IEA Heat Pump CENTRE NEWSLETTER
Volume 26
No. 3/2008

9th IEA Heat Pump Conference

Advances and Prospects in Technology, Application and Markets

IEA
OECD
heat pump centre

IEA **Heat Pump** CENTRE
NEWSLETTER

Volume 26 - No. 3/2008

In this issue

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It is now four months since the Zurich IEA HPC Conference, but it feels like yesterday. I can still remember many of the presentations and the discussions during this event, which clearly showed the great value of coming together and exchanging ideas and experiences. From an organisational point of view the conference must be seen a total success! Those who take on the prestigious task of arranging the next conference in 2011 will therefore have hard work in front of them. In the next issue, we hope to be able to announce the next conference venue.
Thank you all who went to Zurich, and we'll meet again in …??

Roger Nordman
Editor, HPC Newsletter

Heat pump news

Features

Articles

COLOPHON

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Record-breaking IEA Heat Pump Conference

Rune Aarlien
Chairman IOC

All conference organisers are excited to see how many attendees will show up. The conference market is extremely competitive these days, therefore, we had expected 300, but hoped for 350 registrations. *With over 450 participants from 36 different countries, the 9th IEA Heat Pump Conference was the largest IEA heat pump conference ever staged.*

Held in the beautiful city of Zurich, Switzerland, May 20-22, 2008, the Conference entitled *"Advances and Prospects in Technology, Applications and Markets"*, comprised: 72 oral presentations, 154 poster presentations, eight workshops, seven technical tours, and a rich program for accompanying persons. Oral presentations were both invited and chosen, based on a call for papers.

Of particular interest from an organiser's perspective was the strength of the poster program. For one thing, 154 posters is an all-time high, another is that the quality of the posters was generally very high. With enthusiasm and dedication, authors and presenters had a busy time communicating their messages to an interested and large audience. The success of the poster program coincided nicely with the first-time-ever Best Poster Award. It is a pleasure to congratulate the winners: S. Takeda-Kindaichi, K. Nagano, T. Katsura, S. Hori and K. Shibata (Japan) for the poster *"System performance of HVAC in a low-energy house in the cold region of Japan"*, and J. M. Corberan and J. Gonzalvez Macia (Spain), C. Radulescu (Ireland) for the poster *"Performance characterisation of a reversible water-to-water heat pump"*.

One of the Conference highlights was the Ritter von Rittinger Award ceremony. Named after the inventor (Peter Ritter von Rittinger, 1811-1872) of the first heat pump put into practical use (in Upper Austria in 1855), this is a prize associated with prestige and long-term service and dedication to heat pump technology. In fierce competition with several candidates, this year's recipients were Mr. Gerald C. Groff and Professor Pega Hrnjak (both USA), and Professor Eric Granryd (Sweden). Knowing that organisations and individuals throughout the world, and the technology as such, owe you all great thanks for your contributions, I salute each of you!

The Conference was also well organised. I am proud to say that, by and large, this should be credited to the Swiss National Organising Committee (NOC), which consisted of Professor Thomas Kopp (Chairman), Luzia Arpagaus, Fabrice Rognon and Stephan Peterhans. In my opinion, the success of the Conference was for the largest part attributable to the systematic preparations, excellent organisational skills and hard work of these individuals. Thanks a lot!

Finally, let me express our appreciation to participants, sponsors, speakers, chairpersons, regional coordinators and all others who in some way or another contributed. A conference is always teamwork, and small as well as minor contributions make a difference.

I take the success of the Conference as a solid sign that heat pumps, as a technology, have an important role to play in the future. Use of heat pumping technologies is not only necessary, such as in refrigeration and air conditioning, but is also intelligent use of energy, such as in heating. The Conference showed that we are on our way to remedy problems related to global warming and ozone depletion. I am confident that when we meet again in three years' time (probably in Asia) we will have made further steps.

On behalf of IOC, the International Organising Committee
Rune Aarlien (Chairman)

Column

The role of heat pumps in a future energy transition

Dolf Gielen
Senior analyst working in the Sustainable Policy and Technology Office
Head of the IEA ETP project

http://www.iea.org/Textbase/techno/etp/index.asp. IEA/OECD, Paris

A future with reduced carbon dioxide emissions and less dependence on oil can become a reality. This is the central finding of the IEA's study *Energy Technology Perspectives 2008 (ETP 2008)*, recently presented to G8 heads of state by the International Energy Agency (IEA). Commissioned by governments at the G8 Gleneagles Summit in 2005, *ETP 2008* was a core input to leaders of the G8 nations and other major energy-consuming countries at the July 2008 G8 Summit meetings in Japan. ETP 2008 will also be a cornerstone in the technology discussions during ongoing UNFCCC negotiations on a successor to the Kyoto Protocol.

As the IEA underlined in its *World Energy Outlook 2007*, business as usual is no longer a sustainable option. ETP 2008 proposes alternative technology scenarios capable of halving global CO2 emissions by 2050 against current levels, a cut that the United Nations considers necessary if the world is to limit global average temperature change to between 2 ° and 3 ° Celsius.

This latest IEA *ETP* report points to the important role that heat pumps can play in emissions mitigation and energy efficiency. Indeed, it identifies heat pumps as one of seventeen key emission-abatement options. Combined with measures to insulate building envelopes and to recover heat in ventilation air, heat pumps offer an ideal solution: one that is available today. To unlock the potential, however, large efforts will be needed on the part of governments and industry to give heat pumps a major role in homes, manufacturing plant and commercial buildings.

The heat pump industry has a powerful ally in the electricity industry. Because electricity is a CO_2-free energy carrier, an all-electric society could be compatible with deep emission cuts, provided electricity is generated with low emissions. Electric heat pumps have a natural place in such a scenario, where all types of heat pump could contribute to energy-efficient heating.

Technology development has played a key role in enhancing heat pumps' prospects. Air-to-air systems are now available that can operate effectively, even in cold regions such as Sweden. This was not the case as recently as a decade ago. Rapid progress during these past years has significantly improved system efficiency, so that many heat pumps can now routinely achieve a coefficient of performance (COP) in the range of 5 to 7.

Of course, cold-climate use of such air-to-air systems that are also suitable for cooling calls for care to avoid creating additional summer-time energy demand for air-conditioning. In other parts of the world where air-conditioning is a must, maximum system efficiency should be the target. Globally, therefore, combining heat pumps with measures to optimise building envelope efficiency is crucial.

Geothermal heat pumps are still the technology of choice in large parts of Europe. While they are costly to install, the level of ground temperature results in greater efficiency.

But heat pumps are not, of course, confined to space heating. They can be used for water heating, as demonstrated by the Japanese EcoCute system, which is already commercially available today. And we are now seeing application of heat pumps for laundry driers.

In industry, too, the potential of heat pumps is large. Because higher temperatures are generally needed, however, such applications are more challenging technologically.

Further work is certainly needed to broaden the field of applications. Integration of heat pumps into existing buildings equipped with radiators is only possible if the delivery temperature is sufficiently high. Their use can also be limited in certain cases by space constraints. These issues are the topic of ongoing research.

The IEA has been mandated by government leaders to build further on its existing roadmaps work and to sketch transition paths towards a more sustainable energy future. A methodology and a programme are being developed that will eventually encompass all major technologies, including heat pumps. The findings will feed into the 2010 edition of Energy Technology Perspectives. A publication focusing on the role of building technologies, including heat pumps, in the ETP scenario is planned for 2009. Our hope is that they will lead to new policies and investment programmes that will get us closer to the sustainable energy future that the world so badly needs.

Dolf Gielen, Head ETP Project, International Energy Agency

General

IEA calls for major boost in renewable energy use

50% of global electricity supplies will need to come from renewable energy sources by the middle of the century if mankind is to avert the most serious effects of climate change, according to a new report by the International Energy Agency (IEA).

"Moving a strong portfolio of renewable energy technologies towards full market integration is one of the main elements needed to make the energy technology revolution happen," IEA Executive Director Nobuo Tanaka said in Berlin on 29 September while presenting the report, which provides an overview of policies to promote renewable energies in 35 countries, including the US, Germany, the UK and China. "Only a limited set of countries have implemented effective support policies for renewables, and there is a substantial potential for improvement," Tanaka said. In many countries, renewable electricity generators are still having difficulty accessing power grids, while administrative hurdles persist and many markets are poorly designed, according to the IEA. A lack of information and training as well as low levels of public acceptance are also cited as barriers. "Governments need to take urgent action," Tanaka said, pointing to the need for mass market integration based on "predictable, transparent and stable policy frameworks".

According to the IEA study, the most effective renewable energy policies are to found in Germany, Spain, Denmark and Portugal for onshore wind power, and China for its development of solar heating at a competitive cost.

The UK, which press reports have accused of attempting to dilute ambitious EU renewable energy promotion plans, trailed far behind in the ranking, coming just 31st, with high green power costs and low effectiveness.

The EU is expected to finalise its renewables policy framework before the end of the year. Presented on 23 January along with a wider climate and energy package, the renewables proposal is geared towards achieving a 20 % renewable energy share in 2020, with individual renewable energy targets for EU member states

A separate project to further liberalise the EU's energy market, proposed in September 2007, is also in part designed to facilitate better grid access and market development of renewable energies. But Parliament and Council are set to clash over the detail of the plans, in particular on the treatment of formerly state-owned French and German energy giants accused of dominating the energy market at the expense of smaller producers.

The IEA recently appealed to the EU to push ahead with an ambitious energy market liberalisation agenda as a means of promoting renewable energies. The Paris-based agency is also calling on the EU to "significantly" increase funding for non-nuclear clean technology research.

Source:www.iea.org

EU to limit energy use of electronics on stand-by

A range of electric devices in the EU will in future be required to use significantly less energy when idle or in stand-by mode. The rules are expected to be endorsed by Parliament later this year and would take effect as of 2010.

Computers, televisions, printers and similar devices should, by 2010, consume no more than one or two watts when on stand-by. From 2013, that level should then be lowered even further, to 0.5 or maximum 1.0 watt.

The rules, proposed by the Commission in its Regulation to reduce the energy consumption of electrical appliances used in homes and offices, were approved 7 July during a vote in a special regulatory committee composed of member state representatives. The committee was formed as part of the implementation of the EU's 2005 Eco-Design of Energy-Using Products (EuP) Directive.

The electricity savings produced by the lower watt levels would be equivalent to the yearly electricity consumption of Denmark, and would prevent up to 14 million tonnes of CO2 from entering the atmosphere, according to Commission calculations.

Source: Euroactiv Newsletter

New York makes it illegal for businesses to attract passers-by with AC

New York has enacted a law that makes it illegal for businesses purposely to leave entrance doors open to let cool air out. The practice is commonly used by businesses to entice passers-by to come inside on hot days. The Long Island Power Authority has estimated the practice wastes 20 %–25 % of stores' air conditioning. The Natural Resources Defense Council (NRDC) has found that a business with a typical 6 ft x 7 ft (1.8 m x 2.1 m) doorway wastes up to $1000 and about 1 tonne of CO2 over the course of a summer if it leaves its door open with the AC on.

Source: The HVAC&R Industry for September 11, 2008

Wind can produce over 25 % of EU electricity by 2030

Wind energy could easily provide for more than a quarter of the EU's electricity by 2030, provided that wind farms are better connected to existing electricity grids and that a new grid to exploit the offshore wind industry is built, according to a stakeholder action plan detailing research and political priorities for the sector.

"In 2030, wind energy will be a major modern energy source; reliable and cost-competitive in terms of cost per kWh," predicts the European Wind Energy Technology Platform in its long-term strategic research agenda and market deployment strategy published on 25 July. The objective of this action plan is to make wind provide up to 28 % of EU electricity consumption by 2030, corresponding to a total of 300 GW.

Source: Euractiv newsletter

Green building code set for overhaul

LEED, the internationally-recognised voluntary 'green' building rating system, is due to be revamped to take better account of the energy use and environmental performance of buildings.

The revised Leadership in Energy and Environmental Design (LEED) mechanism will be launched in January 2009, according to a statement by the US Green Building Council (USGBC), which introduced the system in 2000.

The original LEED rates buildings according to a points system based on five criteria: sustainable site development, water savings, energy efficiency, materials selection and indoor environmental quality. In the new LEED 2009, "points will be allocated differently and reweighed, and the entire process will be flexible to adapt to changing technology, account for regional differences and encourage innovation," according to the USGBC.

The EU has its own programme for rating the environmental performance of buildings, the 2002 Energy Performance of Buildings Directive (EPBD), which provides member states with an "integrated method" for calculating energy efficiency based on a variety of factors, such as the building's position, heating, cooling and lighting installations. Based on this method, member states are to create their own minimum standards for energy efficiency. But unlike LEED, which has become recognised and popular at international level, the EPBD remains obscure, and member states are behind in implementing the directive. Real and perceived high costs, lack of technical skills and expertise, conflicting national measures and low public acceptance explain why 20 EU member states have yet to implement the EPBD, according to Ursula Hartenberger of the Royal Institution of Chartered Surveyors (RICS). In addition to slow progress in improving building efficiency, EU countries face criticism for failing to improve the energy efficiency of their economies, considered a crucial part of the EU's objective to slash CO_2 emissions by 20 % by 2020.

Source: Euractiv newsletter

Successful Proheat-pump meeting held in Edinburgh

As part of the EU IEE Proheatpump project, a mini-conference under the name of "GSHP Scotland" was held on September 22.

The event was aimed at installers, architects and manufacturers of heat pumps, but of course others were also welcome. The event was divided into an exhibition, where manufacturers and service providers showed their products, and a conference/workshop where many aspects of GSHPs were discussed.

Plenary talks included a brief introduction to the UK GSHP scene, and key organisations introduced themselves, e.g. GSHPA, HPA, EST. Further on, key issues were identified and points of contention and challenges were described.

The four workshops that were held covered topics of interest to architects and builders , installers (technical), installers (marketing) and those concerned with policy and regulation.

In the architects' and builders' workshop, the objective was to provide an opportunity for participants to present experience and cases, to introduce participants not yet involved in GSHPs to the technology and issues, and to identify requirements: policies, support, information, forms of cooperation.

The technical workshop for installers focused on introducing newcomers to key requirements, issues, choices for design and installation of systems, and to industry structure and relations, contracting, etc. In addition, an introduction to accreditation and training systems and requirements was given. Presentations and views on case studies of installations were given, to share early experience and problems and also to allow experienced installers to describe their experience.

Workshop 3, Installer Marketing, was held to summarise general publicity for GSHPs, supplier and installer marketing methods and experience, and to summarise and review results of PHP marketing surveys among installers. It was important to identify requirements and actions concerning policies, support, information, forms of cooperation, national publicity and information initiatives.

Workshop 4, policy and regulation identified and explored key policy and regulatory issues and requirements as well as examined experiences and values of subsidy schemes and other financial support measures.

In total, about 160 participants attended this one-day event. Many voices were heard that this was an excellent organised event that brought many different groups and interests together to share experience.

For more information on the Proheat-pump project, visit www.proheatpump.eu .

Working Fluids

European Commission to revise ODS regulation

The European Commission has presented a proposal to amend current legislation on ozone-depleting substances (ODS), highlighting the interplay between both the ozone-depleting and global warming effect from such substances. However, ensuring that the global phase-out of HCFCs leads to the introduction of climate-friendly alternatives is left for future action.
Source: www.r744.com

Refrigerant talk turns to HFOs

Contractors who have mastered working with CFCs, HCFCs, and HFCs may now want to turn their attention to what is being promoted as the fourth generation of refrigerants - HFOs (Hydro-Fluoro-Olefins). "We are on the verge of a new revolution in refrigerants," said Denis Clodic of the Centre for Energy and Processes at Ecole des Mines de Paris, during a plenary address before 500 engineers from 30 countries at the combined International Compressor Engineering and International Refrigeration/ Air Conditioning conferences hosted by Purdue University. He specifically cited research currently underway with HFO-1234yf that is being developed for mobile air conditioning, but may have applications in stationary equipment.
Source: ACHR News

EPA continues HCFC fine-tuning

Phase-out of HCFCs such as R-22 continues, but the Environmental Protection Agency (EPA) also continues to tweak and fine-tune certain aspects of the process. The agency held a half-day summit for those it called stakeholders in the phase-out, which brought manufacturers and trade associations affected by refrigerant changes to Washington, D.C., on June 16 to hear what's planned and offer input before a final ruling. The issues, based on the meeting, relate to fine-tuning the so-called service tail of HCFCs that might still be allowed in the market from 2020 until the 100 percent elimination in 2030, what HCFCs might be produced in what amounts during the phase-down period, and reclamation efforts.
Source: ACHR News

Continued reliance on R22 causes concern

With only 17 months before the ban on the use of virgin HCFCs, a new study has shown that 65 % of cooling installations still use these refrigerants, primarily R22.
The results of the study by DuPont in nine key EU markets raise questions about the level of preparedness in the market for the EU ban on virgin HCFCs in 2010.
Spanning across industry sectors representing the largest users of refrigerants, the survey revealed various types of cooling installations, all of which still contain significant quantities of HCFCs, ranging from 57 % of chillers to 76 % of air-conditioning installations. Although awareness levels of the legislation are high - 90 % of respondents claimed to be aware of the impending ban - the large quantities of HCFCs that remain suggest that this has not been coupled with a sense of urgency to ensure compliance. Of those who have not yet taken action, 17 % claim to have no intention to do so.
DuPont estimates that with several million installations across the EU using R22, tens of thousands of these installations will need to be serviced every week in order to ensure compliance by 2010, putting a significant strain on contractor services.

This, allied with the scaling down of HCFC production from mid-2009, is likely to create a bottleneck for businesses, with serious financial implications for companies that delay their response.
Source: www.acr-news.com

Technology & Applications

DOE to pursue zero-net-energy commercial buildings

U.S. Department of Energy (DOE) Deputy Assistant Secretary for Energy Efficiency David Rodgers today announced the launch of DOE's Zero-Net-Energy Commercial Building Initiative (CBI) with establishment of the National Laboratory Collaborative on Building Technologies Collaborative (NLCBT). These two efforts both focus on DOE's ongoing efforts to develop marketable Zero-Net-Energy Commercial Buildings: buildings that use cutting-edge efficiency technologies and on-site renewable energy generation to offset their energy use from the electricity grid by 2025.
Source:www.doe.gov

CGC participating in important standards revision for GeoExchange™

The Canadian GeoExchange Coalition (CGC) is working with and as a member of the Canadian Standards Association's (CSA) Technical Subcommittee to support a substantial revision of Standard CAN/CSA

C-448-02, Design and Installation of Earth Energy Systems. CGC seeks to ensure that industry concerns with the current standard will be addressed. C-448 is the only standard for the installation of ground-source heat pump systems in North America whose development and revisions are executed within the strict framework of an independent standards body. The impartiality of such an independent body is essential to ensure that the final product is rigorous, fair and unbiased. CSA is accredited as a Standards Development Organisation by the Standards Council of Canada (SCC). The SCC represents Canada on the International Organisation for Standardization (ISO). Among its principal tasks, the CSA Subcommittee will examine and work to incorporate Direct Expansion (DX) and Standing Column Well (SCW) technology, two well-understood but long-neglected areas of North American practice. Installation techniques for both technologies have never been included in a relevant CSA standard. CGC has been working over the past 24 months to rectify this situation.
Source:www.geoexchange.ca

Largest secondary CO_2 system unveiled in North America

Loblaw, Canada's leading retailer, has opened the country's first superstore using a secondary CO_2 system for low-temperature refrigeration and space heating. The company and governmental partners are sure that the energy-efficient store will "change the face of Canadian grocery retail."
The Scarborough Superstore, opened on 7 May in Ontario, is the first in Canada to use a low-temperature secondary CO_2 system, and the largest in North America. The refrigeration system is estimated to reduce the store's carbon footprint by 15 %. Additional space heating through rejected heat from the refrigeration system will lower the carbon footprint by a further 7 %. The 11 000 square metres store is thus seen as a major environmental flagship in the Canadian retail sector, to serve as an example for other retailers.
The refrigeration system, using CO_2 (R744) for frozen food, reduces the amount of the chemical refrigerant R507 by 90 % - from 3,600 pounds in a standard store to only 350 pounds in the Scarborough Superstore. Moreover, the synthetic refrigerant will not be circulated throughout the store, but will remain in the plant room. Two secondary loops, with either propylene glycol for medium temperature or CO_2 for low temperature application, will manage the cold distribution.
To reduce the energy associated with store heating, the retail sales area will be entirely heated in winter with condenser heat from the refrigeration system. Using plate-to-plate heat exchangers, all rejected heat is effectively captured to circulate back to the store via a fluid loop. Combining an energy-efficient refrigeration system, intelligent lighting and waste reduction, Loblaw's Scarborough store is the largest one to have earned the "Leadership in Energy and Environmental Design Green Building Rating System" (LEED) certification – a third-party certification programme and the nation's accepted benchmark for the design, construction, and operation of high-performance green buildings.
Source: www.r744.com

A/C, heat pump rule-making begins: AHRI testifies at DOE public meeting

Citing poor cost predictions made during past rulemakings for central air conditioners and heat pumps, AHRI called on the U.S. Department of Energy (DOE) to be more "thorough and vigorous" during its three-year rule-making process that began in June.
DOE will use the rule-making process to determine whether the minimum efficiency standards, which were increased in 2006, should be revised again by 2016.
AHRI Vice President for Regulatory Policy and Research Karim Amrane testified June 12 during a public meeting in Washington, D.C., that DOE severely underestimated the cost increase from a 10 SEER to a 13 SEER system.

Amrane called on DOE to perform thorough analyses in three areas:

- Cost increases associated with higher efficiency standards.
- Potential cost impact from an HFC cap as part of climate change policy.
- Feasibility of various enforcement mechanisms for possible regional efficiency standards.

"DOE needs to step back and review past analyses to understand where improvements need to be made," Amrane remarked at the public meeting. In addition, Amrane emphasized that if regional standards are adopted, that they will present unique enforcement challenges.
Source: AHRI newsletter

Danfoss: high-power VFDs

The VLT® Series variable-frequency drives (VFDs) are now available with ratings up to 1350 horsepower (hp) in 460 and 690 vac. F1 frame models deliver power up to 1050 hp, while F2 frame models deliver up to 1350 hp. They facilitate ease of use by sharing a common platform with the company's smaller drives. A Rittal TS8 enclosure system in IP21 (NEMA 1) or IP54 (NEMA 12) enables easy system expansion, while a Class A1 RFI filter (optional Class A2) reduces EMI/RFI without the need for external filters. The high-power F frame-size VFDs feature a unique cooling design that utilizes a ducted back channel to pass cooling air over heat sinks, with minimal air passing through the electronics area.
Source: ACHR News

Denso tests heat pumps using bio-diesel fuel

Denso Corp. announced that it has begun testing industrial heat pump air conditioners using biodiesel fuel at its headquarters in Kariya, Aichi, Japan.

"Denso expects to significantly reduce carbon dioxide (CO_2) emissions from industrial heat pump air conditioners by using biodiesel fuel," said Kenji Ohya, senior managing director in charge of Denso Corp.'s Sales Group. "We aim to introduce biodiesel engine heat pump air conditioners for industrial use by 2010."
In its testing, Denso is examining the system's compatibility and durability when using biodiesel fuel, refined from cooking oil used at the company's cafeterias, to power heat pump air conditioners. Denso estimates that a heat pump air conditioner using biodiesel fuel can reduce CO_2 emissions by approximately 90 percent compared to the company's conventional kerosene heat pump air conditioner, currently marketed only in Japan.
Denso said it is focusing on reducing CO_2 emissions in all company activities, including product development, manufacturing, procurement, and distribution, to reduce its impact on global warming.
Source: ACHR News

DOE offers $90 million for enhanced geothermal systems

DOE issued a Funding Opportunity Announcement (FOA) last week for the research, development, and demonstration of enhanced geothermal systems (EGS), an advanced geothermal technology that drills deep wells into hot rocks, fractures them, and circulates a fluid through the fractures to extract heat. EGS technologies can be used to create new geothermal reservoirs or to revitalise existing geothermal reservoirs that are underperforming. The FOA offers up to $90 million over four years, of which $40 million will go toward research and development (R&D) projects for the technologies needed to commercialise EGS, and $50 million will go toward demonstration projects that revitalise existing unproductive geothermal reservoirs.
The R&D projects will target the technologies needed to create reservoirs at temperatures up to 300 °C (572 °F) and depths as great as 10 kilometres (6.2 miles). They will address specific needs identified in a recent DOE report, "An Evaluation of Enhanced Geothermal Systems Technology." According to a recent study by the Massachusetts Institute of Technology, a reasonable R&D investment in these technologies could create the opportunity to develop 100 000 MW of geothermal power in the United States by 2050, an amount equal to 20 % of the current U.S. generating capacity. See the reports from DOE and MIT in the EGS technology section of DOE's Geothermal Technologies Program web site.
Source: eere network news

Researchers developing miniature refrigeration system for computers

Researchers at Purdue University are developing a miniature refrigeration system to fit inside laptops and personal computers. The research focuses on miniature compressors and evaporators. The researchers have developed a model for designing tiny compressors that pump refrigerants using penny-size diaphragms. According to the researchers, miniature refrigeration would dramatically increase how much heat could be removed compared to conventional cooling systems, which use a fan to circulate air through heat sinks attached to computer chips.
Source: The HVAC&R Industry for June 26, 2008

"geoJETTING" drilling technology developed by the Bochum University of Applied Sciences

A team at the University of Applied Sciences Bochum (Germany) in cooperation with Vaillant has developed a new technology for drilling.
The team of Prof. Rolf Bracke from the university's geothermal centre received the prestigious "Ruhr 2030 award" for this innovation. The technology uses water at high pressure to "cut" the hole into the ground. The soil is completely broken up by water and pressed into the surrounding area.
This technology increases the efficiency of geothermal drilling. It will be distributed by Vaillant GmbH, Remscheid.
Source: www.ehpa.org

Heating and cooling with thermal energy: A large-scale project in Bielefeld

The Cornelsen Publishing House GmbH & Co. KG has built a new 5000 m2 storehouse and house for commissioning works with geothermal heating and cooling, which serves as a heater for the halls in winter and as an air conditioner in the summer for cooling. With costs of 400,000 EUR plus 150,000 for the floor heating, the investment costs for geothermal energy facilities are 30 % more than those of natural gas heating systems. However, 45 % reduced operating expenses give a payback time of about seven years or, with the high increase of energy prices, proportionately earlier. The heat exchange is provided by 28 ground probes that are sunk 130 meters into the Jurassic mudstone (German). The total primary demand is just 38 kWh/m².
Source: energy-server newsletter | Issue 88

Fraunhofer ISE and SorTech develop a solar cooling system for series production

Fraunhofer ISE and SorTech develop a solar cooling system for series production
A solar-powered adsorption cooling machine in connection with geothermic probes has been in operation since 2007 at the Fraunhofer ISE in Freiburg. The machine, which produces 5.5 kW of power, comes from the first prototypes production series from SorTech AG. The machine cools the kitchen of the institute's cafeteria in the summer and heats it in the winter, much to the satisfaction

of the solar researchers in Freiburg. The solar part of the system is a 20 m² large flatbed solar collector panel on the roof of the institute building, which provides about 60 % of the necessary operating energy. If there is not enough sunlight, the heating network delivers additional heating to keep the cooling systems running. Due to the low operating temperatures of around 60 °C, this adsorption cooling machine is perfect for solar cooling and cooling using district heat.

Source: energy-server newsletter | Issue 89

Markets

ITRE committee votes on heat pumps in RES Directive

The EU Parliament's ITRE Committee has voted on the text of its report, which is later to be presented to the Parliament.

The focus of the vote was more than 1000 amendments to the DIRECTIVE OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL on the promotion of the use of energy from renewable sources (the RES Directive) presented by the Commission earlier this year.

The vote resulted in agreement (among others) to amendments that enhance the definition of renewables (Article 2) with paragraphs on aerothermal, hydrothermal and geothermal energy sources. In addition, the ITRE Committee report on the RES Directive will include all types of heat pumps as a technology that uses renewable sources (Article 5).

The next steps in this process are the vote on the ITRE Report in the parliament (scheduled to take place early October) and in the Council.

Source: www.ehpa.org

DOE to guarantee $10 billion in loans for efficiency, renewables

DOE is offering $10 billion in loan guarantees for projects involving energy efficiency, renewable energy and advanced transmission and distribution. The agency is seeking projects relating to biomass, geothermal, solar, and wind energy, as well as projects involving hydro power, alternative fuel vehicles and energy efficiency. In addition to general energy efficiency projects, the solicitation specifically requests projects relating to energy-efficient building technologies and efficient electricity transmission, distribution, and storage. DOE intends to issue loan guarantees for stand-alone projects, as well as projects relating to manufacturing technologies and the large-scale integration of renewable energy, energy efficiency and energy storage technologies into the electrical grid. The agency issued a solicitation on Monday for the loan guarantees, along with two solicitations for nuclear power that increase the total loan guarantee package to $30.5 billion.

Source: EERE Network News

Groundwater and geothermal associations join forces

The National Groundwater Association and the Geothermal Heat Pump Consortium Inc. have signed a memorandum of understanding to foster cooperation in addressing shared interests involving groundwater and geothermal heating and cooling.

Specifically, the agreement between the groups is intended to serve common interests in:

- Promoting national energy efficiency and environmental protection
- Protecting groundwater from contamination
- Creating business opportunities for qualified drilling contractors
- Creating business opportunities for manufacturers and distributors of drilling equipment and related products.

NGWA executive director Kevin McCray, CAE, and GHPC executive director John Kelly signed the agreement.

"With the geothermal heating and cooling market on the rise, it makes sense for our two organisations to work closely together to use and protect groundwater," McCray explains. "Both our organisations are committed to high standards of professionalism that will help ensure the health and vitality of this market."

Kelly adds, "Drilling contractors are the key to the proper installation of the ground heat exchangers that allow geothermal heat pump systems to outperform other heating and cooling technologies. By working together, we can grow the capacity needed to meet the accelerating demand for these systems."

Source: National Driller June 2008 e-Newsletter

Canadian Geoexchange Coalition to release residential design manual

The Canadian GeoExchange Coalition (CGC), Canada's industry association for ground-source heat pump technology, will release a 380-page manual entitled "Design of Residential Ground Source Heat Pump Systems" on October 15th, 2008. The manual, developed through a CGC partnership with a group of Canada's leading technical Colleges and industry specialists, reflects over eight months of work from CGC and its partners.

This addition to the training portion of CGC's Global Quality GeoExchange™ Program® is intended to help answer North American stakeholder desires, expressed over more than a decade of consultations at all levels of the geoexchange industry, to have state-of-the-art, current training materials which reflect the reality of Standards and best practices in north America.

"The Residential Designer manual fills an important gap in available information for heat pump designers. The lack of quality, current training information for small system designers and installers was actually the genesis of the entire CGC quality program" said CEO Denis Tanguay. "The lack of Canada-relevant training material has been a critical barrier for about fifteen years in Canada, and this is but another CGC step in addressing this barrier."

Source: http://www.geo-exchange.ca/en/UserAttachments/news248_PR%2002-09-2008_E.pdf

IEA Heat Pump Programme

Conference Proceedings from Zurich IEA Heat pump conference now available on CD

You can order the conference proceedings from the 9th International Energy Agency Heat Pump Conference in Zurich (20-22 May 2008) in our publication section

Link: http://www.heatpumpcentre.org/publ/HPCOrder/default.aspx?ReportId=649#Report

Annex 29 Ground-source heat pumps – Overcoming market and technical barriers

Workshop in Zurich, May 19, 2008

A workshop on Annex 29, "Ground-source heat pumps – Overcoming market and technical barriers" was held on May 19, 2008, in conjunction with the 9th IEA-HPP Heat Pump Conference "Advances and Prospects in Technology, Application and Markets", in Zurich, Switzerland. All the participating countries, and China, gave presentations on developments on ground-source heat pumping technologies, demonstration projects and market development. More than 50 people from all over the world attended this workshop.

Hermann Halozan, OA of Annex 29, gave an overview of the findings of the project. Ground-source heat pumps are one of the best heating and cooling systems presently available. They are suitable for small systems such as single-family houses, as well as for large commercial buildings. They can be used for new construction sites as well as for retrofit. They offer low operating costs, high efficiency, and the main part of the energy delivered for heating purposes is energy from the environment - renewable energy. If the electricity is from renewable sources – hydro, solar, wind or biomass – heat delivery is fully renewable. However, although the operating costs are extremely low, the first cost can be a barrier for installing such systems, especially in the retrofit case.

Hermann Schranzhofer's presentation, "Buildings and buildings characteristics", gave an overview of the boundary conditions for heating and cooling. Monica Axell and Roger Nordman described the situation in northern Europe in their "GSHP for single-family houses in Sweden - Present status and trends". Sayaka Takeda Kindaichi presented "The best practical GSHP system for a modern low-energy house in the northern part of Japan", while Vasile Minea presented "A residential heat pump with staged vertical ground-coupled direct expansion heat exchanger", describing an excellent installation in Canada. Moonis Ally and Van Baxter's paper, "Enhanced Ground Coupling of Heat Pumps with Solid Water Sorbents", described a new approach for small installations. Rene Rieberer told us of "CO_2 ground heat exchangers" and Katsunori Nagano of "A novel design and performance prediction tool for GSHP and Estimation of ground thermal conductivity using optical fibre thermometers". The final presentations were concentrated on large systems, Jörn Stene presented "Large-Scale Ground-Source Heat Pump Systems in Norway", Göran Hellström "Large-Scale Heat Pump Applications in Sweden", and Xu Wei "Ground source heat pumps in China".

Ground-source heat pumps have the capability of significantly reducing energy demand in the building sector. They are one of the key technologies for reducing CO_2 emissions and energy demand by using renewable energy sources in a highly efficient way.

Hermann Halozan
OA Annex 29

IEA HPP Annex 30 Workshop "Retrofit Heat Pumps for Buildings" 19. May 2008, Zürich/Switzerland

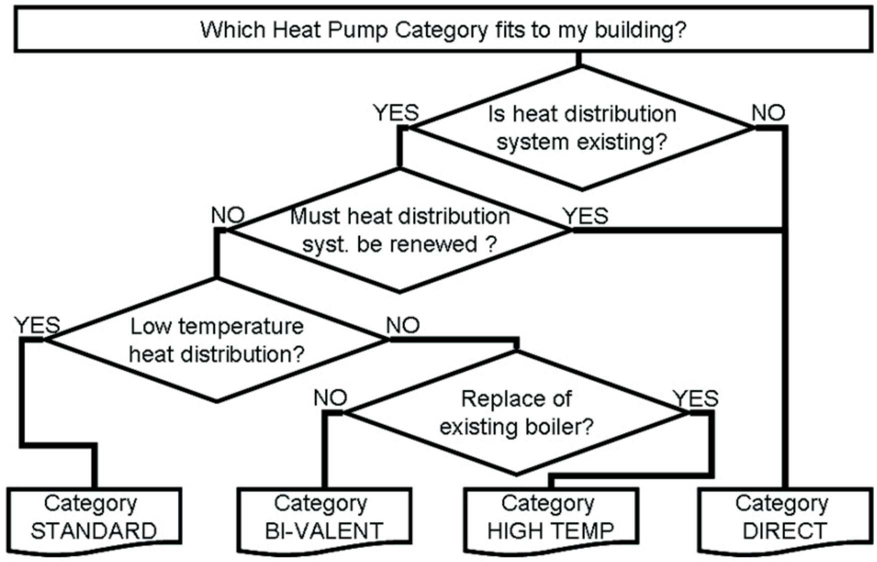
The workshop was organised in connection with the 9th IEA Heat Pump Conference, and presented and discussed the main results of the HPP Annex 30 study on the use of heat pumps for retrofit of buildings. Why do we need a special annex on retrofit heat pumps?

There are more than 150 million dwellings in Europe. Around 30 % were built before 1940, 45 % between 1950 and 1980, and only 25 % after 1980. Around 70 % of them have an energy performance inferior to that of new heating systems, and will have to renew their fossil fuel-based heating equipment in the next few years.

Energy efficiency, reduction of CO_2 emissions and the use of renewable energy in the built environment are the most important aspects that must be addressed in the near future. The Energy Performance of Buildings Directive is therefore an additional driver for heat pumps. However, with the exception of Sweden, the heat pump market in Europe is still concentrated on new one- and two-family houses.

Recognising the potential of the retrofit market, the IEA Heat Pump Program (HPP) added Annex 30, an international collaboration on Retrofit Heat Pumps for Buildings. The annex started in spring 2005 and will run until the end of 2008. A main challenge for Annex 30 has been the limited availability of heat pump technology suitable for retrofitting the different situations in existing buildings.

The primary focus of this annex has been on domestic buildings. In order to reach the goals of the annex, solutions have been found and experience gained on:

- The application of available heat pumps in standard buildings that have been improved, resulting in a reduced heat demand.
- 

  The development and market introduction of new high-temperature heat pumps that use a compact source for application in existing buildings.
- The use of reversible (heating-cooling) heat pumps (air-to-air or air-to-water) in buildings without centralized heat distribution systems.

The presentations at the workshop were concentrated on country reports from Germany, The Netherlands, France and Sweden. Selected case studies of retrofit heat pumps in existing buildings demonstrated the technical capability of heat pumps already today able to meet an essential part of the heating and hot water demand in Europe.

A flowchart was presented, providing a guideline for determining roughly which category of heat pump would be suitable in each case.

The workshop clearly demonstrated the importance of retrofit heat pumps as an important step toward reduction of fossil fuel consumption and related CO_2 emissions in buildings.

The following final steps have been agreed:

There will be a final workshop of Annex 30 in connection with the new Chillventa – International Trade Fair Refrigeration • Air Conditioning • Heat Pumps on 14 October 2008 in Nürnberg/Germany, Exhibition Centre, CCN Ost. See http://www.chillventa.de/en/visitor/supporting_program/

The detailed results of the annex will summarized in a final report, subdivided in

- Executive Summary, to be published as open literature, with the most important, but not confidential, results of the Annex
- Special reports for information to defined target groups, e.g.: Politicians, heat pump industry, R&D organisations, sponsors and housing cooperatives.

Prof. Dr.-Ing. Hans-Jürgen Laue
OA, Annex 30

Annex 31 workshop in Zurich, May 19, 2008.

Some recent outcomes of the Annex 31 activities have been presented in this workshop. Annex 31, "Advanced Modelling and Tools for Analysis of Energy Use in Supermarket Systems", was officially started in January 2006 and is scheduled to continue until December 2008. However, an extension has been proposed to the IEA Executive Committee in order to benefit from the extensive collection of supermarket energy use data collected by US EPA. Sweden is the designated Operating Agent, acting through the Royal Institute of Technology. Participating countries are Sweden, Canada, Germany, the United Kingdom and United States.

The workshop included presentations of case studies of energy-efficient supermarkets from the United States, Sweden, Germany and Canada. Modelling and simulation tools used in the energy analysis of supermarket systems have been described and discussed. Special attention has been paid to the topic of heat recovery and improving the utilization of the refrigeration system in supermarkets to meet simultaneous cooling and heating needs.

The workshop offered an excellent opportunity for those involved in supermarket systems research, development and design to keep track of the latest developments and to exchange ideas with other experts in the field.

Current activity in the annex is devoted to collecting experience from modelling of supermarket systems. Each country will report on modelling activities and on available software (whole supermarket computer models), as well as on useful component models specific to supermarkets.

A series of web-based electronic meetings will be hosted during the autumn. The theme will be modelling and simulation of full supermarket energy systems and of component models.

Please visit the Annex web site on address www.energy.kth.se/annex31

Per Lundquist
OA, Annex 31

Summary Workshop IEA HPP Annex 32 – Economical heating and cooling systems for low-energy houses

IEA HPP Annex 32 started at the beginning of 2006 with nine participating countries: AT, CA, CH (operating agent), DE, JP, NL, NO, SE and US. Annex 32 is committed to the further development of heat pump systems in low-energy houses (LEH). Obtaining practical experience by field testing of heat pump systems for LEH is further objective. The workshop, that was held in conjunction with the 9th IEA HP Conference in Zurich on May, 19-22 2008, presented interim results of the national projects.

As an introduction, C. Wemhöner of the Institute of Energy of the University of Applied Sciences of Northwestern Switzerland presented features of LEH and the current market state. Most participating countries have a strong growth of both built low energy house and heat pump markets. An exception is the Netherlands, where any house on the market is sold,, with the result that there is generally a low thermal quality of building envelopes. However, as outlined by O. Kleefkens of SenterNovem, niche market players are currently specialising in LEH in combination with integrated heat pump and ventilation system solutions. The Netherlands' participation in Annex 32 is a driver of market introduction initiatives, by establishing guidelines and best practice examples in the LEH field.

The first part of the workshop was dedicated to heat pump system developments for LEH. R. Rieberer of the Institute of Thermal Engineering of Graz Technical University described the Austrian development of a heat pump in the 3-5 kW capacity range. A system analysis resulted in a decision for a brine-to-water heat pump with CO_2 refrigerant. A prototype system will be built and lab-tested, and system simulation using the prototype results will yield the seasonal performance factor (SPF) of the overall system design. T. Afjei of the Institute of Energy in Building of the University of Applied Sciences of Northwestern Switzerland presented standard system solutions with heat pumps for space heating and cooling. The first system solution, a ground-source heat pump with free cooling, is currently being field-tested in an ultra-low-energy house. The winter operation shows an overall SPF of about 3.8 for space heating and domestic hot water production. Field monitoring of the space cooling operation and further system configurations will be evaluated in

the ongoing project. M. Voss Lapsa of the Oak Ridge National Laboratory reported on the US development of a highly integrated heat pump providing space heating, domestic hot water, space cooling and ventilation (including humidification and dehumidification). Preliminary annual system evaluation based on prototype laboratory testing results and simulations yielded a reduction in the range of 46 %-67 % of energy consumption of the air-source heat pump, and 52 %-66 % of the ground-source heat pump, compared to a state-of-the-art system for typical sites in the US climate for a net-zero-energy house (NZEH). Initial cost estimates indicate simple payback times of 5-10 years for the air-source heat pump, and 6-14 years for the ground-source heat pump, including the borehole heat exchanger. S. Ruud of SP Technical Research Institute of Sweden pointed out in his presentation that implementation of revised building regulations by 2009 could require higher levels of thermal insulation of buildings and installation of heat pumps, with a resulting specific energy use in new single-family houses with heat pumps being in the range of ultra-low-energy houses. S. Takeda-Kindaichi of the University of Tokyo described common Japanese system solutions of single or multi-split heat pumps for space heating and cooling in reverse operation for the moderate climate zone, where 80 % of the Japanese population lives. However, system simulations revealed that common design proce-

dures lead to oversized heat pumps in LEH applications. The project will determine appropriate design methods. J. Stene of SINTEF Energy Research in Norway described the design of a CO_2 heat pump water heater for use in LEH blocks of flats. System design and simulation show that the unit is both an economical and energy-efficient solution, resulting in energy saving of about 75 % compared to direct electrical water heating and about 25 % compared to common solar domestic hot water heating systems in Norway.

The second part of the workshop was dedicated to the results of field monitoring of systems. M. Miara of the Fraunhofer Institute of Solar Energy systems presented preliminary results of a German field test in co-operation with seven manufacturers and two utilities, including about 110 state-of-the-art heat pumps installed in LEH. All systems showed an SPF of 3 or more for space heating and domestic hot water production. Ground-source heat pumps showed themselves to be the most efficient systems, as water-source heat pumps have a quite high energy consumption for the auxiliary components, in particular the source pump. The results of the air-source heat pump are based mainly on winter operation. Follow-up work includes the installation of further systems and a more detailed evaluation of single systems.

The Canadian contribution was dedicated to a field test of 2 EQuilibrium NZEH in the frame of a nationwide field test of 13 Equilibrium house led by the Canada Mortgage and Housing Corporation. José Candanedo of the Concordia University in Montreal described the field monitoring of two NZEH in co-operation with Hydro-Québec. Both houses incorporate a heat pump and solar components, e.g. heat recovery from ventilation air of ventilated PV-system. J. Stene presented results of field monitoring of an innovative prototype propane heat pump using lake water, which is installed in an ultra-low-energy house in southern Norway. Results of the first winter season show an SPF of 3.3 for space heating and domestic hot water production. However, further potentials for optimisation of the evaporator and expansion valve have been discovered. K. Nagano of Hokkaido University in northern Japan reported the results of field tests of a ground-source heat pump in a low-energy house in the cold climate region. The first field test shows a CO_2 reduction of about 60 %, in comparison with that of a state-of-the-art system. An optimised system is being investigated in a second field test.

C. Wemhöner concluded the workshop. Expected results of the Annex 32 are new HP system concepts for LEH in low capacity range and with natural refrigerants. Field monitoring results will identify field-proven best practice solutions. Design recommendation for standardised system solutions will be derived from both the system simulations and the field trials.

After the workshop, France joined IEA HPP Annex 32, represented by Electricité de France R&D. Work in Annex 32 will continue until the end of 2009. Further information and all workshop presentations for download can be found on the Annex 32 website at http://www.annex32.net, under publications.

Carsten Wemhöner
OA, Annex 32

9th IEA Heat Pump Conference Report on Workshop Annex 33: Compact Heat Exchangers in Heat Pumping Equipment

A workshop covering the work of Annex:33 - Compact Heat Exchangers in Heat Pumping Equipment - was held during the 9th IEA Heat Pump Conference. Delegates from the participating countries - UK, Sweden, USA and Japan - described the work being undertaken. Austria also attended the workshop, and was welcomed as a new participant in the Annex. Some 25 other delegates from the conference attended the workshop.

Short presentations were given by each participant:

Introduction and background to the Annex. *Dr Peter Kew, Heriot-Watt University, Edinburgh, UK.:* The objective of the annex is to present a compilation of the roles that compact heat exchangers (CHEs) can play in heat pumping equipment, directed at minimising both direct and indirect effects of the equipment on the local & global environment as a result of manufacture, operation and/or disposal of the plant. Dr Kew outlined why compact heat exchangers were attractive from a heat transfer point of view. He then went on to summarise some of the market studies which have recently been carried out in the UK, which assess the potential for heat pumps and compact heat exchangers in UK industry.

Development of non-fluorinated energy-saving refrigeration and air conditioning systems in Japan. *Professor Eiji Hihara, The University of Tokyo, Japan:* Prof. Hihara described how the Japanese Ministry of Economy, Trade and Industry is giving considerable support through the New Energy and Industrial Technology Development Organisation(NEDO). The use of compact heat exchangers is likely to play a part in the development of high-efficiency heat pumps using high pressure and/or flammable fluids, for which a low fluid inventory is important from a safety point of view.

Heat transfer characteristics of carbon dioxide flowing in a microfin tube. *Professor Shigeru Koyama, Kyushu University, Japan:* Heat transfer involving supercritical CO_2 provides new challenges, not least that reliable correlations are not available. Professor Koyama described his experimental study of heat transfer to CO_2 flowing in small diameter (6 mm OD) tubes with internal microfins of various geometries, and presented suggested correlations for both heat transfer and pressure drop.

Research and development on compact heat exchangers in Sweden P*rofessor Bjorn Palm, Royal Institute of Technology, Stockholm, Sweden.* this presentation discussed a number of projects related to compact heat exchangers including; optimisation of plate heat exchangers, a CO_2 heat exchanger, development and test of an aluminium minichannel heat exchanger, boiling heat transfer enhancement using a porous surface, and visualization and correlation of heat transfer in single minichannels. Professor Palm concluded that heat transfer and pressure drop during flow boiling in minichannels is still not well understood, and the influence of surface structure and surface tension needs to be investigated further. For practical applications, the low heat transfer coefficient observed at high vapour fractions must be considered in the design of compact heat exchangers.

Update on the US contributions. *Professor Clark Bullard, University of Illinois at Urbana-Champaign, USA.* Professor Bullard gave a précis of the projects which will form the basis of the US contribution to the Annex. Two projects have been completed: State-of-the-Art and Potential Design Improvements, for Flat-tube Heat Exchangers; and Flat-Tube Heat Exchangers in Air-Conditioning and Refrigeration Applications. Projects underway are 1) supercritical gas cooling and near-critical pressure condensation of refrigerant blends in microchannels; 2) void fraction measurement and modelling for condensing refrigerant flows in small diameter tubes; 3) novel materials for heat exchangers; and 4) modelling and analysis of heat exchangers with carbon-fibre structures. Details of these projects may be obtained from ARTI (www.arti-research.org)

Proposed input from Austria. *Dr Michael Monsberger, Arsenal, Austria:* Dr Monsberger, attending his first meeting of the Annex, introduced Arsenal's capabilities and facilities, in particular those of the Heat Pump Test Center, and explained how these activities could contribute to the Annex. The principal proposed input will be the analysis on data gathered from heat pump measurements.

The presentations were followed by a brief open discussion.

All presentations are available on the Annex website: www.compactheatpumps.org

Peter Kew
Deputy OA, Annex 33

Ground Reach: International workshop on Ground-Source Heat Pumps

The GROUND-REACH project ("Reaching the Kyoto Targets by means of a wide introduction of ground-coupled heat pumps (GCHPs) in the built environment"), is supported by the Intelligent Energy for Europe programme of the European Commission. It started on January 1st, 2006 and will run for 36 months. Its aim is to support and promote GCHPs by a series of studies and a large-scale promotional campaign to key professional groups. It intends effectively to assist in implementing EU policy for both short and long term market penetration of ground-coupled heat pumps. The purpose is to facilitate:

– A better understanding of ground-coupled heat pumps' merits and benefits, and their importance in fulfilling Community policy objectives in relation to Kyoto targets and the Building Performance Directive.
– An increased awareness, improved knowledge and perception of ground-coupled heat pumps technology among key European professional groups for short-term market penetration.

The different activities of the project address the following points:

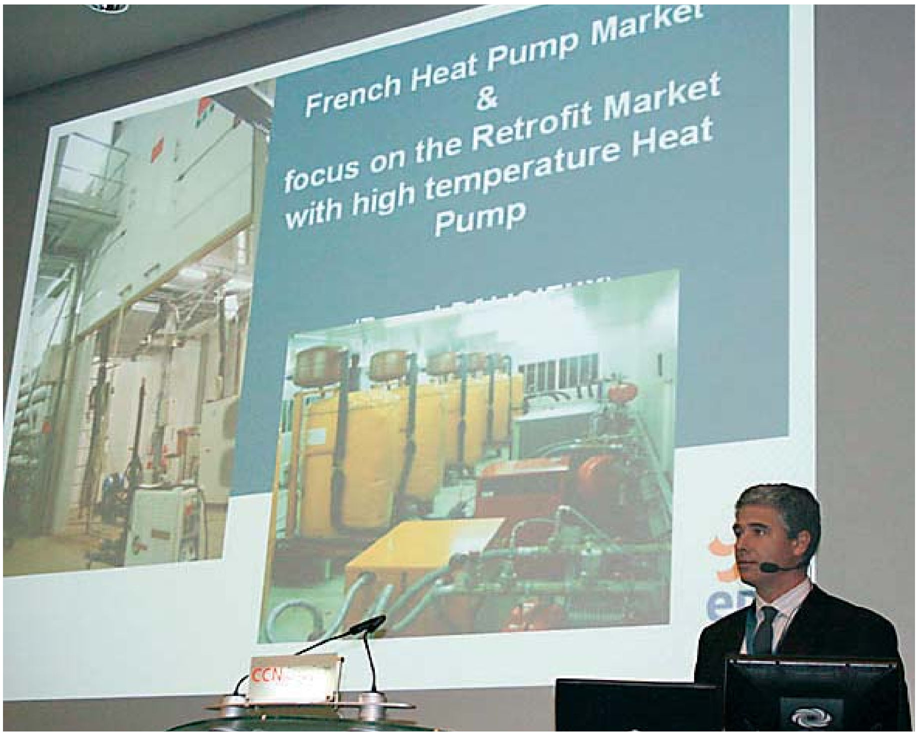
- Estimating the potential of ground-coupled heat pumps for reducing CO_2 emissions and primary energy demand for heating and cooling purposes in the built environment: evaluation of available statistical information, definition of competing heating/cooling technologies, analysis of existing calculation tools, CO_2 emissions calculation.
- Compiling and evaluating existing ground-coupled heat pumps best practice information in Europe: identifying and updating information from all European member states, including case studies, and technical guidelines.
- Analysing the contribution of ground-coupled heat pump technologies for meeting the objectives of the Building Performance Directive: Analysis of the technical, environmental and economic feasibility of ground-coupled heat pump technologies; Guidelines for supporting planners and architects with detailed technical aspects and general questions; Standards review, evaluation and proposals.
- Defining measures to overcome bar-

riers for broader market penetration and setting up a long term dissemination plan: identification of market barriers including legal/regulatory, economic and technical, proposals for long-term EU level interventions to overcome them, including a new directive on RES-Heat.

- Launching a large-scale promotional campaign at European level: brochure, poster, promotional text, presentations, interactive Internet site, setting up the European Geothermal Heat Pump Committee, publications, international conference and exhibition, a series of regional meetings with key professional groups.

The GROUND-REACH consortium consists of the following 21 organisations across Europe:

– Centre for Renewable Energy Sources (CRES) - Greece
– European Geothermal Energy Council (EGEC) - Belgium
– European Heat Pump Association (EHPA)
– Österreichisches Forschungs- und Prüfzentrum Arsenal Ges.m.b.H (Arsenal research) - Austria
– Bureau de Recherches Géologiques et Minières (BRGM) - France
– Ecofys B.V. (ECOFYS) - Netherlands
– The Energy Efficiency Agency (EEA) - Bulgaria
– Escola Superior De Tecnologia De Setubal (EST Setubal) - Portugal
– Fachinformationszentrum Karlsruhe GmbH (FIZ) - Germany
– Geoteam Technisches Büro für Hydrogeologie, Geothermie und Umwelt Ges.m.b.H (GEOTEAM) - Austria
– Associazione Rete di Punti Energia (PUNTI ENERGIA) - Italy
– SVEP Information & Service AB (SVEP) - Sweden
– University of Oradea (UOR) - Romania
– BESEL S.A. (BESEL) - Spain
– COWI A/S - Denmark
– Ellehauge & Kildemoes (E & K) - Denmark
– Flemish Institute for Technological Research (VITO) - Belgium
– Agence de l'environnement et de la Maîtrise de l'energie (ADEME) - France
– Narodowa Agencja Poszanowania Energii S.A. (NAPE) - Poland
– EnPro Engineers Bureau Ltd (ENPRO) - Estonia
– GFE Energy Management (GFE) - Italy



The Ground Reach international workshop (Zurich, Friday, May 23rd, 2008)

The workshop was held as a whole day international event presenting the up-to-date results of the Ground Reach project and dealing with the merits and benefits of GCHP and possible actions to develop their market across Europe.
The workshop was organised in the frame of the HPC 2008 in Zurich, on Friday, May 23rd.
The workshop was divided into three different parts, with the first one describing the heat pump markets across Europe, the benefit of increasing these markets, and the possible way to do so. The second part described several GSHP best practice cases all over Europe, to show that GSHP are an available efficient solution that can be adapted to many configurations. The third part provided technical information on ground heat exchangers and their design rules.
Project website: www.groundreach.eu

Johan Ransquin, ADEME

The Heat Pump Centre at Chillventa

The Heat Pump Centre exhibited at the Chillventa exhibition in Nürnberg, Germany, October 15 – 17. Chillventa is an international trade fair of refrigeration, air conditioning and ventilation and heat pumps, with more than 1500 exhibitors. It offered a good opportunity for HPP to promote the programme to professionals in the field.

International Heat Pump Symposium and final Annex 30 workshop

In conjunction with the exhibition, an International Heat Pump Symposium was held on "Experiences and possibilities of industrial heat pumps for heating, cooling and air conditioning". It was arranged by DKV – Deutscher Kälte- und Klimatechnischer Verein – and a number of interesting presentations was made.
Karl-Heinz Stawiarski from the German Heat Pump Association (BWP) started off with some background information about the German heat pump market, highlighting the potential and challenges for the industry. Among other things, he pointed out that the future market lies in existing building stock where energy efficiency measures will be needed.
Martin Forsén, from the Swedish Heat Pump Association and the European Heat Pump Association, pointed out that industrial heat pumps are a long-neglected market segment with great potential. He showed examples of large heat pump applications, e.g. how heat pumps form part of the district heating and cooling systems in Stockholm.
Jochen Lambauer from the Institut für Energiewirtschaft & Rationelle Ener-

gieanwendung at Stuttgart University presented a study named "Industrielle Großwärmepumpen-Potenziale, Hemmnisse und Best-Practice Beispiele".

Laurent Levacher from Électricité de France (EdF) described the opportunities for high-temperature heat pumps in French industry. He also mentioned the European Centre and Laboratories for Energy Efficiency Research (ECLEER), which EdF has created together with Ecole des Mines de Paris and Ecole Polytechnique Fédéralede Lausanne. The Centre has a heat pump programme of which the main targets are to decrease costs and to increase COP and the integration of heat pumps.

Eberhard Wobst from Thermea. Energiesysteme GmbH, which specialises in high-temperature heat pumps for industrial processes, described CO2 as a refrigerant for these applications.

Klaus Mantel from Ochsner company presented the company's large heat pumps for industrial and residential applications.

Peter Schäftlein from Johnson Controls Systems & Service presented the company's high-temperature heat pump, using water as refrigerant.

Jörg Saar from Danfoss described the choice of compressor technology for different applications.

A final workshop of the IEA HPP Annex 30 Retrofit Heat Pumps for Buildings was also held.

The audience learned that there are more than 150 million dwelling units in Europe, of which 30 % were built before 1940, 45 % between 1950 and 1980, and 25 % after 1980. Many of these dwellings will have to renew or replace their fossil fuel-based heating equipment in the next few years. The project has been focusing on three different solutions:

1. The application of existing heat pumps in already improved standard buildings with reduced heat demand
2. The development and market introduction of new high-efficiency heat pumps with higher heat distribution temperatures
3. The use of air-to-air units in buildings without centralised heat distribution systems

The participating countries, Germany, Netherlands, Sweden and France, described their respective market statuses. In addition, a number of case studies were presented.

A final report of the Annex is expected at the end of this year, with an executive summary being available on http://www.heatpumpcentre.org/.

Ongoing Annexes

Bold text indicates Operating Agent.

Annex 29 Ground-Source Heat Pumps - Overcoming Market and Technical Barriers	29	AT, CA, JP, NO, SE, US
Annex 30 Retrofit heat pumps for buildings	30	DE, FR, NL
Annex 31 Advanced modelling and tools for analysis of energy use in supermarkets.	31	CA, DE, SE, UK, US
Annex 32 Economical heating and cooling systems for low-energy houses.	32	CA, CH, DE, NL, SE, US, JP, AT, NO
Annex 33 Compact Heat Exchangers In Heat Pumping Equipment	33	UK, SE, US, JP
Annex 34 Thermally Driven Heat Pumps for Heating and Cooling	34	AT, **DE**, NL, US

IEA Heat Pump Programme participating countries: Austria (AT), Canada (CA), France (FR), 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.

SYSTEM PERFORMANCE OF HVAC IN A LOW ENERGY HOUSE IN THE COLD REGION OF JAPAN

Sayaka TAKEDA-KINDAICHI, Assistant Professor, The University of Tokyo, Tokyo, Japan
(Formerly Assistant Professor, Hokkaido University, Sapporo, Japan)
Katsunori NAGANO, Professor, Hokkaido University, Sapporo, Japan
Takao KATSURA, Director, Fujiwara Environmental Science Institute Ltd., Sapporo, Japan
Shogo HORI, Graduate Student, Hokkaido University, Sapporo, Japan
Kazuo SHIBATA, President, Nissin Techno Incorporated, Sapporo, Japan

Abstract: This study describes a field test on a low energy house built in the cold region of Japan. The floor area is about 200 m^2. The building envelope consists of well-insulated walls and windows and a floor with high thermal capacity. The total Q-values and C-values are 0.96 W/(m^2·K) and 0.4 cm^2/m^2 respectively.

Heating and cooling are provided by a ground source heat pump unit made in Japan. Continuous heating operation means that the supply temperature for floor heating can be mostly kept low between 30 ºC and 35 ºC throughout the season. The daily average heating output is 3.0 kW, with higher COPs being achieved due to the partial load operation. We can also observe a high seasonal performance. The system COP including the pumping power reaches 4.45. Acceptable indoor environment conditions are maintained by an air change rate of 0.5 hr^{-1}. The room temperature is kept stable above 20ºC all day. The vertical temperatures also show uniform distributions.

From the monthly electric consumption, the largest consumption occurs in January due to the heating operation. The power production by the solar photovoltaic (PV) system nearly meets the load in the summer season. The total amount of annual electricity consumption consists of 37 % for heating, 10 % for DHW and 53 % for ventilation, lighting and other purposes. The real consumption - that is, the difference between production and consumption - is 24 kWh/m^2, which is equivalent to 236 MJ/m^2 in the primary energy quantities. 48 % of the electric power demand can be covered by the PV system.

Keywords: ***low energy houses, HVAC systems, ground-source heat pump systems, COP, SCOP, annual energy balance.***

1 INTRODUCTION

Originally, “how to protect against the summer heat” was the most important issue to be considered in the design of conventional residential buildings for most area of Japan. Many openings on the outer walls can provide natural ventilation through the room space in summer. However, it also means that such construction tends to cause cold and uncomfortable thermal environments in winter. In order to achieve healthy and acceptable environment for occupants throughout the year, improvement of insulation and airtightness of building envelopes have been a fundamental and necessary design concept.

Insulation and airtightness requirements for residential buildings in Japan are set out in the Next-Generation Energy Conservation Standard. Residential buildings in the coldest region of Japan, for example, should be designed not to exceed a heat loss coefficient (Q-value) of 1.6 W/(m^2·K) and an equivalent clearance area (C-value) of 2.0 cm^2/m^2. The insulation levels can be modified for passive solar houses in the standard, which also specifies a ventilation requirement of at least 0.5 air changes per hour.

The heat load can be decreased in well insulated and airtight buildings. In terms of the air conditioning system, low-energy houses require sufficient efficiency even for partial heat load. The use of heat pumps has been recognized to be a solution for further energy savings in low energy houses. In particular, floor heating systems using heat pumps may allow lower supply temperatures due to larger radiation area, thus improving the efficiency of heat pumps.

This study describes a field monitoring on a low-energy house built in the cold region of Japan. The building envelope consists of highly insulated walls and a floor with high thermal mass. A large window area, glazed with low-E argon-filled triple-glazing units faces mainly south. In total, the Q-value and C-value are 0.96 W/(m^2·K) and 0.4 cm^2/m^2 respectively, which are considerably lower than the values in the standard. For heating and cooling, single U tubes are installed into two boreholes and connected to a ground-source heat pump (GSHP) unit made in Japan. The heat from the GSHP is radiated by floor heating in the room space. The variation of supply water temperature needed by the user was monitored by measurements during the heating period. The system performance and indoor thermal environment are also discussed. Moreover, the monthly electricity consumption and production by the solar photovoltaic (PV) system are compared, and finally the annual energy balance is evaluated.

2 SYSTEM SPECIFICATION

2.1 Physical properties of the building envelope

Figure 1 is an external view of the low-energy house. The house is in *Naganuma*, which is located at lat. 43ºN and long. 141ºE and 40 km southeast from *Sapporo*. It has a concrete basement, with wooden construction for the living space. The total floor area is 200 m^2, and the house is generally occupied by two people.

Table 1 lists the physical properties of the building envelope. The thicknesses of the thermal insulation are 240 mm in the ceiling and 186 mm in the outer walls. Floor insulation is provided by 50 mm thick foamed polystyrene boards under the floor slab, which is about 300 mm thick. Cross-linked polyethylene tubes with 13A are buried in the slab for floor heating. 28 mm plywood is laid on the concrete slab and is then covered with 15 mm wooden flooring as the top layer. The total window area is 83 m^2, of which 63 % are south-facing. Passive solar gain is expected in winter. The window area consists mainly of low-E argon-filled triple glazing units. The average heat transfer coefficient including the wooden frame is about 1.3 W/(m^2·K).

The Q value is calculated as 0.96 W/(m^2·K), including heat loss by ventilation of 0.5 hr^{-1}. C value is also measured at 0.42 cm^2/m^2 by an airtightness test.

Table 1; Physical properties of the building

	Insulation	Area [m^2]				
Ceiling	240 mm	132				
Outer walls	186 mm	E	W	S	N	Total
		47	38	55	94	234
Floor	50 mm (with concrete slab of 300 mm)	200 (Base: 23, First: 132, Second: 45)				
Windows	Low-E Triple filled with Argon gas (1.3 W/(m^2K))	E	W	S	N	Total
		6	9	53	14	82
Q value	0.96 W/(m^2K)					
C value	0.42 cm^2/m^2					

Figure 1: External appearance

2.2 HVAC system

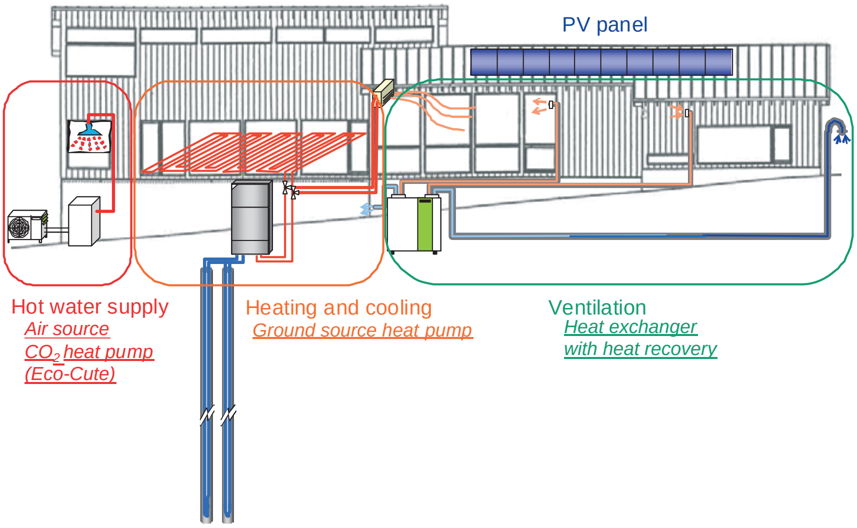


Figure 2: System configuration

2.2.1 Heating and cooling

Figure 2 shows the system configuration of the low-energy house. A ground-source heat pump (GSHP) system is used for heating and cooling. Two boreholes were drilled 5 m apart. Each is about 100 m deep, and a single U-tube was inserted in each one. The groundwater level was measured at -14.7 m from the ground level, and the water temperature was at 10.8 ºC. The stratum consists mainly of mudstone under the groundwater level. The effective thermal conductivity of the ground is evaluated as 1.4 W/(m^2·K) by a thermal response test.

The heat pump unit has an inverter-controlled rotary compressor, so that the heat output can be varied by controlling the compressor speed. Each plot in Figure 3 shows the measured partial load efficiency of the heat pump for the heating operation. Coefficient of performance (COP) is obtained by dividing the heating output (Q_2) by the power consumption of the compressor. At maximum heat output of 10 kW, with an inlet temperature on the primary side (T_{1in}) of 0 ºC and an outlet temperature on the secondary side (T_{2out}) of 35ºC, the COP is 3.7. COP varies with Q_2 and can be increased to 4.5 at a heat output of 4 kW. The figure also shows lines approximated by a multi-regression analysis, which is useful for a performance prediction for the longer term.

The heat produced by the heat pump is distributed to the room space through floor heating. Cooling is also possible through fan coil units in summer. The heat pump provides a constant supply temperature T_{2out}, which is set by the user depending on the heating load.

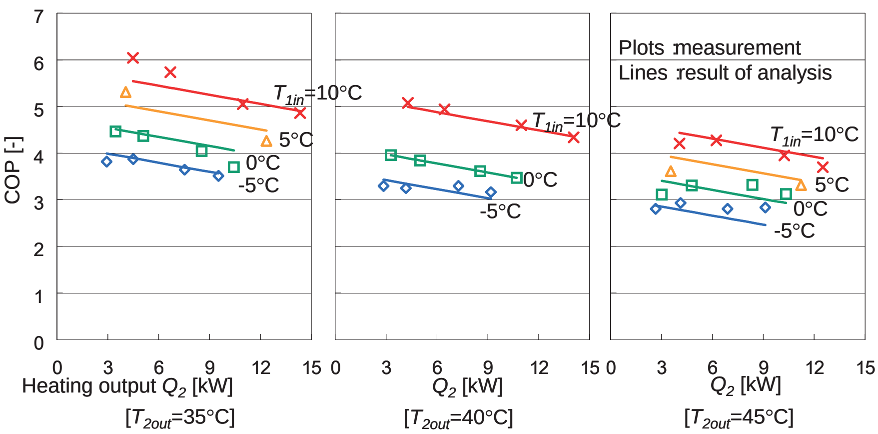


Figure 3: Partial load efficiency of the GSHP (Data from SUNPOT Co., Ltd.)

Solutions of ethylene glycol and propylene glycol are used as heating media in the primary and secondary sides of the heat pump. Volume densities are 30% and 20% respectively.

2.2.2 Ventilation

First, outside air for the ventilation is preheated in a heat tube, which is buried 1.55 m below the ground surface, and is about 50 m long. Ribbed PVC pipes with a diameter of 150 mm or 200 mm are used for the heat tube. In order to prevent problems with condensation during the summer season, the heat tube slopes slightly and condensed water can be drained from the lowest point.

Next, a mechanical ventilation unit with heat recovery is applied. The unit has a temperature exchange efficiency of 90 %.

2.2.3 Hot water supply

An "*Eco cute*" heat pump water heater, developed for the cold region, has been installed. It is an air-source heat pump, using CO_2 as the refrigerant. The hot water produced in the heat pump is stored in a 460 l tank.

2.2.4 Others

The building also has a solar photovoltaic (PV) power generation system. Polycrystalline PV modules integrated with a metal roofing material are mounted as shown in Figure 1. The total generating capacity with 1000 W/m^2 insolation is 6.8 kW.

3 HEAT LOAD CALCULATION

3.1 Conditions for calculation

Firstly, the heat load of the low-energy house is calculated using the SMASH software program. The actual size, direction and physical properties of the building and the other operating method are given for the calculation. The weather data for *Sapporo* is used as the input. The heating operation is continuous all the day, in which room temperature is set at 22 ºC. Cooling operation is intermittent, at a set temperature of 26 ºC.

3.2 Results and discussion

The upper diagram in Figure 4 shows variations of the hourly heating and cooling load, while the lower diagram shows operation of the heating supply.

A total of 6,316 kWh of heat is needed for space heating in a season. The peak load reaches 5.5 kW, but is needed for only a small proportion of the time. 97% of the heating load is less than 4 kW. The result suggests the possibility of using a lower supply temperature, thus improving the COP even in the cold region of Japan.

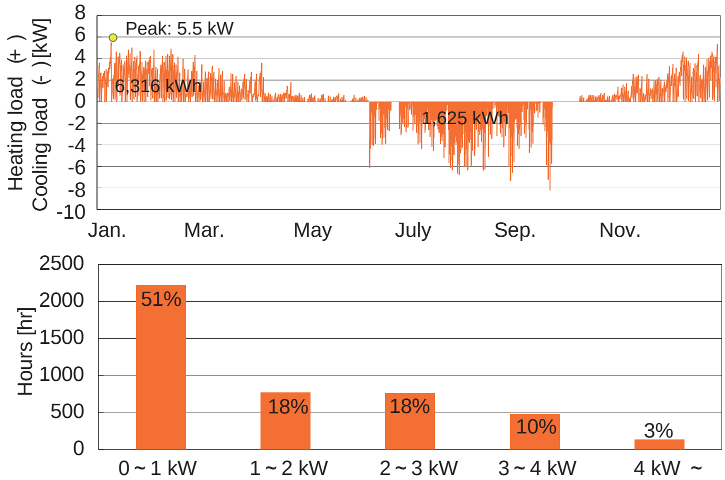


Figure 4: Variations of heating/cooling load (upper) and frequency of appearance of the heating load (lower)

4 RESULTS AND DISCUSSION ON THE FIELD MONITORING

4.1 Temperature variations

The measurement was mainly conducted in the winter season from 2006 to 2007. Figure 5 indicates seasonal variations of temperatures and daily average heating output from the heat pump. The temperatures are the daily averages during operation.

This was a relatively mild winter, with a lowest ambient temperature of -6.8 ºC. The supply temperature to the floor heating T_{2out} was changed by the user depending on the thermal comfort or the weather forecast. The highest temperature was around 40 ºC for a few days at the beginning, but generally ranged between 30 ºC and 35 ºC.

On the primary side, it can be seen that the return temperature from the ground T_{1in} is mostly higher than the outdoor air temperature, indicating the advantage of using the ground as a heat source. The temperature is also relatively stable around 0 ºC, although it decreases to -3 ºC at the lowest when the set value of T_{2out} is changed. This suggests that there is little drop of temperature throughout the season, despite the continuous heat extraction from the ground.

The figure also shows the room temperature, which is measured at a ventilating opening for exhaust in the room. It can be seen that the daily average temperature can almost be kept above 20 ºC. Detailed room temperatures measured in the room space on January 17th in 2007 are shown in Figure 6. T_{2out} is set at 30 ºC during the day. The vertical distributions of the room temperatures are kept constant at all times. The lowest distribution is seen in the early morning, when the outdoor air temperature is -4.1 ºC. However the room temperatures can be almost 20 ºC even in the severe condition. Therefore, comfortable thermal environment conditions can be maintained even though the supply temperature is quite low, around 30 ºC, in such a building with a high thermal insulation standard. On the other hand, the room temperature increased to 28 ºC due to solar heat gain in the daytime. Such overshooting would be improved by heat storage, in an additional storage device for example.

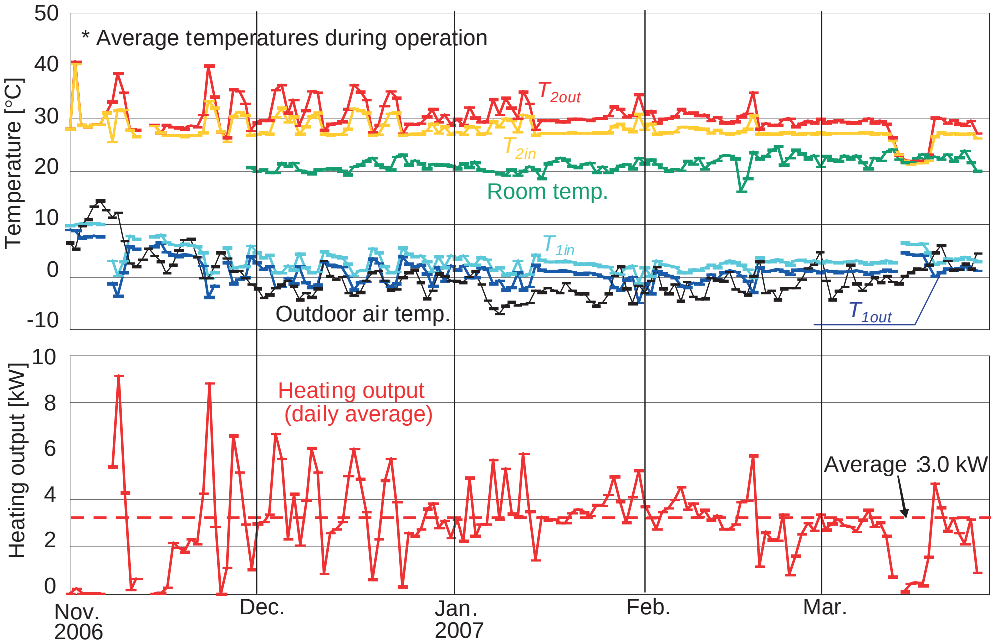


Figure 5: Seasonal variations of temperatures (upper) and daily average heating output (lower)

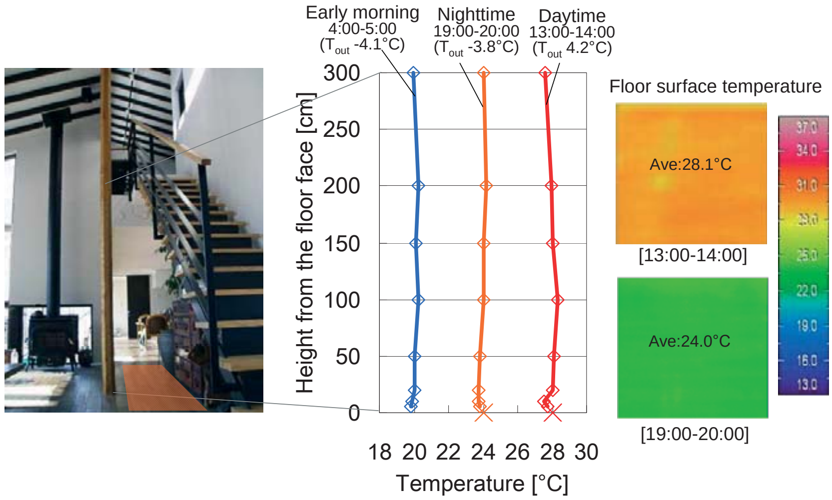


Figure 6: Vertical distributions of room temperature and floor surface temperatures measured by an infrared camera

4.2 Performance of the GSHP system

The daily average heating output ranges from nearly 0 to 9 kW as shown in Figure 5. The lower ones from 2 kW to 6 kW occur more frequently than the higher one above 6 kW. The seasonal average heating load is calculated as 3.0 kW, which is about one third of the heating capacity of the heat pump mentioned before. Such partial load operation can lead to improvement of the performance.

Figure 7 shows the monthly integrated heating output and the monthly average COP and $SCOP_1$. The seasonal heat balance of the GSHP system is also shown in Figure 8. $SCOP_1$ is an index to show a system performance of the GSHP, in which the power consumption from the circulation pump in the primary side is included in the calculation of the COP.

The monthly heating output varies, depending upon the outdoor air temperature. The maximum is 2,591 kWh, and occurred in January. Assuming that the system operated for 24 hours every day during this period, the daily average heating output is calculated as 3.5 kW, which means that the heat pump can work with higher efficiency due to the partial load operation even in the coldest season. The monthly average COP ranges from 4.66 to 5.72. The total electrical power consumption is 2,455 kWh, the heating output is 12,624 kWh and then average COP is calculated as 5.14. This high performance seems to be due to the following three reasons.

1) Primary temperature condition: The temperatures on the primary side can be mostly kept higher than 0 ºC since the system has enough length of boreholes at 200 m.
2) Secondary temperature condition: The supply temperature can be set quite low, around 30 ºC, even in the coldest period, since the low-energy house is well insulated and has a large area of radiation through floor heating.
3) Partial load operation: The heat pump can be operated in the partial load range, giving the higher COP as shown in Figure 3.

The seasonal $SCOP_1$ reaches 4.45. The high system performance may be the result of the use of the adequate circulation pumps. It should be noted that excess designs of the circulation system may cause decrease the system performance.

Figure 8 also shows comparisons of CO_2 emissions from different heat source systems. The thermal efficiencies in the boiler systems are assumed 0.85. It can be seen that the GSHP system provides a CO_2 reduction effect of 58% compared to the oil-fired boiler system in the estimation.

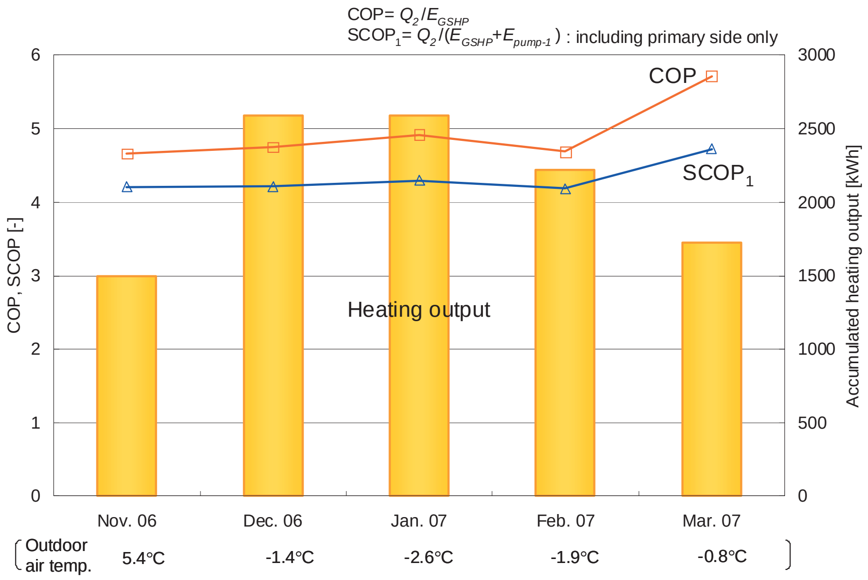


Figure 7: Monthly integrated heating output and monthly average COP and SCOP$_1$ of the GSHP system

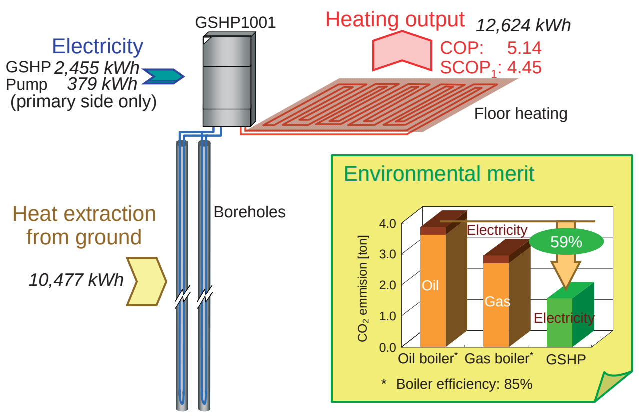


Figure 8: Seasonal heat balance of the GSHP system

4.3 Preheating effect of the heat tube

The temperature at the outlet of the heat tube - that is, at the inlet of the heat recovery unit - was 9.7 ºC at the beginning of the heating season, gradually dropping to 2 ºC at the lowest, as shown in Figure 9. The average heat extracted from the ground during the heating period was calculated as 220 W, which is equivalent to 4.4 W per unit length of the heat tube. This is used effectively for heating up the supply air and results in 14 % reduction of the heating load for the ventilation.

In addition, the authors recognized that the heat tube has a big advantage in avoiding defrost operation of the ventilation unit, as the outlet temperature of the heat tube never dropped below 0ºC.

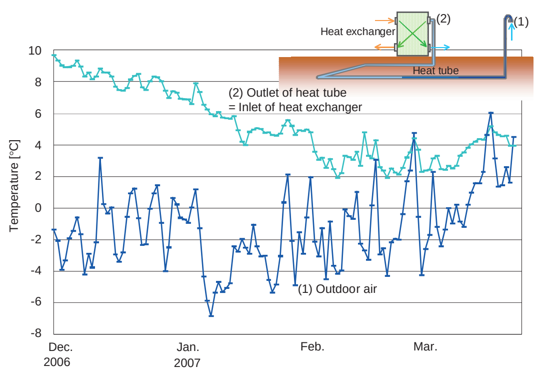


Figure 9: Variation of outlet temperature from the heat tube

4.4 Air change rate

According to the building management code, mechanical ventilation equipment which provides a capacity of ventilation rate of 0.5 air changes per hour has to be installed for residential buildings. In this case, the air volume of the house is about 500 m^3 and the measured centralized supply and exhaust air volumes through the ducts were 231 m^3/h and 224 m^3/h respectively. The measured constant ventilation rate meets the required level when additional local exhaust ventilations are taken into account.

4.5 Annual energy balance

The monthly and annual energy balances in the low-energy house are shown in Figure 10. The electricity consumption consists of that for heating, hot water supply and other purposes, including ventilation. The electricity production is supplied by power generation by the PV system.

The larger consumption, shown mainly in the winter season, is due to the heating operation of the heat pump unit. The decrease in solar power generation in winter seems to be due to snow covering on the roof. On the other hand, the solar power generation nearly reaches or exceeds the consumption from May to October.

The total annual consumption is 9,379 kWh, and is composed of 37 % of heating, 10 % of hot water supply and 53 % of electricity for other purposes. On the other hand, the total produced power is 4,534 kWh, and the energy self-sufficiency rate, which is shown as the proportion of the production to the consumption, can be 48 %. Almost half of the electric power demand can be met by the PV system. The real consumption for all purposes, which is obtained by calculating the difference between the consumption and production, is 4,845 kWh (24 kWh/m^2). On the other hand, it is very obvious that the real electric power consumption for space heating, hot water supply and mechanical ventilation is much less than 20 kWh/m^2. From this point of view, this low-energy house has an energy performance comparable to that in the requirement of the MINERGIE® standard in Switzerland, for instance.

The real power consumption could be reduced by development of a solution for the winter snow problem of the PV system. Effective utilization of the exhaust heat from the ventilation or the solar heat entering the room space by introducing short-term thermal storage will lead to further improvement of the performance of the HVAC system.

5 CONCLUSIONS

Indoor thermal environment and the performance of the heating system using a ground source heat pump (GSHP) are verified, as is the actual annual energy balance of the target low energy house located in the cold region of Japan. The following summarizes the results reported in this paper.

1. Room temperature can be kept above 20 ºC almost all the day even at lower supply temperatures of about 30 °C. In this case, the daily average heating output from the heat pump is about 3 kW, which resulted from the highly efficient operation of the inverter-controlled heat pump in the partial load range. The total seasonal average SCOP of the heat pump system reaches 4.45. The results show that the GSHP system with inverter control is an important element in achieving better seasonal performance for the space heating purpose.

2. The heat load for ventilation can be drastically reduced by the use of a 50 m long heat tube and mechanical ventilation units with heat recovery from the indoor exhaust air. The pre-heating effect of the heat tube is estimated as 220 W (4.4 W per unit length of the tube) in the winter season on average, giving a 14 % reduction in the ventilation load for the low-energy house. The use of the heat tube also has an advantage in preventing the need for defrosting the heat exchanger unit, even during the coldest season.

3. The total electric power demand for all purposes is 9,379 kWh (47 kWh/m^2) in a year. The real electric power consumption - that is, the difference between the demand and the 4,534 kWh of power generation by the solar photovoltaic system - is calculated as 4,845 kWh (24 kWh/m^2). In total, almost half of the energy demand can be covered by the solar system in this low-energy house.

6 ACKNOWLEDGEMENTS

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PERFORMANCE CHARACTERISATION OF A REVERSIBLE WATER-TO-WATER HEAT PUMP

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Abstract: This paper presents a detailed performance characterisation of a reversible water-to-water heat pump operating in cooling mode (SHERHPA 2007). It was designed to be integrated in a building air conditioning system in order to produce cold or hot water as required (Urchueguia et. al. 2006, GEOCOOL 2005). Characterisation of the heat pump behaviour with regard to both cooling capacity and the coefficient of performance (COP) under variable load conditions on secondary fluid external loops has been obtained by laboratory testing. It was possible to simulate the conditions of the real application based on: i) variations of water inlet temperatures on both evaporator and condenser sides and ii) variations of flow rates on the water loops. This experimental performance study has been carried out for water flow rates between 2000 and 4200 kg/h at 10.8ºC and 13ºC water temperatures at the inlet of evaporator. For further analysis, IMST - ART simulation software has been used to extend the range of input parameters up to a minimum water mass flow rate of 1000 kg/h and water inlet temperatures on the evaporator side between 8ºC up to 18ºC, as opposed to experimental tests that have restricted possibilities. As a direct result of the comprehensive understanding of the system response under different load conditions, it has been possible to improve system efficiency by means of optimum control of the water mass flow rates on both evaporator and condenser side.

Key words: ***heat pump, water mass flow rate, optimum, cooling capacity, COP***

1 INTRODUCTION

Sustainable energy systems, such as heat pump technologies, provide efficient use of renewable energy from ambient conditions. Typical applications of a heat pump are: space heating, domestic hot water and processes with combined heating and cooling. A significant research effort for optimised system design with direct application in industry has been undertaken over the last decade (Jacobi et. al. 2005, Corberan et. al. 2006, Primal et. al. 2006), aimed at developing heat pumps that are cost- and energy-efficient and compliant with the environmental regulations that led to the phase-out of conventional refrigerants for protection of the ozone layer. As a result of Kyoto agreement, a series of refrigerants such as R22 are scheduled to be replaced by 2010 (Kyoto protocol 1997). Therefore “natural refrigerants” such as CO2, ammonia and hydrocarbons have to be adopted even if this involves a change in the components' technology and control [Pearson 2006, Jakobsen et. al. 2006, Huff et. al. 2006).

The 6th European Framework Programme research project entitled: “Sustainable Heat and Energy Research for Heat Pump Applications – SHERHPA” aimed to address these challenges and solutions with immediate application in industry. The work presented in this paper is part of this project, and the main objective is to offer a complete experimental

characterisation of a prototype heat pump (Figure 1) working with propane (R290). The work aims to determine the optimum operating parameters for increased efficiency and to prepare the first steps in the control strategy. The main components of the prototype are as follows: i) a Bitzer 4EC–6 semi-hermetic compressor (1), ii) a four-way valve to switch between heating and cooling modes (2), iii) an electronic expansion valve - Carel EV 24 (3), and iv) SWEP V80, B80 evaporator and condenser (4).
The heat pump is designed to provide a nominal cooling capacity of 15 kW and heating capacity of 17 kW. The final purpose of the project is to commission and install the prototype as part of a geothermal building air conditioning system (Figure 1), used to provide the desired temperature control based on both cooling and heating as necessary.

The building air conditioning system (GEOCOOL 2005) has the following components: i) an external loop that consists of a ground heat exchanger (GSHX), a circulating pump, a storage expansion tank and the corresponding piping. This loop takes the heat exchanged at the condenser side of the heat pump (when working in cooling mode) and dissipates it to the ground. ii) an internal loop that distributes the chilled water (when working in the cooling mode) produced at the evaporator side of the heat pump to a series of twelve fan coils installed in the building, and iii) the "engine" of the whole system: the water-to-water reversible heat pump working with propane (R290).

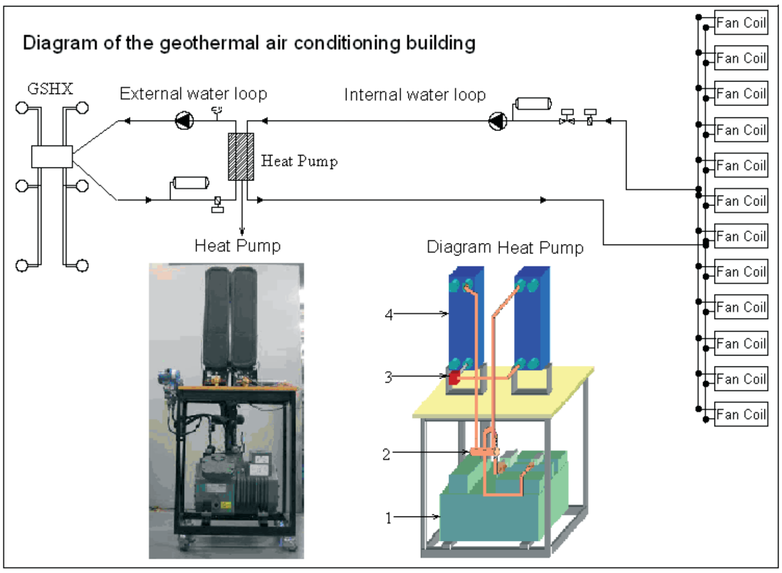


Figure 1: Building air conditioning system

The preliminary stage of the project consisted of an extended experimental test plan of the prototype in a dedicated test facility (Figure 2) capable of reproducing the real service conditions in the laboratory. Considering that the development of highly efficient units is one of the crucial challenges for the future success of refrigeration and air conditioning equipment, the conclusions of the experimental results are intended to provide a clear understanding of the optimum response of the unit for specific operating conditions, and to present an empirical correlation that could be used for future analysis in the development of improved control algorithms.

2 EXPERIMENTAL SET - UP

Considering that the main demand for the heat pump is to work as a chiller, the laboratory experimental stage is concentrated on efficiency maximisation during the cool mode.
The dedicated test rig was designed to be used for both cooling and heating performance testing of reversible water-to-water heat pumps in accordance with the existing EN-255 (1998)

and EN-14511 (2004) standards. Figure 2 shows the schematic design of the test rig, consisting of two water hydraulic groups (loops) capable of maintaining the desired setpoint temperatures at both evaporator and condenser inlet by means of mass flow rate control. The test rig was designed to be able to simulate the load conditions encountered during typical field operation on both sides of the unit. Each water loop included a corresponding secondary plate heat exchanger (PHE) (1), pump (2), by-pass valve (3), inertia tank (4), expansion chamber (5), and pressure valves and gauges.

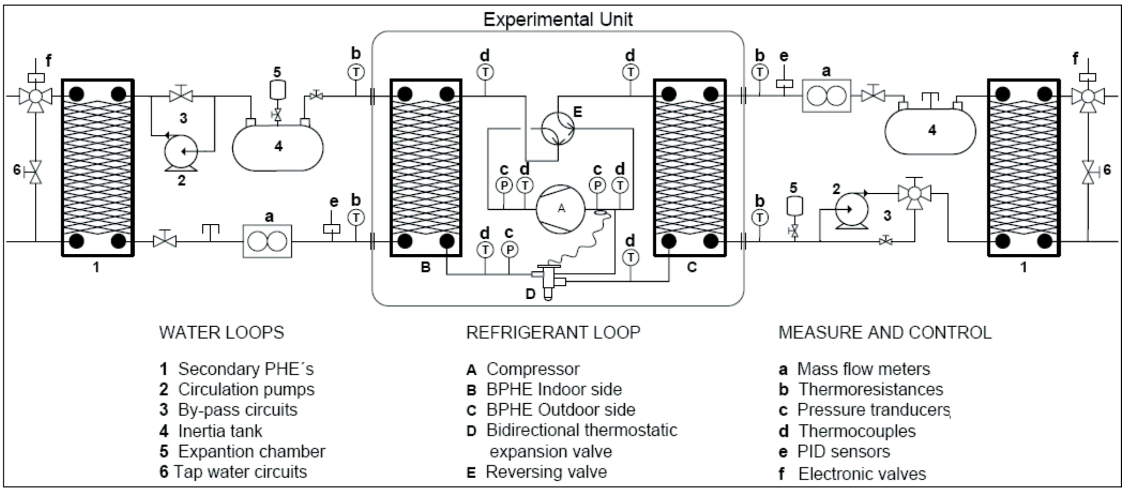


Figure 2: Heat pump laboratory experimental test rig – Technical University Valencia (UPV)

Heat pump operating conditions are set by on control of the outlet or inlet water temperatures in the inner and outer heat exchangers (evaporator and condenser respectively). This control strategy is based on variations of the flow rate of the city water (6) through the secondary heat exchangers (1). This is accomplished by a three-way regulating valve (f) controlled by a PID regulator that receives the signal of the water inlet temperatures (e). During the experimental tests, the inlet temperatures of the secondary fluid were controlled within ± 0.2K from the set point. The temperature difference of the water between the inlet and outlet of the primary heat exchangers was maintained at 5K, controlling the water flow rates by means of a by-pass circuit (3).

The behaviour of the heat pump under test (the experimental unit) is monitored using dedicated sensors installed in the refrigerant and water circuits (Table 1). Two Coriolis mass flow meters were used to measure the water flow (a). The temperature difference through the brazed plate heat exchangers (BPHEs) was monitored with calibrated Pt-100 RTDs (b). Inside the refrigerant loop, pressure transducers (c), located on the suction and discharge line of the compressor, measured the evaporation and condensation pressures. The temperatures along the refrigerant circuit for evaluation of the superheat and subcooling were obtained with T-type calibrated thermocouples (d). A power meter was used to measure the electrical consumption of the compressor in order to evaluate the COP.

Table 1; Specifications of the measuring sensors

Instrument	Type	Range	Precision
Thermocouple	T	-270 – 400C	±0.5 K
Thermoresistance	Pt – 100	-220 – 850C	±0.1 K
Mass flow meter (Coriolis)	Danfoss	0 – 5500 kg/h	±0.05%
	Danfoss	0 – 5500 kg/h	±0.05%
Pressure transducer	Fisher Rosemount	1000 kPa	±0.2%
	Fisher Rosemount	1000 – 2500 kPa	±0.2%
	Fisher Rosemount	0 – 5500 kPa	±0.2%
Power meter	Quantum D - 200	30 kW	±0.1%

Data acquisition is performed using a Hewlett-Packard data logger, continuously recording the measurements. A complete set of all parameters is recorded every ten seconds, once steady state conditions have been reached, i.e. when all the measured values stay constant, without setpoint modifications. Periodic fluctuations of these magnitudes due to operation of the control system are allowed, provided that the average values of these fluctuations do not exceed the permissible deviations between the setpoints and the arithmetic average of the real values as indicated: i) water inlet–outlet temperatures between ± 0.3K, and ii) volumetric flow ± 2%. As far as the capacities are concerned, EN-14511 (2004) permits a maximum uncertainty of 5 % independently of the uncertainties of measurement instruments used, including the uncertainties due to the properties of the fluids. The calculation procedure for both Cooling Capacity ($Q_{cooling}$) and COP ($COP_{cooling}$) is based on the heat balance applied to the heat exchangers, through the measurement of the mass flow rate (m_{evap}), and the inlet and outlet temperatures (ΔT_{water}), as shown in *Eq. 1*. The coefficient of performance (COP) was obtained as a calculation *(Eq. 2)* from the cooling capacity and compressor power consumption - *P* (Incropera and deWitt 2000) that was measured directly by the data logger, with a test duration estimated at 30 minutes for each test.

$$Q_{cooling} = m_{evap} c_p \Delta T_{water} \qquad Eq.1$$

$$COP_{cooling} = \frac{Q_{cooling}}{P} \qquad Eq.\ 2$$

3 VALIDATION IMST–ART SOFTWARE

IMST-ART© is an advanced computer-aided engineering design system that combines accurate and fast algorithms, easy-to-use graphical interface and powerful analysis capabilities into a software package suitable for modelling any vapour-compression system operating with several types of refrigerants and secondary fluids. The software is based on a global numerical model that consists of individual mathematical equations for each component of the heat pump. The solution is given in a steady-state format through the Newton Raphson algorithm (Corberan and Gonzalvez 2002). The technical characteristics of each component, together with information regarding the operation parameters for the desired regime, are introduced as an input and the software is capable of providing information regarding the detailed operating parameters for each component, including the pipes and fittings, together with information on capacity, COP, compressor efficiency, superheat, subcooling and heat transfer data for the heat exchangers.

The comparison between the experimental results and the predictions of IMST–ART obtained for the test conditions listed in Table 2 are shown in Figure 3.

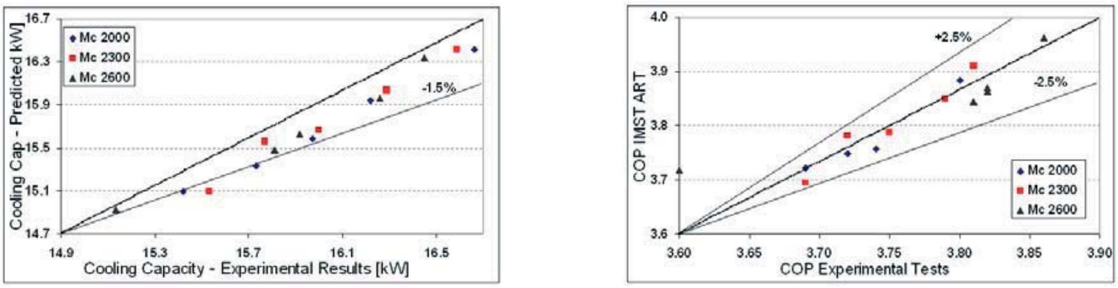


Figure 3: Validation of the IMST–ART predictions versus experimental results

It can be seen that a very good agreement of ±2.5 % is obtained for COP of the heat pump when compared with the experimental results. An even better prediction, of only -1.5 %, is obtained for the cooling capacity of the heat pump under consideration. The validation of the IMST–ART predictions was investigated in detail (Gonzalvez et. al. 2007), and a maximum average difference of only 4 % was obtained for all operating parameters.

4 RESULTS AND DISCUSSION

The main objectives of the experimental work were to determine the optimum response of the system based on variations of the water flow rates, as follows: i) test condition I: based on control of evaporator water flow rates Me = 2000, 2300, 2600, 3100 and 4200 kg/h for constant condenser water flow rates at three rates, Mc = 2000, 2300 and 2600 kg/h, and ii) test condition II: the water flow rates on the evaporator side were maintained constant at 2000, 2300 and 2600 kg/h, while variations were imposed on the condenser side (Table 2). The experimental tests were performed based on constant set–point control of the water temperatures at the evaporator and condenser inlet as required during field operating conditions. Considering that the heat pump was primarily designed to operate in cooling mode as part of the building air conditioning system, the aim of the test matrices was to determine the conditions for optimum cooling capacity and coefficient of the performance (COP).

Table 2; Test conditions – variations of water flow rates.

Evaporator water inlet temperature	Condenser water inlet temperature	Condenser water flow rate **Condition I**	Evaporator water flow rate **Condition I**	Evaporator flow rate **Condition II**	Condenser flow rate **Condition II**
ºC	ºC	kg/h	kg/h	kg/h	kg/h
10.8±0.7	22±0.7	2000, 2300, and 2600	2000	2000, 2300, and 2600	2000
			2300		2300
			2600		2600
			3100		3100
			4200		4200

The cooling capacity for each experimental condition was calculated based on the heat balance applied for the heat pump evaporator (*Eq. 1*). The coefficient of performance (COP) was obtained as the ratio between the cooling capacity and the compressor power consumption (*Eq. 2*).

Experimental Results

This section presents the most significant results obtained during the experimental campaign, together with an extended analysis of the system behaviour based on IMST–ART predictions.

4.1.1 Test condition I

Figure 4 shows the results obtained for Test Condition I. Condenser water flow rates are maintained constant at 2000, 2300 and 2600 kg/h, while evaporator water flow rates are controlled to 2000, 2300, 2600, 3100 and 4200 kg/h.

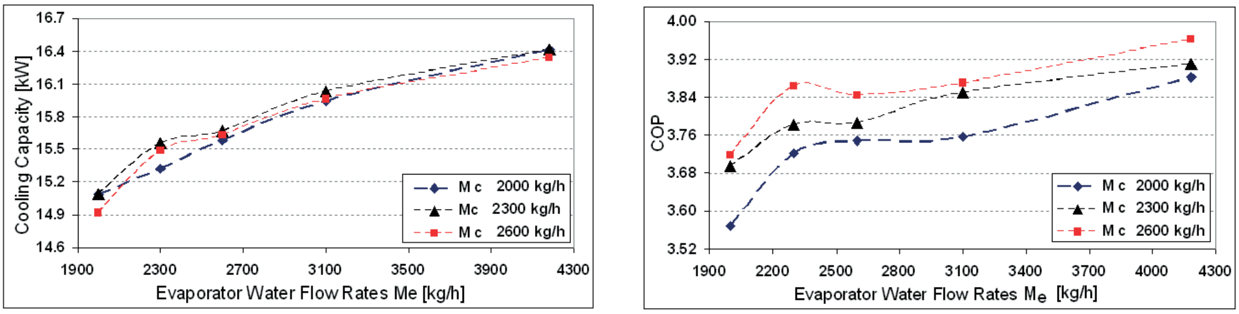
It can be seen that higher evaporator water flow rate results in a similar trend in the cooling capacity variation. An increase of 100 kg/h in the secondary fluid mass flow rate provides an average 2.5 % higher cooling capacity. It can also be seen that the magnitude of the condenser water flow rate has less impact on the cooling capacity, especially when the water flow rate on the evaporator side is higher than about Me = 3100 kg/h.

The combined effect of the evaporator inlet water temperature and the condenser flow rate is also shown in Figure 4. While the setpoint temperature was 10.8 °C for both Mc = 2000 and 2300 kg/h, the 1 % increase in the cooling capacity is the result of 300 kg/h higher water flow rate on the condenser side. For the Mc = 2600 kg/h case, the evaporator water inlet temperature was maintained at 0.4 °C lower value. For this condition, it is concluded that the effect of increasing the water flow rate by 300 kg/h is offset by the lower water temperature, as the cooling capacity for this condition is an average of 1 % lower than at 2300 kg/h mass flow rate. Further analysis of the combined influence of both temperature setpoint and flow rate for COP is shown in Figure 6.

It is concluded that the minimum and maximum cooling performance of the heat pump is between 14.9 kW and 16.4 kW, depending strongly on the variations of water mass flow rates on the evaporator side.

Figure 4 (b) shows that the coefficient of performance (COP) of the heat pump also experiences a uniform increase with both evaporator and condenser water flow rates.

The maximum COP of the unit is obtained for high evaporator flow rates (Me>3100 kg/h). However, it is difficult to take this region into consideration, as the consumption of the circulation pumps is included in the COP of the entire system during heat pump operation in the dedicated building air conditioning system (Figure 1). This overall COP decreases at high evaporator and condenser flow rates (>3100 kg/h), as this regime also implies higher power consumption of the circulation pumps. As a result, for further detailed analysis of the optimum operating parameters, it is necessary to consider the region with lower water flow rates (between 2000 and 3100 kg/h) on the evaporator side. A more detailed review of the test results for this region (Figure 4 b) shows that it is possible to obtain an optimum for COP when both Mc and Me are controlled between 2300 and 2600 kg/h.



a) Cooling capacity at Mc = constant

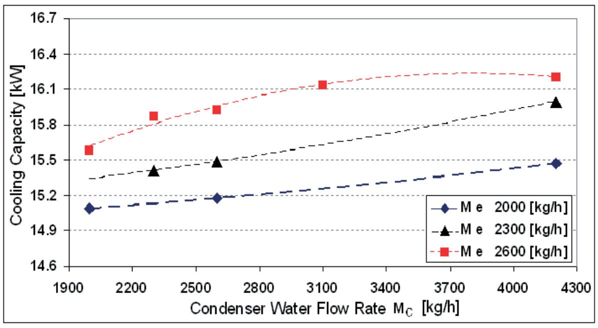
b) COP at Mc = constant

Figure 4: Experimental results obtained for Test Condition I

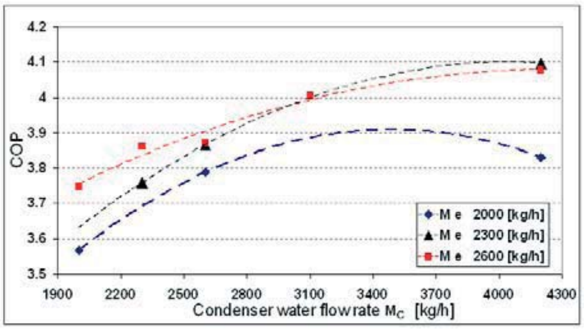
As far as optimised control of the unit is concerned, the target is to develop an algorithm that can decide between periods of time when the heat load is very high and it is necessary to operate at maximum cooling capacity (maintain water flow rates > 3100 kg/h), even if this means less efficiency of the system, and periods of time when it is wished to operate efficiently and the maximized cooling capacity is no longer a priority (maintain water flow rates between 2300 and 2600 kg/h).

4.1.2 Test condition II

Figure 5 shows the experimental results obtained for both cooling capacity and COP of the system during Test Condition II, when the evaporator water flow rates were maintained constant at 2000, 2300 and 2600 kg/h while the condenser flow rate was controlled at 2000, 2300, 2600, 3100 and 4200 kg/h.



a) Cooling capacity at Me = constant



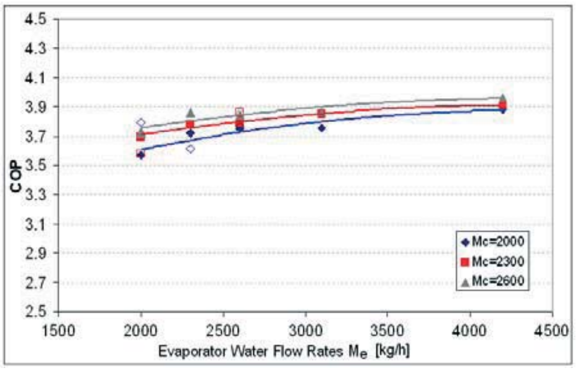
b) COP at Me = constant

Figure 5: Experimental results obtained for Test Condition II

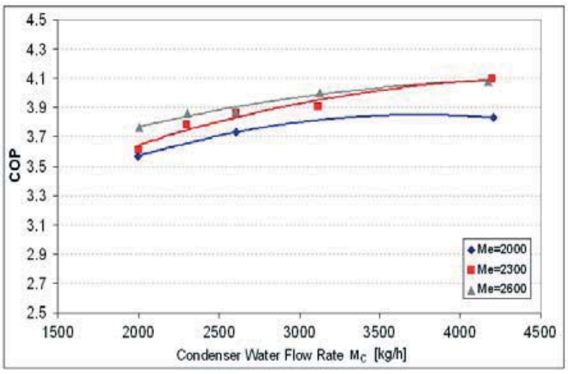
The results shown in Figure 5 confirm the conclusions stated in the previous section. The general trend of both cooling capacity and COP reflects a uniform increase once higher water flow rates occur on both evaporator and condenser sides. For variations of Mc between 2000 ÷ 4200 kg/h and Me between 2000 ÷ 4200 kg/h, the maximum and minimum limits of the heat pump coefficient of performance are 4.1 and 3.58 respectively. An average increase of 300 kg/h in Mc result only in an average 1.2 % higher cooling capacity. It is concluded that the variations of the secondary fluid mass flow rate on the condenser side have a significant effect only on the COP, while the cooling capacity is influenced significantly by the evaporator side water flow rates.

From Figure 5 (b) it can be seen that, for controlled condenser water flow rates in the range 2000 ≤ Mc ≤ 4200 kg/h, the optimum system COP occurs at constant evaporator mass flow rate of Me = 2300 kg/h. This effect is more pronounced at condenser water flow rates higher than 3100 kg/h. However, tacking into account the overall air conditioning system COP, it is apparent that it is desirable to maintain the condenser flow rate between 3100 and 3300 kg/h in order to overcome the negative impact resulting from the high power consumption of the water circulation pump in the external ground loop circuit.

Further analysis of the COP is shown in Figure 6 (a, b) at both test conditions I and II. Several tests were performed in order to quantify the combined influence of evaporator and condenser water inlet temperatures with variations of the mass flow rate. It is noted that, during experiments, it is difficult separately to quantify the influence of these two parameters on the cooling capacity and COP, due to the feedback between them. With the evaporator operating in cooling mode, an increase in the mass flow rate results in a higher heat transfer coefficient and therefore an increase in both capacity and COP as the temperature difference between the refrigerant and the secondary fluid is reduced (countercurrent configuration). However, if we keep the evaporator water inlet temperature constant, as an increase in the flow rate also leads to a decrease in temperature drop across the heat exchanger, then the mean water temperature increases and therefore the refrigerant senses a higher secondary temperature. This effect is beneficial for the COP. Conversely, if the outlet temperature is maintained constant, then the mean water temperature will decrease and so the refrigerant senses a lower secondary temperature, which will result in a reduction of COP.



a) COP trend for Test Condition I



b) COP trend for Test Condition II

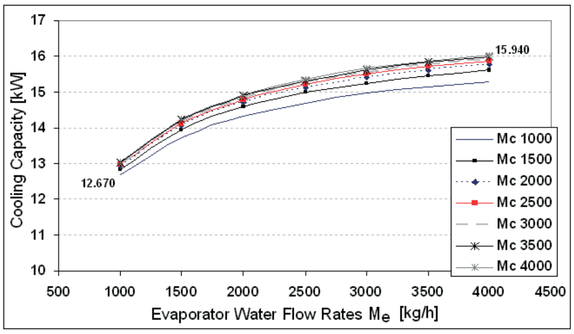
Figure 6: General trend of COP for test conditions I and II

The combined water temperature and flow rate interaction is highlighted during the test campaign for Me = 2600 kg/h and Mc = 4200 kg/h. While maintaining these parameters constant, two tests were performed at different condenser water inlet temperatures: first, at 22 °C and then, for the second test, this temperature was 5 % higher. The increase in this parameter had a significant effect on COP, as a reduction of 9 % occurred. It is concluded that the heat pump COP appears to be very sensitive to water temperature variations on the condenser side while operating in the cooling mode. The cooling capacity is less influenced by the condenser inlet temperature, as it experiences only a 3 % decrease for the same test conditions. Figure 6 shows a similar test comparison for Mevaporator of 2000 kg/h and Mcondenser of 2000 kg/h. It can be seen that, at a higher condenser inlet temperature, with the same ratio of 5.4 %, the COP was reduced by 3.8%.

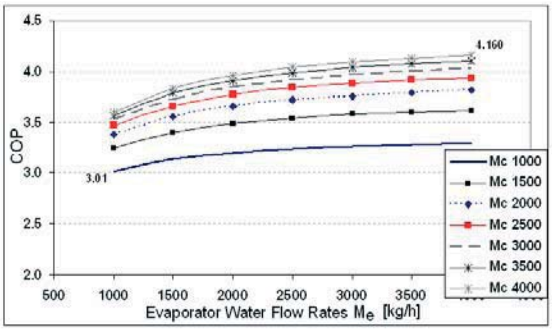
In the real application of the heat pump (Figure 1), the water temperature is highly dependent on the ground loop. The closer the ground and condensation temperatures are, the higher is the increase in COP.

IMST–ART Results

This section presents an extended analysis of the heat pump behaviour, based on supplementary information offered by the IMST–ART software predictions. The test range was extended, starting with a minimum of 1000 kg/h water flow rates with plotted data at each 500 kg/h increase. As a novelty, the influence of evaporator water inlet temperature was quantified for a variation range of 8.1 °C to 18.9 °C.
Figure 7 shows the change in cooling capacity and COP for Test Condition I, with constant condenser water flow rates and varying evaporator water flow rates. In comparison with the experimental results, it can be seen from Figure 7b that a slightly greater increase in COP occurs at the lower range of the condenser water flow rates (<2500 kg/h), while the influence on the cooling capacity is less significant at the same rate.
With regard to evaporator mass flow rates, Figure 7 shows that a variation of 300 kg/h in flow rate when Me < 2000 kg/h has a greater effect on the increase of both COP and cooling capacity; of 4.1 % and 7.1 % respectively.



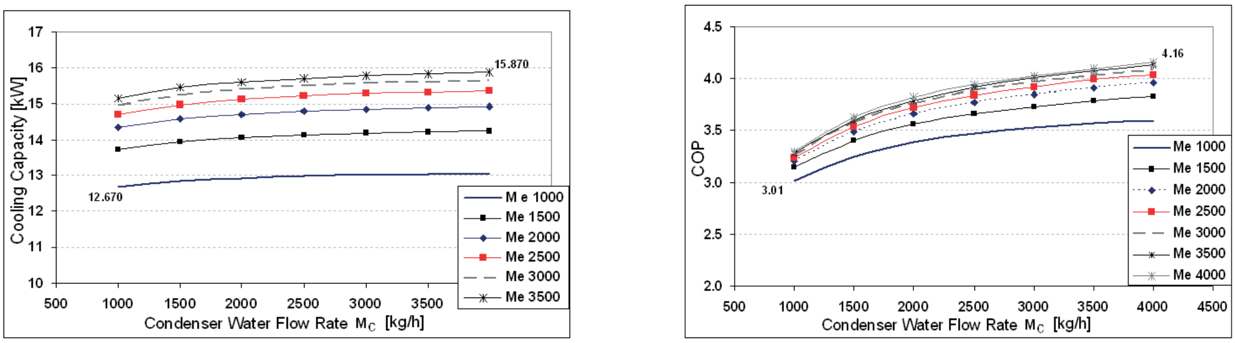
a) Cooling capacity with constant Mc



b) COP with constant at Mc

Figure 7: IMST–ART predictions for Test Condition I

Figure 8 shows corresponding curves for cooling capacity and COP for Test Condition II. The evaporator mass flow rate is maintained constant at six levels between 1000 and 4000 kg/h, against variations of 500 kg/h for each data plotted for the condenser water flow rate. It can be concluded that, for both cooling capacity and COP, the significant increase takes place at high water mass flow rates on the evaporator side (> 2500 kg/h).

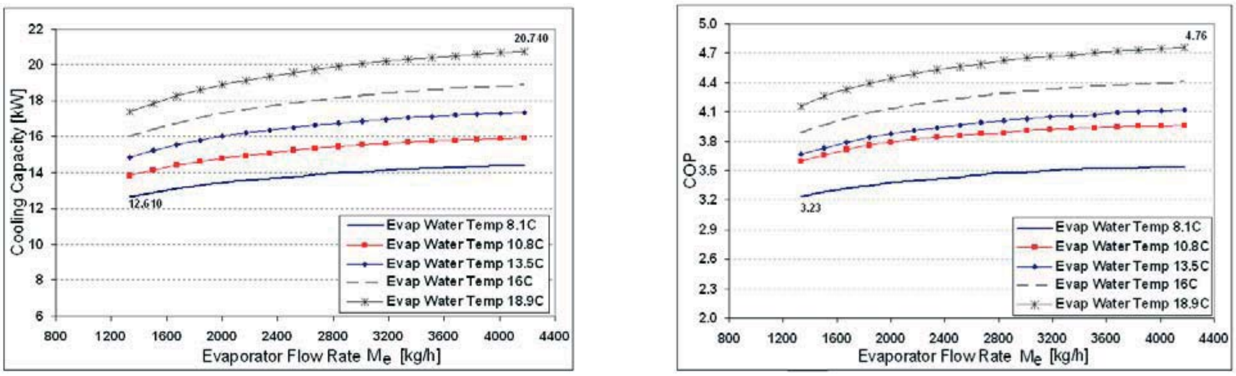


a) Cooling Capacity at Me constant

b) COP at Me constant

Figure 8: IMST–ART predictions for Test Condition II

A comparison between the results shown in Figures 7 and 8 confirms that the variations of the condenser mass flow rates have a greater effect on the COP, while the effect on the cooling capacity is diminished as it is more strongly influenced by changes in the evaporator flow rates. The results are consistent with the experimental data presented in the previous section. It has to be mentioned that, in the real application, the increased flow rate on the condenser side results in a higher heat transfer coefficient on the water side of the ground heat exchanger. However, the heat transfer to the ground is dominated by the heat transfer through the soil, so water side variations will have negligible influence on the global heat transfer to the soil.
As a novelty, Figure 9 shows the results for both COP and cooling capacity when the system is operated at several setpoints for evaporator inlet water temperatures: 8.1 °C, 10.8 °C, 13.5 °C, 16 °C and 18.9 °C. It can be seen that, while the variation of evaporator water flow rate is plotted for the whole range examined, the condenser flow rate is maintained constant at 2600 kg/h.



a) Cooling Capacity at Mc = 2600 kg/h

b) COP at Mc = 2600 kg/h

Figure 9: Experimental and IMST–ART results for evaporator water Inlet temperatures

A uniform average increase of 8 % in the cooling capacity, and 6.8 % for COP, is obtained for 2.7 °C temperature variation. The optimum evaporator inlet temperature obtained, mainly based on the response of COP, is considered to be at 10.8 °C.
The cooling performance and COP operating limits are quantified. The cooling capacity has a minimum of 12.6 kW at 1000 kg/h on both condenser and evaporator sides, while the maximum cooling capacity is 20.74 kW at 4200 kg/h and 18.9 °C water temperature.

5 CONCLUSIONS

This work describes a detailed experimental performance characterisation of a reversible water-to-water heat pump operating in cooling mode, and intended to be integrated in a building air conditioning system, mainly to produce cold water (Geocool 2005, Urchueguia et. al. 2006). The characterisation of the heat pump behaviour with respect to the cooling capacity and COP is obtained under controlled variations of water flow rates on the secondary external loops. It is intended to identify the optimum water flow ranges in order to obtain increased efficiency of the system. The experimental test range is extended and the influence of water temperature at the evaporator and condenser inlet is quantified through the predictions of the IMST–ART simulation software. The general evolution obtained for both cooling capacity and COP reflect a uniform increase once higher water flow rates are used. The evaporator flow rates have a strong impact on the cooling capacity increase, while the condenser flow rates have a more pronounced effect on the system COP. In quantified terms, an increase of 100 kg/h in the evaporator mass flow rate increases cooling capacity by an average of 2.5 %, while the same increase in the condenser flow rate has a greater impact on the COP, increasing that by an average of 5 %. The optimum COP of the heat pump occurs when both condenser and evaporator mass flow rates are between 2300 and 2600 kg/h. This range includes not only the heat pump response but also the requirements and conditions of the real air conditioning system application. The potential for improved control is based on the increase in secondary fluid mass flow rates that lead to higher COP, while taking into account the power consumption of the water circulation pumps in the external loops.

The preliminary experimental test campaign presented in this paper is important, as it provides the results for the optimum operating conditions. At present, the heat pump is installed in the building air conditioning system in order to asses the real advantage of controlling the water flow rates on external loops that leads to minimum consumption of the entire system.

6 ACKNOWLEDGEMENTS

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EFFICIENCY ASPECTS OF HEAT PUMP SYSTEMS - LOAD MATCHING AND PARASITIC LOSSES

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Abstract: Heat pumps often provide efficient solutions for refrigeration, cooling, heating and hot water. Although there still remains much to improve on the efficiency of the heat pump per se, in many cases much larger energy savings can be achieved by improving the design and component efficiencies of the systems that connect the heat pump to the load. Also, when building envelopes improve and control systems better match supply and demand, the active operation of the heat pump unit is reduced. Even now, some applications see more electric energy going into distribution and control than to the operation of the heat pump compressor. Examples show how the efficiency of existing heat pump systems may be greatly improved with no changes to the heat pump, how new HVAC system designs may drastically reduce drive energy to pumps and fans and how the use of ambient conditions and ground storage may provide cost-effective and efficient means for space conditioning.

Key Words: ***control, efficiency, fans, heat pump systems, parasitic losses, pumps***

1 INTRODUCTION

Heat pumps often provide efficient solutions for refrigeration, cooling, heating and hot water. Although there still remains much to improve on the efficiency of the heat pump per se, in many cases much larger energy savings can be achieved by improving the design and component efficiencies of the systems that connect the heat pump to the load.

1.1 Heat Pump Applications

Heat pump systems range from applications with few alternative solutions, such as low temperature refrigeration, to those in fierce competition with other options, e.g. heating only heat pumps. Specific conditions of different applications affect the possibilities of utilizing natural ambient sources and sinks. The specific application also affects the utilization factor (relative operating time) of the heat pump and hence the relative importance of parasitic drive energies and heat losses. Applications may be classified in relation to the temperature deviation of the conditioned space from normal ambient conditions (see figure 1):

1. Frozen and deep-frozen food; large negative deviation, difficult to find natural sinks.
2. Chilled food; moderate negative deviation from room temperature but with natural low temperature sinks in outdoor Nordic climates.
3. Space conditioning (cooling only) in warm climates; moderate negative deviation, few natural sinks.
4. Space conditioning (cooling and heating) in moderate and cold climates; small negative to large positive deviations, ample opportunities for low temperature sinks and sources (see example in 4.1).
5. Space conditioning and hot water (heating only) in moderate and cold climates; large positive deviations (see example in 4.2).

This paper will address some general efficiency aspects of heat pump systems providing two examples (categories 4 and 5).

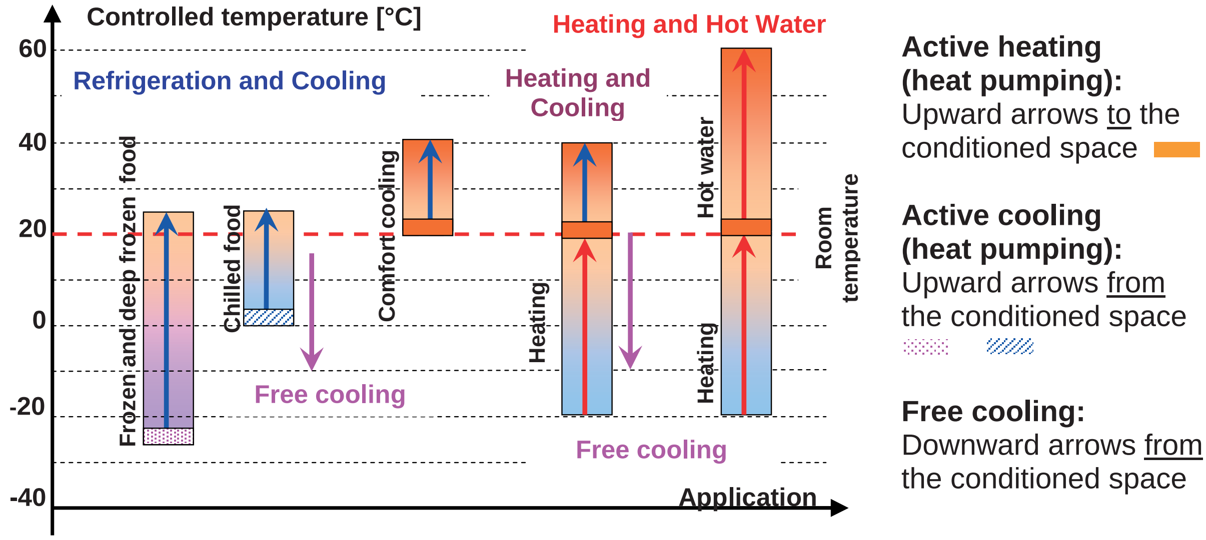


Figure 1: Temperature ranges of typical heat pump applications.

1.2 Heat Pump System Efficiency Aspects

The theoretical energy use of a heat pump system depends primarily on the application and the system design. The application decides the required temperature of the conditioned space and the location provides the ambient climatic condition. In many cases, there are ample opportunities to complement the heat transport achieved by the heat pump (upward arrows in figure1) by heat flows driven by positive temperature differences (free cooling, downward arrows in figure 1). For a specific application there are a number of possibilities to improve the overall System Performance Factor (*SPF*; see 2.3) by optimal uses of free energy and drive power. Some examples (see also section 3) are to:

- Reduce demand by load matching and storage
- Reduce purchase of energy by means of natural sources and sinks
- Reduce drive energy for
  - heat pumping through reduction of temperature lift
  - heat transfer by suitable heat exchangers (new designs!) and optimized flow control
  - heat transport by optimal system design and optimized flow control
  - terminal units by suitable choice (new designs available!) and optimized flow and temperature control

2 BASICS

In the discussion of more efficient use of energy it is essential to distinguish between energy demand and energy use. Energy use typically exceeds actual demand by substantial margins. Actual demand is decided by the quality requirements of the conditioned space, the user pattern, thermal loads, design of the heating, ventilation and air-conditioning (HVAC) and energy supply systems. However, discussion and comparison of energy use and energy efficiency has little meaning unless one has defined the relevant system boundaries.

2.1 Building System Boundaries and Key Numbers

Key numbers for expressing the system energy efficiency will vary depending on purpose and application. This may be confusing, especially for systems using substantial amounts of unpaid-for energy such as direct solar and heat pump systems. Common “efficiency” measures are for instance kWh_{heat}/m^2/year and kWh_{el}/m^2/year (m^2 useable floor area).

The minimum use of heat and electricity is decided by the user demand specification. Real systems will always use more and, depending on whether one includes “free” energy flows or not, there may be widely differing figures for the supply of energy. Typically, building statistics only include purchased energy and hence is rather misleading concerning actual use. Figure 2 provides a simple illustration of energy flows through some alternative system boundaries and the difference between demand, use, net supply, gross supply and purchased supply.

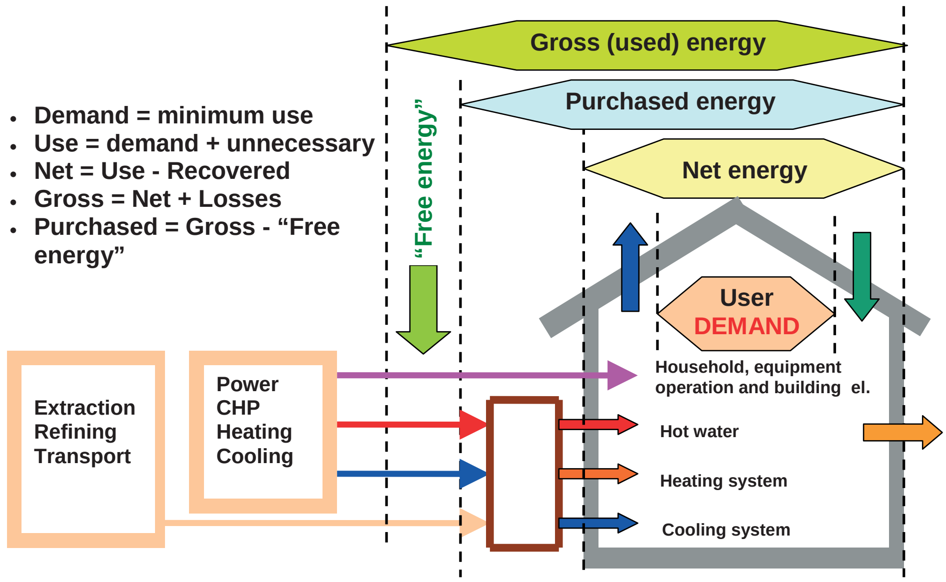


Figure 2: Alternative system boundaries used for efficiency key numbers.

2.2 Demand, Net Use, and Gross Supply in relation to the Load Factor

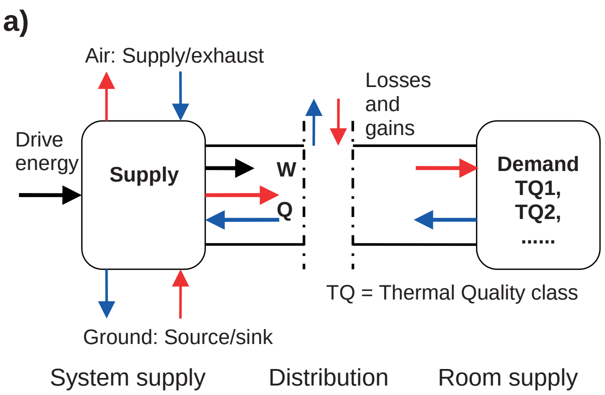
Looking for improvement of future heat pump heating and cooling systems, one must consider the ongoing developments of improving building envelopes, the use of storage to level out heat deficits and surpluses over time, the use of Building Automation Systems with feed-forward and control-on-demand (COD) to minimize supply in relation to demand etc. Such developments, COD in particular, will reduce the use of energy much more than it will reduce the design capacity. This, in turn, will substantially reduce the thermal load factor L_Q of the HVAC system and hence increase the demand factor D_Q. These factors are defined as:

$$L_Q = \frac{Q_0}{\tau_{year} \cdot \dot{Q}_{nom}} \text{ and } D_Q = \frac{Q_{net}}{Q_0} \text{ (see figure 3)} \qquad (1)$$

where Q_0 = demand (minimum net use), $\dot{Q}_{nom}$ = nominal capacity and τ_{year} = 8760 h.

Adding *h* or *c* to the subscripts will indicate heating or cooling. Demand factors may also be correspondingly defined for the ratio D_W between used drive energy, W_{net}, and the minimum net use, W_0.

It is a common experience that it is more difficult to exactly match supply and demand in systems with a low load factor. When a large part of the energy balance is supplied from uncontrolled heat flows (internal loads, heat stored in a building structure etc.), then it often happens that the conditioned space is overheated or overcooled in relation to demand. Based on simulations and measurements, the German standard VDI2067:20 was developed to describe this. Figure 3 shows an example of how the heat demand factor D_Q increases as the load factor decreases (it has the character of $D_Q = k_1 + k_2 / L_Q$ with k_1 and k_2 being constants). The diagram also indicates that the use of distributed pumps, as indicated in 3.5 and 3.6, will better match demand than one central unit.



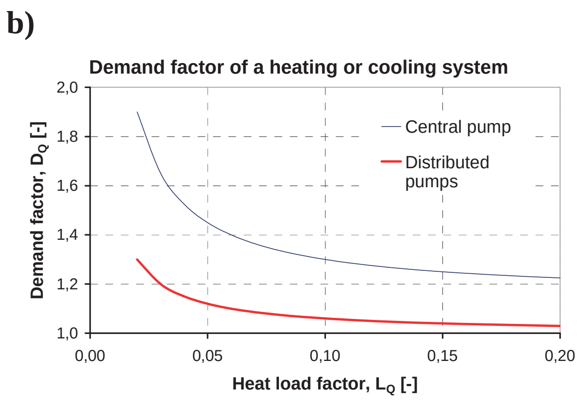


Figure 3: a) System supply, distribution and room supply of a general heat pump system. b) Demand factor of a heating or cooling system as a function of the heat load factor.

Overall use of energy can be reduced by means of reduced demand or improved efficiency. Demand may be tuned by reducing the heating or cooling loads or by changing the indoor environment specifications (e.g. less rigid minimum and maximum temperatures). Figure 3 indicates that there are three major parts of a heat pump heating and cooling system; system supply, distribution and room supply. These are the parts that affect system efficiency and hence will be discussed in this paper.

2.3 Heat Pump System Boundaries and Key Numbers

Key performance indicators of heat pump systems are the Coefficient of Performance (*COP*) and the Seasonal Performance Factor (*SPF*). Both are measures of the delivered heating or cooling energy of the system in relation to the demand for input drive energy. *SPF* is usually the integrated mean value of the instantaneous *COP* over a year, see eq. 2 below. Just as in the case with alternative system boundaries of a building, there will be different *COP*s and *SPF*s depending on the system boundary of the heat pump system. Figure 4 provides an illustration of alternative definitions of *COP*. Using the system boundaries and designations of figure 3 we have in the case of a heating application:

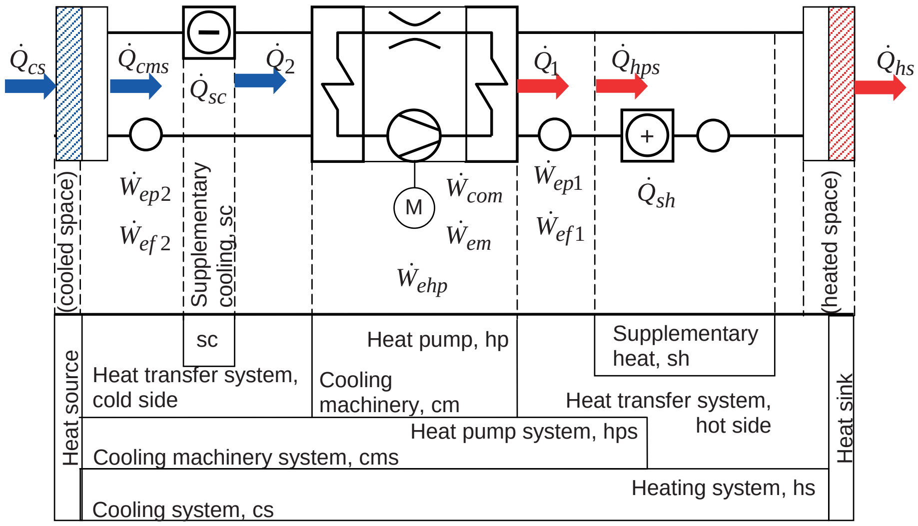
$$COP_{hp} = \frac{\dot{Q}_1}{\dot{W}_{e,hp}} \text{ and } SPF_{hp} = \left[\frac{Q_1}{W_{e,hp}}\right]_{annual} \tag{2}$$

$$COP_{hps} = \frac{\dot{Q}_1 + \dot{W}_{e,p1}}{\dot{W}_{e,hp} + \dot{W}_{e,p1} + \dot{W}_{e,p2}} \text{ and } SPF_{hps} = \left[\frac{Q_1 + W_{e,p1}}{W_{e,hp} + W_{e,p1} + W_{e,p2}}\right]_{annual} \tag{3}$$

$$COP_{hs} = \frac{\dot{Q}_1 + \dot{W}_{e,p1} + \eta_{sh} \cdot \dot{Q}_{sh}}{\dot{W}_{e,hp} + \dot{W}_{e,p1} + \dot{W}_{e,p2} + \dot{Q}_{sh}} \text{ and } SPF_{hs} = \left[\frac{Q_1 + W_{e,p1} + \eta_{sh} \cdot Q_{sh}}{W_{e,hp} + W_{e,p1} + W_{e,p2} + Q_{sh}}\right]_{annual} \tag{4}$$

Corresponding definitions for cooling applications can be obtained by changing subscripts. However, in the heating case, the condenser pump adds to the heating capacity whereas in the cooling case, the evaporator pump detracts from the cooling capacity.

It is obvious from the definitions of SPF_{hp} and SPF_{hps} that the relative influence of the parasitic drive powers increases quickly when improvements regarding the heat pumping process reduces W_{ehp} ($W_{ehp} = W_{em}$ + controls etc.). In the equations 2-4, W_{ep1} is the sum of all parasitic drive powers to pumps and fans on the condenser side and correspondingly for W_{ep2}. Without specifying the system boundaries, comparisons of *COP* may be quite misleading (c.f. figure 9a where at +10 °C COP_{hp} = 3.7 and COP_{hps} = 2.5!).



$$W_{ep1} = \sum_{i=1}^{n} W_{ep1,i} \quad \text{and} \quad W_{ep2} = \sum_{j=1}^{n} W_{ep2,j}$$

Original Swedish indices have been replaced.

1 = condenser side
2 = evaporator side

hp = heat pump
hps = heat pump system
hs = heating system
sh = supplementary heating system

cs = cooling system
cms = cooling machinery system

e = electric
m = motor
f = fan
p = pump

Figure 4: Heat pump system boundaries based on NT VVS 076, 115 and 116 (Fahlén, P. 1996).

3 SYSTEM DESIGN AND OPERATION

Prior to the discussion of system design and operation, it is obvious that the Thermal Quality (TQ) requirements of the conditioned space must have been decided on. These will set the performance requirements of the technical systems in terms of temperature levels, control deviations and thermal capacities. It then remains for the designer to minimize purchased energy by reducing the demand factors through *load matching* (COD), by maximizing the use of free energy (minimize W_e) through storage and the use of *natural sources and sinks*, and to minimize the *drive power* for heat transfer and distribution.

3.1 Load Matching by Control-on-Demand (COD) and Storage

The heat balance of modern residential and office buildings to a large extent relies on removal of internal heat loads by means of ventilation air. With heat recovery, very little supplementary heat is needed and maximum capacity for cooling will also be reduced (c.f. figure 5a). Central supply units can only handle the building as a total whereas actual demand is created at room level. Individual COD can drastically reduce demand but will also reduce the thermal and drive energy load factors L_Q and L_W. Hence load matching becomes important, otherwise the demand factors D_Q and D_W will become unnecessarily high. For a heat pump system with a given design capacity there are two main possibilities:

- Adapting the supply capacity to match demand (capacity control)
- Adapting the HVAC system to accommodate any excess capacity (storage).

Capacity control: With perfect load matching, the heat pump and heating system operating times will be equal, $\tau_{hp} = \tau_{hs}$ (see eq. 6; correspondingly for cooling). With continuous capacity control of the compressor, the operating time τ_{hp} for the heat pump will increase if a given amount of heat *Q* is to be delivered at a lower mean capacity. From eq. 6 it is obvious that if τ_{hp} goes up, then the drive energy for p11 and p12 will increase unless the respective drive powers are reduced. Karlsson (Karlsson, F. 2007) has shown that controlling the condenser and evaporator pumps linearly in relation to the controlled drive power to the compressor comes very close to an optimal control (see also 3.4).

Storage: With active storage, the same principles apply as described above. In this case, load matching is transferred from a variable capacity compressor to a variable capacity pump that controls thermal output from the storage. In both alternatives there will be parasitic losses of, for instance, frequency inverters and pumps. In the case of storage there will also be thermal losses. Active storage has a general potential for levelling of surplus and deficit (spatial as well as temporal redistribution). Passive storage in the building structure is a more complex issue.

3.2 Use of Natural Sources and Sinks

Obviously the use of free-cooling relies on a natural ambient sink with a temperature that is lower than the required conditioned space temperature (air, lake, sea, ground, stored snow etc.). Mattsson and Malmberg (Mattsson, C.-J. and Malmberg, T. 2008) have demonstrated how the design temperatures of cooling systems affect possible use of free cooling. A case-study in Gothenburg indicates that by designing for a supply cooling water temperature of 14 °C instead of the traditional 7 °C, it would be possible to reduce the annual drive energy for heat pumping from 19.2 to 1.5 kWh_{el}/m^2 using a nearby river as the sink. The reduction is partly due to increased evaporator temperature but mainly from a substantial increase of direct free-cooling. In the example of 4.1, the ground is used as combined storage, heat source and heat sink for a heat pump based heating and cooling system. In this case the fractional supply by free cooling could have been further increased with a different HVAC system.

3.3 Temperature Lift - Drive Energy for Heat Pumping

In the discussion of drive energy to a heat pump heating and cooling system, one should distinguish between drive energy for temperature lift (heat pumping), drive energy for heat transfer between heat pump and the sink or source, drive energy for heat transport and drive energy for heat transfer in the terminal units for air-conditioning. This facilitates the understanding of where savings on drive energy are most easily accomplished.

The temperature levels of the heating system - heat source and cooling system - heat sink will decide the possible *COP* and thus the drive energy for temperature lift. Assuming a constant Carnot efficiency of the heat pumping process, within limited temperature variations ΔT, we have:

$$\frac{\Delta COP_{hp}}{COP_{hp}} = \frac{\Delta COP_{1C}}{COP_{1C}} = \left[\frac{\Delta T_2}{T_1 - T_2} - \frac{T_2}{T_1} \cdot \frac{\Delta T_1}{T_1 - T_2}\right] \approx -\frac{\Delta \dot{W}_{e,hp}}{\dot{W}_{e,hp}} \qquad (5)$$

Typically *COP* improves by 2-3 % for each degree of reduced temperature difference T_1 - T_2 between condenser and evaporator (relatively more so the smaller the difference becomes). Typical ways of reducing this difference is finding better sources or sinks and upgrading the terminal units by improving their heat transfer capacities. Regarding the latter alternative, it is very often the most cost effective method but it should be noted that there may be a drive power penalty if this requires higher flows and/or pressure drops. However, it pays to increase the drive energy to pumps and fans as long as the corresponding reduction of drive energy to the compressor is larger (see 3.4).

3.4 Heat transfer - Drive Energy to Pumps and Fans

Optimal flow for heat transfer at the heat pump may be quite different from the optimal heat transfer flow in terminal units. Also, demand in terminal units may not coincide with supply, neither for clock time nor for duration. Therefore, it is useful to distinguish between flows and pressure drops for heat transfer of system supply, of room supply and for distribution.

Regarding parasitic drive energy for heat transfer in the condenser or evaporator of heat pumps, Granryd (Granryd, E. 1998) has shown that for optimum *COP* these drive powers should be related to the cooling capacity by fairly simple relations. His conclusion regarding refrigerating applications was that typically pumps and fans were sized for optimum capacity, not optimum efficiency. The former criterion will yield drive powers up to 8 times the optimal for efficiency. Supporting Granryd's findings, Karlsson (Karlsson, F. 2007), in his work on capacity controlled heat pumps, has further underlined the importance of controlling not only the compressor but also the pumps and fans. If not, *COP* is likely to actually decrease instead of the hoped for increase from, for instance, a variable speed drive (VSD).

To achieve the required heat transfer at a lower drive power penalty, new developments of laminar flow design of air-coils show great promise (Haglund Stignor, C., Fahlén, P. and Sundén, B. 2007). However, in the search for the optimal heat transfer flow and drive energy, one must also consider the effect on drive energy for distribution where an increased flow is a penalty with little or no benefit. Fahlén has described the possibilities of reducing substantially the distribution pressure drops by alternative system designs (see 3.5).

Equation 6 illustrates the importance of parasitic drive powers, both regarding maximum input but perhaps more important in terms of the operating times. The parasitic drive powers are separated in components that are heat transfer related, p11 and p21, and distribution related, p12 and p22 (the respective flows and operating times may be quite different; p22 may for instance be a pump for a recharging system = rcs of a ground storage).

$$\overline{COP}_{hs} = \frac{\int_0^{\tau_{hp}} (\dot{Q}_1 + \dot{W}_{e,p11}) \cdot d\tau + \int_0^{\tau_{hs}} \dot{W}_{e,p12} \cdot d\tau}{\int_0^{\tau_{hp}} (\dot{W}_{e,hp} + \dot{W}_{e,p11} + \dot{W}_{e,p21}) \cdot d\tau + \int_0^{\tau_{hs}} \dot{W}_{e,p12} \cdot d\tau + \int_0^{\tau_{rcs}} \dot{W}_{e,p22} \cdot d\tau} \qquad (6)$$

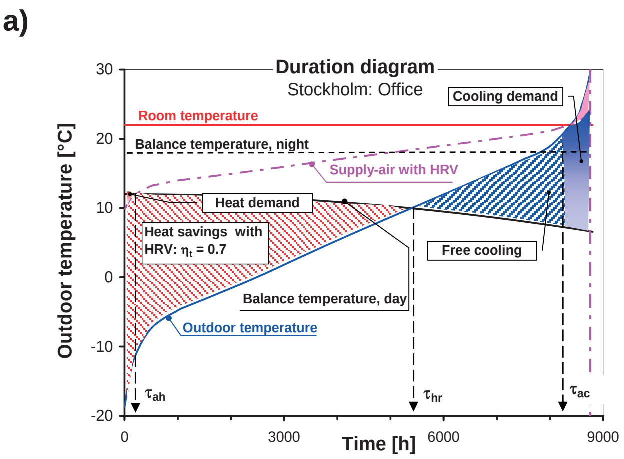
Eq. 7 illustrates in a slightly different way how the heat pump system *COP* is reduced by W_{ep1} and W_{ep2}, more so the higher the heat pump *COP* (drive powers are multiplied by COP_{hp}).

$$COP_{hps} = \frac{\dot{Q}_1 + \dot{W}_{ep1}}{\dot{W}_{e,hp} + \dot{W}_{ep1} + \dot{W}_{ep2}} = COP_{hp} - \frac{\left(COP_{hp} - 1\right) \cdot \dot{W}_{e,p1} + COP_{hp} \cdot \dot{W}_{e,p2}}{\dot{W}_{e,hps}} \qquad (7)$$

3.5 Heat Transport - Heat Loss and Drive Energy to Pumps and Fans

Efficient heat transport to or from the heat pump should minimize the parasitic losses of drive energy and heat. The effect of such losses is much more important in terms of energy than in terms of power due to the difference in operating times of heating and cooling systems and that of the heat pump (see eq. 6). Heikkilä (Heikkilä, K. 2007) noted in a comparison of alternative air-conditioning systems for office buildings that the drive energy for air distribution totally dominated the environmental impact. In another study, Haglund-Stignor (Haglund Stignor, C., Fahlén, P. and Sundén, B. 2007) discusses the importance of not unduly increasing the distribution flows, and hence the transport work, in the search for minimized supply and terminal unit work (the terminal unit in this case being the cooling coil of a display cabinet). Furthermore, Fahlén (Fahlén, P., Markusson, C. and Maripuu, M.-L. 2007) has indicated the possibilities of new system designs and points out that there are three basic ways of reducing the parasitic drive energy for distribution:

- Improve efficiency. Component development of pumps, fans, motors and motor drives.
- Reduce flow rate. System design, e.g. COD.
- Reduce pressure drop. System design (see for instance figures 5 and 6).



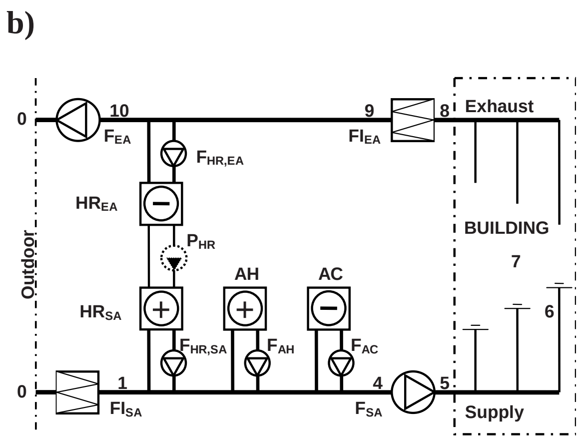


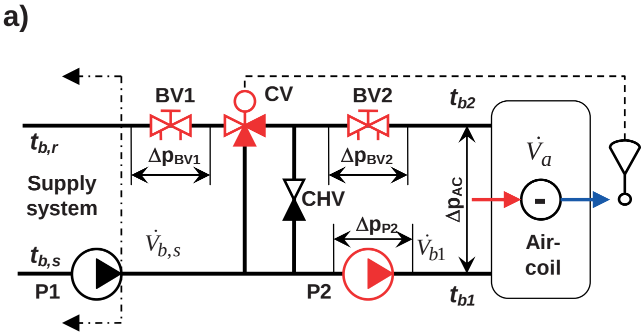
Figure 5: a) Duration diagram of the temperatures of outside air t_{oa}, supply-air t_{sa2} (after heat recovery) and the balance temperature t_{bt} for an office building. b) Parallel design of an air-handling unit.

Figure 5a shows the duration diagram of outdoor and supply-air to an office building. Variable ventilation flows handle most of the space conditioning and the consequence is that the supply-air heater and cooler operate less than 2 % and 9 % of the time respectively. The rest of the time they constitute unwanted pressure drops. Going from an in-line to a parallel design of the air-handling unit is a possible solution to reduce parasitic drive power on the air-side (Fahlén, P., Markusson, C. and Maripuu, M.-L. 2007). Further gains are possible on the liquid side, see 3.6 below.

3.6 Terminal Units - Drive Energy to Pumps and Fans

Terminal units have a decisive influence on the overall efficiency of a heat pump heating and cooling system. The design and sizing will determine the temperature lift of the heat pump and the possibilities of using direct free-cooling (see 3.2 and 3.3). Cooling beams and actively controlled air-supply devices, which permit wide ranges of air flow rate and supply temperature, are such examples. In all cases, however, it is not sufficient to look only at the thermal side. One has to consider parasitic drive powers as well.

Figure 6 shows how new technology can simplify design and at the same time save on drive energy. By choosing direct flow control of a heater or cooler with a VSD pump instead of the traditional shunt group, the number of components goes down (no control or balancing valves) and the drive energy may be reduced by a factor 10 in COD systems. Examples of possible applications of direct flow control by decentralized pumps are, for instance, indirectly cooled display cabinets, supply-air coolers and heaters with integral pumps and fans (figures 5 and 6) and radiators or fan-coil units with integral VSD pumps.



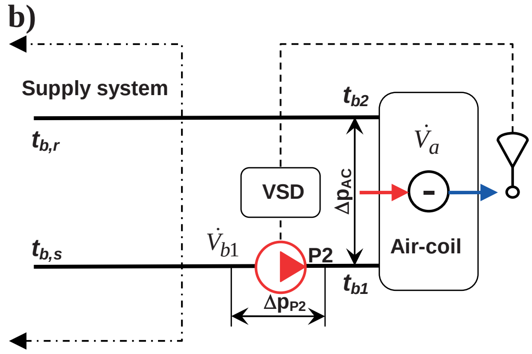


Figure 6: Control by means of a) variable inlet temperature and b) variable coil flow.

4 APPLICATION EXAMPLES

This section provides two examples of how free-cooling and new system designs may reduce the need for drive energy to electric motors of compressors, pumps and fans. The first is a case-study of a ground-source heat pump with combined heating and cooling including seasonal storage and the second a case-study of how the *SPF* of a standard ground-source heat pump can be upgraded.

4.1 An Office Building - Cooling and Heating Application

Space conditioning of office buildings nowadays requires cooling during a large part of the year, in daytime even at outdoor temperatures as low as -10 °C. Using the ground as heat source, heat sink or storage, in combination with a heat pump, is an energy efficient way of satisfying alternating or simultaneous demands for heating and cooling. In the heating mode, the heat pump cools the ground. After the heating season, the cold ground may be used for cooling simply by pumping brine through a heat exchanger in the building. This will heat the ground and at the end of the cooling season the ground temperature may not be sufficiently low. Then the heat pump may have to operate to reduce the brine temperature. Unless there is alternative use for condenser heat it will have to be wasted to the ambience.

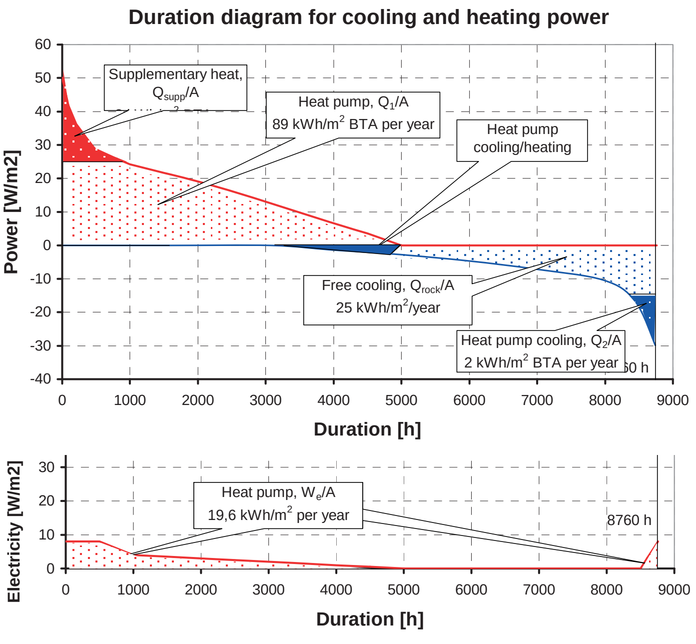
4.1.1 System Specifications

Building:	Office building of 5300 m^2 at Lund University, Sweden.
Ventilation:	VAV/CAV with heat recovery.
Heating and cooling:	Radiators and low temperature supply-air.
Supply system:	Heat pump with ground as source, sink and storage; supplementary heat by district heating.

Savings by reducing the purchase factor through heat recovery, variable ventilation flow, free-cooling with ventilation (outside) air and seasonal ground storage. No HVAC system modifications.

4.1.2 Results

Figure 5 provides measured results from February 2002 to February 2003.



Annual energy use	**MWh**	**MWh/ m^2**
Heating demand	515	97
Heating supply		
Heat pump	475	89
Supplementary	40	8
Cooling demand	155	29
Cooling supply		
Free cooling	130	25
Heat pump[1)]	15	3
Heat pump[2)]	10	2
Electricity		
Compressor	104	19.6
Pumps	7	1.3

1) Simultaneous heating and cooling
2) Cooling only

Figure 7: Results from a combined heating-cooling-storage system in Lund (Naumov, J. 2005).

4.1.3 Discussion

Measured results indicate that the parasitic ratio, $R_p = W_p/W_{hp}$, is close to optimal (7 %; see 3.4) for maximizing *COP*. As noted in 3.2, however, much greater use of direct free-cooling would have been possible with high supply temperature cooling water and chilled beams. This would also have resulted in higher brine temperatures during the winter heating season and hence less drive power to the heat pump for heating.

4.2 A Residential Building - Heating Only Application

This single-family house was the reference house of the Nordic heat pump competition in 1995 and was retrofitted with the winning low-cost ground-source heat pump system (around 4600 € for heat pump, borehole, one fan-coil unit, one radiator and installation). The system has subsequently been upgraded with a number of modifications (see figure 8a). The order of the modifications differs in practice, c.f. figure 9a, from the planned actions of figure 8a.

4.2.1 System specifications

Building: Timber-framed, single-family house, 150 m^2 in Boras, Sweden.
Ventilation: Mechanical exhaust, CAV.
Heating and cooling: Direct-acting electric plus hydronic fan-coil and radiators (retrofit 1).
Supply system:
1) Heat pump with ground as source, heating only.
2) Recharging of the borehole from an exhaust-air heat recovery coil.
3) Storage tank for heating and hot water, new control system.
4) Addition of 4 more radiators.

Modifications aim to reduce the purchase factor through a combination of recharging the ground storage, load matching with a storage tank and better control, increased room heater capacity, and reducing the parasitic drive power ratio.

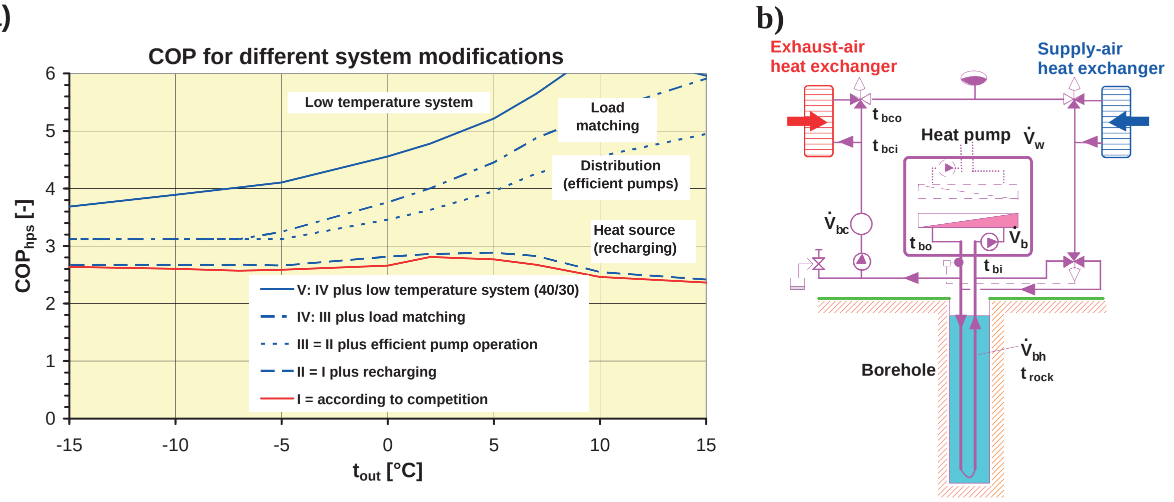


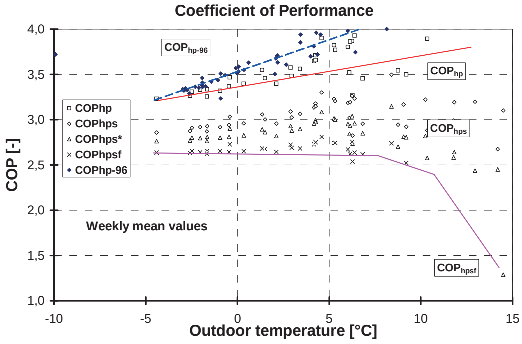
Figure 8: a) Predicted COPhps for alternative modifications of the original installation. b)Schematic of the recharging system with exhaust-air and supply-air heat exchangers.

4.2.2 Results

Figure 8a illustrates the effects of alternative modifications to the original system (I). Neither in theory (fig. 8a), nor in practice (fig. 9a) does the addition of a recharging system (II) have any noticeable effect on *COP*. However, together with more efficient pump operation (III), load matching (IV; a storage tank) and improved room-heater capacity (V) there are drastic improvements. SPF_{hps} goes up from 2.7 to around 4.2 in theory and 3.7 in practice. The latter figure, however, also includes hot water and also the pumps have not yet been upgraded.

Figure 9a shows the *COP* as a function of the outdoor air-temperature with recharging during the winter 2000/2001 (as a comparison, the diagram includes COP_{hp} without recharging during 1996). COP_{hps*} includes the recharging pump and COP_{hpsf} includes also the fan in the fan-coil unit.

a)



b)

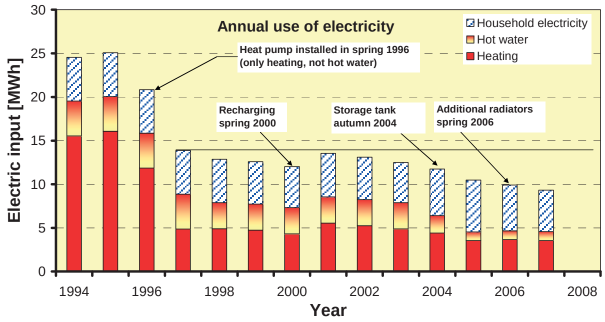


Figure 9: a) Measured values without and with recharging of COP_{hp}, COP_{hps}, and COP_{hs} respectively. b) Measured values of electricity for heating, hot water and household purposes.

4.2.3 Discussion

The results of figure 9a confirm the prediction in figure 8a that performance will not directly improve from raising the borehole temperature by recharging with heat from the exhaust air. Unless something is done regarding the heat transfer capacity of the heating system the actual gains of recharging are miniscule. It is like fitting a larger heat pump to an existing system, the condensing temperature will rise, on-times will be shorter and the parasitic ratio will increase. The quickest savings are simply to go for the best available pump technology and control. By this, the original parasitic ratio of R_p = 0.38 can be reduced to 0.08. No change in the heat pump as such is likely to provide an efficiency improvement of this order (30 %). Figure 9a also illustrates the importance of defining the system boundary (c.f. figure 3). With or without pumps and fan makes a tremendous difference.

The end result, after all modifications, is quite satisfactory. With no change to the heat pump, SPF_{hps} has improved by 30-40 % while at the same time providing both increased energy coverage for heating and now also including hot water (both factors will normally reduce *COP*). The total annual purchase of energy has been reduced from 25 to < 10 MWh and the specific purchase is now 67 W/m^2/year. This is lower than in most modern passive houses being built in Sweden with super insulation, high efficiency heat recovery and solar heat.

5 CONCLUSIONS

The efficiency of *heat pumps* has developed and continues to develop through a number of improvements of components in the refrigerant system such as compressors and compressor motors, condensers and evaporators, expansion devices, but also with new process concepts etc. However, future improvements on the efficiency of *heat pump systems* will to an ever increasing extent depend on component and system developments outside the heat pump per se. Some important conclusions from previous and ongoing research are that:

- Improved design of conditioned spaces such as buildings, display cabinets etc. will reduce the demand for heating and cooling.
- Demand reduction tends to be larger in terms of energy than in terms of power. This will reduce the system load factor.
- Reduced load factors will reduce the HVAC system efficiency and increase the importance of storage and capacity control for load matching.
- Improved system efficiency is possible by reducing the drive powers and heat losses for heat transfer and distribution.
- Distinction between heat transfer and distribution pressure drops facilitates an optimization of the flows and drive powers to pumps and fans (the relative operating times differ).

- New system designs with distributed pumps and fans can drastically reduce the HVAC system drive power (full control authority with no additional control pressure drops).
- Heat pump systems tend to have parasitic energy ratios much greater than optimal, in some applications over 50 %.

Summing up, it is possible by means of heat pump technique to drastically reduce purchased energy in numerous applications for space conditioning and hot water. This, however, requires attention to engineering details in order to fully benefit from the theoretical potential. Many of these details relate to the minimization of parasitic drive energy and heat loss.

6 NOMENCLATURE

	Symbols, Latin letters		
COP	coefficient of performance [-]	*T*	thermodynamic temperature [K]
D	demand factor [-]	$\dot{V}$	volume flow rate [m^3/s]
L	load factor [-]	*R*	ratio [-]
p	pressure [Pa]	*W*	work, mechanical or electric [J]
Q	heat [J]	$\dot{W}$	power, mechanical or electric [W]
$\dot{Q}$	power, thermal [W_{th}]		**Symbols, Greek letters**
SPF	seasonal performance factor [-]	Δ	difference, change (e.g. pressure, temp.)
t	celsius temperature [°C]	η	efficiency, temperature [K/K_{max}]

	Subscripts				**Abbreviations**
1	condenser	cm	cooling mach.	CAV	Constant Air Volume flow rate
2	evaporator	cs	cooling sys.	COD	Control-On-Demand
a	air	hcs	heating and	HVAC	Heating, Ventilation and Air-Conditioning
ac	air-coil		cooling system	TQ	Thermal Quality
b	brine	hp	heat pump	VAV	Variable Air Volume flow rate
e	electric	hps	heat pump sys.	VSD	variable-speed-drive
f	fan	hr	heat recovery		
m	motor	rcs	recharging sys.		
out	outdoor	Q	heat		
p	pump, parasitic	W	work		

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FIELD EXPERIENCE WITH GROUND-SOURCE HEAT PUMPS IN AFFORDABLE LOW ENERGY HOUSING

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Abstract: The tangible end-user benefits of low energy housing are needed most at the entry level, where high energy costs can be least afforded. Yet, this sector also provides the greatest challenge in managing the associated construction cost increase. Lasting structural and behavioral changes in the marketplace are required to overcome the barriers created by an over-emphasis on first cost. A large-scale demonstration was initiated with just such a goal. This affordable housing project utilizes ground-source heat pumps, low energy construction techniques, and in some cases, on-site solar electricity production to reduce energy consumption by 60-80% from current practice. This paper discusses the creation, goals and funding of the project, reviews the energy reduction measures employed, and presents the estimated and actual results in terms of energy consumption, carbon emissions, and economics.

Key Words: ***heat pump, geothermal, ground-source, low energy, zero energy, solar, photovoltaic, affordable, low income, Habitat***

1 INTRODUCTION

Habitat for Humanity International (HFHI) is a nonprofit Christian housing ministry that has built more than 250,000 homes around the world, providing more than 1 million people with safe, decent, affordable shelter. HFHI is ranked the third largest private home builder in the U.S., completing on average more than 5,000 homes per year and another 20,000 homes per year in other countries. Although they depend on volunteer labor and donations of money and materials, Habitat is not a giveaway program. In addition to a down payment and the monthly mortgage payments, these limited-income homeowners invest hundreds of hours of their own labor into building their own Habitat house and the houses of others.

Like most HFHI affiliates, Central Oklahoma Habitat for Humanity (COHFH) has focused on minimizing the initial costs of construction to keep homeowner mortgage payments low. Their modest wood-frame single-family detached homes have traditionally been built to code-minimum insulation specifications and have utilized standard-efficiency natural gas forced-air furnaces, split-system central air conditioners, and gas-fired storage water heaters. With U.S. natural gas prices doubling in recent years, an overlooked aspect of the COHFH mission to provide affordable home ownership emerged… the monthly energy costs, which in peak months could rival the mortgage payment for their lowest-income homeowners, because the payments are based on their income level.

COHFH installed its first ground-source (or geothermal) heat pump (GHP) system in 2005, with the technical and financial support of a local GHP manufacturer. During 2006, 10 more GHP systems were installed, with 9 as part of the COHFH “Home Builder Blitz” in Spencer, Oklahoma. The Blitz homes were sponsored by a group of local professional home builders and were all constructed from start to finish in 5 days as a promotional event.

In late 2006, COHFH reached an agreement with the local GHP manufacturer to incorporate GHP systems into all of their homes on an ongoing basis, an average of 50 per year. For the next 5 years, most of these homes will be built in Hope Crossing, a new (first phase 2007) 240 lot COHFH development located on 24 hectares in Oklahoma City. In early 2007, the local electric utility added its support to the Hope Crossing project. The additional funding provided for envelope, lighting, and appliance upgrades that were desired to further reduce the energy consumption of these homes. Thus, a private-sector collaborative team consisting of COHFH, the electric utility, and the GHP manufacturer was formed with the ambitious goal of making Hope Crossing a showcase large-scale demonstration of affordable low energy housing.

1.1 Hope Crossing Project Goals

- Install GHP systems in all homes

- Reach the lowest energy consumption feasible in all homes by using cost-effective and generally available measures such as insulation and window upgrades, and high efficiency lighting and appliances

- Track the energy consumption of all homes to establish a baseline of performance, and install a proportion of “smart” data-recording meters to develop detailed electricity demand profiles

- In a limited number of homes, integrate on-site solar photovoltaic (PV) grid-connected systems to demonstrate the benefits of zero peak energy (no net grid electricity use during summer peak periods) and eventually zero net energy (no net grid electricity use over a full year)

- Utilize this large-scale, affordable, low energy housing demonstration as a market transformation publicity tool, as a platform to generate spin-off projects with other HFHI affiliates, and as a means to attract additional COHFH donations

1.2 Hope Crossing Project Funding

The average COHFH house costs about $85,000 to build, which is also the selling price to the homeowner. However, COHFH directly provides its homeowners a zero-interest mortgage loan, so the selling price is actually received over a 20-30 year period. So, even if the costs of reducing the energy consumption of their homes were included in the selling price, they would actually be recovered slowly over many years as the mortgage was paid. For COHFH to proceed with the project, the funding for the additional costs had to be found elsewhere.

The GHP manufacturer and its local installer agreed to install the GHP systems for COHFH at the same price they were paying for a gas furnace and air conditioner. Several steps were taken to reduce the installation costs to make this more feasible:

1. Integrate the ground-loop pumping and purging valves into the heat pump to save on space required, equipment cost, and field labor content

2. Drill a single 120m heat exchange bore directly under the floor slab instead of the typical practice of drilling two 60m heat exchangers in the lawn, which requires a separate excavation step to manifold them and route the piping into the house.

These measures put the installer in a better position, but to meet the cost goal still requires the manufacturer to donate the GHP. The electric utility is funding the incremental costs for the envelope, lighting, and appliance upgrades as part of its low-income weatherization program. Both of these parties shared the costs for the 2 homes equipped with solar PV systems. There is good justification for this level of financial support beyond the project demonstration value: it is an act of corporate stewardship, but excels over a one-time gift in continuing to provide financial and environmental benefits over many years; in addition, the project provides valuable experience in an unexplored segment of the housing market.

2 THE PROGRESSION TO LOW ENERGY

Oklahoma City is considered a mixed-humid climate that requires significant amounts of both heating and cooling, and that has significant humidity levels throughout the year. Air conditioning is considered a must in all new homes, which makes air-based distribution systems the norm. Most new homes use either a central forced-air gas furnace coupled with a split-system air conditioner, or a split-system central air-source heat pump and supplemental resistance heat. In low-cost single-story houses typical of the COHFH homes, the central duct system is usually located in the unconditioned attic, perhaps the worst possible location from an energy-efficiency standpoint.

Table 1 outlines the characteristics of the 4 types of COHFH homes that have been constructed to date, with the “Standard Gas” home being typical of all COHFH homes built up until recently. The first step taken to reduce energy consumption was to install a GHP system in this same type of home, which is designated “Standard GHP” in Table 1. Later, the envelope, lighting, and appliances were upgraded leading to the designation of “Low Energy GHP” home. Finally, 2 homes to date have been equipped with solar PV systems sized with a goal of being zero-peak, or nearly “off-grid” during the utility peak load period, which occurs during summer afternoons. In addition, these “Low Energy GHP + PV” homes should not require any grid electricity (on a net basis) to operate the GHP system over the year.

Figure 1 illustrates the major changes made to the homes in the progression from Standard Gas to Low Energy GHP + PV.

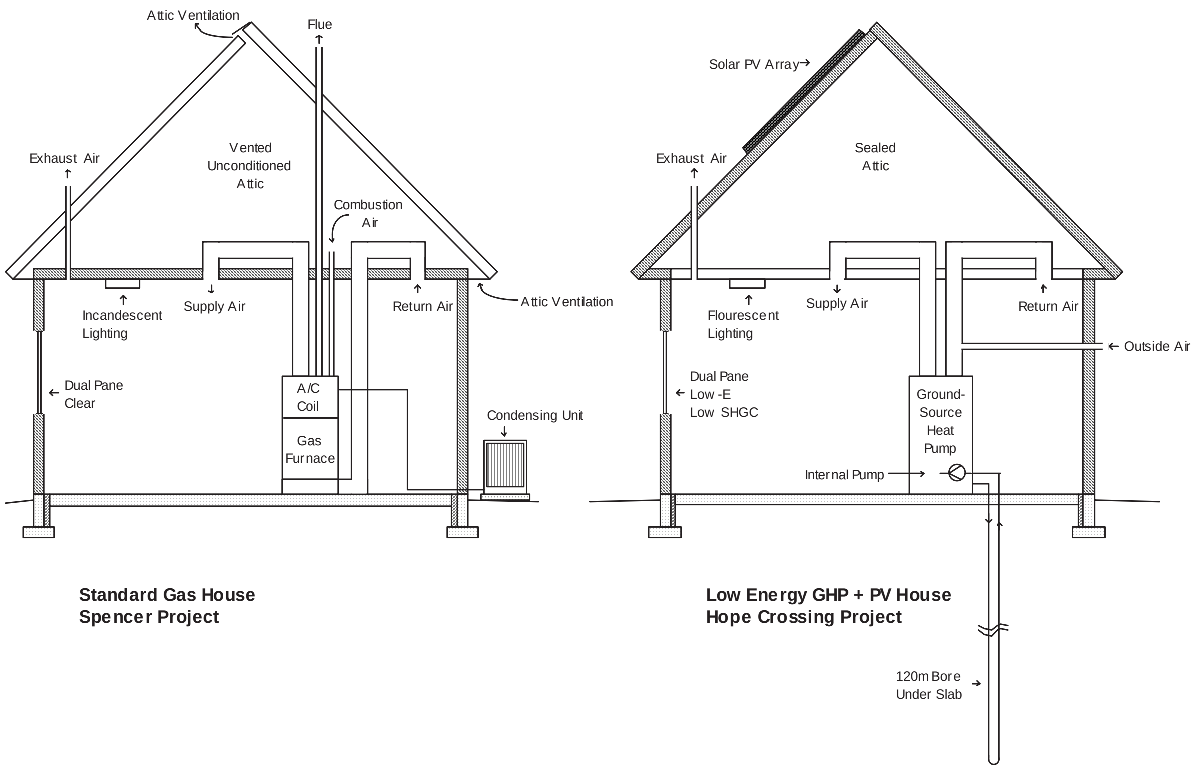

Figure 1: Evolution of COHFH Homes

Table 1: COHFF Housing Characteristics

House Type		Standard Gas	Standard GHP	Low Energy GHP	Low Energy GHP + PV
Project		Spencer	Spencer	Hope Crossing	Hope Crossing
Heating		Gas	GHP	GHP	GHP
Cooling		Split A/C	GHP	GHP	GHP
Hot Water		Gas Storage	Elec Storage[1]	Elec Storage[1]	Elec Storage[1]
Lighting		Incandescent	Incandescent	Flourescent	Flourescent
Appliances		Standard	Standard	Energy Star[2]	Energy Star[2]
Solar PV Array	DC kW				2.3
Construction		Wood Frame	Wood Frame	Wood Frame	Wood Frame
Floor		Slab on Grade	Slab on Grade	Slab on Grade	Slab on Grade
Attic		Vented	Vented	Sealed	Sealed
Insulation Type		Fiberglass	Fiberglass	Spray Foam	Spray Foam
Air Duct Location		Attic	Attic	Attic	Attic
Ventilation		Spot Exhaust	Spot Exhaust	Central Supply	Central Supply
Average Floor Area	m^2	110	110	110	110
Total Envelope UA	W/K	183	183	137	137
U-value - Slab Perimeter	$W/(m^2 \cdot K)$	1.10	1.10	1.10	1.10
U-value - Wall	$W/(m^2 \cdot K)$	0.35	0.35	0.35	0.35
U-value - Flat Ceiling	$W/(m^2 \cdot K)$	0.19	0.19		
U-value - Sloped Roof	$W/(m^2 \cdot K)$			0.26	0.26
U-value - Air Duct	$W/(m^2 \cdot K)$	0.95	0.95	0.71	0.71
U-value - Window	$W/(m^2 \cdot K)$	2.84	2.84	1.99	1.99
SHGC - Window		0.62	0.62	0.40	0.40
Natural Air Change Rate - Htg	1/h	0.78	0.78	0.30	0.30
Natural Air Change Rate - Clg	1/h	0.56	0.56	0.22	0.22
Envelope Air Leakage Rate		Untested	Untested	Tested	Tested
Air Duct Leakage Rate		Untested	Untested	Tested	Tested
Heating Load (-11°C, 21°C)	kW	8.2	8.2	5.3	5.3
Cooling Load (36°C, 24°C)	kW	6.1	6.1	4.7	4.7
Hot Water Load (55°C)	kWh/yr	3200	3200	3200	3200
Lighting Load	kWh/yr	1753	1753	701	701
Appliance Load	kWh/yr	4667	4397	3518	3518

1. GHP desuperheater assist
2. Energy Star appliances meet strict efficiency guidelines

3 ACTUAL ENERGY CONSUMPTION – STANDARD HOUSES

Monthly energy consumption during 2007 was collected from utility meter data for 16 homes in the Spencer project. The characteristics of these homes are outlined in Table 1, with 8 being of type Standard Gas and 8 of type Standard GHP. The occupancy of each house type was essentially the same, with the Standard Gas homes averaging 2.8 people, and the Standard GHP homes averaging 2.6. Figure 2 shows the total energy consumption for each house type, by month and annually, as determined by averaging the monthly utility meter data for the homes of each type.

The heating (E_H) and cooling (E_C) components of the energy consumption for each house type were derived by analyzing the monthly metered energy consumption and corresponding monthly Degree Days (DD_H, DD_C) using linear regression. Figure 3 presents the results of this analysis, normalizing the energy consumption to remove the y-intercept, or base load component, in each case. The high value of correlation (R^2) indicates that the heating and cooling loads of the standard homes are strongly dominated by the outdoor temperature, or in effect, the building envelope.The reduction in heating and cooling energy consumption achieved by the GHP system was:

E_H reduction using GHP = 1- (E_H GHP/E_H Gas) = 1 – (.0134/.0397) = 66.3%

E_C reduction using GHP = 1- (E_C GHP/E_C A/C) = 1 – (.0145/.0284) = 48.8%

Figure 4 shows the total energy cost for each house type, by month and annually, as determined by averaging the actual monthly utility costs for the homes of each type. The reductions in annual total energy consumption (E_{Ta}) and annual energy costs ($E_{\$a}$) achieved by the GHP system were:

E_{Ta} reduction using GHP = 1- (E_{Ta} GHP/E_{Ta} Gas) = 1 – (116.2/234.3) = 50.4%

$E_{\$a}$ reduction using GHP = 1- ($E_{\$a}$ GHP/$E_{\$a}$ Gas) = 1 – (\$1,023/\$1,606) = 36.3%

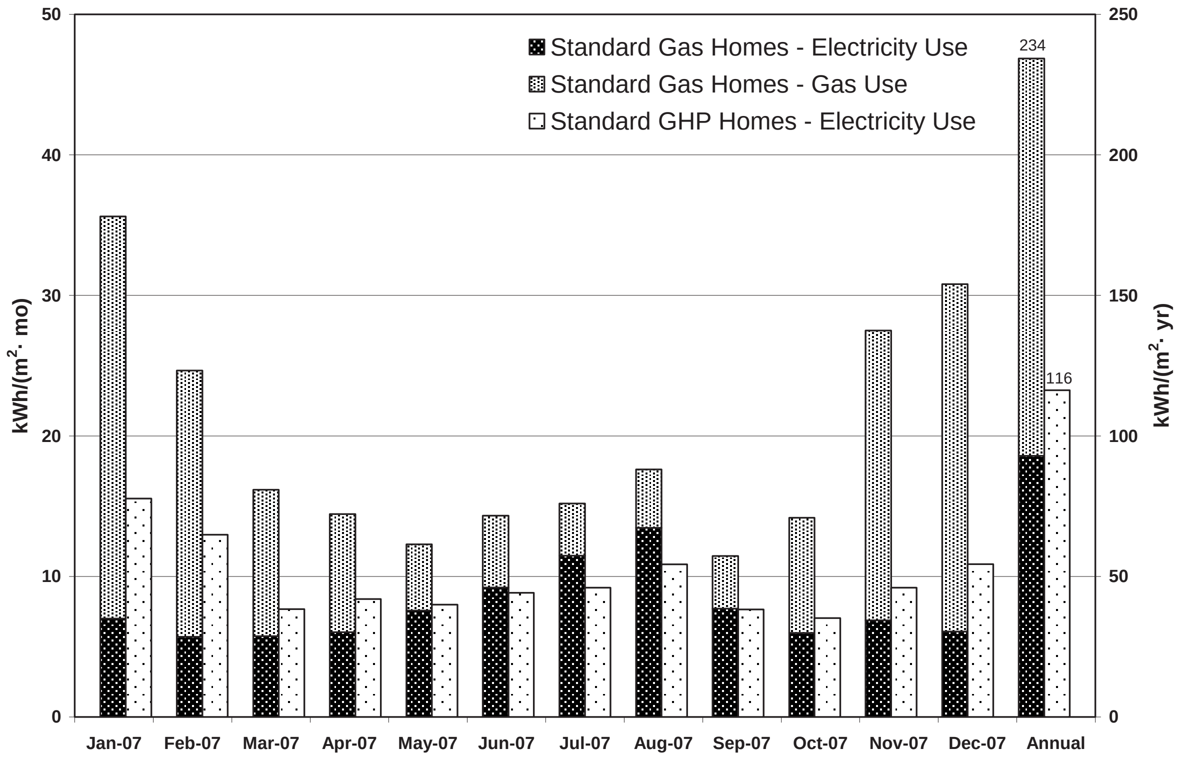

Figure 2: Average Metered Energy Consumption

Although many low energy homes were completed in the Hope Crossing project from mid-2007 onward, this did not provide an adequate time period to gather useful energy consumption data for analysis.

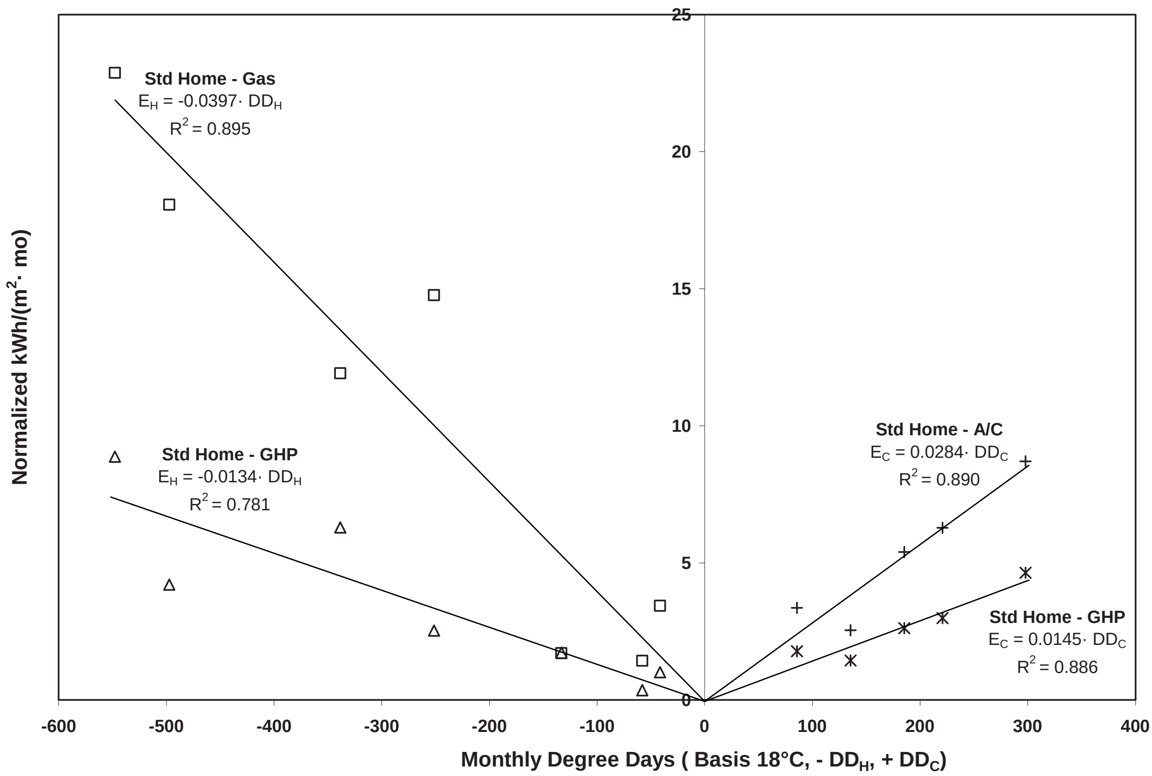

Figure 3: Derivation of Heating and Cooling Energy Consumption

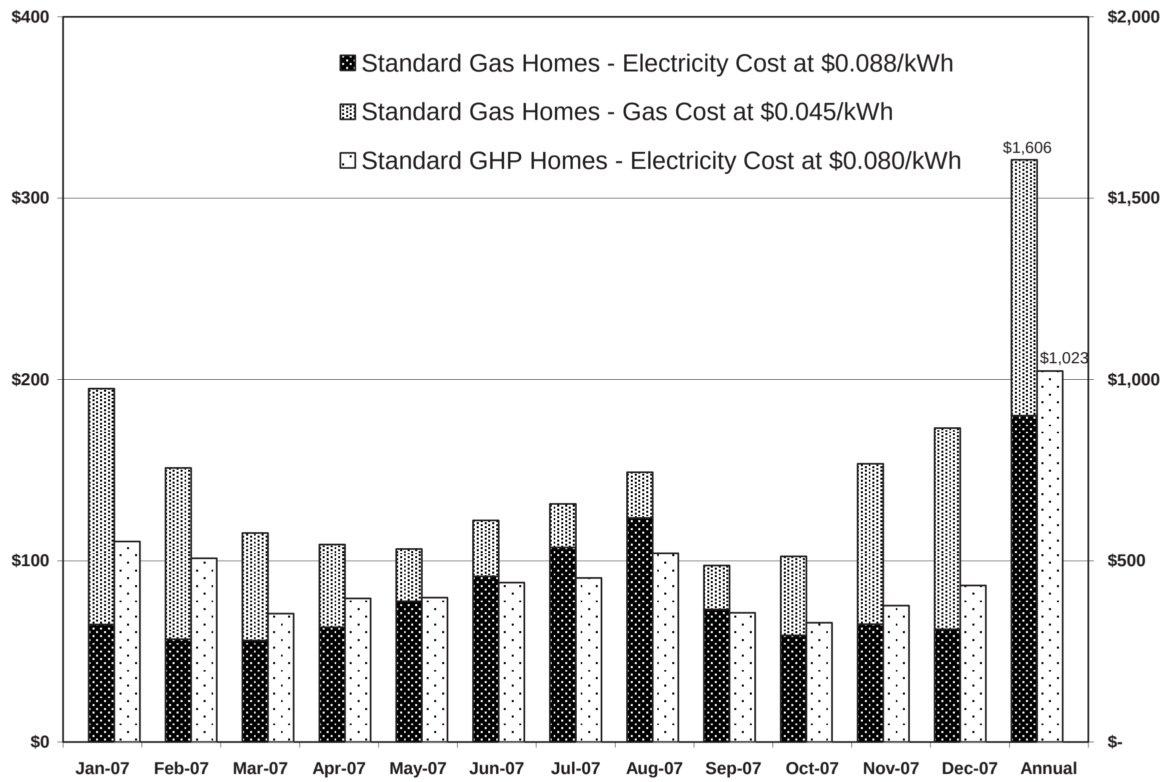

Figure 4: Average Metered Energy Cost

4 ESTIMATED ENERGY CONSUMPTION – ALL CURRENT HOUSES

Energy consumption estimates were generated as part of the project planning process in order to evaluate the benefits of GHP technology, of low energy construction modifications, and of integrating a grid-connected solar PV system. All typical end-use loads were considered in order to estimate the total energy consumption of the homes.

Benchmark hot water, lighting, and appliance loads for the standard house types were estimated using the methodology of Hendron et al. (2004). Following the benchmark procedures, the hot water load is calculated as a function of the number of bedrooms (N_{br}), the supply temperature setting, and the water mains temperature, which is assumed to vary as a function of climate and day of year. The lighting load is calculated as a function the finished floor area (FFA). The total appliance load calculated as a function of N_{br} and FFA.

In the low energy houses, compact fluorescent lighting (CFL) displaces the benchmark incandescent lighting used in the standard houses, reducing lighting energy consumption by an assumed 60%. The low energy homes also use Energy Star appliances, which are assumed to reduce benchmark appliance energy consumption by 20%.

For the GHP homes, all loads are met with electricity. In fact, natural gas distribution lines were not installed in the Hope Crossing project. For the standard homes using natural gas for heating, the hot water heater is also gas-fired, and it was assumed, based on feedback from COHFH, that 75% of the ranges and 25% of the clothes dryers were also supplied by gas. The remainder of the loads in the gas-heated homes are met with electricity.

Residential energy analysis software (ClimateMaster 2004) was used to estimate the annual energy consumption of the heating, cooling, and water heating equipment for all system types. The software uses design load calculations, annual bin weather data, and detailed equipment performance characteristics, including factors such as auxiliary energy use and cycling losses, within a modified bin method calculation. Ground heat exchanger sizing and performance modeling are also incorporated.

The annual energy output of the solar PV system was estimated using PVWATTS software (NREL 2008). The software uses hourly Typical Meteorological Year (TMY) weather data and a PV performance model that incorporates array DC rating, DC to AC derating factors, and array tilt and azimuth to estimate AC energy production (kWh) for a crystalline silicon PV system.

Figure 5 shows the annual site energy consumption obtained using the estimation methods discussed, in total and by end-use, for each of the 4 COHFH house types listed in Table 1. Houses of all 4 of these types have now been constructed. Upon review of Figure 5, it is apparent that as the energy consumption of the homes is reduced by using GHP systems and low energy construction techniques, the hot water and appliance loads become dominant. Looking ahead, there are several methods of addressing the hot water load, leaving the appliance (which includes plug) load as a major challenge.

4.1 Validation of Energy Consumption Estimates

Figure 6 compares the estimated annual energy consumption to the actual metered energy consumption during 2007 for the Spencer project homes. The estimates are calculated using an average weather year for the locale. In 2007, the annual DD_H were 13% less than in the average weather year, and the annual DD_C ran 16% below normal. The adjusted estimates in Figure 6 are corrected to correlate with the actual weather incurred during 2007. As can be seen, the estimated energy consumption for both house types is reasonably close to the actual consumption, deviating by less than 9% in the worst case after adjustment.

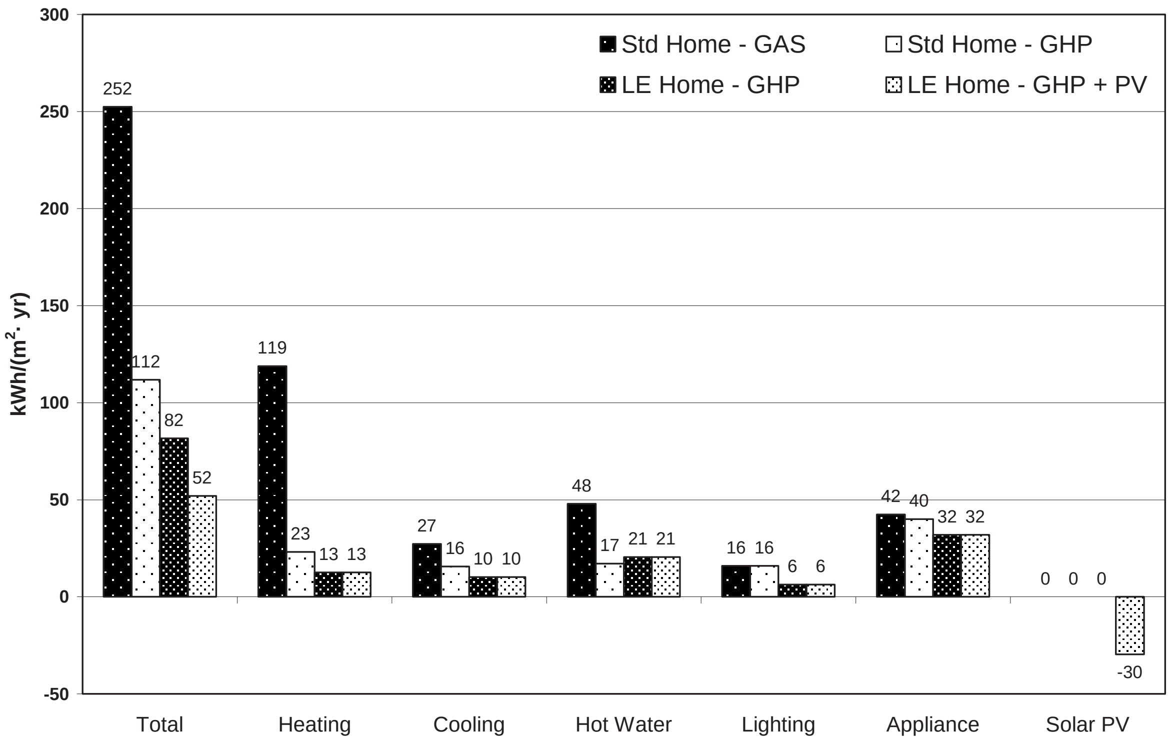

Figure 5: Estimated Site Energy Consumption by End-Use

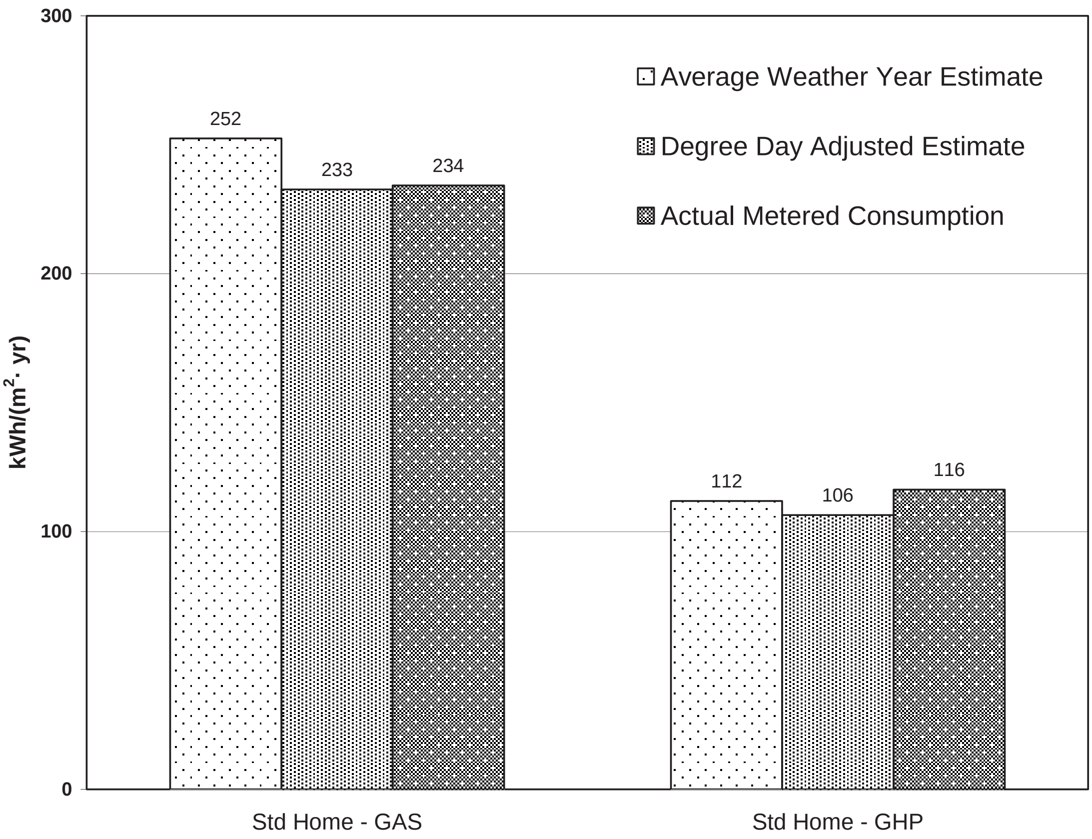

Figure 6: Estimated and Metered Total Energy Consumption

4.2 Source Energy and Carbon Emission Estimates

Source (or primary) energy consumption and carbon equivalent emissions are calculated using the annual site energy consumption estimates for each COHFH house type and factors for electricity and natural gas obtained from Deru and Torcellini (2006). For electricity, these factors account for power plant conversion inefficiencies, and transmission and distribution losses. They also include precombustion effects associated with extracting, processing, and delivering primary fuels to the point of conversion in the power plant. For natural gas, the factors account for both precombustion effects and on-site combustion emissions. The factors used are based on U.S. national averages.

Figure 7 compares the total estimated site and source energy consumption, and the associated carbon equivalent emissions, or global warming potential (GWP), for each COHFH house type. The 240 low energy GHP homes to be constructed in the Hope Crossing project will collectively save nearly 1,100 metric tons of CO_2 per year, or 22,000 metric tons over a nominal 20 year lifespan, compared to the Standard Gas homes that COHFH had been building. If all of the homes had the 2.3 kW solar PV option, another 12,000 metric tons could be saved over 20 years.

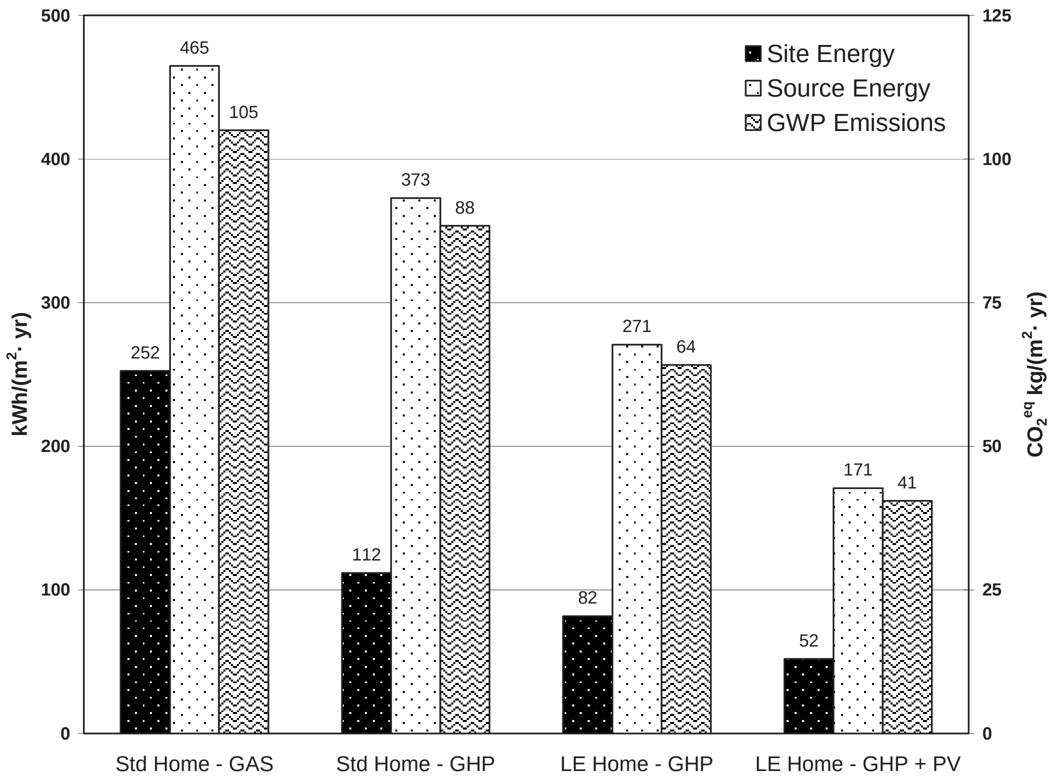

Figure 7: Estimated Total Energy Consumption and Emissions

5 FUTURE IMPROVEMENTS

Several additional improvements to the COHFH houses are under consideration:

- Advanced GHP with inverter-driven variable speed compressor and integrated full-condensing hot water modes in lieu of desuperheating (LE Home - Adv GHP)

- Advanced GHP with grid-connected solar 2.3 DC kW PV system sized for zero peak (LE Home - Adv GHP + PV)

- Full “zero- energy” home with a grid-connected 5.2 DC kW solar PV system capable of producing all the energy required for the home on a net basis over the year (ZE Home - Adv GHP + PV)

The estimated site energy consumption for homes incorporating these improvements are presented in Figure 8, with the current Low Energy GHP home as a baseline. The advanced GHP reduces the energy consumed for heating, cooling, and water heating by 34% compared with the current GHP. Note that the appliance load represents nearly 50% of the total energy consumption of the Low Energy Adv GHP home. No matter what is done to improve the envelope or thermal energy systems in these homes, the appliance and plug load will remain as the limiting factor.

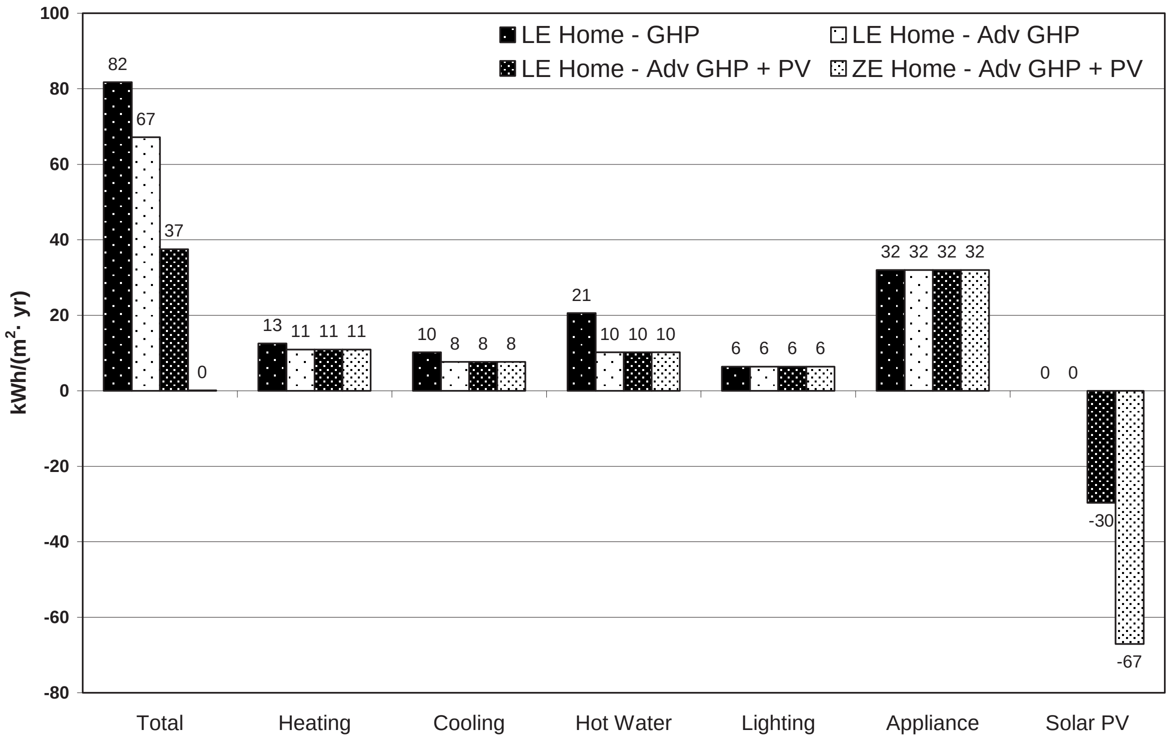

Figure 8: Estimated Site Energy Consumption by End-Use

6 ECONOMIC ANALYSIS

Table 2 provides detail on the additional investment and annual energy cost saving associated with current and future COHFH homes types relative to the benchmark Standard Gas home. The additional costs shown in Table 2 are the normal prices that any builder of COHFH scale would pay in the Oklahoma City market. They do not reflect the discounts and outright donations that COHFH is receiving. There is no valid benchmark for the price of solar PV systems in Oklahoma, so these costs were estimated using information from volume purchases in States with higher activity levels.

Figure 9 shows the return on investment (ROI) provided by each home type relative to the Standard Gas home. The ROI calculation assumes an annual energy cost inflation rate of 2%. In comparing the ROI presented in Figure 9 to alternative investments, it should be noted that energy cost savings are not taxable income, so this would be equivalent to the net after-tax ROI of an income-producing investment. The GHP systems and the low energy construction costs provide a very favorable ROI. The investment in Solar PV is much more difficult to justify on energy cost savings alone. Providing financial benefits for the carbon emission reductions would improve the economics for all of these alternatives.

Table 2: Additional Investment and Annual Energy Cost Savings

House Type:	Standard Gas	Standard GHP	Low Energy GHP	Low Energy Adv GHP	Low Energy GHP 2.3 kW PV	Low Energy Adv GHP 2.3 kW PV	Zero Energy Adv GHP 5.2 kW PV
Additional Investment[3]:							
Heating and Cooling	$ -	$ 4,500	$ 4,500	$ 6,000	$ 4,500	$ 6,000	$ 6,000
Solar Photovoltaic	$ -	$ -		$ -	$ 15,000	$ 15,000	$ 30,000
Insulation, Lighting, Appliance	$ -	$ -	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500
Total	$ -	$ 4,500	$ 7,000	$ 8,500	$ 22,000	$ 23,500	$ 38,500
Annual Energy Cost[1,2]:							
Heating	$ 596	$ 127	$ 69	$ 60	$ 69	$ 60	$ 60
Cooling	$ 270	$ 155	$ 101	$ 76	$ 101	$ 76	$ 76
Hot Water	$ 275	$ 132	$ 158	$ 78	$ 158	$ 78	$ 78
Lighting	$ 175	$ 175	$ 70	$ 70	$ 70	$ 70	$ 70
Appliance	$ 423	$ 440	$ 352	$ 352	$ 352	$ 352	$ 352
Solar PV Contribution	$ -	$ -	$ -	$ -	$ (228)	$ (228)	$ (636)
Total	$ 1,739	$ 1,029	$ 750	$ 636	$ 522	$ 408	$ -
Annual Energy Cost Saving[3]	$ -	$ 710	$ 989	$ 1,103	$ 1,217	$ 1,331	$ 1,739

1. based on electricity at $0.100/kWh for first 600 kWh/month, then $0.050/kWh winter and $0.090/kWh summer for all kWh/month over 600
2. based on natural gas at $0.045/kWh equivalent
3. Relative to Standard Gas Home

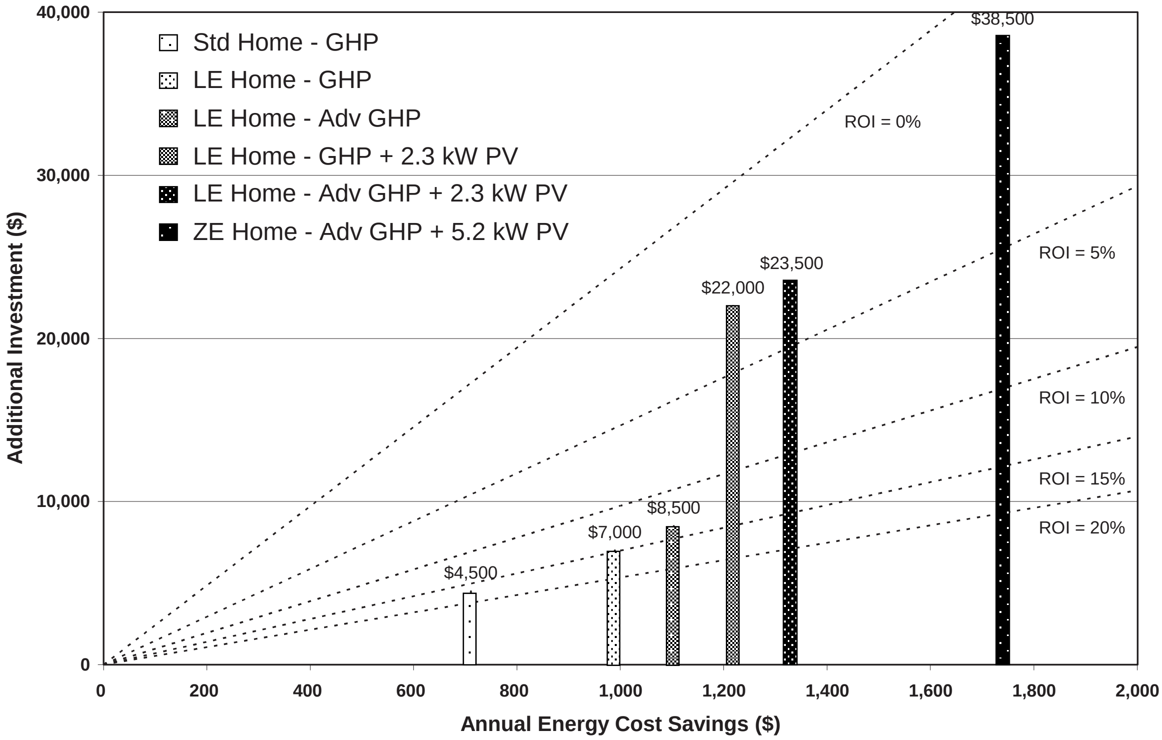

Figure 9: Return on Investment Relative to Standard Gas Home

7 CONCLUSIONS

- The total site energy consumed by the typical new COHFH home can be reduced from 50-75% through the use of GHPs combined with low energy construction techniques

- The 240 low energy GHP homes to be constructed in the COHFH Hope Crossing development will collectively eliminate nearly 1,100 metric tons of CO2 emissions per year, and 22,000 metric tons over 20 years, compared to the standard gas furnace homes typical of prior COHFH developments

- Actual annual site energy savings of 50% were obtained using GHP systems in standard homes during the first phase of this project, which led to a 36% reduction in annual energy costs

- GHPs and low energy construction are both cost-effective, providing a return on investment in the 15% range, and both are generally available in the COHFH locale

- When the energy consumption of a home is brought to these low levels, relatively small solar PV systems can be incorporated that will nearly eliminate the summer peak load imposed on the electric utility, and that will provide all of the energy required to operate the GHP system

- By using a larger solar PV system, zero energy homes are feasible, but expensive

- As the energy consumption of the home is reduced, the appliance and plug load grows to nearly 50% of the total, creating a limit on further reductions

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CHARACTERISTICS OF PRESSURE RECOVERY IN TWO-PHASE EJECTOR APPLIED TO CARBON DIOXIDE HEAT PUMP CYCLE

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Abstract: The aim of this research is to elucidate the characteristics of pressure recovery of a two-phase ejector applied to a carbon dioxide heat pump cycle. The effects of the geometric parameters of the ejector on pressure recovery are investigated in this study. The diameter and length of the constant-area mixing tube are varied within ranges of 2.0–3.6 mm and 2–40 mm respectively. The maximum pressure recovery is achieved at a distance of approximately 20 mm from the inlet of the mixing tube, irrespective of the diameter of the mixing tube, when the diameter of the nozzle throat is 0.82 mm. In addition, pressure recovery in the mixing section with a diameter of 3.6 mm is less in comparison with those with diameters of 2.0 and 2.5 mm; in the two latter cases, almost the same pressure recovery is observed. The clearance between the nozzle outlet and the inlet of the constant-area mixing tube is varied in order to compare the constant-area mixing ejector and the constant-pressure mixing ejector. Pressure recovery decreases when the clearance between the nozzle and the inlet of the mixing tube is large. The constant-area mixing ejector is therefore preferable for the carbon dioxide heat pump cycle. Finally, the performance of the ejector equipped with a diffuser is investigated; the measured ejector efficiency is 46 %, which corresponds to a COP improvement of approximately 23–24 %.

Key Words: ***Carbon dioxide, Two-phase ejector, Heat pump cycle***

Nomenclature

A: Cross sectional area [m^2]; D: Diameter [m]; h: Enthalpy [J/kg];
m: Mass flow rate [kg/s]; P: Pressure [Pa]; q_{in}: Cooling capacity per unit mass flow rate [J/kg];
u: Velocity [m/s]; w_{in}: Compression work per unit mass flow rate [J/kg]; Z: Axial displacement [m];
η: Isentropic efficiency [-]; ρ: Density [kg/m^3]; ω: Entrainment ratio [-];

Subscripts

cmp: Compressor; cnv: Conventional; d: Driving flow;
eje: Ejector; ejo: Ejector outlet; ev: Evaporator;
mx: Mixing section; mxi: Mixing tube inlet; mxo: Mixing tube outlet;
s: Suction flow.

1 INTRODUCTION

After the commercialization of “Eco Cute” in 2001, shipment of carbon dioxide heat pump water heaters has increased steadily. The total domestic shipment volume reached 1 million units in September 2007, and the Government expects that this volume will increase up to 5.2 million units till 2010. The coefficient of performance (COP) of the heat pump unit increased from 3.46 in the first generation to 4.9 towards the end of 2007. This improvement in the COP is attributed to improvement in the efficiencies of the components of the heat pump unit, including the compressor and heat exchangers. However, these improvements in the efficiencies proved to be expensive, and it is not possible further to improve the COP significantly. Since the carbon dioxide heat pump operates under high pressure conditions, the energy lost during the expansion process is considerably larger than that of heat pump cycles with conventional refrigerant. The introduction of an ejector in the cycle is greatly beneficial to the recovery of the energy lost during the expansion process. In addition, since the density ratio of vapour to liquid is close to unity, the two-phase flow ejector applied to a transcritical carbon dioxide heat pump cycle can easily achieve a high efficiency. The introduction of a two-phase flow ejector is therefore expected to be a cost-effective approach for improving the COP of carbon dioxide heat pumps.

Figure 1 shows a schematic of the carbon dioxide heat pump cycle with a two-phase flow ejector, and Fig. 2 illustrates the P-h diagram of the cycle. In this ejector cycle, supercritical carbon dioxide is compressed in the compressor and cooled in the gas cooler, and is then led to the ejector through a driving nozzle. Inside the driving nozzle, it is depressurized into a two-phase flow and accelerated to supersonic speed. The supersonic two-phase flow is directed to the mixing section together with the vapour aspirated from the evaporator. These two flows are mixed in the mixing section and the exchange of momentum takes place, with pressure recovery achieved due to the deceleration of the driving flow. In the diffuser section, the remaining momentum leads to further pressure recovery. The two-phase flow from the outlet of the ejector enters a separator, from which the vapour phase aspirates into the compressor; the liquid phase is depressurized by an expansion valve and flows to the evaporator. Due to the mass balance with regard to both the liquid and vapour phases, the entrainment ratio ω — the ratio of the suction flow rate to the driving flow rate — must satisfy Eq. 1, where x_{ejo} indicates the quality at the outlet of the ejector.

$$\omega = \left(1/x_{ejo}\right) - 1 \qquad (1)$$

Because the suction pressure of the compressor is higher than the pressure at the outlet of the evaporator in the ejector cycle, the COP of the ejector cycle can be improved in comparison with that of a conventional cycle. The cooling COP of the ejector cycle is defined as follows.

$$COP = \frac{m_s q_{out,eje}}{m_d w_{cmp,eje}} \qquad (2)$$

Kornhauser [1] analyzed the improvement in theoretical COP of an ejector cycle using R-12 refrigerant. It was found that up to 21 % of the conventional cycle can be realized under standard conditions, i.e., with evaporator and condenser temperatures of –15 °C and 30 °C respectively, assuming constant-pressure mixing and isentropic components. Nakagawa et al. [2] investigated the influence of geometric parameters on the ejector performance using

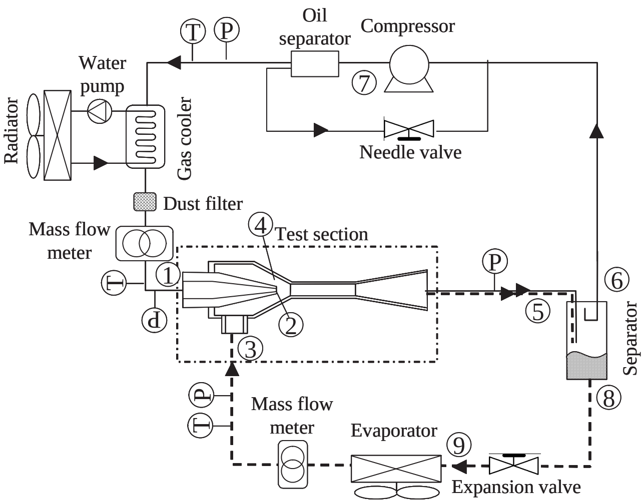

Fig.1: Schematic of the ejector cycle

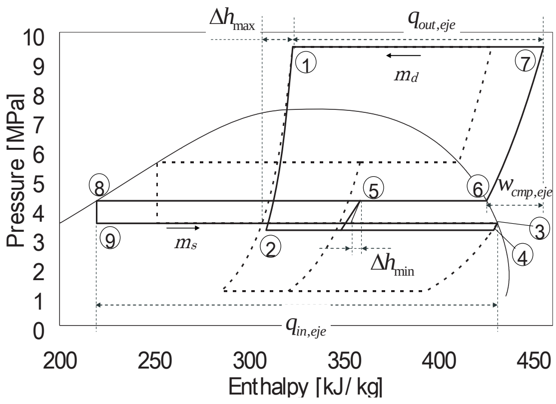

Fig.2: P-h diagram of the ejector cycle

R-12 refrigerant. It was found that the length of the mixing tube must be approximately 200 mm to sufficiently decelerate the liquid phase, which had a large amount of inertia under the following conditions: the driving mass flow rate was 240 kg/h; the mixing tube diameter, 9.5 mm; and the entrainment ratio, 0.2. Harrell [3] showed that when the quality at the inlet of the nozzle was 4 %, pressure recovery increased with the distance between the nozzle outlet and the constant-area mixing tube in an ejector equipped with a convergent mixing cone, constant-area mixing tube, and divergent diffuser. However, when an ejector equipped only with the convergent mixing cone and divergent diffuser was used, pressure recovery decreased by 9 %. Using this ejector, pressure recovery decreased with increasing distance between the nozzle outlet and the constant-area mixing tube. By considering the results of visualization and experiment, it was found that the liquid droplets from the driving nozzle did not enter the surrounding suction flow; however, they remained in the core region, unless recirculation occurred in the mixing section.

Jeong et al. [4] investigated the theoretical COP of the ejector-equipped heat pump cycle using ammonia and carbon dioxide as refrigerants. It was found that if the isentropic efficiencies of the driving nozzle, suction nozzle, and diffuser were 0.9, the COP improvement was 5 % for the ammonia cycle and 22 % for the carbon dioxide cycle. Ozaki et al. [5] compared the effectiveness of the ejector with an expander for the heat pump cycle using carbon dioxide as a refrigerant. It was found that the ejector could improve the COP by a two-stage compression process and recovery of the expansion loss. The maximum improvement in COP was 20 %, which was comparable with that of the expander-equipped heat pump cycle.

Disawas et al. [6] experimentally examined the improvement in COP using an R-134a refrigerant. It was found that the COP of the ejector cycle was higher than that of the conventional cycle for a wide range of experimental conditions, and the effectiveness increased as the temperature of the heat sink decreased. Li et al. [7] investigated the influence of the design parameters of the ejector using cycle simulation. The isentropic efficiencies of the driving nozzle and suction nozzle were determined from the experimental results according to the constant-pressure mixing model. As a result, the COP and cooling capacity increased by 11 % and 9.5 % respectively.

An experimental study has been carried out on the characteristics of the carbon dioxide ejector cycle in our laboratory. A previous report [8] revealed the relationship between the suction flow rate and the pressure drop throughout the suction flow passage. The influence of mixing section diameter and the expansion valve were examined; the optimum diameter of the mixing section was found to be between 2 mm and 2.5 mm. The tested ejector was found to enhance the COP of the cycle up to 11 %. In this report, more experimental results obtained with regard to pressure recovery in two-phase flow ejectors are discussed by focusing on the effect of the lengths of mixing section and nozzle outlet on pressure recovery.

2 OUTLINE OF EXPERIMENTS

Figure 3 shows a schematic diagram of a test ejector, in which the nozzle and mixing section can be changed. The pressure, temperature, and mass flow rate are measured to determine pressure recovery and ejector efficiency. The uncertainties of mass flow rate, pressure, and temperature are 0.2 %, 0.2 %, and 0.1 °C respectively.

In this paper, unless otherwise stated, the driving nozzle was a convergent nozzle with a throat diameter of 0.82 mm; the pressures at the nozzle inlet and entrance for suction flow were 9 MPa and 4.5 MPa, while the temperatures were 35 °C and 12 °C respectively. The distance between the nozzle outlet and the mixing tube inlet was 2.0 mm, and the expansion valve in the suction flow passage was fully opened.

In this experiment, three mixing sections with different mixing tube diameters (2.0, 2.5, and 3.6 mm) were examined to investigate the effect of the mixing tube diameter on pressure recovery. The length of the mixing section was varied within the range of 2–40 mm in order

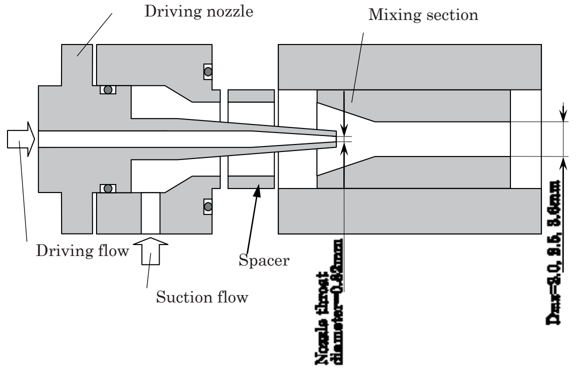

Fig.3: Experimental apparatus of the test section

to determine the optimum length of the mixing tube. Subsequently, the nozzle position was varied by replacing the spacer in order to compare the performances of the constant-area mixing ejector and constant-pressure mixing ejector for the same diameter of the mixing tube. In the constant-area mixing ejector, the nozzle outlet was set in the constant-area mixing tube. On the other hand, in the constant-pressure mixing ejector, the nozzle outlet was set in the convergent mixing cone; in addition, the pressure was assumed to be constant between the nozzle outlet and the inlet of the constant-area mixing tube. Finally, pressure recovery can be measured using the diffuser, and the ejector efficiency and COP improvement are estimated.

In this paper, the ejector efficiency η_{eje} is defined in Eq.3. $\Delta h_{\max}$ denotes the maximum enthalpy change assuming isentropic expansion from the nozzle inlet to the evaporation pressure. $\Delta h_{\min}$ denotes the minimum enthalpy change required to reach the condition of the ejector outlet from the evaporation pressure assuming isentropic compression.

$$\eta_{eje} = \frac{(m_d + m_s)\Delta h_{\min}}{m_d \Delta h_{\max}} \tag{3}$$

3 ONE-DIMENSIONAL SIMULATION MODEL OF EJECTOR

The velocities at the outlet of the driving nozzle and suction nozzle were calculated using Eq.4, where $\Delta h_{isentropic}$ denotes the enthalpy difference, assuming an isentropic process; η_{nzl} is the isentropic efficiency, defined as the ratio of the actual enthalpy change to $\Delta h_{isentropic}$.

$$u = \sqrt{2\Delta h_{isentropic} \cdot \eta_{nzl}} \tag{4}$$

For the mixing cone, the exchange of momentum and energy between the driving flow and the suction flow was neglected because of the small clearance of the nozzle outlet, and the condition of the mixing cone outlet was determined by the same method as that used to determine the condition of the nozzle. The isentropic efficiencies of the driving nozzle, suction nozzle, and mixing cone were assumed to be the same.

In the mixing tube, the outlet condition was calculated using Eqs.5–7, assuming that the sufficient mixing of the driving flow and suction flow takes place and the velocity is homogeneous at the mixing tube outlet. Equations 5–7 are the conservation laws of mass, momentum, and energy respectively. In Eq.6, Z_{mx} denotes the mixing tube length and f_w denotes the frictional force per unit volume obtained using Colburn's equation.

$$\frac{m_d}{\rho_d u_d} + \frac{m_s}{\rho_s u_s} = A_{mx} = \frac{m_d + m_s}{\rho_{mxo} u_{mxo}} \tag{5}$$

$$m_d u_{d,mxi} + m_s u_{s,mxi} + P_{mxi} A_{mx} + f_w A_{mx} Z_{mx} = (m_d + m_s) u_{mxo} + P_{mxo} A_{mx} \ ,$$

$$f_w = -\frac{4}{D_{mx}} \frac{\rho_{mxo} u_{mxo}^{\ 2}}{2} \frac{0.046}{Re_{mxo}^{\ 0.2}} \tag{6}$$

$$m_d \left(h_{d,mxi} + \frac{u_{d,mxi}^{\ 2}}{2} \right) + m_s \left(h_{s,mxi} + \frac{u_{s,mxi}^{\ 2}}{2} \right) = (m_d + m_s) \left(h_{mxo} + \frac{u_{mxo}^{\ 2}}{2} \right) \tag{7}$$

The outlet condition of the ejector without the diffuser was also obtained by using the conservation laws of mass, momentum, and energy on the basis of the assumption that the pressure at the section in which sudden expansion takes place just behind the mixing tube is constant, the frictional loss is negligible, and the velocity is homogeneous at the ejector outlet.

4 RESULTS AND DISCUSSION

Figures 4 and 5 show the variations in pressure recovery and suction flow rate against mixing tube length for three mixing tube diameters (2.0, 2.5, and 3.6 mm) in the case of ejectors without the diffuser. Figure 6 shows the relationship between the suction flow rate and pressure recovery, as obtained from the data shown in Figs. 4 and 5. Pressure recovery in the ejector is the same as the frictional loss along the suction flow passage, which is the flow path from the ejector outlet to the entrance for suction flow passing through the separator and evaporator. Figure 6 also represents the relation between the suction flow rate and the frictional loss along the suction flow passage.

In Figs. 4 and 5, the optimum mixing tube length is approximately 20 mm, regardless of the mixing tube diameter. When the mixing tube length is less than the optimum value, pressure recovery drastically decreases because the two streams exit from the mixing section before they mix sufficiently. On the other hand, in the case of a longer mixing tube, pressure recovery gradually decreases due to the frictional loss in the mixing tube.
A one-dimensional simulation was performed, and its results compared with the experimental results as shown in Fig. 7. The suction flow rate was calculated from pressure recovery using the relation shown in Fig. 6. From the comparison shown in Fig. 7, it was found the calculated results were in good agreement with the experimental results when the comparison is performed under a condition where sufficient mixing takes place ($Z_{mx} \geq 20$ mm) with isentropic efficiencies of the driving nozzle and the suction nozzle assumed in the range of 0.7–0.8. If the mixing tube length is smaller than 20 mm, the calculated results differ from the experimental results due to insufficient mixing.
It can also be seen in Figs. 4 and 5 that pressure recovery in the mixing section with a diameter of 3.6 mm is less than that at diameters of 2.0 and 2.5 mm. In the latter two cases, almost the same pressure recovery is obtained.
The effect of the mixing tube diameter was examined, as shown in Figs. 8 and 9. In Fig. 8, the downward-sloping curve represents the variations in pressure recovery with the suction flow rate calculated using the one-dimensional model under the following conditions: the mixing tube length was 40 mm and the isentropic efficiencies of the driving nozzle and suction nozzle were 0.8. The upward-sloping curve represents the relation between the suction flow rate and the frictional loss along the suction flow passage. The intersection of the downward-sloping curve and the upward-sloping curve indicates the predicted pressure recovery. Figure 9 shows the variation in the velocity at the inlet and outlet of the mixing tube with variation of the suction flow rate.
It can be seen from Fig. 8 that pressure recovery in the mixing tube reduces when the mixing tube diameter increases for a low suction flow rate. This is based on the conservation law of momentum (cf. Eq.6) as follows. From this equation, it is found that pressure recovery in the mixing tube is the difference between the kinetic momentum of the inlet and outlet of the mixing tube divided by the cross-sectional area of the mixing tube. This is the reason why the mixing section with a large diameter shows low pressure recovery.
From Fig. 9 it can be seen that a decrease in the mixing tube diameter or an increase in the suction flow rate results in the velocity of the suction flow increasing at the mixing tube inlet, and that the velocity of the mixed flow increases at the mixing tube outlet. There is therefore a decrease in the loss of kinetic momentum by deceleration of the driving flow; consequently, pressure recovery in the mixing tube reduces. This is the reason why the mixing tube with a

diameter of 2.0 mm shows almost the same pressure recovery as that of the mixing tube with a diameter of 2.5 mm.

Figure 10 shows the variations in pressure recovery with the nozzle outlet clearance. From this figure, it can be seen that pressure recovery reduces with an increase in the nozzle outlet clearance. This means that the constant-area mixing ejector is preferable to the constant-pressure mixing ejector for the same mixing tube. When the nozzle outlet clearance increases, the mixing of the driving flow with the suction flow is not negligible in the mixing cone. As mentioned above, when the loss of kinetic momentum occurs, pressure recovery for a smaller cross-sectional area is greater than that for a larger cross-sectional area. Since the mixing cone has a larger cross-sectional area than that of the mixing tube, the mixing in the mixing cone region — instead of in the mixing tube — leads to lower pressure recovery. This is the reason why the increase in the nozzle outlet clearance has a negative effect on pressure recovery. A detailed comparison between the constant-area mixing ejector and constant-pressure mixing ejector requires two-dimensional simulation of the mixing section, which will be presented in other studies.

In order to estimate the ejector efficiency and the improvement in COP by using the ejector with the diffuser, pressure recovery was measured using this ejector. The diameter and length of the mixing tube were 2 mm and 23 mm respectively, and the diffuser angle was 8°.

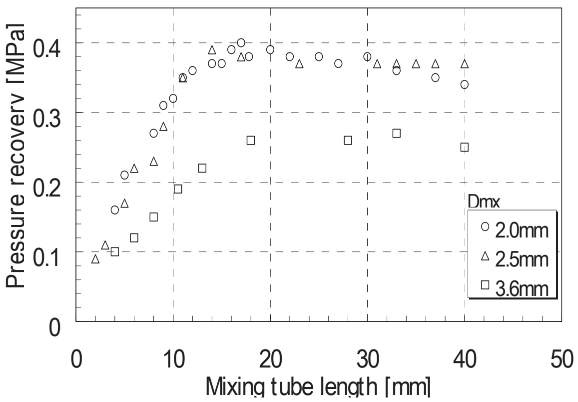

Fig.4: Influence of length of mixing tube on pressure recovery

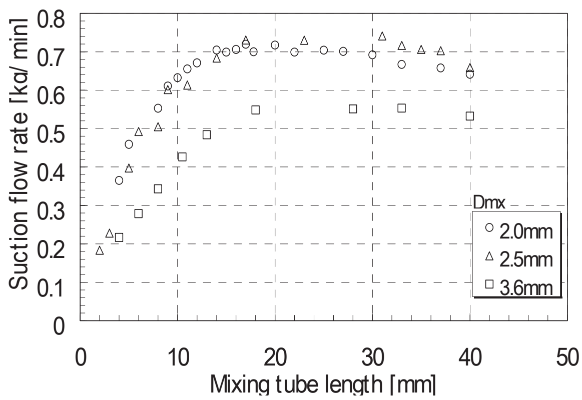

Fig.5: Influence of length of mixing tube on suction flow rate

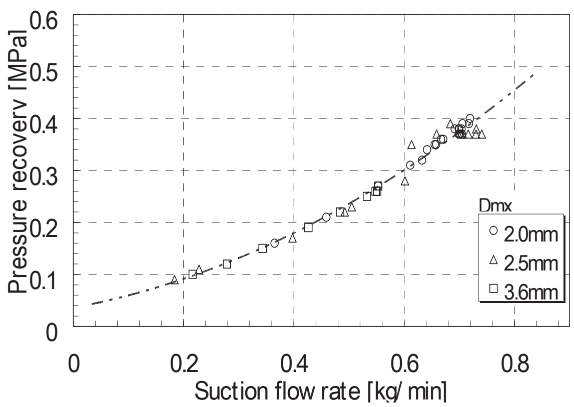

Fig.6: Relation between pressure recovery and suction flow rate with the expansion valve fully opened

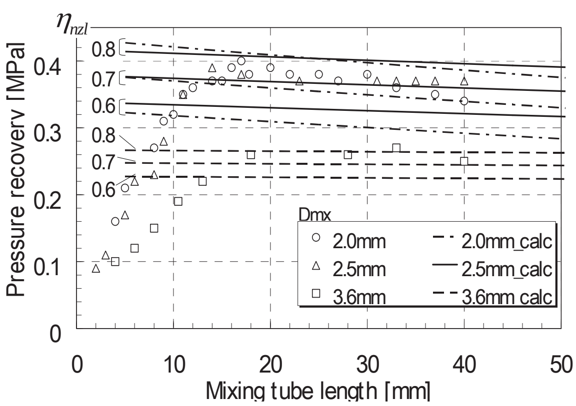

Fig.7: Comparison between experimental and calculated results

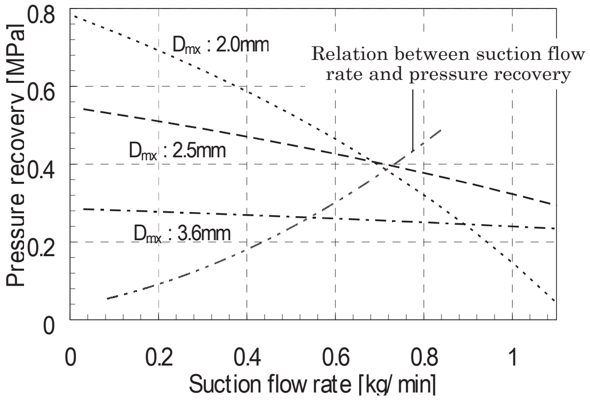

Fig.8: Effect of mixing tube diameter on pressure recovery

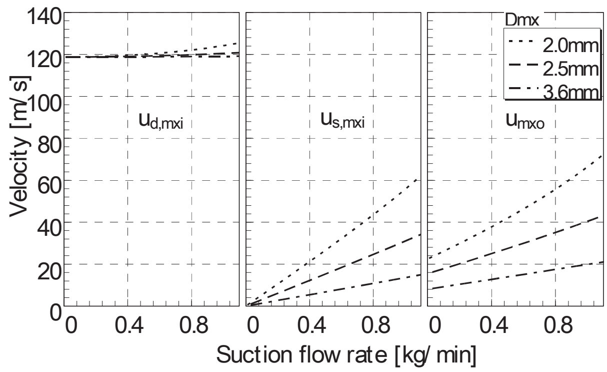

Fig.9: Effect of diameter of mixing tube on velocities at inlet and outlet of mixing tube

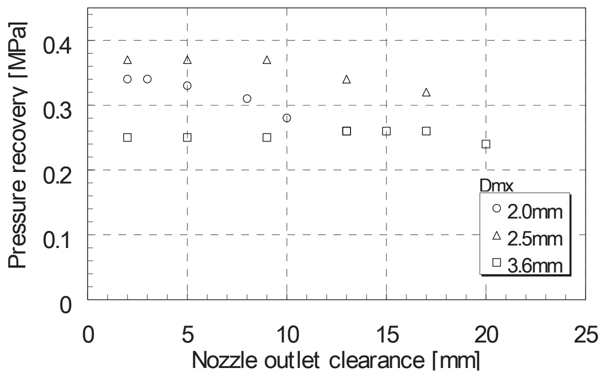

Fig.10: Effect of the nozzle position on pressure recovery

In this experiment, the expansion valve was adjusted in order to satisfy Eq. 1. Table 1 shows the experimental conditions and the results. The COP improvement is estimated on the basis of the assumption that the compressor efficiency is unity; the frictional loss is neglected in the cycle. As shown in this table, the efficiency of the present ejector is 0.46, which corresponds to a COP improvement of 23–24 %.

Table 1; Experimental conditions and results

Parameter	Value		
Experimental conditions			
Pressure at nozzle inlet [MPa]	8.3	9.5	10.5
Temperature at nozzle inlet [°C]	35	40	45
Pressure at entrance for suction flow [MPa]		4.5	
Temperature at entrance for suction flow [°C]		12	
Experimental result			
Driving flow rate [kg/min]	1.24	1.44	1.60
Suction flow rate [kg/min]	0.66	0.72	0.70
Entrainment ratio [-]	0.53	0.50	0.44
Pressure recovery [MPa]	0.57	0.75	0.96
Ejector efficiency [-]	0.46	0.46	0.46
Calculated COP of conventional cycle [-]	4.91	3.72	2.88
Calculated COP of ejector-equipped cycle [-]	6.04	4.61	3.57
Calculated COP improvement [%]	23.0	23.9	24.0

5 CONCLUSIONS

Experimental measurements were conducted to investigate the characteristics of pressure recovery in the two-phase flow ejector, and the following conclusions were obtained.

1. With a nozzle throat diameter of 0.82 mm, pressure recovery is achieved at a length of approximately 20 mm from the inlet of the mixing tube, irrespective of the mixing tube diameter. The experimental results satisfactorily fit the one-dimensional simulation results when the efficiencies of the driving nozzle and suction nozzle are assumed to be in the range of 0.7–0.8.
2. Pressure recovery in the mixing section with a diameter of 3.6 mm is lower than that at diameters of 2.0 and 2.5 mm; in the latter two conditions, almost the same pressure recovery is obtained.
3. The constant-area mixing ejector is preferable to the constant-pressure mixing ejector for the two-phase flow ejector applied to the carbon dioxide heat pump cycle.
4. The ejector efficiency and COP improvement obtained by using the carbon dioxide two-phase flow ejector with the diffuser are 0.46 and 23–24 % respectively.

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This article was not correctly printed in the 2008 IEA HPC Conference Proceedings CD. We therefore reprint this article in the newsletter.

CO_2 HEAT PUMP HEATING AND WATER HEATER SYSTEMS FOR COLD AREAS

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ABSTRACT: By incorporating a first-stage expansion intermediate-cooling cycle (gas injection) into a CO_2 natural refrigerant heat pump, we have managed to achieve a significant improvement in heating capacity and energy consumption efficiency under low ambient temperature conditions. By combining this technology with previously developed technologies for CO_2 heat pump application, including heater-less ice formation prevention technology, "direct heating" technology that provides direct heat exchange between the CO_2 refrigerant and heating antifreeze, and outlet hot water temperature transition control technology, we have developed a prototype CO_2 natural refrigerant heat pump heating and water heater system, which enables both heating and hot water to be delivered by a single system. This report describes the results obtained from performance evaluation tests conducted in our environmental test laboratory and from monitoring tests in a cold region (Hokkaido, Japan), and addresses the possibilities of this system for use in a heat pump system application.

Key Words: ***CO_2 Heat Pump, Heating, Water Heater, Cold Area***

1 INTRODUCTION

Stemming from the growing awareness of environmental concerns in recent years, the adoption of heat pump water heaters and other high-efficiency appliances has been increasing in Hokkaido, Japan's largest cold region. The number of CO_2 natural refrigerant heat pump water heater installations has also risen significantly, from 85 units in 2005 to 237 units in 2006. In addition, the number of new single-family house construction starts in Hokkaido stood at 17 800 units (in 2006), and factors such as the fact that some 50 % (8700 units) of these households have only electrically powered appliances in the kitchen and throughout the home, appear more to be an indication of a tailwind that is helping to boost the introduction of high-efficiency heat pump equipment. Nevertheless, when these figures are compared with the scale of the several hundred thousand units installed in regular-climate areas, the cold region market still remains a small one, and thus market penetration cannot be said to be moving very rapidly.

The consumption of energy (used for heating and cooling, hot water supply, lighting, and so on) per household in Hokkaido is approximately 1.4 times greater than that of the nationwide average. This consumption breaks down to about 50 % for heating and 20 % for hot water supply. In order to achieve energy savings, a two-pronged simultaneous approach to heating and hot water supply, with their high energy demands, is thought to be necessary. Unfortunately, almost all of the electrically powered high-efficiency appliances that have reached practical application to date are single-function types (HP water heaters and HP heaters), and no systems with integrated high-efficiency heating and water heaters, desired by consumers, have been supplied. This is believed to be the principal reason for the lack of growth in the number of units installed.

Acknowledging this situation, several years ago we began our research and development into systems that can supply hot water and provide heating at the same time, and to date have repeatedly assessed the performance of these systems and conducted validation tests (Kobayashi et al. 2002, 2004). We started with the development of a water heater and moved on to the development of technologies for maintaining compressor reliability and preventing ice formation among others, but assuring heating performance at low ambient temperatures has remained an issue at all times (Kuwabara et al. 2006). For some time, the injection system has been known as a technology that improves performance at low ambient temperatures (Goosmann and Zumbro 1928). Recent reports on related research have been increasing (Lambers et al. 2006) (Zha et al. 2006) (Cejka 2007). At this time, we have managed to obtain some specific results for the CO_2 natural refrigerant heat pump technology for cold regions in which the split cycle (gas injection) has been incorporated (Kobayashi et al.2006), (Kuwabara et al. 2007) (Itou 2007). As a result, it is now possible to significantly improve heating capacity and energy consumption efficiency at low ambient temperatures, and the possibility of practically applying this technology in heating and water heater systems has become more viable.

By combining these basic technologies with proven technologies developed for CO_2 heat pump application, including heater-less ice formation prevention technology, “direct heating” technology that provides direct heat exchange between the CO_2 refrigerant and heating antifreeze, and outlet hot water temperature transition control technology, we developed a prototype CO_2 heating and water heater system for cold regions and have been carrying out performance evaluation tests in our environmental test laboratory and monitoring tests in ordinary households. Below, using the data obtained so far, we discuss the current status of CO_2 heat pump heating and water heater systems for cold regions, and examine their future potential.

2 HEATING AND WATER HEATER SYSTEMS

2.1 Overall configuration

The entire system (see **Figure 1**) consists of two 6 kW CO_2 heat pump units (Unit A, used exclusively for heating, and Unit B for both space heating and water heating), a hot water tank unit (370 liter capacity) and a common central equipment unit. The target specifications call for a 6.0 kW hot water supply capacity and 8.0 kW heating capacity, as well as the ability to maintain these capacities, even at an ambient temperature of –20 °C, and to guarantee operation, even at an ambient temperature of –25 °C.

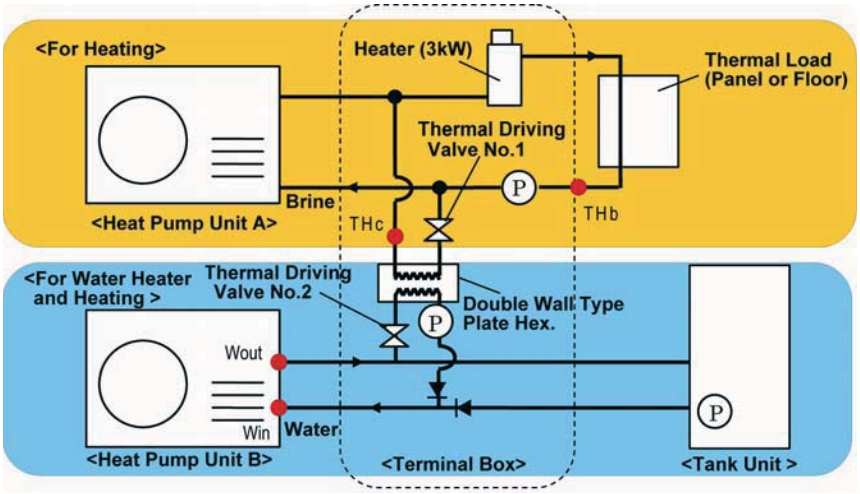

Figure 1: Heating and water heater system

2.2 Direct heating system

A system for ensuring the direct heat exchange between the CO_2 refrigerant and heating antifreeze is used in Heat Pump A, which is used exclusively for space heating. Unlike other space heating systems, which circulate the hot water in the tank through pipes in the house, this system has the advantage that it does not affect the amount of hot water supplied (which means that there are no concerns that the hot water will run out), and continuous heating operation is possible. For this reason, it can meet the demand for full heating loads, and enables floor heating and panel radiators to be operated at the same time.

2.3 Common central equipment unit

A double-wall type plate heat exchanger is contained in the common central equipment unit, allowing the heat from the water circuit on the hot water supply side to be transferred to the antifreeze circuit on the heating side. When water is not being heated for domestic hot water purposes, Heat Pump B can be used to assist the supply of heat from Heat Pump A. We have devised a system that, if the heating load is high and Heat Pump A (used exclusively for heating) cannot meet this load demand alone while domestic hot is being heated, provides support from a 3 kW supplementary heater.

2.4 Safety considerations

Taking into account the fact that the system would be installed in households in the Hokkaido area, we considered characteristics that would make it easy to install the system. For instance, we made the tank unit smaller than door dimensions (through which the tank unit can be carried indoors) and designed the common central equipment unit to be of such a size that it can be installed under the floor, beneath a storage cover. In addition, safety was considered by such means as incorporating a water leak detection device. (**Table 1**)

Table1; Components of space heating and domestic hot water heating system

Heat pump unit	Dimensions (H×W×D)	690×840×290 [mm]
	Weight	66 [kg]
Common central	Dimensions (H×W×D)	417×436×438 [mm]
	Weight	28 [kg]
	Heater	3 [kW]
Tank unit	Dimensions (H×W×D)	1800×650×740 [mm]
	Capacity	370 [L]
	Control	Full auto

3 COLD REGION SUPPORT FEATURES

3.1 Refrigeration circuit

One issue associated with CO_2 natural refrigerant heat pumps has been that when the return load temperature (cold water temperature) is high (40 ˚C or more), the gas cooler outlet temperature (point e) also rises, and energy consumption efficiency (COP) and capacity decline. A high return temperature is an obstacle to applying these heat pumps for space heating. For this reason, we implemented a countermeasure to reduce the gas cooler outlet temperature by adding an intermediate heat exchanger and adopting a single-stage expansion intermediate cooling cycle (**Figure 2**).Some of the gas cooler outlet refrigerant is shunted to the bypass circuit, passing through the second expansion device, and is set to the intermediate pressure, after which it exchanges heat with the refrigerant flowing through

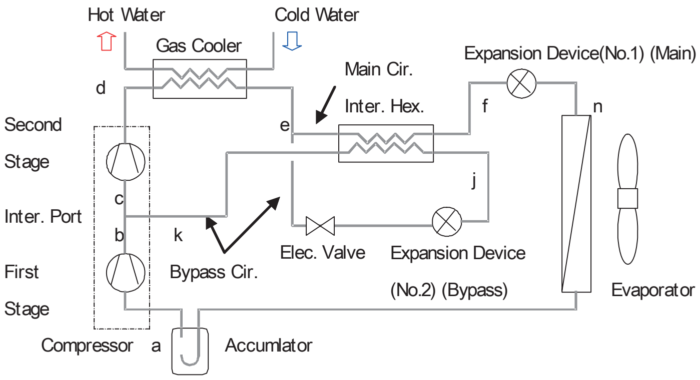

Figure 2: Diagram of heat pump unit

the main circuit, and returned to the intermediate port. The temperature of the refrigerant flowing through the main circuit drops and, since the shunted refrigerant is evaporated and returned, a gas injection circuit is formed. Since this circuit splits off at the gas cooler outlet, it is also called a "split cycle." Hereinafter it is referred to as the split cycle. Unlike a 2-stage expansion cycle, a separating device, such as a gas-liquid separator, is not needed. This circuit has a number of advantages: it is possible to reduce the evaporator inlet quality and enhance the refrigeration effect by cooling the gas cooler outlet high-pressure refrigerant in the intermediate heat exchanger; it is also possible to minimize the temperature rise of the second stage discharge gas by mixing low-temperature gas with the first-stage discharge gas.

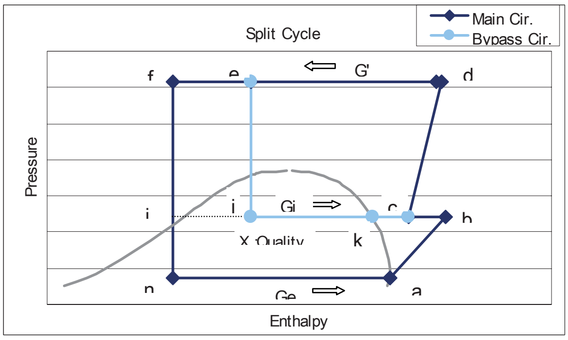

Figure 3: P-h diagram of split cycle

Figure 3 is a pressure vs. enthalpy diagram of the split cycle: here, P is the pressure, h the enthalpy, G' is the gas cooler refrigerant flow, Ge is the evaporator refrigerant flow, and Gi is the bypass circuit refrigerant flow. The superheat at the intermediate heat exchanger outlet point k and evaporator outlet point a for the gas refrigerant was assumed to be zero. In addition, X is the quality at the intermediate heat exchanger inlet point j, and point i was made the point at which middle pressure line j-c is extended and intersects with line f-n. Symbols a, b, c, d, e, f, n indicate the main circuit, and e, j, k indicate the bypass circuit. The gas cooler outlet refrigerant is cooled, and moves from point e to f. The enthalpy at evaporator inlet point n is reduced, and Δh (the enthalpy difference between point n and point a) increases. For the same capacity, therefore, refrigerant flow Ge through the evaporator can be lower than G' and the input can be reduced. Conversely, if refrigerant flow Ge through the evaporator is fixed, the capacity increases. In both cases, COP increases.

The effect of the split cycle is dependent upon quality X: the higher quality X is, the greater the effect of the split cycle. Also, the lower the low pressure, or the higher the gas cooler outlet temperature, the higher quality X will be, so in terms of the operating conditions, the lower the ambient temperature or the higher the return temperature from the heating load, the greater the effect will be.

3.2 Element technology

We developed an intermediate heat exchanger and a 2-stage compression rotary compressor that supports gas injection. (See **Table 2**.) We estimated the capacity of the intermediate heat exchanger while anticipating the highest bypass volume conditions, pursuing a compact construction, and determined its specifications. The compressor is a hermetic internal intermediate-pressure 2-stage rotary compressor, the same system as before, but since the gas refrigerant is injected into the intermediate pressure port via the intermediate heat exchanger, the intermediate pressure rises, and the balance between the high pressure and low pressure changes. As a result of simulations, in line with the new refrigeration circuit, we reset the displacement volume ratio to a point where it is very efficient (around 70 %). Some of the motor winding and valve specifications have been changed to allow the rotational speed to be increased for use in cold regions. Furthermore, in cold regions, there is always the concern of ice forming in or on the evaporator. We therefore incorporated an ice formation prevention coil at the very bottom on the outside of the evaporator, and adopted a construction to run the high-pressure refrigerant in order to prevent ice from forming. (Heater-less ice formation prevention technology)

3.3 Heating outlet water temperature control that supports CO_2 heat pumps

In exercising remote control over the heating, the heating water supply temperature can be set manually to 50 ˚, 60 ˚ or 70 ˚C, after which the control transfers to automatic temperature control. When the space heating return temperature rises to a certain value during a heating operation, this is interpreted as a reduction in the heating load, and the controller reduces the supply temperature setting in steps. Conversely, when the difference between the heating supply and return temperatures becomes greater, this is interpreted as an increase in the heating load, and the controller increases the temperature setting in steps. In this way, by incorporating a function that ascertains the increase or reduction in the heating load and adjusts the heating water supply temperature, system efficiency is improved by the automatic transition to a state in which the efficiency is always high.

Table 2; Specifications of intermediate heat exchanger and CO_2 compressor

Intermediate Hex.	Construction	Direct contact type (welded)
	CO_2 intermediate pressure	OD=6.35 [mm] t=1.2 [mm] (bare tube)
	CO_2 high pressure side	OD=6.35 [mm] t=1.2 [mm] (bare tube)
	Length	2.8[m]
Compressor	Mechanism	Rolling piston type two-stage compression
	Motor type	DC inverter-driven
	Nominal	1300 [W]
	Displacement volume	3.90 / 2.80 [cm^3] (Ratio: 70%)
	Dimensions	Outside diameter:118 [mm]、 height:217[mm]
	Weight	9.3 [kg]

4 SYSTEM EVALUATION

4.1 Independent heat pump performance tests

After connecting the heat pump units to the water supply system, we set the expansion devices to Manual and took measurements in a calorie test chamber. **Figures 4 and 5** show the results of the maximum heating capacity tests in the split cycle and conventional cycle. The test conditions were a space heating outlet water temperature of 55 ˚C and a return temperature of 40 ˚C.

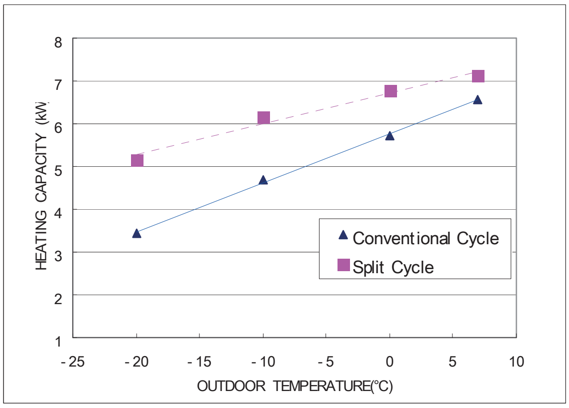

Figure 4: Heating Capacity of split and conventional cycle

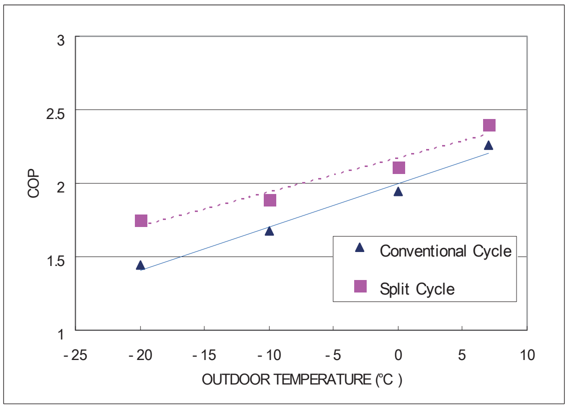

Figure 5: COP of split and conventional cycle

Compared with the conventional cycle, it is clear that the heating capacity in the split cycle increases significantly and that the COP also increases at the same time. Furthermore, the lower the ambient temperature, the higher the rate of increase in the heating capacity and COP. The tests we performed proved the effect to be great at a low ambient temperature.

4.2 Auxiliary heating capacity tests using Heat Pump B

It was possible to deliver a heating capacity of 5 to 7 kW using a single Heat Pump A, but the use of one Heat Pump A is not enough to meet the demands of the highest heating loads. Heat Pump B delivers additional heat for the highest heating loads. (However, when Heat Pump B is heating domestic hot water, the demands of the load are met by the supplementary heater inside the common central equipment unit instead.) We confirmed that when the flow rate of the primary heat medium (water) and the secondary heat medium (brine) of the double-wall type plate heat exchanger in the heating and water heater system shown in **Figure 1** are 2 liters/minute and 3 liters/minute respectively, it is possible to maintain a capacity of 3 kW with an outlet water temperature (W_{out}) of 75 ˚C and a return temperature (W_{in}) of 50 ˚C, and a capacity of 4.3 kW with an outlet water temperature of 75 ˚C and a return temperature of 40 ˚C.

4.3 Simulation results

We calculated the annual average COP while factoring in the switches between the operation modes such as the auxiliary heating operation by Heat Pump B. The conditions for the heating calculations included a hypothetical operation site in Sapporo City, a floor area of 140 square meters, and an insulation performance of 1.24 W/m^{2}K, and we used the SMASH residential building thermal load calculation program. (See **Table 3**.) By calculating the heating load from the weather data and comparing the performance characteristics for the heating load, we extracted the operation modes corresponding to the loads, calculated the total input and divided the total heating load to obtain the annual average COP. We also fixed the flow at 3 liters/minutes and performed this calculation both when the outlet water temperature is set to a constant 60 ˚C, regardless of the fluctuations in the heating load, and when it is switched in five steps from 40 ˚C to 60 ˚C in accordance with the load. For the hot water supply load, we simulated the IBEC-L mode, and calculated the annual average COP in the same way as for space heating. Based on field test data obtained in the past, the figures representing the drop in performance due to the defrosting cycle were also incorporated into the simulation, and consideration was given to bringing these figures into alignment with the actual equipment.

Table 4 shows the results of the simulation. When the space heating water supply temperature is controlled in five steps from 40 ˚ to 60 ˚C, improvements of about 42 % with Heat Pump A used exclusively for heating, of about 38% with the heating system, and of about 26 % for the overall system including domestic hot water heating, were obtained, compared to when the outlet water temperature is fixed at 60 ˚C. It can be understood that the effects of outlet water temperature control on the annual average COP are immense.

Table 3; Calculation conditions of annual average COP

Weather data	Sapporo City
Calculation program	SMASH
Floor area	140 m^2
Insulation efficiency	1.24 [W/m^2K] (Next Gen. Standard Region I)
Heating load (details)	11304 [kWh] (Unit A: 10271, Unit B: 615, Heater: 419)
Water heater load	6774[kWh]

Table 4; Calculation results of annual average COP at Sapporo city

COP	Heating unit (A)	Heating	Water heater	Total
Without outlet water temperature control (60˚C: constant)	1.36 (100)	1.33 (100)	2.44 (100)	1.60 (100)
With outlet water temperature control (40 - 60˚C: 5 stages)	1.93 (142)	1.83 (138)	2.44 (100)	2.02 (126)

5 MONITORING TESTS

5.1 Installation conditions

We installed the systems in ordinary households in Sapporo (**Table 3**) and Asahikawa, which are major districts in Hokkaido, and over a 1-year period, including the winter season, we conducted monitoring tests with a view to assessing system performance and assuring its reliability. An example of the installation conditions is shown in **Figure 6,** and an example of a system on which the monitoring tests were conducted is shown in **Figure 7**.

Figure 6: Heat pump units installed in Asahikawa

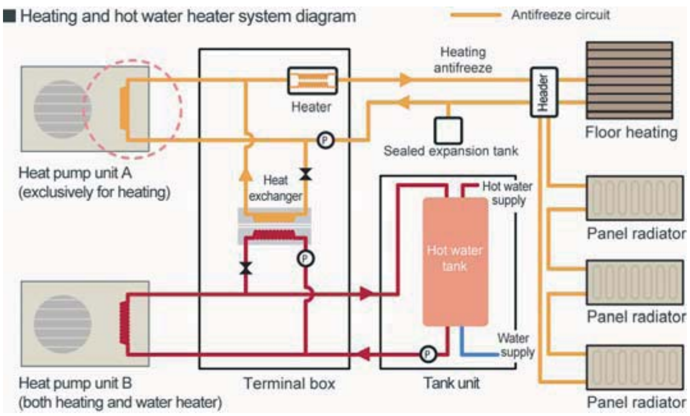

Figure 7: Total system of heating and water heater

5.2 Operation control

From the performance as shown in **Figure 8**, we verified that the water heater capacity of Heat Pump B, used both for space heating and as a water heater, was 5 to 6 kW at temperatures of –8 ˚C and below. Similarly, the performance shown in **Figure 9** verified the heating capacity of Heat Pump A, used exclusively for space heating, to be 4 to 5 kW at ambient temperatures of –15 ˚C and below. The system is designed in such a way that when the ambient temperature drops and the heating load increases to a point beyond the heating capacity of Heat Pump A, the auxiliary heating function starts. If, at this time, Heat Pump B is operating to supply hot water, the supplementary heater turns on first, and provides support by intermittent operation under ON/OFF control. When time passes and the hot water operation of Heat Pump B ends, the supplementary heater turns off, and operation switches over to auxiliary heating provided by Heat Pump B. **Figure 9**.a shows the operation of this sequence.

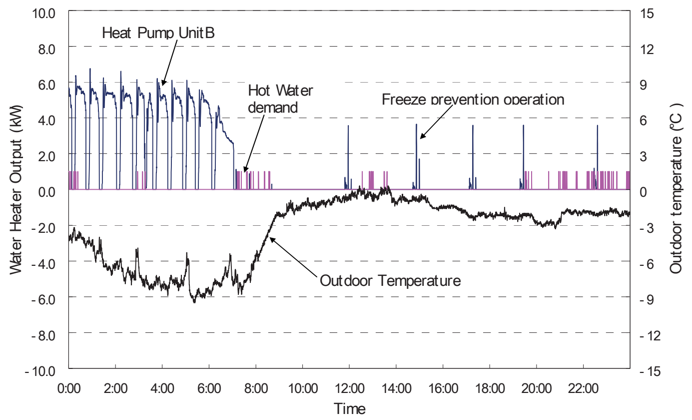

Figure 8: Water heater operation (Sapporo: 2007.Feb.8)

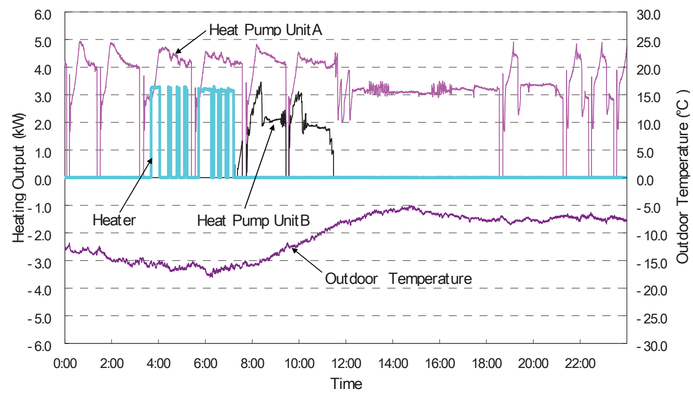

Figure 9: Heating operation (Asahikawa: 2007.Feb.10)

5.3 Room temperature control

The space heating outlet water temperature control works as follows. When the space heating system water return temperature rises to a certain extent, this is interpreted as a reduction in the heating load. The system then reduces the supply temperature in steps. Conversely, when the difference between the space heating supply and return water temperatures increases, this is interpreted as an increase in the heating load, to which the system responds by increasing the outlet water temperature setting in steps. It can be seen from **Figure 10** that the outlet water temperature also shifts slightly upward as the ambient temperature drops and the heating load increases. It can be seen that the system also responds, by reducing the outlet water temperature, to a rise in the room temperature above 20 ˚C.

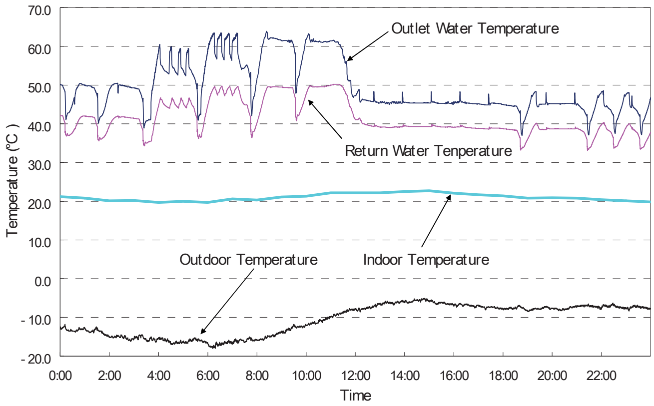

Figure 10: Room temperature control (Asahikawa: 2007.Feb.10)

5.4 Reliability

We performed the monitoring tests on a total of six units, and so far no problems have arisen in the main parts of the system. During these monitoring tests, we were also able to verify that the installation of protective covers on the evaporators in locations where wind and snow are severe is effective in preventing ice formation, and that the software that prevents ice from forming on water pipes during shutdown of the water heaters can be customized using the software for the water heaters currently destined for cold regions. In this way, based on the verification results obtained from both a hardware and software perspective of the system, we believe that we have generally achieved a level of reliability that is thought to be required for the practical application of the system.

6 CONCLUSION

Incorporating a first-stage expansion intermediate cooling cycle (a split cycle) in Heat Pump A, used exclusively for space heating, and Heat Pump B, used both for space heating and as a water heater, has made it possible to drastically improve performance at low ambient temperatures. While laying the foundations for its basic performance, we devised a space heating and water heater system consisting of two heat pump units and a 3 kW supplementary heater, and carried out verification tests on this system in ordinary households in major districts in Hokkaido. As a result, we ascertained that this system can more than adequately support the hot water supply and heating loads anticipated in cold regions. Although issues such as a review of a control system for the partial heating load change need to be resolved item by item in the future, we believe we have generally attained our objectives if we basically assess matters from a practical application standpoint of the system.

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Energy Statistics of OECD Countries, 2008 Edition
This volume contains data on energy supply and consumption in original units for coal, oil, gas, electricity, heat, renewables and waste. Complete data are available for 2005 and 2006 and for the first time in this edition, supply estimates are available for the year prior to publication (i.e. 2007). Historical tables summarise data on production, trade and final consumption. The book also includes definitions of products and flows and explanatory notes on the individual country data.
Source: www.iea.org

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This volume contains data on the supply and consumption of coal, oil, gas, electricity, heat, renewables and waste, presented as comprehensive energy balances expressed in million tonnes of oil equivalent. Complete data are available for 2005 and 2006 and for the first time in this edition, supply estimates are available for the year prior to publication (i.e. 2007). Historical tables summarise production, trade and final consumption data as well as key energy and economic indicators. The book also includes definitions of products and flows, explanatory notes on the individual country data and conversion factors from original units to energy units.
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Deploying Renewables - Principles for Effective Policies
Renewable energy can play a fundamental role in tackling climate change, environmental degradation and energy security. As these challenges have become ever more pressing, governments and markets are seeking innovative solutions. But what are the key factors that will determine the success of renewable energy policies? How can current policies be improved to encourage greater deployment of renewables? What impact can more effective policies have on renewables?
Source: www.iea.org

To be released on 12 November by the International Energy Agency:
World Energy Outlook 2008
Are world oil and gas supplies under threat? How could a new international accord on stabilising greenhouse-gas emissions affect global energy markets? World Energy Outlook 2008 answers these and other burning questions.
WEO-2008 draws on the experience of another turbulent year in energy markets to provide new energy projections to 2030, region-by-region and fuel-by-fuel. It incorporates the latest data and policies.
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Natural Gas Market Review 2008 - Optimising investments and ensuring security in a high-priced environment
Over the last 18 months, natural gas prices have continued to rise steadily in all IEA markets. What are the causes of this steady upward trend?
Unprecedented oil and coal prices, which have encouraged power generators to switch to gas, together with tight supplies, demand for gas in new markets and delayed investments, all played a role. Investment uncertainties, cost increases and delays remain major concerns in most gas markets and continue to constitute a threat to long-term security of supply.
A massive expansion in LNG production is expected in the short term to 2012, but the lag in LNG investment beyond 2012 is a concern for all gas users in both IEA and non-IEA markets. Despite this tight market context, regional markets continue on their way to globalisation. This tendency seems irreversible, and it affects even the most independent markets. Price linkages and other interactions between markets are becoming more pronounced.
The *Natural Gas Market Review 2008* addresses these major developments, assessing investment in natural gas projects (LNG, pipelines, upstream), escalating costs, the activities of international oil and gas companies, and gas demand in the power sector. In addition, the publication includes data and forecasts on OECD and non-OECD regions to 2015 and in-depth reviews of five OECD countries and regions, including the European Union.
It also provides analysis of 34 non-OECD countries in South America, the Middle East, Africa, and Asia, including a detailed assessment of the outlook for gas in Russia, as well as insights into new technologies to deliver gas to markets.
Source: www.iea.org

IEA Energy Policies Review - The European Union 2008
For the first time, the IEA has reviewed the energy policies of the European Union which shape the energy use of almost 500 million citizens in 27 EU member countries. A unique entity, governed under complex and almost constantly evolving structures, the EU constitutes a challenge for energy policy-makers. Its energy policy has a global impact, not only because of its 16 % share of world energy demand, but also because of EU leadership in addressing climate change.
Strong policy drives are under way in the EU to achieve the completion of the internal energy market, increase renewable energy supply, reduce CO2 emissions and make the EU more energy-efficient. Concerns about security of supply have also led to a greater focus on improved energy relations with supplier countries, and new institutional structures are being put in place. How much progress has been made in the field of security, internal market and external energy policies? And in which of these areas has the EU already implemented a fully integrated policy?
IEA Energy Policies Review
Source: www.iea.org

Energy Statistics of Non-OECD Countries - 2008 Edition
This volume contains data for 2005 and 2006 on energy supply and consumption in original units for coal, oil, gas, electricity, heat, renewables and waste for over 100 non-OECD countries. Historical tables summarise data on production, trade, final consumption and oil demand by product. The book includes defini-

tions of products and flows and explanatory notes on the individual country data.
Source: www.iea.org

Electricity Information 2008
Electricity Information provides a comprehensive review of historical and current market trends in the OECD electricity sector, including preliminary 2007 data. This reference document brings together essential statistics on electricity and heat. It therefore provides a strong foundation for policy and market analysis, which in turn can better inform the policy decision process toward selecting policy instruments best suited to meet domestic and international objectives.
Source: www.iea.org

Renewables Information 2008
Renewables Information provides a comprehensive review of historical and current market trends in OECD countries. This reference document brings together essential statistics on renewable and waste energy sources. It therefore provides a strong foundation for policy and market analysis, which in turn can better inform the policy decision process to select policy instruments best suited to meet domestic and international objectives.
Source: www.iea.org

New Publication on Natural Refrigerants
The replacement of R22 in commercial and industrial refrigeration and the air-conditioning sector poses a major challenge for developing countries when planning and implementing the accelerated HCFC phase-out. Natural refrigerants are the only available ozone- and climate-friendly alternatives to HCFCs, and should therefore be given preference over HFCs when looking for sustainable, long-term alternatives to R22.
This is why the federally owned Deutsche Gesellschaft für Technische Zusammenarbeit (GTZ) GmbH now has published a 384-page collection of articles, compiled to highlight the benefits of natural refrigerants. The purpose is to provide guidance to those involved in implementing the HCFC phase-out in developing countries: policy stakeholders, manufacturers of refrigeration and air-conditioning equipment, and end-users of R22 such as supermarket chains or owners of large (commercial) buildings with air-conditioning systems.
The publication is available for download on www2.gtz.de/dokumente/bib/gtz2008-0359en-natural-refrigerants.pdf

Load Calculation Software from Wrightsoft
Wrightsoft® announces the release of the new version of the Right-N® software module. The software enables users to perform calculations and complete HVAC system designs based on the Manual N heating and cooling load calculation method for small and mid-size commercial structures. It incorporates the procedures and applications of the newly revised Fifth Edition of ACCA's Manual N, the first official rewrite of Manual N in more than 20 years. It enables users to draw the floor plan of a project and have the Manual N worksheet automatically determine the load and generate standardized reports.

IEA HPP Annex 32 uses VIP+
IEA HPP Annex 32 uses VIP+ to evaluate different configurations of integrated multifunctional heat pump systems in low and ultra-low energy houses.
The project leader for the Swedish task, Svein Ruud, has chosen VIP+ for the analysis. With VIP+ parameters such as building physics, climate and performance of different heat pumps can easily be changed. VIP+ is validated according to the IEA-BESTEST and ASHREA-BESTEST.
Source: Structural Design Software in Europe AB
http://vip.strusoft.com/index.php?option=com_content&task=view&id=13&Itemid=44&lang=en

2008

17 - 18 November 3rd National GeoExchangeTM Business & Policy Forum
Toronto, Canada
www.geo-exchange.ca

19 - 21 November RENEXPO® South-East Europe 2008
International trade fair and conference for renewable energies and energy efficient building and renovation.
Bucharest, Romania
E-mail: redaktion @ energie-server.de

19 - 21 November Deutsche Kaelte-Klima-Tagung 2008
Ulm, Germany
www.dkv.org/index.php?id=94

20 November UK geothermal
London, UK
www.zenithinternational.com/events/event_details.asp?id=61

20 - 21 November International Symposium on New Refrigerants & Environmental Technology 2008
Kobe, Japan
www.jraia.or.jp/frameset_english.html

23 - 25 November China International Exhibition for Auto Air-conditioning & Transportation Refrigeration
Shanghai, China
www.autocoolexpo.com/en/zhgk.asp

1 - 12 December Conference of the Parties (COP) 14
City of Poznan, Poland
organised by the United Nations Framework Convention on Climate Change (UNFCCC)

2009

24 - 28 January ASHRAE Winter Meeting
Chicago, USA
www.ashrae.org

29 - 30 January European Meetingpoint - Energy Efficient Building and Renovation
Meeting within the CEP® 2009
Stuttgart, Germany
www.cep-expo.com

29 - 31 January CEP Clean Energy Power® 2009
International trade fair and conference for renewable energies & energy efficient building and renovation with 5th Innovation Conference
Stuttgart, Germany
E-mail: redaktion @ energie-server.de
www.cep-expo.com

29 - 31 January AHR Expo
Chicago, USA
www.ahrexpo.com

5 - 7 February MACS 2009 Convention and Trade Show
Dallas, USA
www.macsw.org/AM/Template.cfm?Section=Convention

24 - 26 February RAC 2009 Refrigeration & Air Conditioning Exhibition
Birmingham, UK
www.racexhibition.com

24 - 27 February Climatizacion (IFEMA)
International Air Conditioning, Heating, Ventilation and Refrigeration Exhibition
Madrid, Spain
www.ifema.es/ferias/climatizacion

2 - 3 March DENEX® 2009
Trade Fair and Conference for decentralised energy systems, bioenergy, and energy efficient building and renovation.
Wiesbaden, Germany
E-mail: redaktion @ energie-server.de
www.denex.info

10 - 12 March Cholod Expo
Moscow, Russia
www.cholodexpo.com

10 - 13 March Climateworld
Moscow, Russia
www.climatexpo.ru

10 - 14 March ISH Frankfurt
Frankfurt, Germany
ish.messefrankfurt.com/frankfurt/en/

16 - 19 March Cold Climate 2009
Sisimiut, Greenland
www.coldclimate2009.dk

More events can be found on www.heatpumpcentre.org

In the next Issue

Natural working fluids

Volume 26 - No. 4/2008

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 effi ciency 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 benefi ts 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.

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