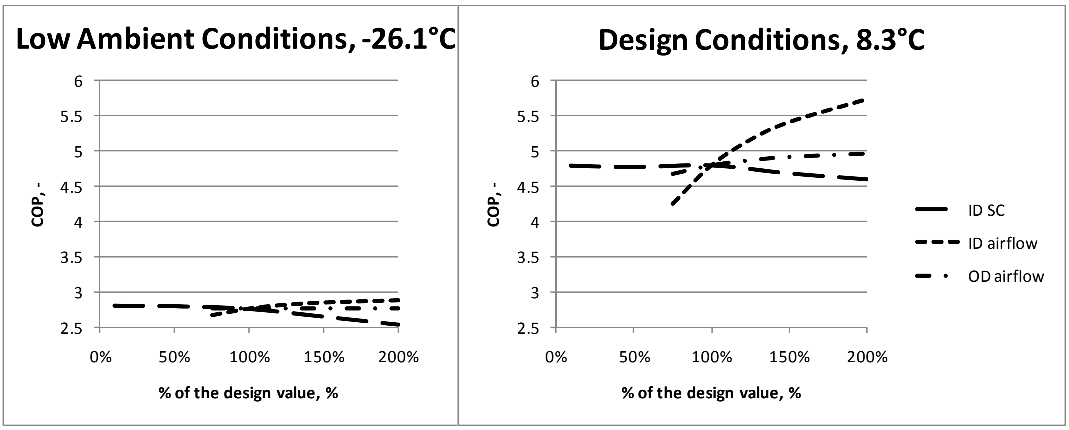

Figure 6: the impact of OD and ID airflow, ID SC on the performance of the flash tank cycle.

4.2 Economizer Cycle

Varying the outdoor airflow rate in the multi-stage economizer cycle of Figure 3 (b) showed strong impact for high ambient conditions; however the performance seemed to be less sensitive to the outdoor airflow rate as the ambient temperature falls below -5°C. On the other hand the refrigerant superheat leaving the outdoor coil had a bigger impact on the system performance. Figure 7 summarizes the impact of OD SH on the performance of the economizer cycle. The results are similar to that of the flash tank cycle. The optimal SH is 5.6°C for the design ambient conditions and 0.6°C for the low ambient conditions. Similar to the flash tank cycle, the performance was insensitive to OD SH below 8°C for the design ambient conditions and 2.8°C for the low ambient conditions. Overall, the economizer cycle showed slight performance degradation varying between 2.6% capacity and 1% COP at design conditions and less than 0.5% at low ambient conditions. This loss in capacity is largely due to having economizer effectiveness of less than 100%.

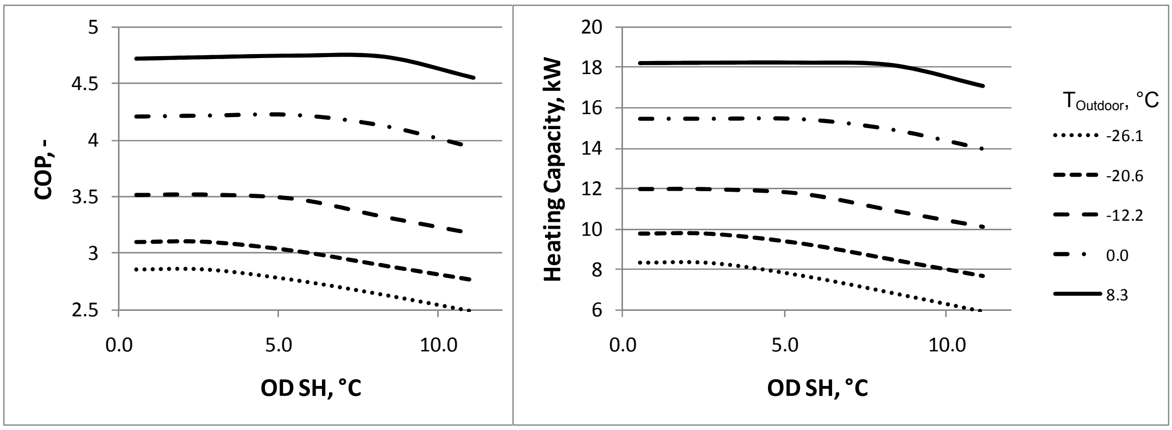

Figure 7: The impact of OD SH on the performance of the economizer cycle.

Based on the previous observation, a parametric study was devised to study the impact of economizer effectiveness on the performance of the economizer cycle. In this study, the economizer overall UA was varied between 75% to 200% of the design value, which is equivalent to 94% to 99.9% economizer effectiveness. This study showed that at low ambient conditions, an economizer with double the overall heat transfer coefficient would result in a heat exchanger effectiveness of 99.9% and would have similar performance to the flash tank cycle. Hence doubling the heat exchanger size resulted in less than 0.4% performance improvement at low ambient conditions. At design ambient conditions, the larger economizers did not improve the cycle performance. This is mainly due to the increased superheat of the injected vapor to the second stage compression chamber. The results of this study are summarized in Figure 8 showing the impact of economizer effectiveness on the cycle COP and heating capacity. These results suggest that smaller economizer can be used at minimal performance degradation as suggested by the tradeoff between added cost of larger economizer and cycle performance improvement.

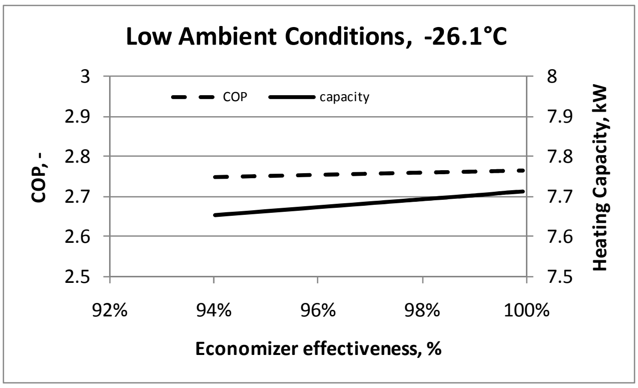

Figure 8: Effect of economizer effectiveness on the system performance.

4.3 Flash Injection Circuit Cycle

The performance of the multi-stage flash injection circuit cycle of Figure 4 was investigated. The power receiver was modeled as a refrigerant-to-refrigerant heat exchanger with constant effectiveness of 0.3 whereas the economizer was modeled as a refrigerant-to-refrigerant heat exchanger with a constant effectiveness of 0.7. The parametric study revealed similar performance sensitivity to that of the flash tank cycle and the economizer cycle. The OD SH was shown to have the most impact on the system performance at low ambient conditions.

The main difference was that 2 refrigerant-to-refrigerant heat exchangers and the required expansion valves were used instead of a single flash tank. Figure 9 summarizes the impact of OD SH on the performance of the economizer cycle. At the design ambient conditions, the multi-stage cycle with flash injection circuit showed better COP than the economizer cycle but at the cost of lower heating capacity. However, at low ambient conditions, the flash injection circuit configuration resulted in lower COP and heat capacity than the economizer cycle. This is largely due to the model assumptions that resulted in economizer effectiveness of 97.6% for the economizer cycle and only 70% for the flash injection circuit.

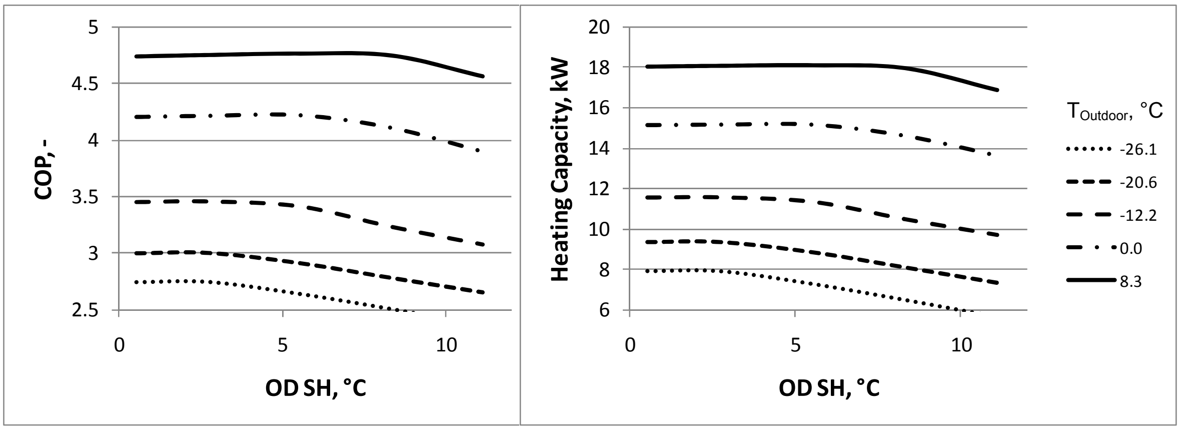

Figure 9: The impact of OD SH on the performance of the economizer cycle.

The impact of the heat exchangers effectiveness on the flash injection circuit cycle performance was further investigated. The power receiver effectiveness was varied between 20% and 70% while the economizer effectiveness was varied between 60% and 95%. The results of this parametric study are summarized in Figure 10. The results indicate that increasing the power receiver effectiveness would improve the cycle COP but would result in lower heating capacity. This is largely due to the decrease in refrigerant mass flow rate associated with the increase in superheat at the compressor inlet. The results shows that changing the power receiver effectiveness from 20% to 70% (about 5 fold increase in heat exchanger size) increased the COP by 0.4% while reducing the capacity by 1.7%. On the other hand, increasing the economizer effectiveness from 60% to 70% (about 3 fold increase in heat exchanger size) increased the COP by 0.4% and increased the heating capacity by 1%. A flash injection circuit cycle with economizer effectiveness better than 75% would surpass the COP of the flash tank cycle but would still suffer from 3% lower heating capacity.

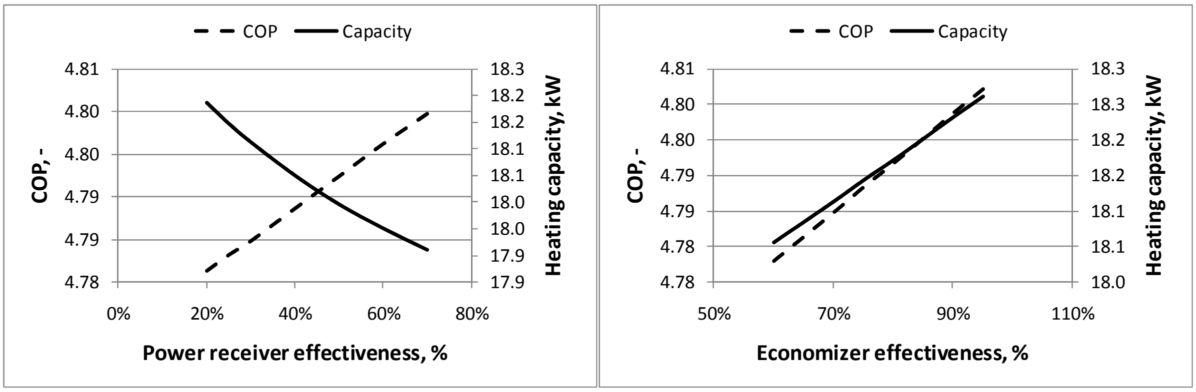

Figure 10: The impact of heat exchangers effectiveness on the performance of the flash injection circuit cycle performance.

5 Discussion

The results shown in section 4 indicated that the flash tank cycle, which is the simplest cycle to model, has the best performance. This is mainly due to the fact that a flash tank would have an effectiveness of 100% and that in our system simulations we relied on the compressor performance map to determine the injection pressure. Flash injection cycles, are simple in analysis but are not simple to implement in a commercial product due to the difficulty of controlling the intermediate pressure and the need for larger system charge and improved refrigerant charge management techniques.

The flash injection circuit cycle provided better COP than the economizer cycle. This cycle can be further optimized by varying parameters such as the expansion prior to the power receiver and the intermediate pressure. This would lead to a new optimized scroll compressor injection ports location. The tradeoff between cost and performance improvements needs to be further investigated in order to have optimal use of materials.

Finally, the compressor performance map used in the current study was developed for fixed injection fluid superheat of 5.6°C and using an economizer with a 5.5°C approach temperature controlled using TXVs (Beeton and Pham 2003). However, in our simulations, the superheat at the injection port was not controlled and the injection ratio was dictated by the system solver. There was no means to provide any corrections to the compressor performance based on the injection ratio and the injection superheat. We believe that there is a need to develop an advanced compressor map for multi-stage injection type compressor that incorporates critical operating parameters such as evaporating, condensing, and intermediate saturation temperatures, injection flow rates, and injection superheat.

6 CONCLUSIONS

Cold climate heat pumps present an opportunity for improved heating efficiency. Multi-stage injection cycles can be used to maintain acceptable system performance at low ambient conditions. A new flexible system simulation tool has been developed and presented in this paper. The new simulation capability was applied to 3 multi-stage injection cycle configurations: flash tank cycle, economizer cycle and flash injection circuit cycle. The results indicated that the simple flash tank cycle showed superior performance. However economizer cycle has negligible performance degradation and is easier to control and design. The flash injection circuit cycle offers additional system flexibility and allows for further performance improvement. These could be designed with higher COP than a flash tank cycle but at the cost of capacity reduction.

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A TWO-PHASE THERMOSIPHON DEFROSTING TECHNIQUE FOR AIR-SOURCE HEAT PUMPS

Paul Byrne, Jacques Miriel, Yves Lénat
Université Européenne de Bretagne
Equipe MTRhéo, Laboratoire LGCGM, INSA de Rennes et Université de Rennes1, France

Abstract: Air-source heat pumps loose performance during winter because of low ambient temperatures and frosting. The most common defrosting technique is to reverse the cycle. Unfortunately it provokes a break in the heat production and the defrosting energy is drawn from the heat stock previously constituted. This article presents the design of a heat pump prototype for simultaneous heating and cooling (named HPS). Its refrigeration circuit involves a piece of refrigeration circuit that could be modified and implemented to standard air-source heat pumps during retrofit or maintenance in order to carry out defrosting with better performance. It uses a water tank that recovers the subcooling energy of the refrigerant at first and is subsequently used as a cold source for evaporation. The second part of the sequence liberates the air evaporator for defrosting. Between two evaporators (air-to-refrigerant and water-to-refrigerant) at different temperatures, a thermosiphon forms. A supplementary amount of vapour flows out of the water evaporator and migrates towards the colder inside surface of the air evaporator tubes in thermal contact with the frost layer. The vapour, while condensing, brings the defrosting energy. The liquid returns back to the water evaporator by gravity. This defrosting system was observed by means of infrared thermography and tested experimentally on a heat pump prototype. It proved very efficient as defrosting time was short. Using this defrosting technique ensures:

- a continuous heat production with even better performance while defrosting thanks to the higher evaporating temperature,
- more frequent defrosting sequences because more easily activated, impacting on lower frost thickness and higher mean heat transfer coefficients.

A numerical study was carried out to assess the performance improvement achieved on a heating sequence. Simulations show a COP and an exergetic efficiency improvement respectively of 12 % and 18 %.

Key Words: defrosting, two-phase thermosiphon, air-source heat pumps

1 INTRODUCTION

Air-source heat pumps are energy-saving heating devices compared to electric convectors or radiators. They are easy to install and offer good comfort when coupled to low temperature heating systems. Moreover some heat pumps can be reversible and provide space cooling during the summer. The main drawback of this type of heat pump is frost formation at the air evaporator (Xia et al. 2006) (Huang et al. 2009) (Shao et al. 2009) during the winter. When the ambient air humidity is higher than the saturation humidity at the evaporator surface temperature, condensation forms on the fins of the air coil. Under low ambient temperatures, condensation turns to frost. Frost increases the thermal resistance and the pressure drop on the air side flow. Because classic defrosting techniques provoke a break in heat production, the system performance strongly decreases when heating demands are at their highest. The defrosting sequence described in this article is carried out without stopping the heat production. A thermosiphon defrosting technique is proposed through the design of a heat pump for simultaneous heating and cooling (HPS) (Byrne et al. 2009). The HPS can carry out space heating, space cooling and domestic hot water (DHW) production for hotels and small office or residential buildings. On the low pressure side of refrigeration plants using several evaporators, thermosiphons are usually cancelled out by non-return valves because they represent risks of refrigerant trapping in the circuit. The HPS takes advantage of this phenomenon to run an “automatic” defrosting. Thermosiphons are known to be efficient means of heat transfer (Lee et al. 2009) (Hakeem et al. 2008) (Dobson 1998). This article first presents the HPS concept and its defrosting technique. Then the thermosiphon is validated and observed using infrared thermography. Finally a sequence involving defrosting during winter was modelled using TRNSYS software (Solar Energy Laboratory 2000). A simulated comparison between the performance of the HPS and a standard reversible heat pump is presented.

2 THE CONCEPT OF THE HPS

2.1 General specifications

The HPS applications are space heating, space cooling and domestic hot water production for hotels or glass fronted buildings in which the proportion of simultaneous demands in heating and cooling is high. This situation occurs in mid-season (spring and autumn) for north-south oriented buildings in which rooms facing north need heating and rooms facing south need cooling. Another situation is encountered in summer when cooling and domestic hot water demands are simultaneous.

The HPS prototype (figure 1) produces hot and chilled water using plate heat exchangers. A balancing air coil works either as a condenser for heat rejection in a cooling mode or as an evaporator for heat suction in a heating mode. The air evaporator and the air condenser are never used at the same time. These functions have been assembled in the same three-fluid air coil (air, high pressure refrigerant and low pressure refrigerant) in order to decrease the finned surface area compared to separate air condenser and evaporator. When the tubes of the air evaporator are used the surface of the fins near the tubes of the air condenser are also used and vice versa. A subcooler is connected to the cold water loop to carry out a short-time heat storage during winter sequences. Depending on the mode of operation, the electric components (compressor, fan and electronic valves named Evr) are managed automatically by a programmable controller or manually by the operator. The thermostatic expansion valves are named TEV1 (connected to the water evaporator) and TEV2 (connected to the air evaporator). Non-return valves named Nrv1 and Nrv2 are placed at the outlets of the air and water condensers to avoid refrigerant trapping in the condensers.

As pressures and temperatures are linked during condensation, a high pressure control system ensures that condensation is completed in the condenser (and does not finish in the subcooler). Moreover it is able to control the condensation temperature and thus the heating capacity. A special liquid receiver is placed on the liquid line. It is connected to the compressor discharge line and the inlet of the air evaporator by copper tubes of smaller diameter on which electronic on-off valves are placed (EvrHP and EvrLP in figure 1). The high pressure control system indirectly controls the volume of liquid in the receiver. The volume of liquid in the different condensers depends on the mode. If the high pressure is below the set point, the electronic valve EvrHP is opened by the controller. The receiver is filled up with gas coming from the compressor discharge line at a pressure higher than the pressure in the receiver until the set pressure is reached. The gas entering the receiver drives the liquid towards the evaporator. The non-return valve closes because pressure becomes higher at the outlet than at the inlet. The subcooling heat exchanger and the bottom part of the water condenser are filled up with more liquid until the appropriate level of liquid is reached. If however the chosen mode is the cooling mode, the condenser becomes the air heat exchanger. The set point for pressure is then the lowest possible. The pressure is reduced by driving the vapour out of the top part of the receiver towards the inlet of the air evaporator. The refrigerant in a liquid phase is sucked out of the water condenser and the subcooler and enters the receiver. The operation of the control system depends upon a special liquid receiver being designed sufficiently high and narrow with the main objective to enhance temperature stratification and to limit as far as possible thermal transfer between the gas and the liquid. When the gas is injected, part of it condenses. When gas is rejected to the low pressure, part of the liquid evaporates. Although these phenomena can reduce the efficiency of the liquid variation in the receiver, it stabilizes the control system. The receiver is also thermally insulated to reduce the heat transfer towards the ambience.

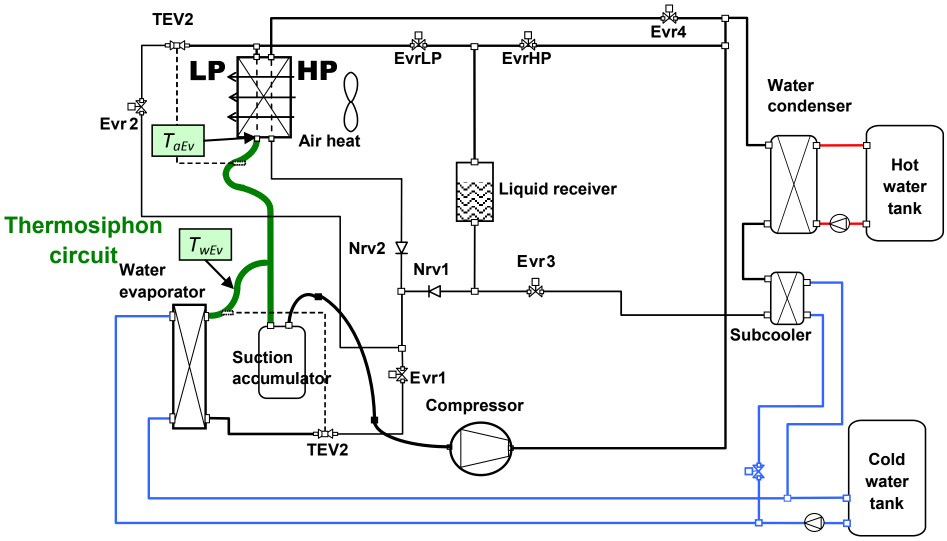

Figure 1: HPS Circuit

Table 1 shows the general specifications of the components. The chosen refrigerant is R407C, which is widely used in heat pumping technology. The refrigerant charge is increased compared to a conventional heat pump system. The calculated minimum refrigerant charge of the HPS is 9.8 kg and the effective charge is 16.3 kg for safety of usage during experiments. As a comparison, Poggi et al. (2008) gives statistical data concerning specific charge of air conditioning systems leading to a maximum of 1 kg per kW of cooling capacity for split systems. Meunier et al. (2005) gives an average ratio of 0.2 kg per kW for conventional heat pumps. Therefore the refrigerant charge of the HPS will be higher than conventional systems and its operation should be restricted with unsecure fluids.

Table 1: Specifications of HPS components

Component	**Specification**
Compressor Brand: Copeland Type: Scroll Ref: ZB38KCE-TFD	Swept volume: 14.5 m³/h Nominal cooling capacity (T_{ev} = 0 °C / T_{cd} = 40 °C): 11.5 kW
Water heat exchangers - condenser - evaporator - subcooler	Type: plate heat exchanger 50 plates, 2.45 m² 34 plates, 0.8 m² 14 plates, 0.16 m²
Air heat exchangers	Type: finned tubes, 68 m² 6 rows of 30 tubes (l = 750 mm , Ø = 10 mm)
Working fluid	R407C

2.2 Operating modes

Three operating modes can be run.

- The simultaneous mode produces hot and chilled water using the water condenser and the water evaporator (electronic valves Evr1 and Evr3 are open).

- The heating mode produces hot water using the water condenser, the air evaporator (electronic valves Evr2 and Evr3 are open) and also the subcooler to store the subcooling energy in the cold water tank.
- The cooling mode only produces cold water using the water evaporator and the air condenser (electronic valves Evr1 and Evr4 are open).

A production ratio r_p can be defined following equation 1 using heating and cooling capacities. For a simultaneous mode this ratio is close to 1.3.

$$r_p = \frac{\dot{Q}_h}{\dot{Q}_c} \tag{1}$$

A demand ratio r_d can be defined following equation 2 using instantaneous heating and cooling demands.

$$r_d = \frac{heating\ demand}{cooling\ demand} \tag{2}$$

Depending on the building demand ratio r_d, two sequences can be run:

- if $r_d > r_p$, heating demand is higher than cooling demand, the sequence starts in the simultaneous mode and continues in the heating mode when the cooling demand is satisfied;
- if $r_d < r_p$, cooling demand is higher than heating demand, the sequence starts in the simultaneous mode and continues in the cooling mode when the heating demand is satisfied.

The variability of operating times in each mode enables the control system to adapt hot and chilled water production to heating and cooling demands.

2.3 Classic winter sequence

During winter the sequence alternates between heating and simultaneous modes. The cold water tank is used as a short-time heat storage. The sequence begins by the heating mode engaging the water condenser, the air evaporator and the subcooler. The cold water tank is heated, usually from 5 to 15 °C, by the refrigerant subcooling energy. Then the HPS switches to the simultaneous mode and uses energy stored in the cold water tank as a cold source for the water evaporator. The cold water tank temperature decreases from 15 to 5 °C.

In the simultaneous mode, the evaporating temperature is higher than in the heating mode from the moment that ambient air is colder than the short-time heat storage tank. Therefore, using the simultaneous mode for a time during the winter sequence enables to produce hot water continuously with improved average system performance compared to standard air-source heat pumps. Besides, in the simultaneous mode, the air evaporator is not used for evaporation and can be defrosted using a two-phase thermosiphon.

2.4 Winter sequence with defrosting

In the heating mode, under cold outside air temperatures, the fins of the air evaporator get frosted. Before frost thickness becomes critical, the cold tank temperature rises to 15 °C and the simultaneous mode is engaged by the controller. In this mode, the air coil is automatically defrosted by a two-phase thermosiphon formed between the two evaporators. A supplementary flow of vapour comes out of the water evaporator and migrates towards the air evaporator where the temperature, and thus the pressure, is lower. The refrigerant exchanges latent heat with the frosted fins and condenses. It finally flows back to the water evaporator by gravity.

A major advantage of this sequence is to carry out defrosting without stopping the heat production. Frost thickness can thus be minimized and mean convection heat transfer coefficients at the evaporator can be maximized. The average heat pump efficiency under frosting conditions is improved compared to the performance of standard air-source heat pumps that use hot gas or reversed cycle defrosting methods (Rajapaksha et al. 2003).

3 EXPERIMENTAL VALIDATION

3.1 Experimental setup

A HPS prototype was built and connected to a water distribution system. The water system is used to limit the temperature variation of hot and cold water. It is composed of two water tanks, a circulation pump and four fan coil units (FCU) that dissipate the heating or cooling energy produced by the heat pump. The heating and cooling nominal power of the FCU is respectively 1.85 kW (50 °C / 40 °C) and 1.5 kW (7 °C / 12 °C). Temperature and humidity of the air circulating through the AHX are controlled, namely in order to obtain frosting conditions at the air coil. Type K thermocouples were placed on every inlet and outlet of the refrigeration cycle components on refrigerant, water and air sides. They were placed in contact with the copper tube and recovered by a lagging tape. The uncertainty on these measurements is ±0.5 K. High and low pressure sensors come from the P299 series of Johnson Controls. Their accuracy is ± 1 %. The sensors were connected to an acquisition unit using a time base of 1 second and to the control computer through which were also managed the different electric components. Electric energy consumption was given by a pulse electric meter. Water mass flow rates were calculated using water meters.

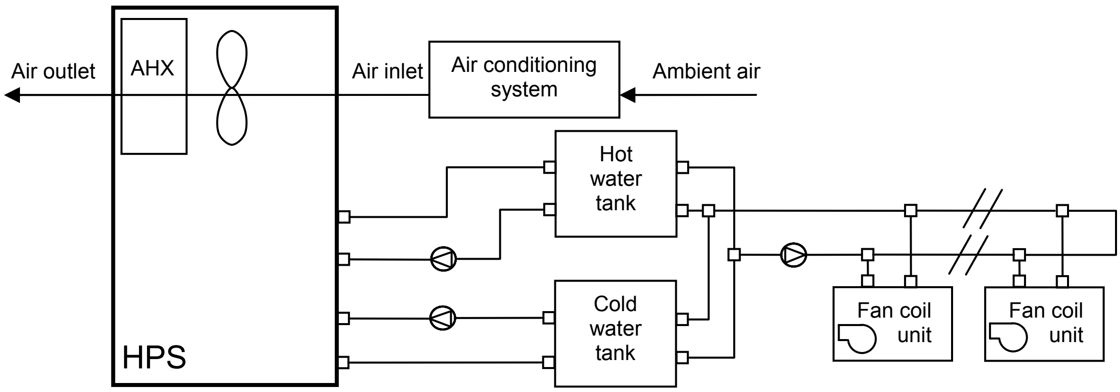

Figure 2: Scheme of the connections between the HPS and the air and water systems

3.2 Two-phase thermosiphon observation

The two-phase thermosiphon circulates in tubes that have large diameters (represented in green in figures 1 and 3). Vapour migrates from the water evaporator towards the air evaporators. Condensed refrigerant returns back by gravity to the water evaporator and the suction accumulator.

Thermographic pictures (figure 4) have been taken before and during the defrosting phase at the position of the white square in figure 3. Before the defrosting sequence, temperature is homogenous. The tube contains vapour flowing from the air evaporator towards the compressor. When the two-phase thermosiphon defrosting sequence is launched, the thermostatic expansion valve TEV2 does not supply refrigerant to the air evaporator anymore because the electronic valve Evr2 is closed. In figure 4 it can be seen that heterogeneity of temperature appears in the tube. The hotter vapour phase flows up the tube in the top part of the section and the colder liquid phase flows down in the bottom part.

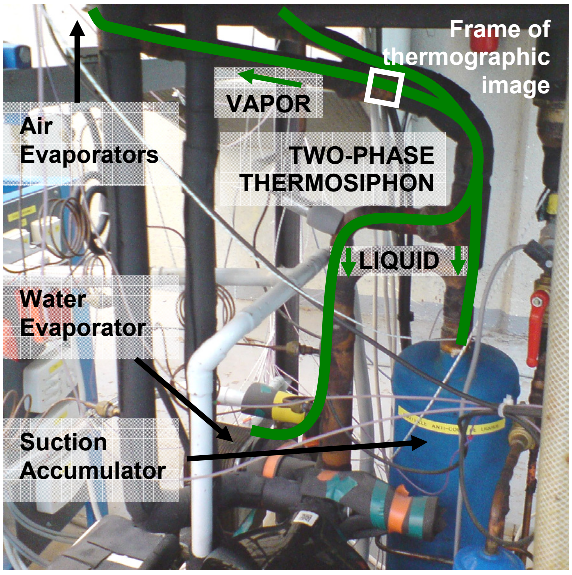

Figure 3: Photograph of the thermosiphon setup

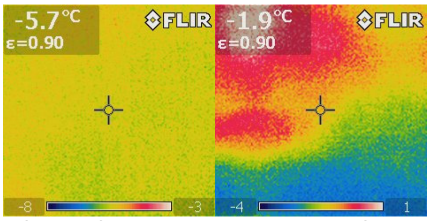

Figure 4: Thermographic pictures of the tube between the two evaporators

3.3 Defrosting technique validation

The validation is based on the observation of the frost layer disappearance during a defrosting sequence using the two-phase thermosiphon. The defrosting time is compared to a frost layer disappearance time by thermal exchange with the ambient air surrounding the test bench. In this second case compressor and fans are stopped.

3.3.1 Frosting phase

The first phase consists in running the heat pump in a heating mode under frosting conditions. Ambient air temperature is controlled around 0 °C. Frost progressively appears on the LP section of the air coil of the prototype. The frost layer thickness increases over a period of 30 minutes. Meanwhile the cold water tank temperature increases from 5 to 15 °C thanks to the amount of energy recovered by subcooling the refrigerant.

3.3.2 Defrosting phase

After 30 minutes, the frost layer is assumed to be sufficient to run the defrosting sequence. The electronic valve Evr2 is switched off and the electronic valve Evr1 is switched on. The water evaporator is used instead of the air evaporator. The simultaneous mode is engaged and the two-phase thermosiphon defrosting starts. The frost layer disappears within 2 minutes (figure 5). In order to compare, after a 30-minute frosting phase, the prototype has been stopped. The defrosting energy was then brought by thermal convective exchange with the ambient air surrounding the test bench. The time of frost disappearance reached 20 minutes. This comparison confirms that the thermosiphon defrosting technique works very efficiently.

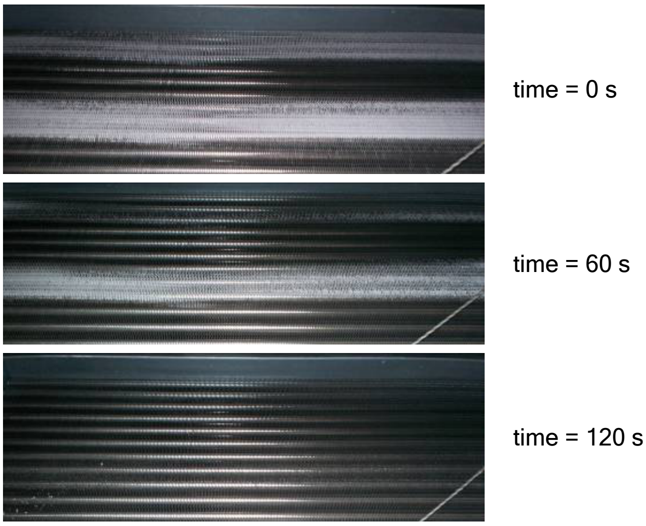

Figure 5: Photographs of the air coil during the thermosiphon defrosting phase

4 SIMULATION STUDY

A simulation study is carried out to evaluate the performance improvement brought by the use of such a defrosting technique. An energetic and exergetic comparison is made between the HPS and a standard reversible heat pump working with R407C and coupled to a hotel of 45 bedrooms located in Rennes (France).

4.1 Modelling assumptions

The simulations are run using TRNSYS software. The time step is of 5 minutes. The reversible heat pump is supposed not to be able to produce DHW because during summer, it would interfere too much with an operation in the cooling mode. The reversible heat pump model uses an electric water heater for DHW production. In the HPS model, the heating capacity devoted to DHW production is deducted from the total heating capacity of the HPS. An electric backup is also implemented to the model in case the HPS cannot satisfy the DHW production on its own. The chosen hypothesis is that the use of a desuperheater (not in place on the prototype but modelled for the simulation study) does not induce any change in the set point for high pressure compared to what it would be without DHW production. Defrosting of the HPS is supposed « automatically » carried out during a simultaneous mode by the two-phase thermosiphon. For the reversible HP defrosting is achieved by an inversion of the refrigeration cycle. Reversing the cycle provokes a drawing of heat in the hot water tank. A defrosting model was implemented to the heat pump modelling.
An electric backup for space heating was also implemented to the HPS and reversible heat pump models.
Table 2 presents the annual needs in space heating, space cooling and DHW production of the hotel of 45 bedrooms located in Rennes.

Table 2: Annual needs of a 45-bedroom hotel in Rennes

Sectors	*Annual needs*
DHW production	74870 kWh/year
Space heating	51680 kWh/year
Space cooling	39184 kWh/year

4.2 Frosting and defrosting model for the reversible heat pump

A simplified frosting model was programmed in order to launch defrosting sequences in the reversible heat pump model. The defrosting model calculates the frost mass on the air coil at time step *i*+1 by equation 3 in function of the specific humidity at the inlet and the outlet of the coil, air mass flow rate and the duration of the time step *dt*. The bypass factor f_{bp} of the air coil is taken constant at a value of 0.7.

$$m_f^{\,i+1} = \left(w_{in} - w_{out}\right)\cdot\left(1 - f_{bp}\right)\cdot \dot{m}_{air}\cdot dt + m_f^{\,i} \tag{3}$$

When the heat pump has worked during 45 minutes with an ambient temperature beneath 7 °C, the cycle is reversed. Evaporation is carried out at the water heat exchanger and condensation at the air heat exchanger. Heat is pumped out from the hot water tank to carry out defrosting. Defrosting sequences take into account the performance loss due to the break in the heat production and to the heat used at the evaporation of the reversed cycle. Therefore in the simulation, there is a break in the heat production during a complete time step but the defrosting time is shorter (equation 4). During the rest of the time step, the heat pump is stopped. It is calculated by the ratio of the defrosting energy (equation 5) over the available heating capacity in the reversed cycle. The energy drawn from the hot water tank corresponds to the defrosting energy Q_{df}. The mean electric power over the time step during which defrosting occurs is calculated using equation 6.

$$t_{df} = \frac{Q_{df}}{\dot{Q}_c + \dot{W}} \tag{4}$$

$$Q_{df} = m_f \cdot L_f \tag{5}$$

$$\dot{W}_{ave} = \frac{t_{df}}{dt}\cdot \dot{W} \tag{6}$$

4.3 Equations for energy and exergy calculations

In the winter, the heating mode or the simultaneous mode is used. The instantaneous COP is calculated over a time step by dividing the heating capacity by the electrical power. When only space cooling is needed, the instantaneous COP is given by the division of the cooling capacity by the electric power. When heating and cooling are needed, the simultaneous mode is used. The instantaneous COP is given by the sum of heating and cooling COPs (equation 7). The average coefficient of performance over a sequence is calculated in proportion to the sum of heating and cooling needs on each time step (equation 8).

$$COP_{inst} = \frac{\dot{Q}_h + \dot{Q}_c}{\dot{W}} \tag{7}$$

$$COP_{ave} = \frac{\sum COP_{inst}\cdot(q_h + q_c)}{\sum (q_h + q_c)} \tag{8}$$

Summing heating and cooling energies appears somewhat tricky since these energies are from different nature. So the scientific community prefers to work with exergy which corresponds to the amount of energy ideally convertible into mechanical work (O'Callaghan et al. 1981). Exergy evaluates the quality of the produced energy (equation 9). For instance, Sarkar et al. (2006) use exergy as a performance factor of a heating and cooling heat pump working with CO_2. Exergetic efficiency is defined as the ratio of the produced exergy over the consumed exergy at compression. As electric and mechanical energy are considered to be pure forms of exergy, the consumed exergy at the compressor corresponds directly to the mechanical work of compression and the exergy consumption of the electric backup corresponds to the electric energy consumption. Equation 10 gives the exergetic efficiency of both machines for each time step. For the reversible heat pump, exergy used for DHW production is zero. The source temperature T_{so} is calculated by the logarithmic mean temperature equation (equation 11). The reference temperature T_0 is the ambient temperature.

$$Ex = Q \cdot \left| 1 - \frac{T_0}{\overline{T}_{so}} \right| \quad (9)$$

$$\eta_{ex} = \frac{Ex_h + Ex_{DHW} + Ex_c}{W} \quad (10)$$

$$\overline{T}_{so} = \frac{T_{in} - T_{out}}{\ln \frac{T_{in}}{T_{out}}} \quad (11)$$

4.4 Simulation results

This section of the article presents the simulation results of coefficients of performance and exergetic efficiencies of the HPS and the reversible heat pump, during a winter heating sequence. The winter sequence chosen corresponds to the evening of January 13th (between 16:00 and 22:30) of the TRNSYS weather data file of Rennes. During this sequence, the ambient temperature decreases from 2.0 °C to -0.9 °C.

4.4.1 Evolution of coefficients of performance during a heating sequence

Figure 6 presents the evolution of the coefficient of performance during the chosen heating sequence.

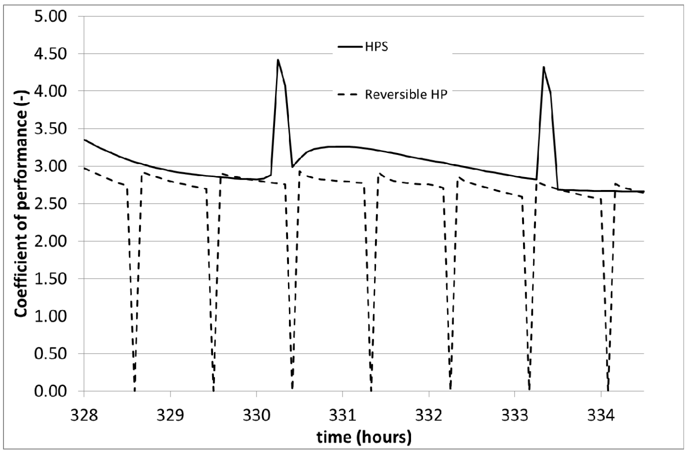

Figure 6: Evolution of coefficients of performance during a heating sequence

A time step during which the COP is zero corresponds to a defrosting sequence of the reversible heat pump. Coefficients of performance are globally better for the HPS and much better during the use of the simultaneous mode. The simultaneous mode COP is higher than the heating mode COP because the source temperature is higher and the discrepancy between the source and the evaporating temperatures is lower for water than for air heat exchangers. Therefore the evaporating temperature in the simultaneous mode is increased even more.

The HPS works in the simultaneous mode only two times during the sequence that lasts 6 h 30 min. The frosting level of the air heat exchanger may be important and could affect the performance of the HPS. This problem is not taken into account in the modelling since it can be easily solved by a correction in the sizing of the cold water tank.

4.4.2 Evolution of exergetic efficiencies during a heating sequence

Figure 7 shows the evolution of the exergetic efficiencies during the chosen heating sequence.

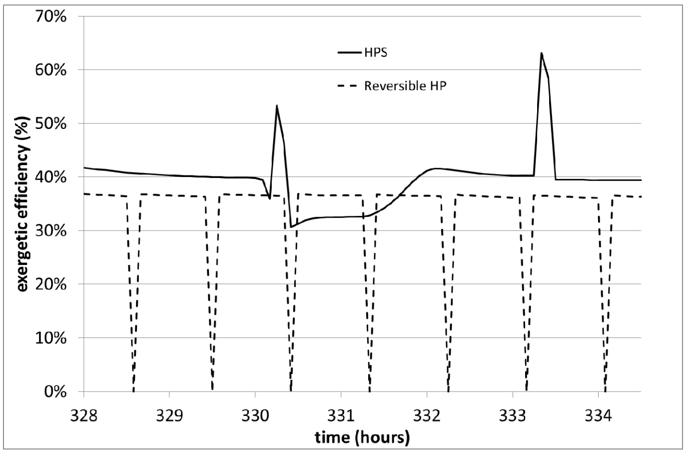

Figure 7: Evolution of exergetic efficiencies during a heating sequence

The exergetic efficiencies almost follow the evolutions of the COPs, except between the hours 330 and 332 which correspond to the interval 18:00 to 20:00 of January 13th. During this 2-hour period, a demand in DHW appears. This extra demand provokes a performance loss in terms of exergetic efficiency. The reversible heat pump is not affected by the DHW demand since DHW is produced by an electric water heater.

4.4.3 Mean performance factors during a heating sequence

Table 3 shows the mean performance factors during the heating sequence presented on figures 6 and 7. The COP and the exergetic efficiency are respectively 12 % and 18 % higher for the HPS compared to a reversible HP.

Table 3: Comparison of performance factors during a heating sequence

Type of heat pump	*HPS*	*Reversible HP*
COP (-)	3.03	2.71
η_{ex} (%)	39.3 %	33.3 %

When taking into account the energy and exergy consumptions of the electric backups for space heating and DHW production, the COPs decrease and the exergetic efficiencies increase (table 4). The exergetic efficiency is higher because electric energy consumed by the electric backups is considered as pure exergy. The COP of the reversible heat pump dramatically decreases because of the complete DHW production is carried out by the electric water heater. The exergetic efficiency of the reversible heat pump increases more because there is no conversion factor (Carnot factor) for DHW production, as there is for thermal energy produced by the HPS.

Table 4: Comparison of performance factors taking into account the backups for heating and DHW production during a heating sequence

Type of heat pump	*HPS*	*Reversible HP*
COP (-)	2.57	1.82
η_{ex} (%)	44.5 %	55.6 %

5 CONCLUSIONS AND PERSPECTIVES

This article presents the design of an air-source heat pump for simultaneous heating and cooling (HPS) that offers improved performance compared to a standard reversible heat pump. Apart from producing simultaneously hot and cold water, the system switches from the air evaporator to a water evaporator during winter sequences. When standard reversible heat pumps are penalized by defrosting (break in the heat production, decrease in COP), the HPS carries out defrosting with enhanced performance thanks to a water source at the evaporation at a higher temperature than ambient air.

In refrigeration circuits, thermosiphon phenomena are usually cancelled out by the use of non-return valves because of possible trapping of refrigerant in some parts of the circuit. On the contrary, the HPS takes advantage of the thermosiphon effect, generated between the two evaporators, to carry out a defrosting sequence. The two-phase thermosiphon was observed by thermographic pictures of a section of the tube between the two evaporators. The defrosting effect was validated by the quick disappearance of the frost layer on the air coil during a defrosting sequence (2 minutes) compared to the defrosting time when the heat pump was stopped (20 minutes). Other experiments must be carried out in real operating configurations to assess the effect of the two-phase thermosiphon defrosting technique under more realistic winter conditions of temperature and humidity.

A simulation study was carried out to evaluate the performance improvement of the defrosting technique during a heating sequence. Results show that compared to a reversible heat pump the HPS works with higher COP (12 %) and higher exergetic efficiency (18 %) without taking the electric backups into account.

The two-phase thermosiphon is a non-penalizing defrosting technique that can be adapted to standard air-source heat pumps with some modifications on the refrigerant circuit. A short-time heat storage has to be connected and used as a heat source first for subcooling and subsequently for evaporation. This defrosting system was tested on a special prototype and further research will be carried out on the simplification of this concept and its adaptation on conventional heat pumps.

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NOMENCLATURE

AHX	air heat exchanger
COP	coefficient of performance
DHW	domestic hot water
dt	time step duration (s)
Evr	Electronic valve of regulation
f	factor (-)
FCU	fan coil unit
HP	heat pump
HPS	heat pump for simultaneous heating and cooling
L	latent heat of fusion (J kg^{-1})
LP, HP	low pressure, high pressure (Pa)
m	mass (kg)
$\dot{m}$	mass flow rate (kg s^{-1})
Nrv	non-return valve
q	demand (J)
Q	thermal energy (J)
$\dot{Q}$	heating or cooling capacity (W)
r	ratio (-)
t	time (s)
T	temperature (°C)
TEV	thermostatic expansion valve
$\dot{W}$	electric power (W)

Greek symbol:

η	efficiency (%)

Subscripts:

aEv	air evaporator
ave	average
bp	bypass
c	cooling
d	demand
df	defrosting
ex	exergetic
f	frost
h	heating
inst	instantaneous
p	production
r	refrigerant
so	source
wEv	water evaporator

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Guide for Achieving Advanced Energy Savings Published by Industry Leaders

Guidance to get you at least halfway to achieving net-zero-energy design is now available in a new publication.

Advanced Energy Design Guide for Small to Medium Office Buildings: Achieving 50% Energy Savings Toward a Net-Zero-Energy Building is the first book in a series of Advanced Energy Design Guide (AEDG) publications that provides recommendations to achieve 50 percent energy savings when compared with the minimum code requirements of ANSI/ASHRAE/IESNA Standard 90.1-2004, Energy Standard for Buildings Except Low-Rise Residential Buildings. The book was developed by a committee representing a diverse group of energy professionals drawn from ASHRAE, the American Institute of Architects (AIA), the Illuminating Engineering Society of North America (IES), the Department of Energy (DOE) and the United States Green Building Council (USGBC).

The series follows the earlier six-book series that provided guidance to achieve 30 percent savings. The ultimate goal is to provide guidance to achieve net-zero-energy buildings, that is buildings that produce more energy than they consume.

http://www.ashrae.org/pressroom/detail/guide-for-achieving-advanced-energy-savings

Report: Global AC market poised for growth

After witnessing a temporary disruption in growth, the global market for air-conditioning systems staged a recovery in 2010 and is now projected to continue growing, according to the recent report "Air Conditioning Systems: A Global Strategic Business Report". Growth in the market over the next few years will be primarily led by increasing adoption of energy-efficient models, particularly inverter-based air conditioners, and by rapidly growing replacement demand.

http://www.companiesandmarkets.com/Market-Report/air-conditioning-systems-a-global-strategic-business-report-692785.asp

2011

25-26 October
SusTEM2011 - Sustainable Thermal Energy Management in the Process Industries
Newcastle upon Tyne, United Kingdom
https://webstore.ncl.ac.uk/browse/extra_info.asp?compid=1&modid=2&prodid=82&deptid=9&searchresults=1

6-9 November
ISHVAC 2011: 7th International Symposium on Heating, Ventilating and Air-Conditioning
Shanghai, China
Contact: Zhangxu-hvac @ mail.tongji.edu.cn
http://www.ishvac2011.org/

7-9 November
China Mobile Air Conditioning Show 2011
Shanghai, China
http://www.autocoolexpo.com/en/

17-19 November
HVACR Indonesia
Jakarta, Indonesia
http://hvacrseries.com/indonesia/

29-30 November
International Heat Pump Development Forum, China 2011
Shanghai, China
http://www.heatpumpchina.cn/

5-7 December
GeoPower Europe 2011
Milan, Italy
http://www.greenpowerconferences.com/EF/?sSubSystem=Prospectus&sEventCode=GE1112IT&sSessionID=0bac6d077be4d98b475eac966f7e3990-5388588EF/?sSubSystem=Prospectus&sEventCode=GE1112IT&sSessionID=0bac6d077be4d98b475eac966f7e3990-5388588

2012

21-25 January
ASHRAE Winter Meeting
Chicago
http://www.ashrae.org/events/page/Chicago2012

23-25 February
ACREX India 2012 "For a greener tomorrow"
Biec, Bangalore, India
http://www.acrex.org.in/pdf/brochure.pdf

12-13 March
High Performance Buildings Conference
San Diego, California
http://www.ashrae.org/events/page/hpbconf

27-30 March
Mostra Convegno Expocomfort
Milan, Italy
http://www.mcexpocomfort.it/en/aspx/showFolder.aspx?Folder=1750

17-20 April
2nd European Energy Conference
Maastricht, the Netherlands
http://energy-conference.eu/

23-27 June
ASHRAE Annual Conference
San Antonio, Texas
http://ashraem.confex.com/ashraem/s12/cfp.cgi

25-27 June
10th IIF/IIR Gustav Lorentzen Conference on Natural Refrigerants
Delft, Netherlands
http://www.gl2012.nl/

16-19 July
21st International Compressor Engineering Conference at Purdue
14th International Refrigeration and Air Conditioning Conference at Purdue
2nd International High Performance Buildings Conference at Purdue
West Lafayette, Indiana
https://engineering.purdue.edu/Herrick/Events/2012Conf/index.html
http://www.conf.purdue.edu/

29 July - 1 August
10th IIR Conference on Phase Change Materials and Slurries for Refrigeration and Air-Conditioning
Kobe, Japan
http://www2.kobe-u.ac.jp/~komoda/pcms/

10-12 October
Chillventa
Nuremberg, Germany
http://www.chillventa.de/en/

25-26 October
3rd IIR Workshop on Refrigerant Charge Reduction in Refrigerating Systems
Valencia, Spain
http://www.imst.upv.es/iir-rcr2012/

2013

7-8 June
15th European Conference - Italy
United Nations Environment Programme
Milan, Italy

In the next Issue

Working fluids for a sustainable future

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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.

IEA Heat Pump Programme

International collaboration for energy efficient heating, refrigeration and air-conditioning

Vision

The Programme is the foremost worldwide source of independent information and expertise on environmental and energy conservation benefits of heat pumping technologies (including refrigeration and air conditioning).

The Programme conducts high value international collaborative activities to improve energy efficiency and minimise adverse environmental impact.

Mission

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.

IEA Heat Pump Centre

A central role within the programme is played by the IEA Heat Pump Centre (HPC). The HPC contributes to the general aim of the IEA Heat Pump Programme, through information exchange and promotion. In the member countries (see right), activities are coordinated by National Teams. For further information on HPC products and activities, or for general enquiries on heat pumps and the IEA Heat Pump Programme, contact your National Team or the address below.

The IEA Heat Pump Centre is operated by

SP Technical Research Institute of Sweden

IEA Heat Pump Centre
SP Technical Research
Institute of Sweden
P.O. Box 857
SE-501 15 Borås
Sweden
Tel: +46 10 516 50 00
Fax: +46 33 13 19 79
E-mail: hpc@heatpumpcentre.org
Internet: http://www.heatpumpcentre.org

IEA
OECD
heat pump
centre

National team contacts

AUSTRIA
Prof. Hermann Halozan
Technical University of Graz
Innfeldgasse 25
A-8010 Graz
Tel.: +43-316-8737303
Fax: +43-316-8737305
Email: halozan@tugraz.at

CANADA
Dr Sophie Hosatte
Natural Resources Canada
CETC – Varennes
1615 Bd Lionel Boulet
P.O. Box 4800
Varennes
J3X 1S6 Québec
Tel.: +1 450 652 5331
E-mail: sophie.hosatte@nrcan.gc.ca

FINLAND
Mr Jussi Hirvonen
Chair of the Board
Finnish Heat Pump Association SULPU ry,
Tel: +358 50 500 2751
Email: jussi.hirvonen @ ivt.fi
Internet: www.sulpu.fi

FRANCE
Jean-Michel Parrouffe
ADEME
Service des Réseaux et des Energies
Renouvelables
500 route des Lucioles
FR-06560 Sophia Antipolis

GERMANY
Prof. Dr.-Ing. Dr. h.c. Horst Kruse
Informationszentrum Wärmepumpen und
Kältetechnik - IZW e.V
c/o FKW GmbH
D-30167 Hannover
Tel. +49-(0)511-16 74 75-0
Fax +49-(0)511-16 74 75-25
E-mail: email@izw-online.de

ITALY
Dr Giovanni Restuccia
Italian National Research Council
Institute for Advanced Energy Technologies
(CNR – ITAE)
Via Salita S. Lucia sopra Contesse
5 - 98126 Messina
Tel.: +39 (0)90 624 229
Fax: +39 (0)90 624 247
E-mail: giovanni.restuccia@itae.cnr.it

JAPAN
Mr Makoto Tono
Heat Pump & Thermal Storage Technology
Center of Japan
1-28-5 Nihonbashi Kakigara-Cho Chuo-Ku,
TOKYO 103-0014
Tel: +81-3-5643-2404
Fax: +81-3-5641-4501
E-mail: tono.makoto@hptcj.or.jp

NETHERLANDS
Onno Kleefkens
Agentschap NL
Divisie NL Energie en Klimaat
P.O. Box 8242
Croeselaan 15
3521 BJ Utrecht
Tel: +31 30 2393 449
Fax: +31 30 2316 491
E-mail: onno.kleefkens@agentschapnl.nl

NORWAY
Mr Bård Baardsen
NOVAP
P.O. Box 6734, Rodeløkka
N-0503 Oslo
Tel. +47 22 80 5006
Fax: +47 22 80 5050
E-mail: baard.baardsen@rembra.no

SOUTH KOREA
Mr Seong-Ryong Park
Korea Institute of Energy Research
Department of Renewable Energy
71-2, Jang-dong, Yuseong-gu, Daejeon
Republic of Korea 305-343
Tel.: +82 42 860 3224
Fax: +82 42 860 3133
E-mail: srpark@kier.re.kr
http://www.kier.re.kr/eng/index.jsp

SWEDEN
Ms Emina Pasic (Team leader)
Swedish Energy Agency
Energy Technology Department
Bioenergy and Energy Efficiency Unit
Kungsgatan 43
PO Box 310
SE-631 04 Eskilstuna
Tel.: +46 16 544 2189
emina.pasic@energimyndigheten.se

SWITZERLAND
Dr Thomas Kopp
Hochschule Rapperswil
On behalf of the
Swiss Federal Office of Energy
Energy Renewable Division
Oberseestrasse 10
8640 Rapperswil
Tel.: +41 55 222 4923
E-mail: tkopp@hsr.ch

UK
Ms. Penny Dunbabin
Department of Energy & Climate Change
(DECC)
Area 6D, 3-8 Whitehall Place, London SW1A
2HH
Tel.: +44-300-068-5575
E-mail: penny.dunbabin@decc.gsi.gov.uk

USA
Ms Melissa Voss Lapsa
Oak Ridge National Laboratory
Engineering Science and Technology Division
Bethel Valley Road
PO Box 2008
Oak Ridge, TN 37831-6054
Tel.: +1-865-576-8620
Fax: +1-865-576-0279
E-mail: lapsamv@ornl.gov