IEA **Heat Pump** CENTRE

NEWSLETTER

Volume 26
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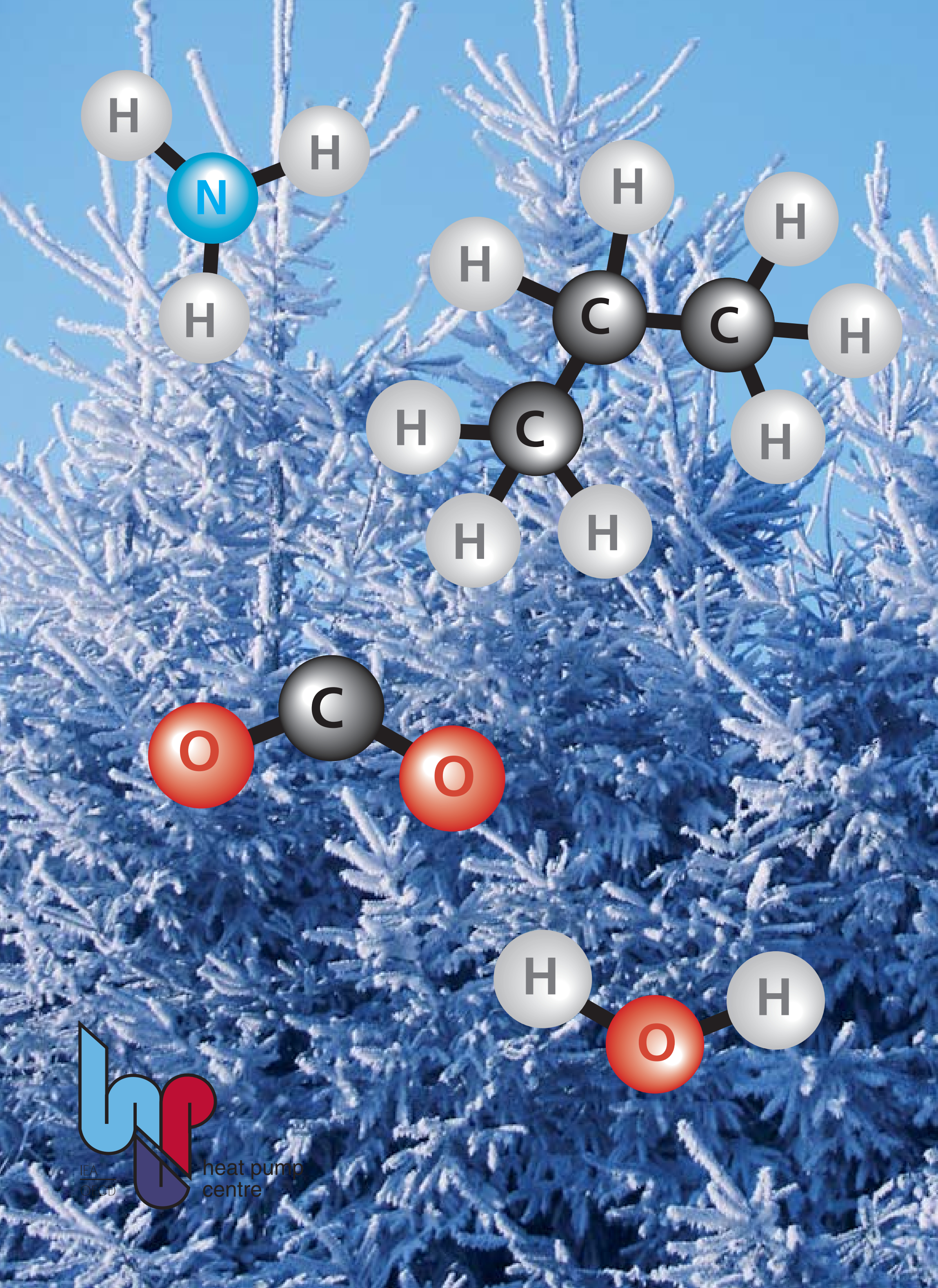
Natural Working Fluids



CO_2 supermarkets making progress in U.S. and Canada

Ammonia as refrigerant in small-capacity systems

Development of non-fluorinated energy-saving refrigeration and air conditioning systems in Japan

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In this issue

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Natural working fluids for heat pumps have had a renewed interest in the past 10 years, mainly through CO_2. In this issue we preent some of the technical improvements that have been made, and show a glimpse of what is going on. The issue contains articles covering not only CO_2, but also ammonia. A non-topical article describes also an exergetic analysis of air/water source heat pumps. Enjoy your reading.

Roger Nordman
Editor, HPC Newsletter

Heat pump news

Features

Topical article

Non-Topical article

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It's only natural...

Dr Andy Pearson,
Managing Director - Contracts,
Star Refrigeration Ltd

In recent years the factors which determine preference for different refrigerants in the minds of purchasers, specifiers, designers and manufacturers have been shifting like the coloured fragments in a kaleidoscope. These factors include technical and commercial considerations but also encompass political issues and, perhaps most elusive of all, public image. It is notable for example that it is almost impossible to buy a domestic refrigerator in Europe without hydrocarbon as the working fluid whereas in North America hydrocarbon fridges are virtually unknown. The recent announcement by GE that they have applied for EPA approval of a product using isobutane as refrigerant and cyclopentane in the insulating foam for the US market is a significant step in the ongoing rehabilitation of "old" refrigerants; those used prior to the introduction of fluorinated gases.

There is a host of reasons why these compounds fell out of favour in the early part of last century. Some are flammable, some are toxic and some operate at relatively high pressures, and the introduction of R-12 provided a substance better suited to the manufacturing and operating technology at that time. Now that we have new materials, new compressor designs and even new refrigeration cycles these old substances are being revisited and in many cases application of modern materials and techniques enables benefits to be gained that are simply not available with fluorinated gases.

So what is a "natural" refrigerant? This term is generally used to encompass substances which are found in nature and therefore do not pose any risk of chronic environmental damage. The "big five" natural refrigerants are ammonia, carbon dioxide, hydrocarbons, air and water. This definition must be used with caution: the ammonia and carbon dioxide used for refrigeration are byproducts of industrial processes, and they can both cause environmental damage if released carelessly. In fact all five natural refrigerants can harm the environment – principally through their use of energy to drive the refrigerating system. Any system which runs inefficiently or which leaks regularly places a burden on the environment and cannot be ignored. The real measure of the underlying attractiveness of these substances is that they create new possibilities in refrigeration, either in the operating envelope – heating to new heights or cooling to low temperatures more effectively - or in the efficiency of the system. All of the case studies highlighted in this edition have demonstrated this in one way or another: the real headline news about natural refrigerants is that they are enabling us not just to match the performance of the current crop of fluorocarbons, but to take refrigeration technology to the next level. This is why they cannot be ignored by specifiers, designers and users. Ultimately it will be technological advancement, not political pressure, marketing hyperbole or fashion consciousness that determines whether the "natural" wave is here to stay.

Column

Buildings Coordination Group

Hermann Halozan is the chair of the IEA Buildings Coordinated Group (BCG)

The International Energy Agency was founded as an autonomous body within the OECD with the goal of working for stability in the world energy markets. Already in 1980, just after the second oil price shock, the IEA Strategy Study came to the conclusion that buildings offered a possibility of reducing oil dependency in a relatively short time frame. In the framework of the IEA, several Building-Related Implementing Agreements (BRIAs) have been established, dealing with energy conservation in the building sector:

- The main IA in the area of buildings was and is Buildings and Community Systems (ECBCS) which, from its very beginning, has been dedicated to buildings and community systems, i.e. covering the requirements of the user (health, comfort etc.) in an energy-efficient and sustainable way.
- District Heating and Cooling (DHC) addresses the heating and cooling demand of communities. Its importance is growing with respect to rapidly growing urbanisation.
- Solar Heating and Cooling (SHC) deals with the application of solar thermal systems, appropriate for both space heating and domestic hot water production as well as for cooling.
- Energy Storage (ECES) was originally focused on large underground storage systems. Its focus now is the strategic and necessary component for the efficient utilization of renewable energy sources and energy conservation.
- Demand Side Management (DSM) is finding solutions to problems such as load management, energy efficiency, strategic conservation and related activities.
- Photovoltaic Power Systems (PVPS) is concentrated on electricity production directly from solar energy, covering grid-connected systems as well as stand-alone systems.
- Energy Efficient Electrical Equipment (4-E) deals with improvement of the efficiency of electrical equipment and appliances in the building sector.
- Heat Pumping Technologies (HPP) covers heating, cooling and air conditioning as well as refrigeration, i.e. all systems based on the second law of thermodynamics adding exergy to the energy source available such as air, groundwater, the ground, or waste heat to reach the temperature level required.

Initially, these Implementing Agreements worked on a technology basis, but during the last decades they became concerned with the larger system aspects. This required co-operation among the different IAs, and the first platform for such a co-operation was the Future Buildings Forum (FBF) initiated by ECBCS. Originally the FBF was a think tank for future Annexes (co-operative projects) of ECBCS, but soon it became, supported by the IEA Secretariat and its desk officers, a link to the other building-related Implementing Agreements.

To make these links more official and to coordinate the activities of the building-related IAs, the Buildings Coordination Group (BCG) was established by Egil Öfverholm, former Vice-Chair for Buildings in the Working Party of End-Use Energy Technologies (EUWP). The BCG is a platform for communication and personal contacts between building-related IAs, not only of the EUWP, but also of the Renewable Energy Working Party (REWP). But this platform is not limited to the IAs. There should also be active communication between the IAs and the IEA Secretariat, which provides a lot of information, which will ensure fruitful communication between the two sources to foster the developing and dissemination of information on end-use energy technologies.

Today, energy use in residential and commercial/public buildings accounts for about 35 % of global final energy consumption – approximately 1.179 Mtoe in the OECD countries; 221 Mtoe are used in the economies in transition; and 1.169 Mtoe in developing countries. The 2005 Gleneagles G8 Summit ended with an agreement on a new Dialogue on Climate Change, Clean Energy and Sustainable Development between G8 countries and other interested countries. It was also noted that, without the Plus-5 countries, such activities cannot be successful.

The rapid and continuous urbanisation creates challenges for the design of communities, but it also provides opportunities. Ageing of people and buildings in industrialised countries creates a need to adapt buildings to life cycles of the population and an increased need for renovation. Lack of skilled resources in industrialised countries necessitates product development towards more prefabricated and industrially produced buildings and systems. This in turn requires changes in education.

Rise living standards in developing countries offers on the one hand increasing business opportunities, but on the other hand new challenges for solving the scarcity of natural resources. The 80 % of the world's population now living in developing countries will soon require the same quality of life as the population of the industrialised countries. The urgent need to solve this equation in a sustainable way calls for close international cooperation. Although many solutions are universal, some specific solutions are needed in the very different circumstances of the developing countries.

There is increasing pressure to reduce significantly the energy consumption of communities and buildings. The industry seeks a new generation of highly efficient buildings with much lower energy use. The goal is to shift the emphasis from technology development to market-orientated development. Many technologies are ready for the market, but the market is not ready for the new technologies. This means a move from first-cost-based business towards life-cycle-performance-based business. This, however, does not mean departure from technological advances. Solutions need to fulfil ecological and economical demands as well as social acceptance and sustainability.

The direction towards near-zero primary energy use and carbon emissions in buildings and communities is clear. This calls for reducing heating, cooling and lighting loads to a minimum, utilising the exergy of renewable and waste energy sources as effectively as possible, and making fossil fuel use as effective and clean as possible. The IAs working together in the BCG have the potential to meet this challenge by covering the technologies required, as well as the systems and deployment activities. And they have the potential together with the IEA Secretariat to achieve solutions that can contribute to reduce the impact of climate change.

Hermann Halozan
Austria

General

President Obama calls for greater use of renewable energy

President Barack Obama's inaugural address called for the expanded use of renewable energy to meet the twin challenges of energy security and climate change. Noting that "... each day brings further evidence that the ways we use energy strengthen our adversaries and threaten our planet," President Obama looked to the near future, saying that as a nation, the United States will "... harness the sun and the winds and the soil to fuel our cars and run our factories." Those were the first references ever to the nation's energy use, to renewable resources, and to climate change in an inauguration speech of a U.S. president. President Obama later circled back to the subject of climate change, proclaiming that "... with old friends and former foes, we will work tirelessly to ... roll back the spectre of a warming planet." See the inaugural addresses of all the presidents, including President Obama's inaugural address, on the American Presidency Project Web site, an effort of the University of California, Santa Barbara.

As the President was being sworn in, the newly revised White House Web site went live, and it prominently features President Obama's agenda for energy and the environment. The President's "New Energy for America" plan calls for a federal investment of $150 billion over the next decade to catalyse private efforts to build a clean energy future. Specifically, the plan calls for renewable energy to supply 10 % of the nation's electricity by 2012, rising to 25 % by 2025. The plan also calls for deploying energy efficiency, including the weatherisation of one million homes each year. It also calls for an economy-wide cap-and-trade program to achieve an 80 % cut in greenhouse gas emissions by 2050. And to help meet that goal, the plan sets a target of placing one million plug-in hybrid cars on the road by 2015, along with a national standard to reduce the carbon emissions from our motor fuels. To help meet the plug-in hybrid goal, the plan calls for a new $ 7000 tax credit for those who purchase advanced vehicles. See the president's New Energy for America plan on the White House Web site.

The U.S. Senate also acted swiftly following the inauguration, and in a brief session the senators confirmed the nominations of Dr. Steven Chu as Secretary of Energy, Tom Vilsack as Secretary of Agriculture, and Ken Salazar as Secretary of Interior. In an Interior Department press release, Salazar noted the importance of federal lands to energy production, promising to help build a clean energy economy for the twenty-first century. He also called President Obama's energy imperative "our moon shot" for energy independence.
Source: EERE Network News

AHRI implements new metric (SI) policy for standards and guidelines

AHRI product sections and their engineering committees will introduce a new metric (SI) policy for standards and guidelines, with effect from Jan. 1 2009. The policy stipulates that when drafting new, revised or reaffirmed AHRI standards, each new document shall have only one set of units, not dual units as in the past. The product section must publish a metric SI-only standard that includes metric (SI) rational units for all testing and rating conditions, or it can publish two separate standards; one issued in metric (SI) rational units, and the other issued in inch-pound (I-P) rational units.

EU member states endorse Commission proposal to improve the energy performance of external power supplies

At the meeting of the Ecodesign Regulatory Committee on 17 October, EU Member States endorsed the European Commission's proposal for a regulation aimed at improving the energy performance of external power supplies. The regulation is expected to cut the EU's electricity consumption of these devices by 30 % by 2020, compared to a "business as usual" scenario.
Source: Energy and Transport in Europe Digest

AHRI joins consortium to advise DOE on high-performance green building issues

The Air-Conditioning, Heating, and Refrigeration Institute (AHRI), along with nine other organisations, has formed a consortium in response to a U.S. Department of Energy (DOE) request for advice on high-performance building issues. The High-Performance Commercial Green Building Partnership (HPCGBP) brings together leading organisations from all aspects of the building community to provide guidance and technical expertise on key sustainability issues to DOE's Building Technologies Program.

In addition to AHRI, members of the HPCGBP's steering committee include representatives from the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), the American Institute of Architects (AIA), the Alliance to Save Energy (ASE), the Building Owners and Managers Association (BOMA), the International Code Council (ICC), the Illuminating Engineering Society of North America (IESNA), the National Association of State Energy Officials (NASEO), the National Electrical Manufacturers Association (NEMA) and the U.S. Green Building Council (USGBC).
The partnership seeks to be recognised as a "Partnership Consortium" by the Department of Energy pursuant to Section 421 of the Energy Independence and Security Act of 2007. Section 421 is part of the formation of the Net-

Zero-Energy Commercial Building Initiative to develop a research, development and deployment strategy aimed at achieving net-zero-energy commercial buildings.

The partnership participants reflect all disciplines necessary to design and build high-performance commercial buildings, including:

Architects and engineers
Development, construction, financial and real estate industries
Building owners and operators
Academic and research organisations
Building code agencies and organisations
Independent high-performance green building associations or councils
Experts in indoor air quality and environmental factors
Experts in intelligent buildings and integrated building information systems
Utility energy efficiency programs
Manufacturers and providers of equipment
Public transportation industry experts
Non-governmental energy efficiency organisations

For more information, please visit www.hpcgbp.org.
Source: www.ahrinet.org

Refrigerants

GE plans isobutane refrigerators for U.S.

Appliance manufacturer GE has submitted a request to the U.S. Environmental Protection Agency (EPA) for approval to use isobutane as a refrigerant in household refrigerators. Upon gaining EPA approval, GE plans to include isobutane in a new GE Monogram® brand refrigerator in development for introduction in early 2010. The refrigerator would use cyclopentane, another hydrocarbon, as the insulation foam-blowing agent to replace commonly used HFC foam-blowing agents. According to the company, isobutane has been widely used in household refrigerators in Europe and parts of Asia for several years. EPA approval would allow, for the first time, use of a hydrocarbon refrigerant as a substitute for HCFC-based refrigerants now widely used in the U.S.
Source: The HVAC&R Industry

Technology

Nanoparticle research could bring energy savings for chillers

According to the National Institute of Standards and Technology (NIST), research that it has conducted into the use of varying concentrations of nanoparticle additives indicates an opportunity to gain major energy efficiency improvements in large industrial, commercial, and institutional chiller systems. These systems account for about 13 % of the power consumed by the nation's buildings, and about 9 % of the overall demand for electric power, according to the U.S. Department of Energy.

NIST researcher Mark Kedzierski has found that dispersing "sufficient" amounts of copper oxide particles (30 nanometers in diameter) in a common polyester lubricant and combining it with R-134a refrigerant improves heat transfer by between 50 % and 275 %. "We were astounded," Kedzierski said.
Source: www.achrnews.com

AHRI, ASERCOM, CRAA work to harmonise efforts for CO_2 standard

AHRI, the Association of European Refrigeration Compressor and Controls Manufacturers (ASERCOM) and the China Refrigeration and Air-Conditioning Industry Association (CRAA) have formed a partnership to harmonise technical standards in the United States, China and Europe. The group's first effort will be to work with AHRI's Compressors and Condensing Units Engineering Committee to develop AHRI Standard 570P, Performance Rating of Positive Displacement Carbon Dioxide Compressors and Compressor Units. Currently, there are no technical performance standards for carbon dioxide compressor technology. Jointly drafting this standard is another step toward global harmonisation, providing manufacturers with a single testing procedure to rate their equipment's performance. While the standard will be the same in China and Europe, the name of the standard will reflect the recognised standards in the country or region. For example, in China, it will be a GB/T Standard and in Europe, a CEN Standard. A panel of experts has been assembled to lead a tri-party harmonisation effort to review draft rating conditions for carbon dioxide compressors. The next tri-party meeting will be held in conjunction with the China Refrigeration Show in April 2009.
Source: www.ahriet.org

DOE says 50 % milestone is reachable toward zero-energy buildings

The U.S. Department of Energy (DOE) and the National Renewable Energy Laboratory (NREL) have released the first technical support documents to show that 50 % energy savings are achievable in commercial retail buildings toward its goal of zero-energy buildings. The two reports provide recommendations on how to achieve 50 % energy savings over American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 90.1-2004 in grocery stores and medium-sized retail buildings.
Source: ACHR News

DOE launches zero-net-energy commercial building initiative

The U.S. Department of Energy (DOE) recently launched the Zero-Net-Energy Commercial Building Initiative (CBI) with the objective of developing, by 2025, new commercial buildings that produce as much energy as they use. Energy-efficiency technologies and on-site renewable energy generation systems, including solar power and geothermal energy, will help make this possible. DOE has also formed the National Laboratory Collaborative on Building Technologies, which will allow the department and five of its national laboratories to work together on the research, validation, and commercialisation priorities that are important to achieving zero-net-energy buildings.

"DOE's Commercial Building Initiative and the Collaborative are urgently needed to accelerate innovation and market adoption in the field of high-performance buildings," says DOE Deputy Assistant Secretary for Energy Efficiency, David Rodgers. "Now we are bringing to bear the unprecedented collaboration in scientific resources of five national laboratories to bring about the needed transformation of the built environment, lower our carbon footprint in buildings and accelerate commercial deployment of clean, efficient building technologies."

To learn more about DOE's Energy Efficiency and Renewable Energy programs in buildings, visit *http://buildings.energy.gov.*
Source: www.ahrinet.org

Florida man invents eco-friendly air system

Keeping your home comfortably cool at night while you sleep is a big waste of energy and money. But a Broward man has designed an air conditioner that cools only where you need it most at night, so you can sleep easily while cutting your power bill

The unit is built into a piece of furniture at the foot of the bed, so it looks like part of the room — no wires or bulky machine. It blows a steady stream of cool air over sleeping bodies to keep them cool, so you can leave your main home air conditioning at a cost-saving higher temperature all night.

Market

EU expands definition of renewable energy

On 17 December 2008, the European Parliament adopted the eagerly awaited EU Directive on the Promotion of Renewable Energy Sources. The final compromise text recognises for the first time aerothermal and hydrothermal energy as sources of renewable energy under EU law.

Under the new legislation, each member state should increase its share of renewable energies to 20 % by 2020.

The European Partnership for Energy and the Environment (EPEE), the voice of the heating, cooling and refrigeration industry in Europe, is delighted that the use of aerothermal energy (energy in the air) and hydrothermal energy (energy stored in water) – both to be utilised by heat pumps - will now be promoted as part of the new EU policy on renewable energy sources.

Frans Hoorelbeke, Chairman of EPEE, commented:
"The potential of aerothermal and hydrothermal energy sources is enormous and can greatly contribute to Member States' success in reaching their renewable energy targets of 20 % by 2020."

Geothermal, aerothermal and hydrothermal energy is currently exploited using heat pumps, which were recognised as such "renewable energy technologies" by the EU, alongside wind turbines and solar panels.

So far, heat pumps have the widest prevalence in Sweden, but selling rates are expanding all over Europe.

Heat pump shipments are up, but overall unitary shipments decline

Heat pump shipments for October totaled 123,170, up 5 % from the same month a year ago, according to statistics from the Air Conditioning, Heating, and Refrigeration Institute (AHRI). However, combined U.S. factory shipments of central air conditioners and air-source heat pumps for October totaled 325 787, down 17 % compared with the same month a year ago.
Source: ACHR News

Residential water heaters can now carry the Energy Star label

DOE has announced that as of January 1, Energy Star-approved residential water heaters are now available. Five types of residential water heaters will be allowed to carry the Energy Star label: high-efficiency, gas-fueled, storage water heaters; gas-fueled condensing water heaters; whole-home, gas-fueled, tankless water heaters; heat pump water heaters; and solar water heaters. After space heating and cooling, water heating is the second largest energy expense in U.S. homes and represents up to 15.5 % of all national residential energy consumption. Energy Star-approved water heaters can reduce residential water heating bills from 7.5 % - 55 %. Over the next five years, the new water heater criteria are expected to save U.S. consumers $823 million in utility costs, avoid 4.2 million tons of carbon dioxide emissions, and achieve cumulative energy savings of more than 3.9 billion kilowatt-hours of electricity and 270 million therms (7.9 TWh) of natural gas, enough energy to power more than 375 000 homes for a year.
Source: EERE Network News

CGC to incorporate direct expansion technology in national quality program

The Canadian GeoExchange Coalition (CGC) today announced that as a result of its work since 2006, direct expansion (DX) technology will be fully included in Canada's national geo-exchange quality program beginning on May 1, 2009. Since late 2006, CGC has worked with DX manufacturers, distributors and installers to define the technology's particular advantages challenges

and requirements; has collaborated with the Canadian Standards Association (CSA) to engage a targeted revision of the current national standard, C-448; has consulted internationally with general industry and DX stakeholders; has prepared a recommendation for DX technology to be considered by the standards writing committee; has successfully gained utility and government funding for standard redevelopment; and is participating on the committee through the CGC members so that the revision will meet the needs of all stakeholders. At the same time, CGC has worked with DX manufacturers to develop installation guidelines. Based on those guidelines, CGC is in the process of developing a full DX Installer's course, to begin in April 2009.
Source: www.geo-exchange.ca

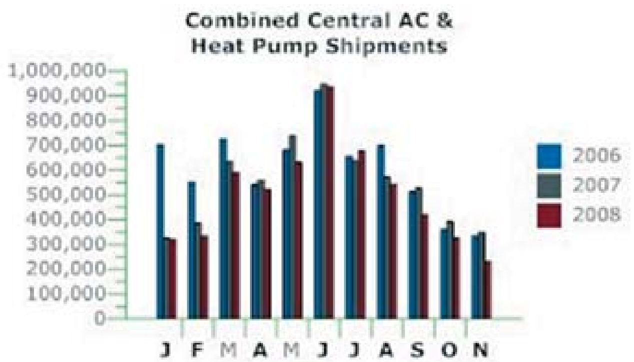
EU announces next eco-design product priorities

The European Commission has unveiled a list of ten priority energy-using product groups for which it wants energy-efficiency standards to be established in the next three years.

According to the Commission's plans, adopted on 21 October, the product groups under investigation will be included in the EU's 2005 Eco-design Directive, which defines binding minimum standards for energy performance. The final list includes such product groups as air-conditioning and ventilating systems as well as food preparation and refrigeration equipment.
This is the second batch of product groups to be selected. A first installment of 19, including heating equipment, lighting, domestic appliances and electric motors, was selected for energy-efficiency standards during the transitional phase following the directive's adoption in July 2005. Eco-design standards are expected to be finalised for five of these by 2009.
Source: www.euractiv.com

November heating and cooling equipment shipment data released

Central Air Conditioners and Air-Source Heat Pumps
Combined U.S. factory shipments of central air conditioners and air-source heat pumps for November totaled 231 995, down 33 % compared with the same month a year ago. For the year to date, combined shipments totaled 5 595 194, an 8 % drop compared with the same period last year.



Heat pump shipments for November totaled 95 506, down 21 % from the same month a year ago. For the year to date, heat pump shipments totaled 1 768 813, a 0.8 % drop compared with the same period last year.

Ongoing Annexes

Bold text indicates Operating Agent.

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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.

CO_2 Supermarkets Making Progress in U.S. and Canada

David Hinde, USA

Implementation of CO_2 in North American Supermarkets is in its beginning stages. Since 2006, several systems utilizing CO_2 as a low-temperature two-phase secondary coolant have been installed and started in various supermarket sizes and formats, with several more installations in progress, and more recently, the first CO_2 cascade system was installed. Significant practical information has been gained through start-up and operation of these systems relating to performance, maintenance, safety, energy consumption, and refrigerant charge. Customer experiences are presented, with details given for a few of the more publicized installations.

Introduction

As of 2008 new supermarket refrigeration systems are primarily designed for use with HFCs, most often using R-404A or R-507. However, data from the U.S. Environmental Protection Agency (EPA) indicates that 64% of our installed base of operating refrigeration systems in the U.S. still use R-22 as the working fluid, and recent surveys indicate the average leakage rate of refrigerant from existing stores is still fairly high at 23.5% per year. Not having been subjected to the pressures of HFC bans or high taxation of synthetic refrigerants as in other parts of the world, system choices here have been driven primarily by energy consumption and system first cost. A recent trend has been established towards equipment with lower refrigerant charge and lower leakage rates. This has for the most part been voluntary, driven by corporate initiatives focused on the increasing awareness of the effects of greenhouse gas emissions on the environment and climate change. Government regulations have been slow to respond to these concerns and maximum leakage rates are still not yet mandated for systems operating with HFC refrigerants, though this is expected to change soon. More recently, the EPA GreenChill Partnership[1], a voluntary government cooperative alliance, has been encouraging low-charge, low leakrate system installations, and has been met with encouraging participation by the industry.

When the reduction of refrigerant charge is desired, two systems are generally considered: distributed direct expansion systems, and secondary coolant systems. Distributed direct expansion systems have been applied successfully and are available from a variety of manufacturers in various forms. Some moderate decreases in refrigerant charge can be achieved; however leakage rates remain a concern due to the increased number of components, connections, and still large amount of refrigerant-containing piping.

Secondary coolant systems offer another alternative which can provide more significant reductions in both charge and leakage rate. Introduced in the U.S. in 1996, medium-temperature secondary coolant systems using propylene glycol are now gaining wider acceptance and represent a significant portion of business with over 500 systems installed, primarily in the U.S. but with a small percentage of installations in Canada and Mexico. Today, these systems are primarily applied not only for the benefits of HFC charge reduction, but enhanced product quality and decreased system maintenance. Concerns about increased energy consumption have largely been overcome through proper design practice.

Low-temperature secondary systems were introduced in the late 1990's using various potassium-based salts. Dozens of systems were installed by multiple manufacturers but some difficulties were experienced with leakage of the secondary fluid and resulting corrosion of surrounding materials. Although the potassium-based fluids exhibit superior performance to other single-phase fluids, material compatibility continues to be a concern. Use of the potassium salts has slowed dramatically in the last several years with only a handful of new installations. However recent introduction of cost-effective plastic piping materials and components may allow these systems to become a viable alternative for some applications in the future.

In searching for alternative fluids suitable for low-temperature application, it became clear that CO_2 as a two-phase secondary coolant showed several advantages compared to the single-phase salts. Primarily these were lower pumping power, smaller pipe sizes, excellent heat transfer

[1] For more information on the EPA GreenChill Advanced Refrigeration Partnership, please see www.epa.gov/greenchill/

properties, and good material compatibility with the additional benefit of the low cost of the fluid.

A third option in charge reduction, using CO_2 as a direct expansion cascade refrigerant, also shows promise as a cost effective and efficient alternative to CO_2 secondary systems and continues to be evaluated today.

CO_2 SUPERMARKET INSTALLATIONS IN NORTH AMERICA

In 2001, laboratory testing of low-temperature CO_2 secondary systems was initiated. After extensive investigation of the system operation, display-case and unit-cooler performance, and various piping configurations and control methods, the first U.S. system was installed in the field in mid-2006.

As of late-2008 at least 10 low-temperature carbon dioxide systems have been installed in the U.S. and Canada. Nine of the systems utilized CO_2 as a low-temperature two-phase secondary fluid with stores ranging in size from small markets to large supercenters and warehouse-style stores. The nine systems also included a primary refrigeration system using an HFC (R-404A or R-507). All but one installation included a medium-temperature secondary coolant system using propylene glycol.

CO_2 SECONDARY COOLANT INSTALLATIONS

Wal-Mart introduced the use of CO_2 secondary systems for the low-temperature refrigeration loads in their Sam's Club warehouse-style stores, first at a location in Savannah, Georgia, and later at a site in Fayetteville, Arkansas. Figure 1 shows the interior of the mechanical center at one of the two nearly-identical sites where 110 kW (375 kBtu/Hr) of refrigeration is provided by the CO_2 secondary system. In both sites, the medium-temperature systems use propylene glycol secondary systems in ABS plastic piping, which has proven to both speed up installation time and eliminate a significant portion of increasingly expensive copper piping. An energy savings of 2-3% has been reported for the low-temperature system compared to similar formats with DX HFC systems.

Figure 1. CO_2 Secondary Installation for a Wal-Mart Warehouse-Style Store

Figure 2. CO_2 Secondary Coolant System Installation at Food Lion

Food Lion introduced CO_2 secondary systems to the first supermarket-format stores in Montpelier and Portsmouth, Virginia. The first site in Montpelier was installed with a DX HFC system for the medium-temperature loads in order to isolate the energy impacts of the CO_2 system. At around 60 kW (200 kBtu/Hr) low-temperature load, comparison with other similar sized stores in the vicinity showed a small average energy savings associated with the CO_2 system. The second site in

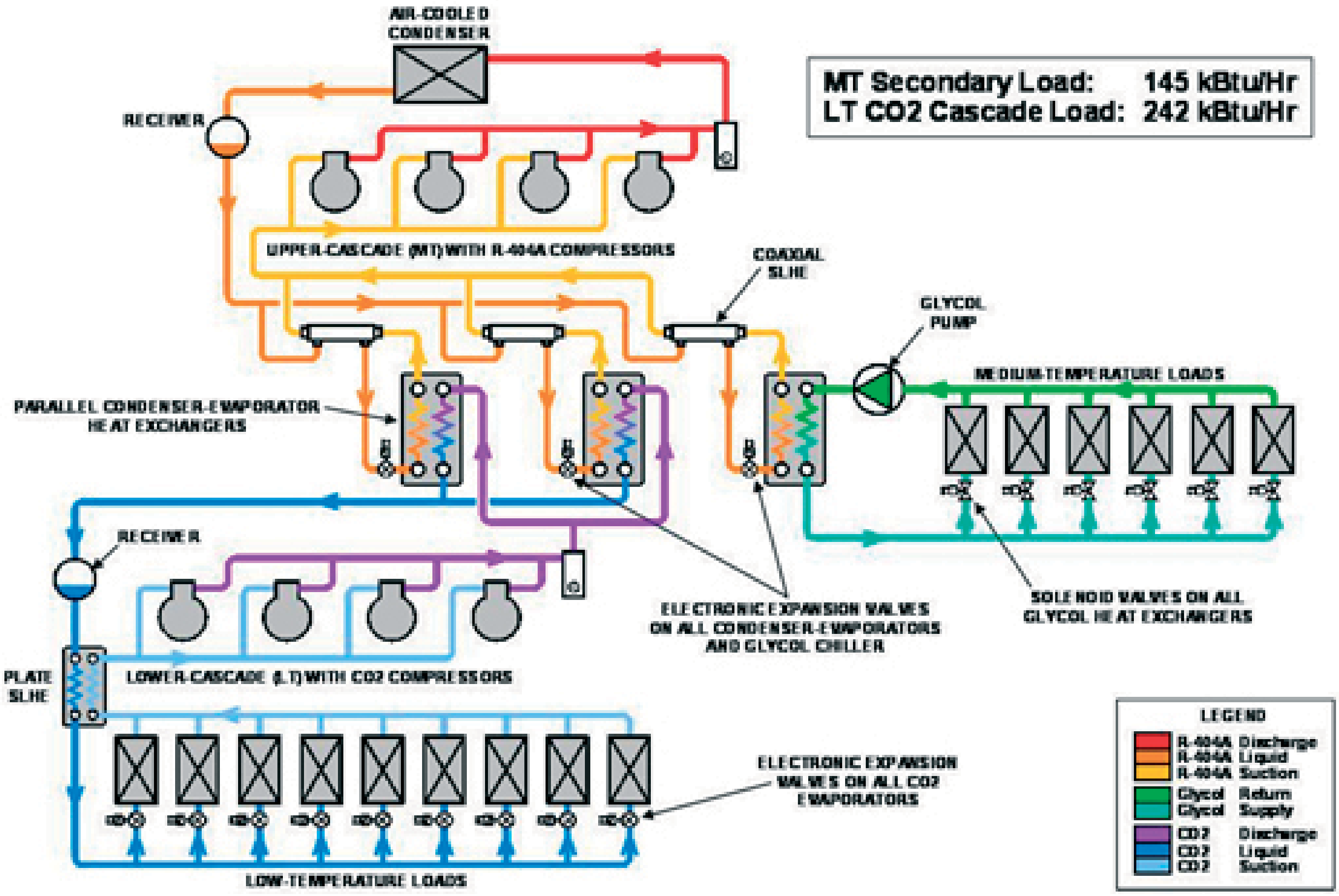

Figure 3. CO_2 Cascade System Installation in Price Chopper

Portsmouth, of similar size, included a medium-temperature propylene glycol secondary system also installed with ABS plastic piping. The CO_2 system, shown in Figure 2, is a typical arrangement with multiple condenser-evaporator heat exchangers above a common liquid-vapor separator.

The largest system installed in North America is in a Loblaw's site in Scarborough, Ontario and features a low-temperature CO_2 secondary system with a capacity of 160 kW (550 kBtu/Hr). Also featuring a medium-temperature propylene glycol secondary system, the store achieved an 85% reduction in HFC charge by using plate heat exchanger water-cooled condensers. The condenser-water system was uniquely integrated with the stores HVAC system to increase heat reclaim efficiency.

CO_2 CASCADE INSTALLATIONS

In early-2008 the first CO_2 cascade system was installed and started in a Price Chopper supermarket in Saratoga, New York. This system utilized CO_2 as low-temperature direct expansion refrigerant with a system load of 70 kW (240kBtu/Hr). As shown in Figure 3, the CO_2 was condensed by an R-404A upper-cascade system which also chilled propylene glycol for about 35 kW (135 kBtu/Hr) of medium-temperature loads. Reciprocating compressors were chosen for the CO_2 cascade system (Figure 4) due to compressor availability; though more recently scroll compressors have become available to the U.S. market.

Questions still remain regarding system cost compared to today's HFC direct expansion systems and low-temperature CO_2 secondary systems, and laboratory investigations will continue to optimize energy consumption and determine how to best commercialize this technology.

SYSTEM EXPERIENCES

Higher pressures and availability of components has not proven to be problematic. Concurrent to the introduction of these systems, significant introduction of components suitable for application with R-410A, increasingly becoming the domestic A/C refrigerant of choice for replacement of R-22, made the majority of these components readily available.

Successful installation of the CO_2 secondary coolant and cascade systems has relied heavily on comprehensive contractor training programs developed specifically for these applications and will continue to be critical to the implementation of these and other types of CO_2 systems in the future.

The selection of CO_2 grade or purity-level has been carefully considered. Initial systems used CO_2 gas of 99.99% purity (Coleman- or Instrument-Grade) however some systems have started using 99.5% industrial-

Figure 4. CO_2 Cascade Compressor

grade materials when better grades are not readily available. Charging the CO_2 through liquid filter-driers and a purge of non-condensable gases during start-up seems to make specific requirements unnecessary. CO_2 costs and purity-level availability appear to vary widely throughout the country, however most installations have been able to obtain the materials around 1.10 $US/kg (0.50 $US/lb).

CO_2 systems are susceptible to the same types of mistakes that can plague any field-installed refrigeration system – problems have included contractors not evacuating 100% of the piping network, and charging of impure refrigerant. CO_2 cylinders are not subject to the same clear labeling requirements as the synthetic refrigerants, and contaminated or improperly labeled cylinders have been problematic at some sites. Non-condensable gases left or charged into the system will collect in the condenser-evaporator heat exchangers but can be relatively easily purged from the system.

Since a CO_2 secondary coolant system is essentially a recirculated liquid system with wet returns (circulation rate greater than 1), balancing the flow between loads had not been a problem. Proper application of the piping network combined with careful coil design has shown that balancing can be done during the design-phase of the project and removes any complicated field-balancing procedures from consideration.

Back-up or auxiliary refrigeration units have been installed on some stores to provide a source of cooling for the CO_2 during extended power outages or maintenance procedures and opinion remains divided on future application of this cooling depending on customer experiences, reliability of the power supply, and availability of a back-up power-supply.

A detailed study of the impact of system choice on energy consumption, carbon emissions, and material requirements for the distribution piping network was performed and found that the CO2 alternative systems can provide a dramatic reduction in equivalent carbon emissions without a penalty on energy consumption, and in some cases can result in energy savings up to 12% [1]. Additional data from field-installed systems is needed to fully document the impacts on energy and long-term leak-rates of these systems and will be gathered as additional systems are installed; several projects are now in the planning stages which may provide this necessary information.

Conclusions

As attention is focused on addressing the impacts and mitigation of climate change, the increasing effort at reducing equivalent carbon emissions will require a more structured move toward systems with lower refrigerant charge and lower leakage rates. Since introduction to the U.S. in 2006, low-temperature CO_2 secondary systems and more recently cascade systems have shown that they can be successfully installed in North American supermarkets and can better reduce refrigerant emissions compared to other technologies available today. As CO_2 systems reach full commercialization, they are poised to become an increasingly important part in addressing the role of commercial refrigeration systems in this global effort.

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Improving the performance of transcritical CO_2 cycles using a controllable ejector

Fang Liu and Eckhard A. Groll, USA

CO_2 is being advocated as one of the natural refrigerants to replace CFCs and HCFCs in vapor compression cycles based on its environmentally friendly characteristics. However, the perceived lower efficiency of the basic transcritical CO_2 refrigeration cycle compared to vapor compression systems using HFC and HCFC refrigerants is a major hindrance for the technology to make progress towards practical applications. In order to recover the expansion losses and increase the cycle efficiency, it has been proposed to replace the expansion valve with either a work producing expansion device or an ejector expansion device. A comparison of the two devices shows that the ejector expansion device offers the advantages of simplicity, reliability, and availability while work producing expansion devices have the potential to yield higher cycle efficiencies. In the paper presented here, it is shown that the transcritical CO_2 cycle with an ejector expansion device outperforms a basic transcritical CO_2 cycle in both COP and cooling capacity by up to 38 and 41%, respectively, for the application of a military Environmental Control Unit at high ambient temperatures. It is also shown that a careful design of the ejector and control of the motive nozzle mass flow rate are critical in order to improve the performance of ejector expansion transcritical CO_2 cycles.

Introduction

Research studies on transcritical CO_2 refrigeration cycles have drastically increased since 1990 because carbon dioxide is being advocated as one of the natural refrigerants to replace CFCs, HCFCs, and HFCs in vapor compression systems. This is mainly due to the high latent heat of vaporization, good transport properties, and other environmentally friendly characteristics of carbon dioxide. However, the potentially lower coefficient of performance (COP) of the basic transcritical CO_2 refrigeration cycle, whose state points are shown in a pressure-enthalpy diagram in Figure 1, compared to the COP of the vapor compression cycle using halogenated fluorocarbons is a major hindrance for the technology to make progress towards practical applications. To improve the efficiency of transcritical CO_2 cycles, various innovative ideas and techniques have been proposed, including the use of microchannel heat exchangers, internal suction-to-liquid line heat exchangers, optimal control of high side pressure, expansion work recovery machines, and two-stage compression with intercooling among others (Kim and Groll 2007).

One of the largest sources of total cycle irreversibility of transcritical CO_2 cycles is the expansion process (Robinson and Groll 1998). Thus, recovery of the expansion losses is one of the key issues to improve the system efficiency. Among various expansion losses recovery schemes, an ejector expansion device has the advantages of simplicity, reliability, and availability compared to other devices. Although ejectors have been widely used in the refrigeration and other industries for many years (Chunnanond and Aphornratana 2004), most ejector applications use single-phase working fluids. In comparison, only few studies can be found in the literature on two-phase flow ejectors, as used in ejector expansion refrig-

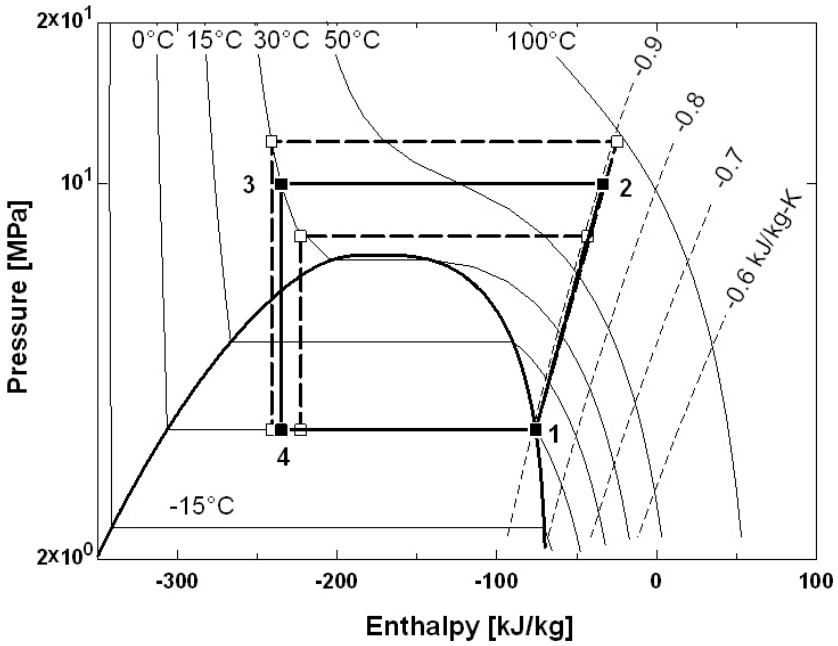

Figure 1: State points of transcritical CO_2 cycle in a pressure-enthalpy diagram (Properties from EES; Klein 2004)

eration cycles (Liu and Groll 2008). However, the design parameters and the operation conditions of transcritical two-phase flow ejectors are significantly different than the ones for single-phase applications. In addition, the interaction of the ejector expansion device with other system components such as compressor, gas cooler and evaporator is not well understood.

Ejector Expansion Transcritical CO_2 Cycle

The ejector expansion refrigeration cycle was first proposed by Kornhauser (1990). The ejector expansion transcritical CO_2 cycle consists of a two-phase flow ejector, separator, compressor, gas cooler, evaporator, and control valves as shown in Figure 2. Figure 3 shows the corresponding state points in a CO_2 pressure-enthalpy diagram.

A typical ejector consists of a motive nozzle, a suction nozzle or receiving chamber, a mixing section and a diffuser. High pressure motive stream expands in the motive nozzle and its internal energy converts to kinetic energy. The high speed motive stream entrains the low pressure suction stream into the mixing section. Both streams exchange momentum, kinetic and internal energies in the mixing section and become one stream with almost uniform pressure and speed. The stream converts its kinetic energy into internal energy in the diffuser to reach a pressure higher than the suction stream inlet pressure. When an ejector is used to replace the expansion valve in a transcritical CO_2 cycle, the expansion work lost during isenthalpic expansion process is recovered by the ejector to increase the evaporator outlet pressure to a higher compressor suction pressure. The compression work may be reduced due to the lower pressure ratio, which increases the COP of the system. In addition, a low quality, almost saturated liquid, stream enters the evaporator providing better refrigerant distribution and increased effectiveness of the evaporator.

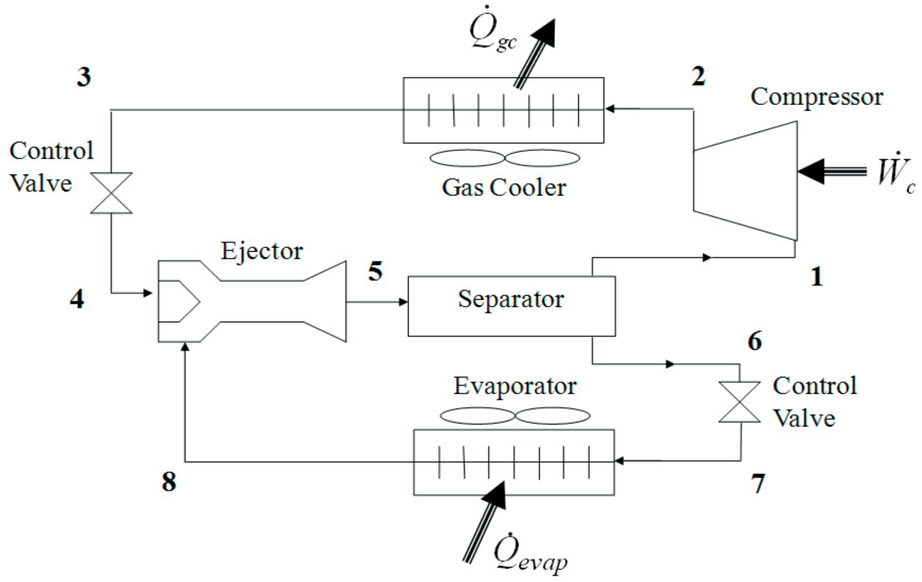

Figure 2: Schematic of ejector expansion refrigeration cycle

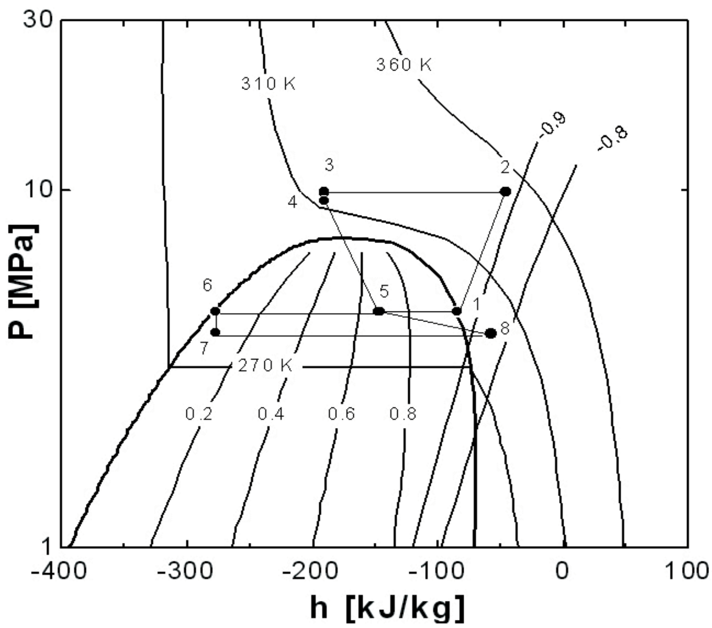

Figure 3: State points of ejector expansion transcritical CO_2 cycle in a pressure-enthalpy diagram (Properties from EES; Klein 2004)

The working processes of an ejector are illustrated in Figures 4 and 5. The motive stream expands in the motive nozzle from the high pressure P_1 to the receiving chamber pressure P_b. The enthalpy reduces from h_1 to h_{mb} and the velocity increases to u_{mb}. The suction stream expands in the suction nozzle from pressure P_2 to P_b. The enthalpy reduces from h_2 to h_{sb} and the velocity increases to u_{sb}. The two streams mix in the mixing section and become one stream with pressure P_m and velocity u_{mix}. This stream further increases its pressure to P_3 in the diffuser by converting its kinetic energy into internal energy. The fluid stream leaving the ejector is separated. Saturated vapor at the ejector discharge pressure enters the compressor, while saturated liquid expands onto the evaporation pressure and enters the evaporator.

Only limited work can be found with respect to transcritical ejector expansion devices although it has been shown that the COP can be improved by using them in refrigeration systems.

Elbel and Hrnjak (2004) studied the effect of an internal heat exchanger on the performance of a transcritical CO_2 system with an ejector. The authors used an experimentally validated system simulation model for a mobile air-conditioning system for a typical mid-sized car to evaluate the ejector expansion cycle performance. The modeling of the ejector within the system model was based on several idealized assumptions. Their results indicated that the use of an ejector significantly increases the performance compared to systems without ejector and without internal heat exchanger. In comparison to a conventional system with internal heat exchanger, the utilization of an internal heat exchanger in the ejector system yields less performance increase than the ejector system without an internal heat exchanger.

Jeong et al. (2004) constructed simulation models of a two-phase flow ejector and a vapor compression cycle with an ejector. The authors investigated the characteristics of the ejector and the performance of the ejector expansion cycle. The working fluids were ammonia and CO_2. Based on the simulation result, an optimum mixing section inlet pressure exists which maximizes the performance of the ejector. Using an ejector efficiency of 90%, the COP of the ejector expansion vapor compression cycle using ammonia is 5% higher than that of the conventional cycle and the COP of the ejector expansion cycle using CO_2 is 22% higher than that without an ejector.

Ozaki et al. (2004) studied the regeneration of expansion energy by using an ejector in a CO_2 cycle. The COP improvement by employing an ejector cycle was compared with that for an expander cycle for ideal and real cases. An experiment was carried out in order to verify the potential of COP improvement. When the COP improvement of the ejector cycle was compared to that of the expander cycle under the condition that the recovered expansion power was used ideally, the ejector cycle provides the COP improvement of less than half of that of the expander cycle due to the unavoidable losses caused by the irreversible mixing in the mixing section of the ejector. However, if the efficiencies of the ejector and the expander were taken into consideration, the COP improvement of the ejector cycle was equal to, or better than that of the expander cycle. It was determined that the refrigerant flow in the nozzle reaches supersonic flow conditions. The critical flow rate of the CO_2 coincided with the value calculated by the IHE (Isentropic Homogeneous Equilibrium) model. The gas cooling pressure of the CO_2 cycle was controlled by changing the throat area of the nozzle. Finally, the experiment using the ejector in a mobile air-conditioner system verified the COP improvement of approximately 20%.

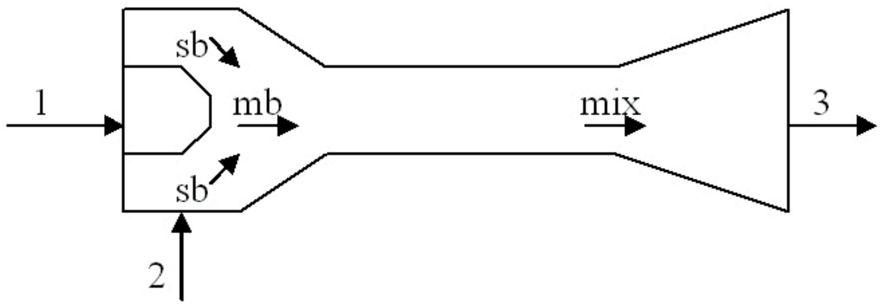

Figure 4: Schematic of the ejector expansion device

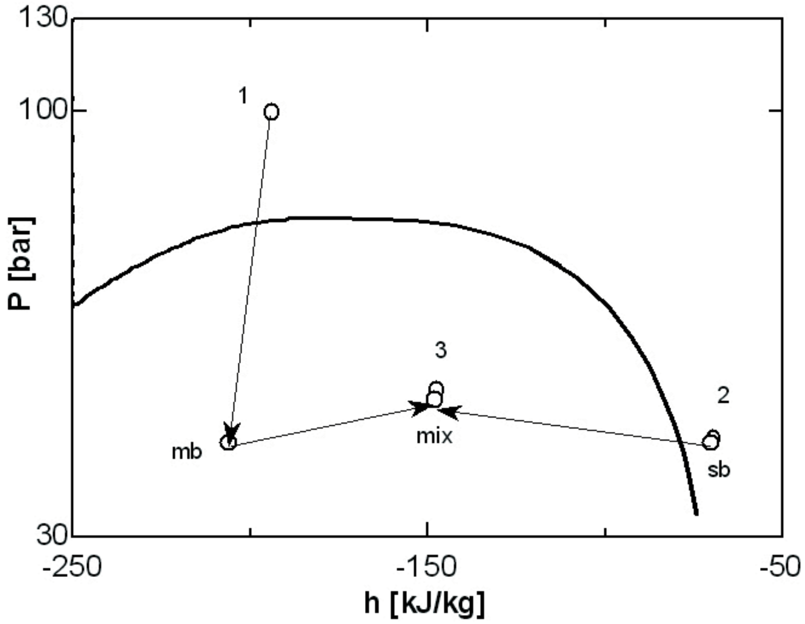

Figure 5: State points of the ejector working processes in a CO_2 pressure-enthalpy diagram (Properties from EES; Klein 2004)

Li and Groll (2005) theoretically evaluated the performance of ejector expansion transcritical CO_2 refrigeration cycles. A constant pressure mixing model for the ejector was established to perform a thermodynamic analysis of the ejector expansion cycle. The effect of the entrainment ratio and the pressure drop in the receiving section of the ejector on the relative performance of the ejector expansion transcritical CO_2 cycle was investigated for typical air conditioning operation conditions. The effect of different operation conditions on the relative performance of the ejector expansion transcritical CO_2 cycle was also investigated using assumed values for the entrainment ratio and pressure drop in the receiving section of the ejector. It was found that the COP of the ejector expansion transcritical CO_2 cycle can be improved by more than 16% over the basic transcritical CO_2 cycle.

Ksayer and Clodic (2006) used a constant pressure-mixing-zone model for the ejector and found that the COP of the ejector expansion transcritical CO_2 can be improved by more than 15% compared to the conventional transcritical cycle for typical air conditioning operating conditions.

Deng et al. (2007) conducted a theoretical analysis of a transcritical CO_2 ejector expansion refrigeration cycle which uses an ejector as the main expansion device instead of an expansion valve. It was found that the maximum cooling COP of the ejector expansion cycle is up to 22% better than the cooling COP of a conventional vapor compression refrigeration cycle and the ejector expansion cycle cooling capacity is 11.5% better than the conventional refrigeration cycle cooling capacity. In addition, the ejector expansion cycle performance was found to be very sensitive to operating conditions.

The experimental data in Elbel and Hrnjak (2008) showed that the ejector simultaneously improved the COP and cooling capacity by up to 7% and 8%, respectively, in the CO_2 system. Values of 0.8 were assumed for the individual ejector component efficiencies in the ejector calculation routine to get the results that served as the basis for the design of the experimental prototype ejector. An overall ejector efficiency based on standard pressure, temperature, and mass flow rate measurements was defined. Experiments showed that the ejector performed with a higher efficiency when the high-side pressure was relatively low. However, it was also found experimentally that despite lower ejector efficiencies, the COP increased as the high-side pressure increased as a result of using the integrated needle to reduce the motive nozzle throat area in the ejector.

Ejector Design

Domanski (1995) showed that the COPs of the ejector expansion refrigeration cycles are very sensitive to the ejector efficiency. Thus, maintaining high ejector efficiencies for all operating conditions is critical to achieve high system performance. The efficiencies of two-phase flow ejectors are significantly affected by the geometric parameters of the ejector and the system operation conditions (Liu and Groll 2008). In particular the following geometric parameters need to be considered: (1) motive nozzle throat diameter, (2) motive nozzle exit position relative to mixing section entrance, (3) mixing section constant area diameter, (4) mixing section length, (5) diffuser exit diameter, and (6) diffuser angle.

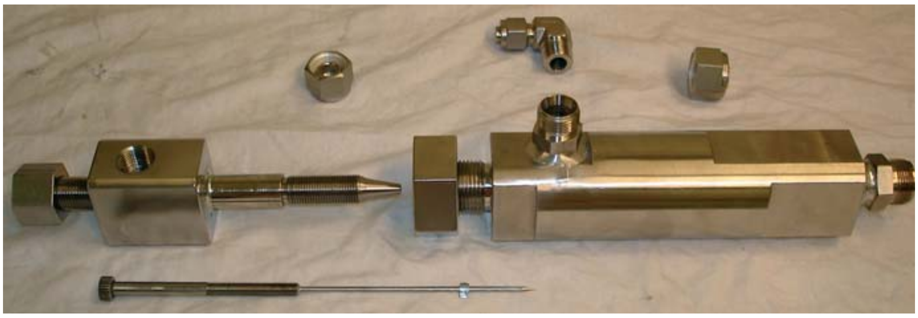

Figure 6: Photograph of controllable ejector expansion device

Literature studies showed that two of these six geometric parameters are related to others, while one can be identified directly. Keenon et al. (1950) showed that the optimum length between the motive nozzle exit position and the diffuser entrance is 6 to 8 times of the mixing section diameter. Rusly et al. (2005) found that the distance between the motive nozzle exit and mixing section entrance should be 1.5 times the constant area diameter of the mixing section to obtain optimum performance. Owen et al. (1992) suggested that the optimum diffuser angle for diffusing two-phase flow is 7°.

Liu and Groll (2008) developed a two-phase flow ejector model in which the other three geometric parameters, i.e., motive nozzle throat diameter, mixing section constant area diameter, and diffuser exit diameter, can be determined based on the choice of working fluid and operating conditions. Using this model and the information provided in the literature, a stainless-steel controllable ejector expansion device as shown in Figure 6 was constructed for operation in a transcritical CO_2 air conditioning system with 10 kW cooling capacity as follows:

1 - motive nozzle throat diameter: 2.7 mm
2 - motive nozzle exit relative to mixing chamber entrance: 1.5 times the mixing chamber diameter
3 - mixing chamber constant area diameter: 4.0 mm
4 - mixing chamber length: 6.5 times the mixing chamber diameter
5 - diffuser entrance to exit diameter ratio: 3
6 - diffuser angle: 7°

The ejector expansion device has two control options. The motive nozzle exit position relative to the mixing section inlet and thus, the suction nozzle throat area is adjustable through a thread mechanism. In addition, the throat area of the motive nozzle is adjustable by positioning a needle in the nozzle via a threat mechanism. Detailed drawings of the ejector expansion device, including the drawings of the motive nozzle, suction nozzle, mixing section, diffuser, needle, and connectors, are presented in Liu and Groll (2008).

Energy Savings Potential

The controllable ejector described above was installed in a transcritical CO_2 air conditioning system. Details of the system configuration and components can be found in Liu and Groll (2008). Comparisons of the cooling COP and cooling capacity between the test results of the ejector expansion transcritical CO_2 cycle and model predictions of the basic transcritical CO_2 cycle at the same external operating conditions are shown in Figures 7 and 8, respectively, as functions of outdoor temperature and ejector geometries. It can be seen from Figures 7 and 8 that both the cooling COP and cooling capacity ratios increase with an increase in outdoor temperature. At an outdoor temperature of 37.8°C, the ejector cycle outperforms the basic cycle regardless of the ejector geometries used. This is not the case at temperatures below 37.8°C. However, the number of ejectors with different geometries that result in cooling COP and cooling capacity ratios larger than 1 increases with the outdoor temperature. This trend indicates that benefits of using ejectors increase with the outdoor temperature and could be substantial at temperatures above a certain outdoor temperature. Using an ejector geometry of motive nozzle throat diameter D_t = 1.8 mm and suction nozzle exit diameter D_b = 5.5 mm (i.e., the distance from motive nozzle exit to the mixing section constant-area entry is 1.5 times of the diameter of the mixing section constant-area) at an outdoor temperature of 37.8 °C (100.0 °F), the comparison indicates that the cooling COP and cooling capacity of the ejector expansion cycle is approximately 38.3% and 40.8% larger than the ones of the basic cycle, respectively.

The cooling COP and cooling capacity improvements of the ejector expansion transcritical CO_2 air conditioning system compared to the basic transcritical CO_2 system that were obtained in this study are greater than those found by other researchers. This is primarily due to the fact that the experimental system chosen for the comparison is a military standard environmental control unit. The COP of the basic transcritical CO_2 system is poor, which is indicated by very high approach temperatures between the gas cooler CO_2 outlet temperature and air inlet temperature. The potential to improve the COP of this cycle by using an ejector is much greater than if the basic cycle would be optimized for the given application.

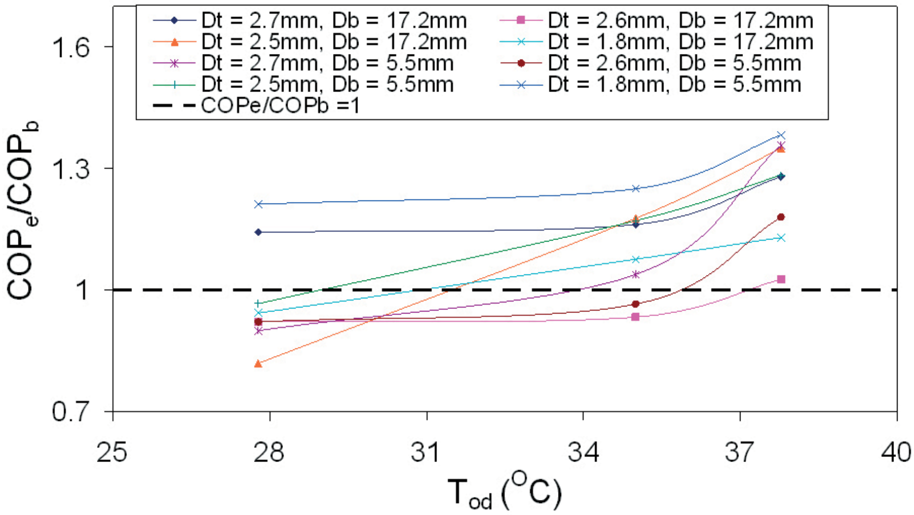

Figure 7: Cooling COP ratio between test results of the ejector expansion transcritical CO_2 cycle and model predictions of the basic transcritical CO_2 cycle versus outdoor temperature (Indoor temperature of 26.7 °C and indoor relative humidity of 50%)

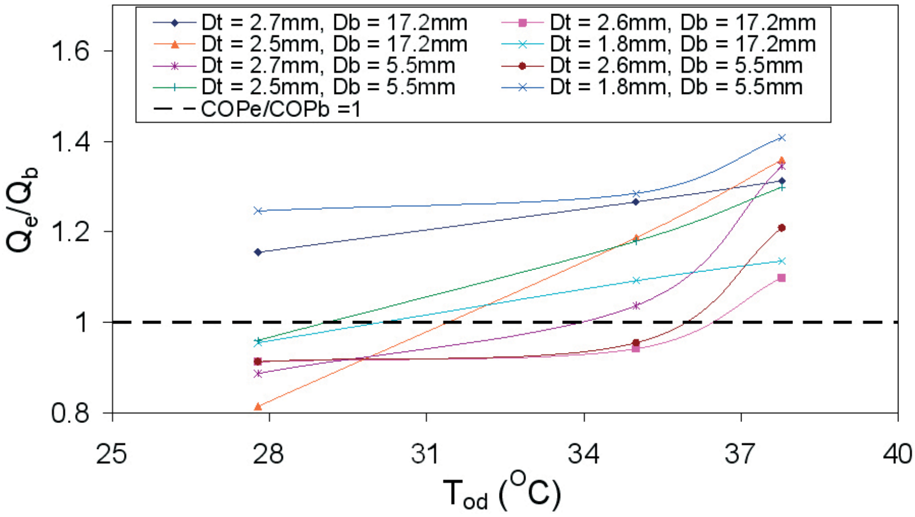

Figure 8: Cooling capacity ratio between test results of the ejector expansion transcritical CO_2 cycle and model predictions of the basic transcritical CO_2 cycle versus outdoor temperature (Indoor temperature of 26.7 °C and indoor relative humidity of 50%)

Conclusions

The following conclusions can be drawn:

- Using an expansion work producing device to reduce the throttling losses in a transcritical CO_2 vapor compression refrigeration cycle involves complex mechanical construction, difficulty in control, restricted system integration and other issues that make the integration difficult and expensive in practical applications.
- Using an ejector expansion device is attractive because the ejector is simple to construct and provides robust operation without moving parts while still yielding significant performance improvements.
- Literature studies have shown that the COP of the ejector expansion transcritical CO_2 cycle can be 7 to 22% larger than the basic transcritical CO_2 cycle for typical air conditioning operation conditions using overall ejector efficiencies or ejector components efficiencies of 0.7 to 0.9.
- The ejector geometry has to be carefully selected for the given operating conditions and has to be adjusted as the operating conditions change, or a performance

penalty for the ejector expansion transcritical CO_2 system may result.

- The potential performance gains of using ejectors in transcritical CO_2 cycles tended to be larger at higher outdoor temperatures. The highest predicted improvements in cooling COP and cooling capacity for a military Environmental Control Unit were found to be 38.3% and 40.8%, respectively, at an outdoor temperature of 37.8°C, an indoor temperature of 26.7 °C, and an indoor relative humidity of 50.0%.

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Development of Non-fluorinated Energy-saving Refrigeration and Air Conditioning Systems in Japan

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In order to develop alternative refrigerants and promote the use of products that employ alternative refrigerants, the Ministry of Economy, Trade and Industry (METI) of Japan supports R&D activities of industries, non-profit research laboratories and universities. This article describes details of the "Development of Non-fluorinated Energy-saving Refrigeration and Air Conditioning Systems" project.

HFCs, PFCs, and SF6 have been widely used in a variety of applications, as alternatives to ozone-depleting substances. However, all of these gases are major contributors to the greenhouse effect and so, along with carbon dioxide, methane, and nitrous oxide, have been recognised as belonging to the group of gases of which emissions must be reduced, in accordance with the Kyoto Protocol adopted in December 1997. HFCs are alternative fluorinated fluids that are used as refrigerants, foaming agents for thermal insulating materials, and propellants.

Figure 1 shows the trend of emissions of HFCs, PFCs, and SF6 in key countries. Although emissions of these gases in Japan have been reducing steadily in recent years, predictions of future emissions are not optimistic, as shown in Table 1. For example, predicted emissions of these gases in 2010 are higher because HCFC refrigerants are expected to be replaced by HFCs, and the emission of HFCs from discarded refrigeration and air conditioning equipment is expected to increase sharply. Refrigeration and air conditioning equipment will be a major source of greenhouse gases other than carbon dioxide.

In order to develop alternative substances and promote the use of products that employ alternative substances, the Ministry of Economy, Trade and Industry (METI) supports R&D activities of industries, non-profit research laboratories, and universities. As shown in Figure 2, the New Energy and Industrial Technology Development Organization (NEDO) manages METI-subsidised projects. NEDO is now also responsible for R&D project planning and execution, project management, and post-project technology evaluation functions.

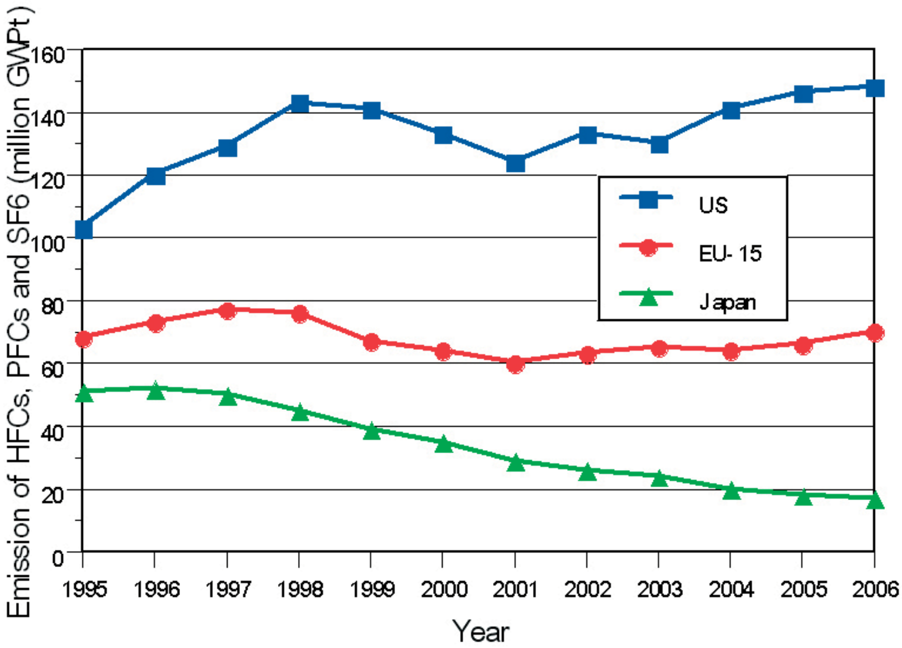

Figure 1 Trend of emissions of HFCs, PFCs, and SF6 in key countries.

Due to the regulation of specific CFCs and HCFCs as part of the Montreal Protocol for protecting the ozone layer, Japan is obligated to phase out the production and use of CFC and HCFC refrigerants. For this reason, HFCs have been developed as alternatives to ozone-depleting substances. At an early stage, manufacturers of refrigeration and air conditioning equipment successfully replaced CFCs and HCFCs with HFCs in their most common models. Air conditioners with non-fluorinated refrigerants that contribute minimally to the greenhouse effect have been partly commercialised; however, their use is not yet widespread, due to safety concerns and poor energy efficiency. In general, it is extremely challenging technically to use non-fluorinated refrigerants in air conditioners, and refrigerants have not advanced considerably beyond the research stage. In order to commercialise them, basic

Table 1 Results and predictions of emissions of HFCs, PFCs, and SF_6 in Japan. Unit (million tons-CO_2)

Year	1995 base	1996	1998	2000	2002	2004	2006	2010 estimated
Total emission	51.4	52.2	46.0	34.7	25.6	20.0	17.3	31.4
HFC production	22.9	19.1	17.1	14.9	8.4	3.2	3.3	3.1
Cellular porous medium and insulator	0.5	0.4	0.4	0.4	0.4	0.6	0.3	0.3
Aerosol	1.4	2.1	2.9	2.8	2.7	2.2	1.1	1.0
Refrigeration and air conditioning equipment	0.8	1.2	1.8	2.5	3.4	4.0	4.2	18.5
Detergent and solvent	10.6	10.4	7.4	2.8	2.3	2.3	2.2	2.6
Semiconductor and liquid crystal production	4.1	5.0	6.4	7.4	5.7	5.8	4.6	4.5
Electrical insulation gas	11.0	11.8	9.1	2.8	1.5	1.0	0.7	0.8
Metal production	0.2	0.2	0.5	1.0	1.1	1.0	0.9	0.5

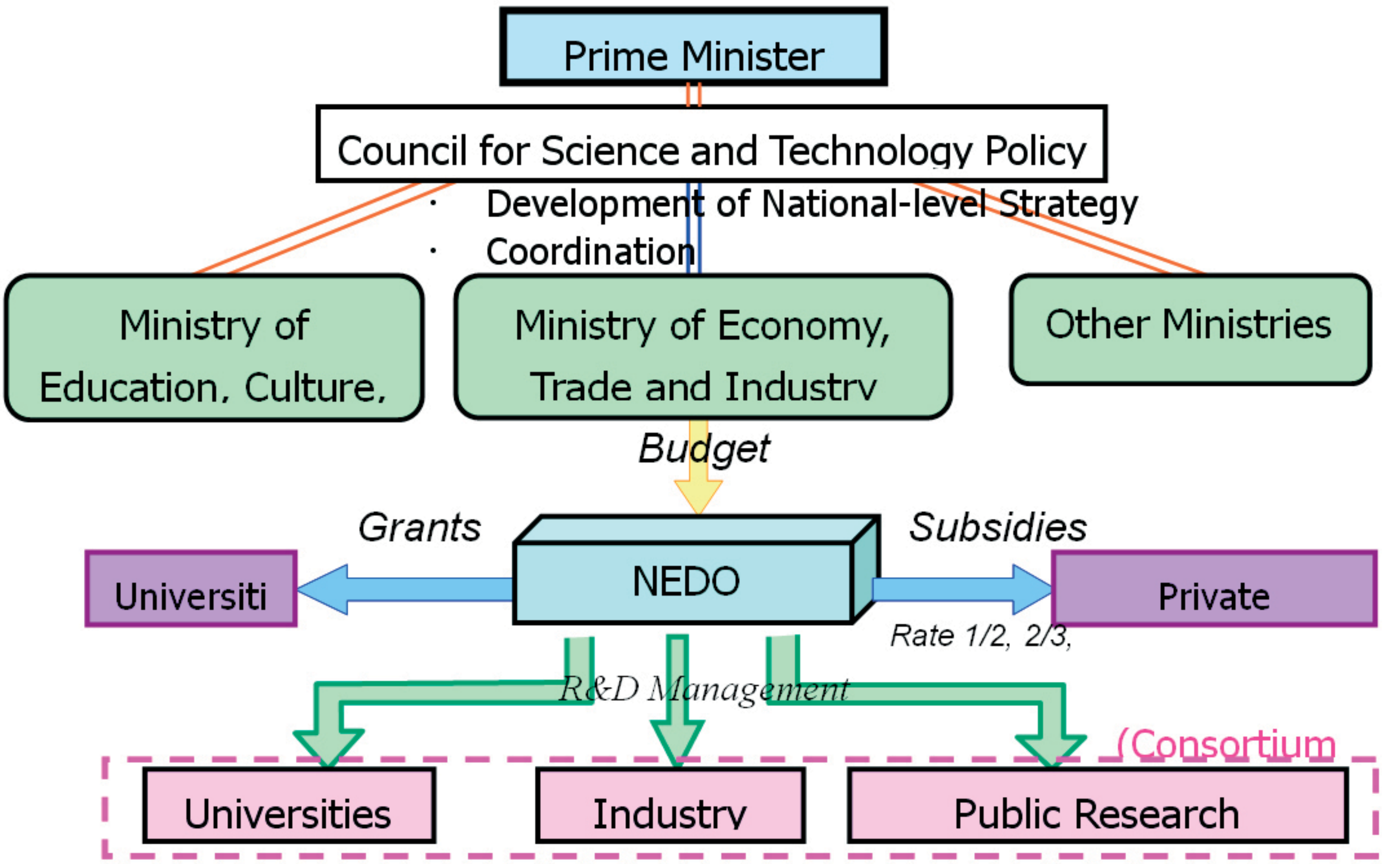

Figure 2 R&D Promotion Scheme in Japan.

equipment and energy-efficient, safe, and secure processes must be comprehensively developed. NEDO has developed and/or improved refrigeration and air conditioning systems with excellent safety and functionality. These systems use non-fluorinated substances as refrigerants, which do not deplete the ozone layer and have a low global warming potential (low GWP). Objectives of the project also include conservation of energy and reduction in overall environmental impact. As mentioned earlier, the project is called "Development of Non-fluorinated Energy-saving Refrigeration and Air Conditioning Systems," and its duration is from FY2005 to FY2009. The author is currently serving as the project leader. Table 2 shows the themes, companies, periods, and refrigerants developed under this project. R&D of refrigeration and air conditioning systems that employ a wide variety of natural refrigerants has been carried out. Development of low-GWP refrigerants, including HFO1234yf and its mixture, was begun in 2008.

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Table 2 Details of the project

Sector	Theme/Company/Period	Refrigerant
Residential	Development of a residential VRV system/Daikin Industries/2005–2007	CO_2
	Development of a ventilation air conditioning system/Shin Nippon Air Technologies/2005–2009	Desiccant
	Humidity control equipment for residential use/SINKO Industries/2008–2009	Desiccant
	Development of an energy-conservation air conditioner using a low-GWP refrigerant/Daikin Industries/2008–2009	Low-GWP refrigerant
	Development of an energy-conservation air conditioner using a low-GWP refrigerant/Mitsubishi Electric/2008–2009	Low-GWP refrigerant
	Development of an air conditioner using a low-GWP refrigerant/Panasonic/2008–2009	Low-GWP refrigerant
Commercial	Development of a hydrocarbon-based refrigerant air conditioner for commercial use/Mayekawa Mfg./2005–2007	HC
	Non-fluorinated refrigerator/Mac/2005–2007	HC and CO_2
	Non-fluorinated energy-conservation refrigeration and air conditioning system for convenience stores/Sanden/2005–2007	Ammonia
	Development of high-efficiency technology of CO_2 refrigeration cycle/Sanyo Electric/2005–2007	CO_2
	Development of high-efficiency heat pump chiller with a hydrocarbon refrigerant/Zeneral Heat Pump/2005–2006	HC
	Propane/carbon dioxide cascade system for freezing, refrigeration, and cold air conditioning/ Mitsubishi Heavy Industries Air Conditioning & Thermal Systems/2005–2007	HC and CO_2
	Air conditioning system capable of simultaneous heating and cooling operations/Mitsubishi Electric/2005–2007	CO_2
	Development of a magnetic refrigeration system at room temperature/Chubu Electric Power/2005–2007	Solid magnet
	Development of air conditioning equipment using supercritical CO_2 as a secondary refrigerant/ Mitsubishi Heavy Industries/2005–2007	HC and CO_2
	Development of high-efficiency technology in CO_2 refrigeration system using subcooling process/Sanyo Electric/2008–2009	CO_2
Transportation	Development of air cycle for mobile air conditioners and a desiccant system/Earthship USA/2005–2007	Air
	Development of a waste-heat-utilizing mobile air conditioning system/Honda R&D/2005–2007	CO_2
Evaluation	Practical performance evaluation and safety standard/Japan Refrigeration and Air Conditioning Industry Association/2005–2007	
	Environment influence of refrigeration and air conditioning system/National Institute of Advanced Industrial Science and Technology/2005–2009	

Ammonia as refrigerant in small-capacity systems

Björn Palm, Sweden

Ammonia is widely used in large systems because of its excellent thermodynamic and transport properties, resulting in energy-efficient systems with little environmental impact. Nevertheless, despite its favourable properties, ammonia has not been used in small systems. The present paper reports on work on small ammonia systems being performed as part of the EU SHERHPA project, aimed at developing heat pumps with natural refrigerants.
First, the availability, and lack, of components for small ammonia systems is considered. Second, potentials and problems specific to small ammonia systems are discussed. Finally, the paper describes the design of a 9 kW water-to-water heat pump that contains as little as 100 g of ammonia, producing domestic hot water at close to 60 ° C while having a condensing temperature of below 50 °C.
The text is partly identical to that in previously published reports to the Commission.

Introduction

Ammonia has been used continuously as a refrigerant for about a hundred years. In addition to being used in vapour compression cycles, it is also used in absorption refrigeration systems. During this time, much experience has been collected concerning the use of ammonia as refrigerant. At first sight, it would seem that there ought not to be any difficulty in finding components for designing refrigeration systems with ammonia. However, although ammonia has been used for a long time, it has been mainly in large commercial and industrial applications. One of the few exceptions is the small- to medium-sized absorption-type refrigerators based on the Platen-Munters principle, operating without moving parts and utilising an inert gas (hydrogen) to maintain a partial pressure difference between the evaporator and the condenser.

The historic reason for not using ammonia in small vapour compression systems is probably the strong smell of ammonia, in combination with the difficulty of maintaining leakproof systems while using open compressors. Any such system, having even the slightest leakage of ammonia, would result in a bad reputation for the product, even if the leakage resulted in concentrations far below the limit where flammability or toxicity might become a problem. With the introduction of the odourless, non-flammable, non-toxic “safety refrigerants” - CFCs and HCFCs - in the 1930s, all other refrigerants were soon ousted from the household market for vapour compression systems.

A second reason may be that ammonia is strongly corrosive to copper. This means that while it has been possible to design hermetic compressors for CFCs and HCFCs, where the electric motor driving the compressor is located inside a hermetic shell and where the motor is cooled by the refrigerant vapour, this has not until recently been possible for ammonia.

Most small and medium-size refrigeration and heat pump systems use CFCs, HCFCs or HFCs, and these systems have been built using copper tubing as it is soft and easy to shape, while still able to withstand the necessary pressure levels. Due to the non-compatibility of ammonia and copper, and the differences in terms of component materials, the refrigeration industry has been divided into two branches, one dealing primarily with the halogenated hydrocarbons using copper tubing and hermetic or semi-hermetic compressors, and one dealing with ammonia, using steel tubing and open compressors, concentrating on the large-capacity systems.

Since the discovery of the CFC-refrigerants’ negative effect on the ozone layer and their contribution to the global warming, interest in using natural fluids - i.e. substances already present in the natural environment - as refrigerants has increased tremendously. One important step in this direction is to increase the range of applications for ammonia. As a part of this broadening of the use of ammonia it is important to identify components available on the market. For small-scale applications, there is a lack of components specifically designed for ammonia. The next step is therefore to try to identify components designed for other applications which may also be suitable for ammonia. For large-capacity applications, it is important to try to identify components which can increase the reliability and safety of the systems, thereby allowing the use of ammonia in applications where it is presently not used because of safety concerns or safety regulations. One important aspect here is the total charge of refrigerant: Reducing the charge is an important step in increasing safety.

The aim of one of the Work Packages in the EU SHERHPA project has been to design a small, single-family house heat pump using ammonia as refrigerant. A first step in the design is to investigate the availability of the necessary components on the market. The components should be designed for a heating capacity of up to 10 kW. The focus has been on identifying components for a water-to-water heat pump. The heat pump should be designed both for space heating, being connected to a hydronic heating system, and for producing domestic hot water. A sketch of the basic design is shown in Fig. 1. The figure shows the design of one of several test rigs used, but includes all components necessary for the actual heat pump. The figure includes also the instrumentation and auxiliary equipment for testing the heat pump in the lab.

As ammonia has a strong smell which may cause panic, and as it may be flammable and toxic in high concentrations, the system should be designed to be gastight and to operate with a minimum amount of refrigerant. This has influenced the focus of the search for components to such with small internal volume. Possible choices of components are presented below.

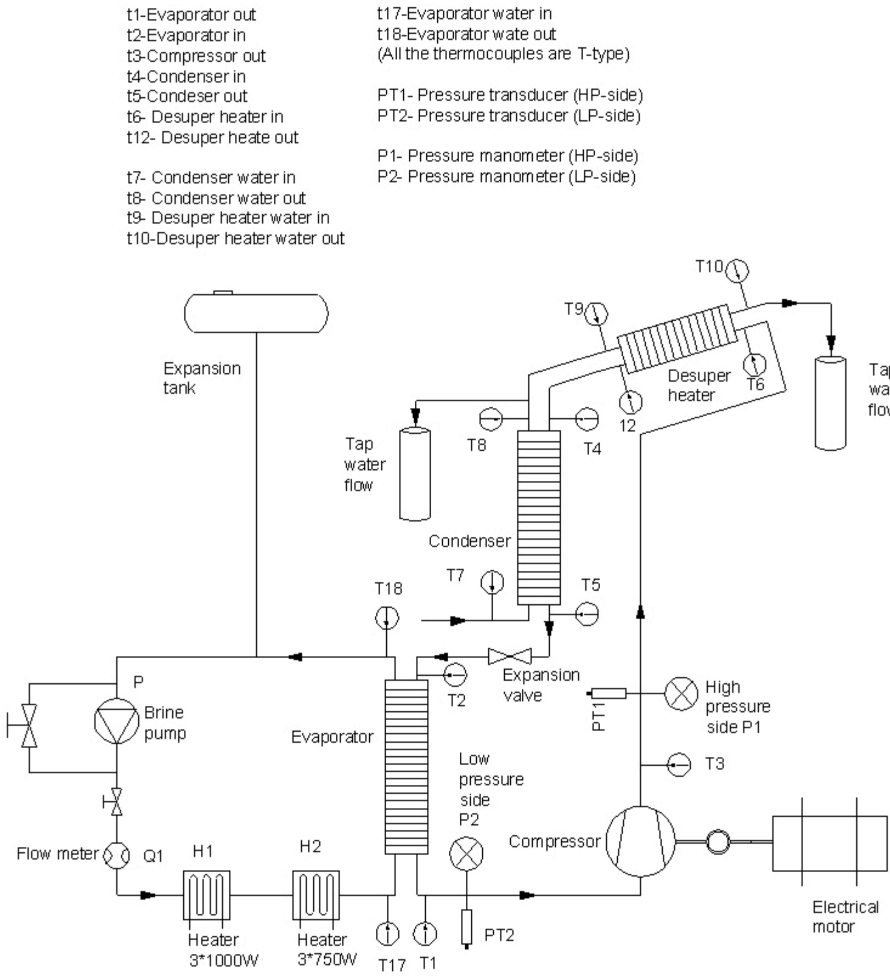


Figure 1: Schematic diagram of heat pump test rig with heaters simulating heat source

Compressors

As already noted, ammonia is highly corrosive to copper and copper alloys, and for this reason it is difficult to design a hermetic compressor for ammonia as the electric motor, usually having copper windings, would be in contact with the refrigerant. However, a hermetic compressor may not be absolutely necessary for the design of the heat pump. Small size open compressors, available on the market, could be suitable for this application. Another option is to use a motor where the rotor contains no copper, and where the copper-containing stator windings are located outside a pressure-proof hood. This type of compressor is referred to as a separating hood compressor or canned motor compressor. A third option is to find a hermetic compressor designed with aluminium instead of copper in the motor windings. The electrical insulation of the wires in the motor would also have to be chosen to be compatible with ammonia.

Open compressors

Open compressors designed for ammonia are available from a few different manufacturers even for small sizes. One example, the F2 compressor from Bock, is shown in Figure 2. In most cases, the same compressors, or very similar designs, may be used for different refrigerants. The size of the motor as well as the heating or cooling capacity depends on the refrigerant used and the running conditions. If belt drive is used, the capacity of open compressors can easily be varied by the choice of the sizes of the pulleys in the transmission. Constant or variable-speed motors may be connected to the compressor as desired. All these factors contribute to the fact that a smaller number of compressor types are necessary

Figure 2: Example of F2open compressor for ammonia.

to cover a large range of demands, compared to hermetic compressors where the combination of motor and compressor is fixed.
Among the possible choices, the model F2 compressor, shown in Figure 2, was chosen for the pre-prototype. This compressor is designed

for ammonia, and the heating capacity, when run at the lowest allowable speed, is about 7 kW. Similar compressors are also available from other manufacturers.

Separating hood compressors

In a separating hood compressor, or canned motor compressor, the electrical windings of the stator of the compressor motor are located on the outside of a thin gas-tight shell. The rotor, which has no windings but only permanent magnets, is located inside the shell. This means that

Figure 3: Separating hood compressor.

there are no electrical connections or shaft from the refrigerant side to the outside, thus minimizing the risk of refrigerant leakage. Also, there is no direct contact between the refrigerant and the windings. Canned motors have been used in other branches of industry where the windings need to be shielded from a harsh environment. Canned motor compressors are not widely used in the refrigeration industry. In fact, there seem to be only one manufacturer of compressors for refrigeration and heat pump applications of this type, the Austrian company Frigopol (see Figure 3). The company manufactures such compressors in different sizes ranging from about 3 kW to about 21 kW cooling capacity from a single compressor. Canned motor compressors are also mentioned on Mycom's web page, but there is no information if this type of compressor is still in production.

Aluminium-motor compressors

A few years ago, Mayekawa (Mycom) announced that it had developed Japan's first "high-efficiency semi-hermetic canless motor for ammonia refrigerant". According to their web page, the motor is compatible with ammonia by using aluminium in the motor windings and by the use of Teflon® for electrical insulation of the windings. The compressor is of scroll type, and the cooling capacity ranges from 5 to 15 kW. It is connected directly to the motor, which runs at 3600 r/min at 60 Hz. The scroll compressor per se seems to be a development of products previously presented by Hitachi. Previous models, for other refrigerants, have been supplied with a liquid injection system. It is not clear if the new type of compressor is also supplied with this system. Even though this compressor was described on the web page, it seems that it has still not been made commercially available. However, a larger size compressor of similar motor design was recently presented by Mycom.

The lack of hermetic or semi-hermetic compressors would at present be an obstacle to the introduction of small ammonia heat pumps on the market.

Heat exchangers

The heat pump needs three heat exchangers: condenser, evaporator and desuperheater. For the application with ammonia, a low charge of refrigerant is desired. This is achieved primarily by the use of indirect systems, i.e. a water (brine) to water heat pump system, by which design the refrigerant-containing parts may be housed within a small volume without long lines connecting the parts. With short lines, the diameters of the connecting lines can be small, without much penalty from pressure drop, which will contribute to reduction of the charge. In such compact systems, the main part of the refrigerant will be in the heat exchangers, and to achieve a low total charge the heat exchangers should be chosen to have as low internal volume as possible.

Figure 6: Gasket plate heat exchangers

Heat exchangers for HFC refrigerants are frequently designed with copper tubing or copper surfaces, and are therefore not suitable for use with ammonia. However, there are options available that are suitable for this refrigerant. A very compact type of heat exchanger is the plate heat exchanger (Figure 6). This type of heat exchanger is found in different designs for different applications: Originally, plate heat exchangers were designed as gasketed plates inserted into a frame with solid end-plates held together by connecting rods. In large ammonia systems, it is common to use semi-welded plates, where the plates constituting the ammonia side are welded together, while the water side is kept tight by gaskets. In small to medium-size heat pump and refrigeration systems with HFC and HCFC as the refrigerant, it has been common for the last 15 years or so to use brazed plate heat exchangers. Normally, the brazing metal is copper. However, for special applications, nickel is used as the brazing material, and is fully compatible with ammonia. Nickel brazing is not as strong as copper brazing, but fully sufficient for the pressure levels necessary for heat pump applications. Several companies manufacture nickel-brazed plate heat exchangers in sizes suitable for the present project.

Some years ago, Alfa Laval introduced a new type of fully welded heat exchangers called Alfa Nova (Figure 7). This type contains only stainless steel and is thus fully compatible with ammonia. The plates are joined by a novel technique called *AlfaFusion*, based on the use of a filler which, at high temperature, melts

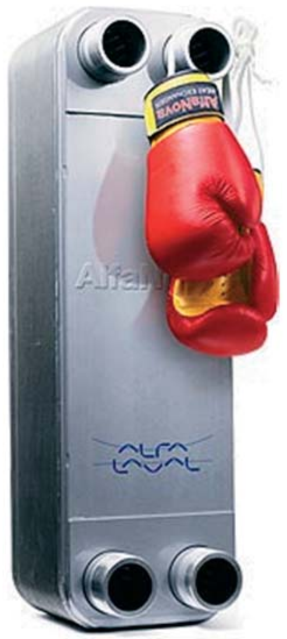


Figure 7: Fully welded stainless steel heat exchanger. (AlfaNova)

and forces the surface of the plate material to melt, thus creating a complete stainless steel bond.

Another interesting type of heat exchanger for the present application is the spiral-type heat exchanger, manufactured by Spirec. This is a fully welded stainless steel heat exchanger used for many different applications. The principle of the design is shown in Figure 8. One important advantage of this heat exchanger is that the flow is not divided into parallel channels as in the plate heat exchangers.

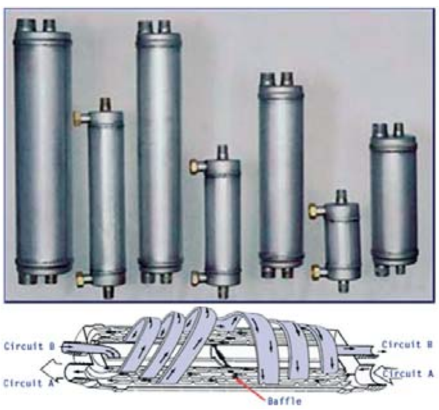


Figure 8: Spirec heat exchangers

Several other types of heat exchangers are found, made either of plain steel, of stainless steel or of aluminium, and all these may be used with ammonia, if designed for the pressure levels necessary (min 20 bar). An interesting possibility is to use heat exchangers made from extruded multichannel aluminium tubes. These tubes may be used both for air-to-refrigerant heat exchangers and for water-to-refrigerant heat exchangers. They are already used in the automotive industry for AC condensers and evaporators with HFC as refrigerant. The small diameters of the channels result in very small internal volumes and thereby low refrigerant charge.

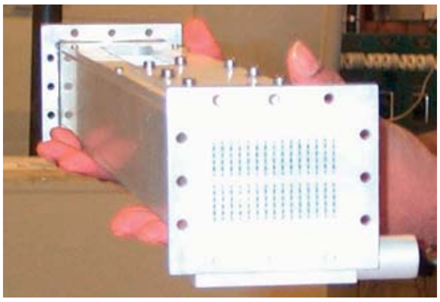


Figure 9: Prototype water to refrigerant heat exchanger

A prototype water-to-refrigerant heat exchanger has been designed and tested with propane as the refrigerant at the Royal Institute of Technology (KTH) in Stockholm (Figure 9). Two such prototype heat exchangers were used in the final test set-up.

Expansion devices

Thermostatic valves

Most heat pumps in the 3-10 kW size range use thermostatic expansion valves. Such valves are of course designed specifically for a given refrigerant, as the bulb is filled with the same fluid as in the system, or with a fluid having a similar vapour pressure curve.

We have scanned the market for a valve designed for ammonia and have had problems finding one for capacities below 15 kW. The smallest ammonia thermostatic expansion valve we have found on the market is Danfoss TEA 20 (Figure 10). This valve is designed for a wide range of capacities, from about 3 kW to 28 kW, which is achieved by selecting one out of several possible orifices.

Figure 10: TEA 20 thermostatic expansion valve

Even though the valve is designed fits the capacity of the small heat pump being developed in the SHERHPA project, it still does not seem to be the best solution. It has clearly been designed for larger systems, and its large mass of valve housing and bulb may have a negative influence on the stability of the system. The valve was tested in the first pre-prototype, and did not perform well in this system.

Electronic valves

Electronic expansion valves have so far been used mainly in relatively large commercial and industrial systems. One important reason for this is that the cost of these valves has been too high to be motivated in smaller capacity systems. Some smaller valves have also been introduced to the market, but these have mainly been intended for use with HFC refrigerants.

A few years ago, a small electronic valve suitable also for use with ammonia was introduced by the Italian company Carel. The valve is a needle valve, regulated by a small canned stepping motor, and has no electric

Figure 11: Electronic expansion valve previously also available for ammonia

parts in contact with the refrigerant. The first series of valves had the needle and all other parts made of stainless steel, thus making it compatible with ammonia. The electronic driver has the vapour pressure curve of ammonia programmed, so that all parts are prepared for the use in ammonia systems. An additional feature of this valve is that it is bi-directional, which may be very important in heat pumps used for cooling in the summer time. This valve was tested in the prototype heat pump with at least some success.

Unfortunately, the valve design was later changed by the manufacturer, so that the needle of later models is made of brass, which is not compatible with ammonia. The electronic driver may still be used, but we have not been able to find a valve suitable for low capacity with ammonia that can be connected to this driver.

Capillary tubes

Refrigerators, freezers and low-capacity air conditioning systems frequently use capillary tubes as expansion devices. This may be a viable option also for low-charge heat pumps, if designed in the correct way. It may be argued that the necessary capacity of the expansion device of a heat pump varies and that the constant constriction of a capillary tube would not be suitable. However, the capacity of a capillary tube adapts automatically to that of the compressor as the inlet condition of the capillary tube changes from highly subcooled to vapour /liquid mix: If the capacity of the capillary tube is too low to match the flow through the compressor, liquid will start collecting in the condenser, thereby increasing the subcooling and thus postponing the flashing in the capillary tube, which will reduce the flow resistance and increase the mass flow. A capillary tube would have to be combined with a low-pressure receiver to prevent liquid from reaching the compressor. However, the size of this receiver is directly related to the amount of charge in the system, and could be very small. The use of capillary tubes in direct-expansion ammonia systems needs further investigation.

Prototype heat pump design

The aim has been to design a heat pump system extracting heat from a secondary loop, as would be the case in a ground source heat pump. Likewise, heat is assumed to be delivered to a hydronic heating system in a building. The heat pump should be able to supply both space heating and heating of domestic hot water. As the temperature of ammonia increases during compression much more than does that of other fluids, the hot gas temperature sets a limit to the pressure ratio, and so to the temperature gap which can be bridged in one single stage. This is often seen as a drawback of this refrigerant. However, in a heat pump of the type designed here, this specific property can be utilized to generate domestic hot water at temperatures above the saturation temperature in the condenser. The high hot gas temperatures should thus be seen as a great advantage in this case. For this reason, the system design includes a hot gas heat exchanger and, as shown in Figure 11, the hydronic water is first passed through the condenser and then part of the flow is allowed to pass through the desuperheater. By adjusting the flow rate through the desuperheater, it is possible to adjust the temperature level of the outgoing water to any value up to the hot gas temperature of the refrigerant, which is above 100 °C. In an actual design, the heat pump would heat a tank with hydronic water. Water from the bottom of the tank would be fed into the condenser. The main flow of water after the condenser would be returned to the middle of the tank, while the water heated in the desuperheater would be fed into the top of the tank. Water to the heating system would be taken from the middle of the tank and returned to the bottom of the tank. In this way a stable stratification of water would be achieved in the tank. Domestic hot water would be heated by passing it through a spiral copper tube inside the tank, with the water entering at the bottom and exiting at the top.

Plotting the process in an h-log(p) diagram, it is clear that the heat delivered in the desuperheater is about 20 % of the total. As the domestic hot water is preheated in the bottom of the tank, these 20 % should only be used for raising the temperature of the water from the condensing temperature. Typically, the condensing temperature would be +40 °C and the desired domestic hot water temperature +60 °C. Assuming a water inlet temperature of 15 °C, less than half of the power for heating the domestic hot water is supplied in the desuperheater. This means that even if the hot water demand is as much as 40 % of the total power demand, the condensing temperature would not need to be increased to maintain a domestic hot water temperature of 60 °C.

Throughout the development of the prototype, the same compressor was used, a model F2. This is an open compressor, and it was connected to a standard electric motor with a nominal power of 4 kW. This is more than necessary, and this has probably influenced the performance in terms of COP of the system. The motor was speed-controlled by an inverter, although the final laboratory tests were performed at nominal constant speed.

As ammonia is slightly flammable and is toxic in high concentrations, the design was focused on minimising the charge of refrigerant in the system. Minimum charge was achieved by use of compact heat exchangers and by designing the system without a receiver. Also, all tubing was selected with the smallest suitable diameter, and the system layout was made so as to get short tubing distance, particularly in the liquid line.

The ammonia used in the tests was refrigerant grade, with a water content less than 200 ppm.

First tests

In the first set of tests, all heat exchangers were AlfaNova plate heat exchangers. These heat exchangers are diffusion-welded and are available in all sizes down to units suitable as a desuperheater in this system. Tests were run with a mineral oil, and no special means was provided for returning the oil to the compressor. After only a few hours of operation the performance degraded and the system became unstable. When opening the evaporator it was found that a considerable amount of oil had accumulated in this part of the system. It was obvious that the velocities in the evaporator were not enough to force the oil out of the heat exchanger.

Figure 11: Combined receiver and sight glass

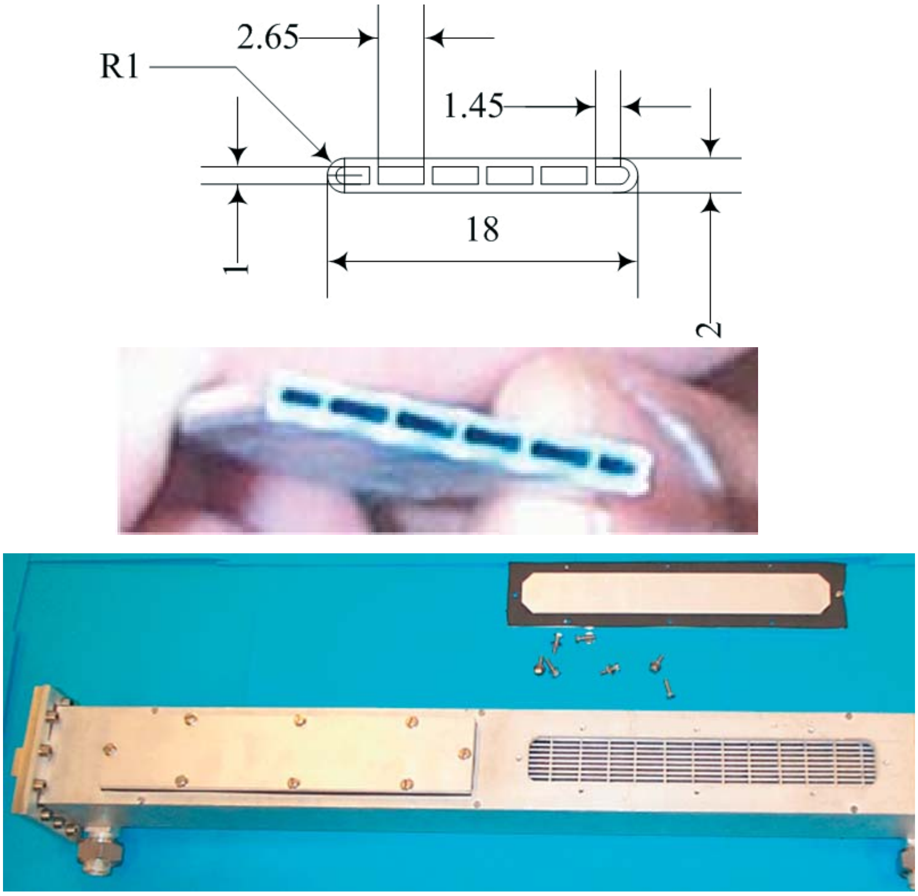


Figure 12: Prototype aluminium heat exchanger tested as both condenser and evaporator. The top part of the figure shows details of the flat aluminium tubes used for the refrigerant flow. All numbers in mm.

Final prototype

Because of the problems with oil return and the fact that feeding the evaporator from the top did not solve the problem of oil return (because of the bad performance of the heat exchangers when fed from the top), it was decided to re-design the system so that the evaporator could be fed alternatively from the top or the bottom, and to include an oil separator in the system. The basic system layout is still the same as before, with the additional ability to pass the refrigerant through the oil separator if necessary.

Several changes were introduced at this stage, the most important of which are the following:

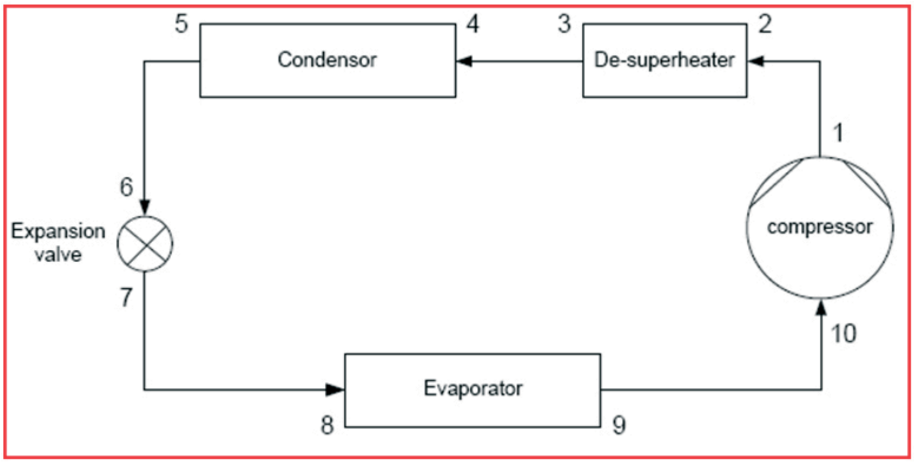
- The introduction of the oil separator. As no small oil separators specifically designed for ammonia are available on the market, a unit designed for HFC, was used. The shell of the unit is of steel, but there are probably parts made of brass inside it.
- A combined receiver and sight-glass was introduced in the liquid line. This unit was designed and made at the department of ROYAL INSTITUTE OF TECHNOLOGY and consists of a glass tube with compression fittings with Teflon gaskets at the two ends: see Figure 11). Using this glass receiver, it was possible to see any changes in the colour of the ammonia, indicating wear of the system. It was also possible to see the liquid level and thus determine if the charge was sufficient or too large under the specific running conditions. It was also very helpful in controlling the function of the expansion valve.

  The water temperatures out of the condenser and out of the desuperheater were regulated by electrically operated valves set by electronic PID controllers. A three-way valve was used to mix cold mains water with the water from the condenser so as to keep the return temperature of the water to the condenser at 40 °C. A two-way valve restricted the water flow through the desuperheater so as to keep the outlet water temperature at 60 °C. A second two-way valve was installed in the line from the condenser to the three-way valve, but this valve was kept at a constant opening during the test and acted only to provide a slight pressure drop to force the water through the desuperheater. These modifications and additions greatly assisted the running of the system.
- As blockage of the expansion valve had occurred on some occasions, a filter (strainer) was introduced in the liquid line just before the expansion valve. This particular type, has a very small internal volume and can be easily disassembled for cleaning.

- Different types of expansion valves were tested, both manual and electronic. The most successful, in this setup, was the small electronic valve from Carel (E^2V). The valve itself is a needle valve, where the position of the needle is determined by a stepper motor. It is designed to be used with any of several refrigerants. As mentioned above, the first versions of this valve used were designed to withstand ammonia, but later during the project it was found that the manufacturer had altered the design so that parts of the interior were made of copper alloys and therefore the valve was no longer recommended for ammonia. Still, we have used this valve with some success, with the assumption that the possible corrosion would not influence the results during a short period of operation.

The valve is connected to an electronic driver, which controls the position of the valve based on measurements of pressure and temperature.

- All refrigerant tubes were made of stainless steel. The tube lengths and tube diameters of the system are shown in Figure 12.



Tube section	Tube length m	Tube OD mm	Tube thickness mm	Working pressure bar
1-2	2.65	12	1	200
3-4	0.49	10	1	240
5-6	0.8	6	1	420
7-8	0.22	10	1	240
9-10	1.2	12	1	200

Figure 13: Tube lengths and tube diameters for pre-prototype 3

Tests were run with this system with different types of heat exchangers used as evaporators and as condensers. The most important general experience from these tests is as follows:

- A prototype plate heat exchanger with extra-low pressing depth was tested briefly as evaporator. The unit was operated with the refrigerant entering at the bottom. The performance was not as expected (high temperature differences between the fluids), probably because of oil build-up and perhaps partial blockage of the channels in the plate heat exchanger.
- Prototype heat exchangers designed at our department at the Royal Institute of Technology in another project were tested both as condensers and evaporators. The design is shown in Figure 2. The heat exchangers are built up as shell and tube type, but with flat multichannel aluminium tubes. The refrigerant flows inside the flat tubes, and the water is led back and forth in the narrow slots made up of the flat tubes, the flow being governed by baffle plates. The refrigerant surface areas of these heat exchangers were 0.78 m² for the evaporator and 0.94 m² for the condenser. This is similar to the surfaces of the originally used plate heat exchangers. The experience from this heat exchanger was mainly good. The temperature differences as a condenser were equal to or better than the plate heat exchanger. As an evaporator, the performance was much better than for any of the other types of heat exchangers tested. In addition, the system could be operated stably for long periods of time, indicating that the oil was being returned to the compressor. However, the performance was considerably lower than for the same heat exchanger when used with propane as refrigerant. This indicates that the oil is still adversely affecting heat transfer, either by blocking parts of the channels or by covering the heat transfer surfaces with an oil film, constituting a thermal resistance.
- The oil return was obviously only partly solved even with the best of the heat exchangers tested. In an effort to resolve the oil return problem, the system was cleaned out completely and a miscible oil was loaded into the compressor. The oil used was a PAG oil with the name of CP 412-100. With it, the evaporator performance was enhanced considerably and the system could be run stably. Still, the temperature differences in the evaporator were larger than with propane in the same heat exchanger under similar conditions. This result is not expected if based only on the thermodynamic properties of the two refrigerants.
- The use of the oil separator made no major difference to the charge, or to the performance of the evaporator.

In all of the tests with this final prototype, the ammonia charge was about 100 g. The capacity was about 9 kW at the conditions expected in typical heat pump applications. This gives a specific charge of 11 g refrigerant per kW heating capacity.

The following section presents the test results from this prototype for the tests using the aluminium mini-tube heat exchanger and the miscible PAG oil.

Test results

The results presented here are for the final prototype using the miscible PAG oil and the aluminium micro-channel heat exchangers as condenser and evaporator and a plate heat exchanger as desuperheater. Other test conditions are given in the table below:

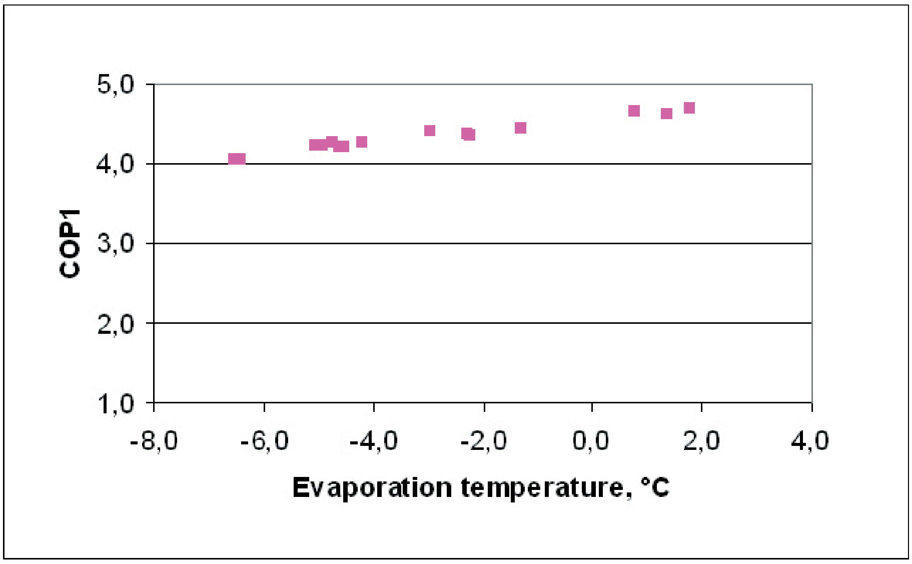


Figure 14: COP1 of the system as a function of the evaporation temperature. Values not including electric motor efficiency.

Test conditions:
- Ammonia charge: ~100 g
- Condensing temperature: 48±2 °C
- Desuperheater outlet temp: 63±1 °C
- Superheat: 5±1 K
- Subcooling: 3±1 K
- Evaporator loads: 5 – 7 kW

Figure 14 shows the measured COP_1 for the system at different evaporation temperatures. The values are calculated from the ratio between the heat output and the compressor work input. The heat output is calculated from the water flows and temperature changes through the condenser and desuperheater. The compressor work is calculated as the product of the specific enthalpy change across the compressor, based on the measured pressures and temperatures at the compressor inlet and outlet, and the mass flow of refrigerant calculated from a heat balance across the condenser and desuperheater. To be more specific, the COP is calculated as can be seen below.

This means that the efficiency of the electric motor is not included in the calculation. The reason for using this definition is that the electric motor was slightly too large for the application and therefore did not have a high electrical efficiency under the particular conditions. As the choice of electric motor is not a representative characteristic of ammonia heat pumps, it was considered more interesting to show values excluding the performance of the motor. As the efficiency of a good electric motor of this size is around 90 %, it is easy to correct the values in the diagram by multiplying by 0.9. This indicates that the COP_1 of a system working under the conditions given here should be in the range 3.6 to 4.3.

The results show that the system functions well and gives highly competitive COPs.

$$COP_1 = \frac{\dot{Q}_1}{\dot{E}_k} = \frac{\dot{Q}_{c,w} + \dot{Q}_{d-s,w}}{\dot{E}_k} = \frac{c_{p,w} \cdot \left(\dot{m}_{c,w} \cdot \Delta T_{c,w} + \dot{m}_{d-s,w} \cdot \Delta T_{d-s,w}\right)}{\dot{m}_r \cdot \Delta h_{k,r}}$$

where

Q_1 = heat rejected in the condenser and in the desuperheater

$Q_{c,w}$ = heat rejected in the condenser, calculated from the water side.

$Q_{d\text{-}s,w}$ = heat rejected in the desuperheater, calculated from the water side.

E_k = compressor power, calculated from refrigerant flow and enthalpy change across the compressor

$c_{p,w}$ = specific heat of water

$m_{c,w}$ = mass flow of water through the condenser

$\Delta T_{c,w}$ = temperature change of the water passing through the condenser

$m_{d\text{-}s,w}$ = mass flow of water through the desuperheater

$\Delta T_{d\text{-}s,w}$ = temperature change of the water passing through the desuperheater

m_r = mass flow of refrigerant, calculated from a heat balance across the condenser and desuperheater

$\Delta h_{k,r}$ = enthalpy change of the refrigerant across the compressor, based on pressures and temperatures at inlet and outlet

Figure 15 shows the heating capacities measured at the condenser and at the desuperheater. As expected, and as discussed above, the desuperheater power is about 20 % of the total power. Under the tested conditions, the hot water flow rate out of the desuperheater was about 1.5 litres per minute at 63 °C. This temperature is high enough to ensure killing of any Legionella or other bacteria. This means that the heat pump should be able to produce all necessary hot water without increasing the condensing temperature above 48 °C during much of the year. The final outcome in this respect will depend on the necessary temperature levels in the hydronic system, the use of hot water by the individual family and the climate where the building is located.

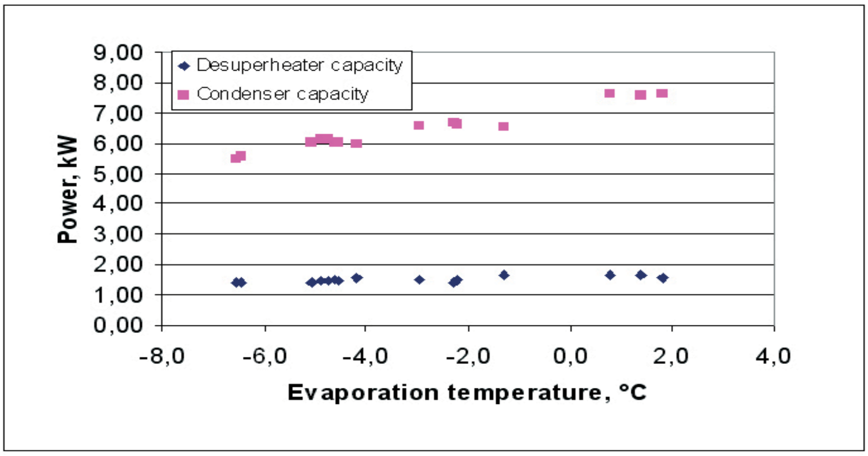


Figure 15: Heating capacities from condenser and from desuperheater as a function of the evaporation temperature

Figure 16 shows the compressor outlet temperature (hot gas temperature) as a function of the evaporation temperature, still with the condensing temperature of 48 °C. It is clear that this condensing temperature is at the limit of what can be reached with evaporation temperatures that can be expected in ground-source heat pumps in northern Europe.

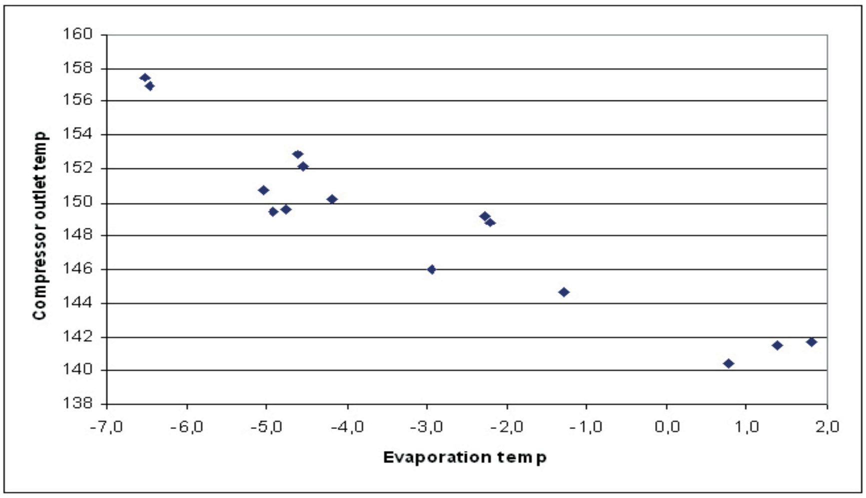


Figure 16: Compressor outlet temperature as a function of the evaporation temperature

Conclusions

A pre-prototype of a water-to-water heating-only domestic heat pump for heating both domestic hot water and water for space heating, using ammonia as the refrigerant has been designed, built and tested. The tests show that this type of system is possible: The hot gas temperatures are within limits under conditions which could be expected in a domestic heat pump. The relatively high hot gas temperature allows heating of domestic hot water to temperatures well above the condensing temperature, which is an advantage of using ammonia as refrigerant.

It has also been clearly demonstrated that the amount of refrigerant can be kept at a very low level if compact heat exchangers are used. 100 g of ammonia was sufficient for the 9 kW heat pump using mini-channel heat exchangers, and 120 g with plate heat exchangers.

The greatest obstacle for commercial introduction of this technology is the lack of components. However, open compressors are available, as well as one electronic expansion valve of suitable size (but containing copper parts). Welded-plate heat exchangers are also available which should function well as desuperheaters and condensers. The evaporator is more crucial. Four different evaporators were tested, and none of them performed well with non-miscible oil. A low pressing depth plate heat exchanger and a prototype mini-channel heat exchanger were tested with a miscible PAG-oil. Of these, only the mini-channel evaporator gave acceptable mean temperature differences.
In conclusion, the lack of components is a significant problem for the development of this technology. A prototype heat pump would have to be designed from prototype heat exchangers and probably also a prototype expansion valve.

Future plans

The SHERHPA project is now ended and therefore also the funding for this research. At Royal Institute of Technology we plan to continue the work started, using the components still available as well as the knowledge collected during the SHERHPA project. The components in the prototype made for the laboratory tests will constitute the base for this work. The planned heat pump prototype will use the prototype mini-channel heat exchangers and an open compressor, together with a synchronous permanent magnet motor.

Heat Transfer Characteristics of CO_2 as Refrigerant for Heat Pump Water Heaters

Shigeru Koyama and Ken Kuwahara, Japan

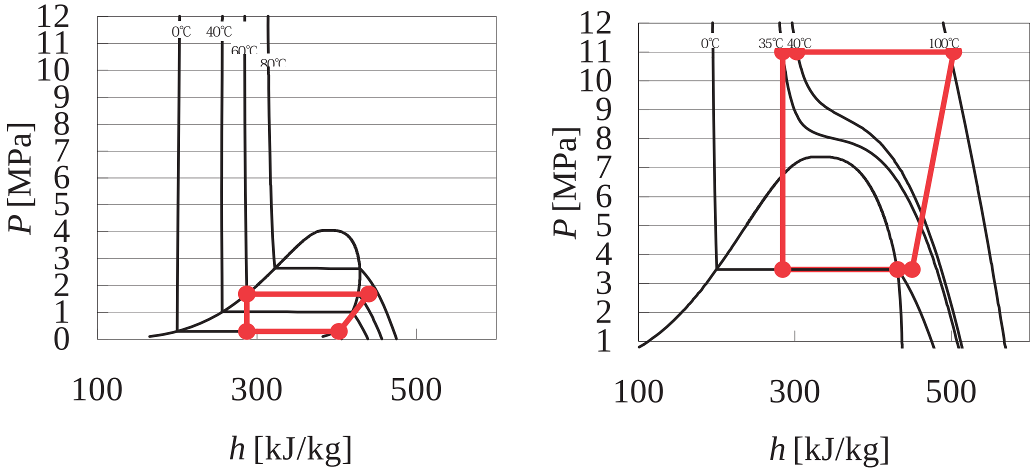
1. Introduction

In the air-conditioning and refrigeration fields, systems based on CFC and HCFC refrigerants have been used for the last several decades for heat pumping and refrigeration applications. However, since Molina and Rowland [(1)] reported the destruction of the stratospheric ozone layers due to CFC refrigerants in 1974, the international movement to prevent ozone destruction has been enhanced. As a result, CFC refrigerants have been totally eliminated, and HCFC refrigerants have been restricted in the Montreal Protocol in 1987. This has been followed by promotion of HFC refrigerants, that have no effect on ozone layers, as replacements for CFCs and HCFCs. Moreover, at the Kyoto Conference on Prevention on Global Warming (COP3) held in 1997, HFC refrigerants were also subject to regulations from the viewpoint of prevention of global warming. In response to such an effort to reinforce international regulations, the development of high-performance heat pump and refrigeration systems using naturally occurring substances as working fluids such as carbon dioxide, ammonia, water, isobutane and so on, which have no or only negligible effect on global warming, has become the most important challenge recently. Heat pump water heaters that use carbon dioxide as the refrigerant, residential-use refrigerators that use isobutane as the refrigerant and vending machines that use carbon dioxide or isobutane as the refrigerant, are already available on the market, and the possibility of using carbon dioxide (CO_2) refrigerant for air conditioners such as automotive air conditioners, packaged air conditioners, room air conditioners and others is now being considered.

In such a stream of technology development, our research group is now engaged in experimental research concerning heat transfer and flow characteristics in various heat transfer processes (forced convection cooling at supercritical pressure, boiling and evaporating heat transfer, and condensation) of CO_2 in spiral-grooved tubes with the aim of obtaining basic data to design heat exchangers for heat pump and refrigeration systems. This report introduces and analyses the results obtained by our research group. Before describing the heat transfer characteristics of CO_2, the report starts with an overview of a trans-critical heat pump cycle that uses CO_2 as a refrigerant, complemented by a presentation of the basic thermophysical properties of CO_2.

2. The basic characteristics of CO_2

The pressure / specific enthalpy diagrams of heat pump cycles (an ideal cycle with the evaporation temperature assumed at 0℃ and the outside air temperature at 40℃) that use R134a and CO_2 as refrigerants are shown respectively in Fig. 1 (a) and Fig. 1 (b). As shown in Fig.1 (a), the R134a cycle, a conventional heat pump cycle consists of a compressor, a condenser, an expansion valve and an evaporator. In this cycle, the process of heat absorption from a low-temperature heat source is effected by boiling and evaporation of refrigerant, and the process of heat rejection to a high-temperature heat source is effected by condensation of refrigerant. On the other hand, the CO_2 cycle, which is shown in Fig. 1 (b), consists of a compressor, a gas cooler, an expansion valve and an evaporator. Here, the process of heat absorption from a low-temperature source is the same as in the case of conventional refrigerant that is shown in Fig. 1 (a). However, as the critical temperature of CO_2 is 31.05 °C, which is significantly lower than that of the conventional refrigerants, the process of heat rejection to the high-temperature source is carried out at a pressure higher than the critical pressure, which causes the forced convection cooling process that does not involve condensation. For this reason, the CO_2 cycle is called a trans-critical cycle.



(a) R134a

(b) CO_2 (R744)

Fig. 1 Heat pump cycles on *P*-*h* diagrams

Tables 1 (a) and 1 (b) compare the physical properties of R134a and CO_2 on the low-pressure side and on the high-pressure side respectively. As for the physical properties on the low-pressure side in Table 1 (a), the saturation pressure of CO_2 is about twelve times higher than that of R134a, the vapour density of CO_2 is about 6.7 times higher than that of R134a, while the latent heat of CO_2 is comparable and about 1.16 times higher than that of R134a. This means that the refrigerating capacity of CO_2 per unit of piston displacement is higher than that of R134a. In addition, the specific heat at constant pressure of CO_2 is higher than that of R134a in both vapour and liquid phases. As for the liquid thermal conductivity, which primarily affects heat transfer characteristics, that of CO_2 is also higher than that of R134a. Furthermore, as for the viscosity, which strongly influences the flow characteristics, that of CO_2 is lower than that of R134a. This means that the pressure drop of CO_2 is less than that of R134a. As for the physical properties on the high-pressure side, which are shown in Table 2 (b), CO_2 reaches a supercritical state. Though it is difficult directly to compare physical properties, the pressure of CO_2 is about 6.5 times higher than the saturation pressure of R134a, and the enthalpy difference between inlet and outlet of gas cooler of CO_2 is about 1.4 times higher than the latent heat of R134a.

As mentioned above, CO_2 is a refrigerant that has excellent thermodynamic and transport properties, with much lower loads on the environment than conventional CFC refrigerants, as the ozone depletion potential of CO_2 is zero and the global warming potential is as low as 1. For comparison, the heat rejection process of R134a on the high-pressure side is by condensation, which operates at essentially constant temperature, whereas that of CO_2 is by single-phase cooling that involves a large change in temperature (the appliances that effectively utilize these properties of CO_2 are on the market as heat pump water heaters).

3. Experimental apparatus

Table 1 Comparison of physical properties of R134a and CO_2

(a) low-pressure side

		R134a (Evaporator)		CO_2 (Evaporator)	
		Saturated liquid	Saturated vapour	Saturated liquid	Saturated vapour
Pressure	MPa	0.29	0.29	3.48	3.48
Temperature	℃	0	0	0	0
Density	kg/m^3	1,295	14.43	928.1	97.32
Latent heat	kJ/kg	198.6		230.9	
Specific heat at constant pressure	kJ/(kg·K)	1.341	0.897	2.535	1.826
Viscosity	μ Pa·s	271.1	10.73	105.4	14.31
Thermal conductivity	W/(m·k)	0.09201	0.01151	0.09755	0.00679
Surface tension	N/m	0.01156	-	0.00455	-

(b) High-pressure side

		R134a (Condenser)		CO_2 (Gas cooler)	
		Saturated liquid	Saturated vapour	Outlet	Inlet
Pressure	MPa	1.682	1.682	11	11
Temperature	℃	60	60	40	100
Density	kg/m^3	1,053	87.38	685.0	214.3
Latent heat	kJ/kg	139.13		193.5*	
Specific heat at constant pressure	kJ/(kg·K)	1.660	1.387	4.074	1.617
Viscosity	μ Pa·s	124.2	13.79	54.63	23.46
Thermal conductivity	W/(m·k)	0.06609	0.01831	0.05870	0.01192
Surface tension	N/m	0.00372	-	-	-

* Specific enthalpy difference between inlet and outlet of gas cooler

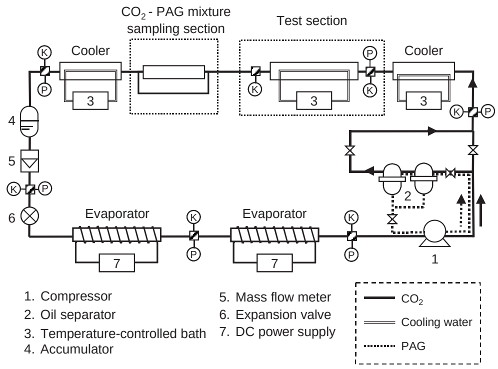


Fig. 2 Experimental apparatus

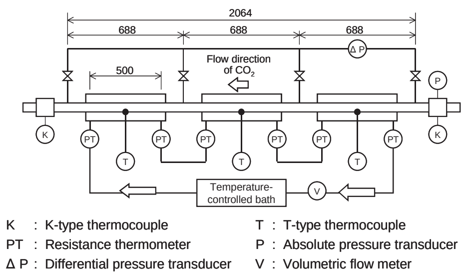


Fig. 3 Test section

Fig. 2 is a schematic diagram of the experimental model used in the experiment of cooling heat transfer of CO_2 under supercritical pressure. The experimental model is a vapour compression heat pump system that consists of a refrigerant loop, PAG oil loop and heat source water loop. Major elements of the refrigerant loop include a compressor, two oil separators, a gas pre-cooler, a heat transfer test section, a CO_2-PAG oil mixture sampling section, a gas post-cooler, an accumulator, a mass flowmeter and two electrically heated evaporators. The heat source loop is used as the cooling heat source of CO_2 in the test section.

Fig. 3 shows a schematic diagram of the test section used in the experiment of cooling heat transfer of CO_2 under supercritical pressure. The test section is a double-tube counter-flow heat exchanger. CO_2 flows in the inner tube and cooling water flows in the annular part surrounding the inner tube. The overall length of the test section is 2064 mm, consisting of three subsections, each of which is 688 mm. The effective heat transfer length of each subsection is 500 mm.

Experiments of heat transfer of CO_2 bycondensation can be done by using the experimental apparatus shown in Fig. 2 and the test section shown in Fig. 3. Experiments of boiling and evaporating heat transfer of CO_2 are also done by inserting the test section, which is identical to the one shown in Fig. 3, between the two electric heating evaporators in the refrigerant loop in Fig. 2.

4. Cooling heat transfer characteristics of CO_2 in spiral-grooved tubes in the supercritical pressure range

4.1 Experimental heat exchanger tubes

Table 2 shows the shapes and sizes of experimental heat exchanger tubes. For the experiments, three types of copper spiral-grooved tubes of different groove shapes, and one type of smooth copper tube, were used. The outer diameters of both grooved tubes and the smooth tube are 6.02 mm. As for the shapes of the spiral-grooved tubes, the fin height is 0.15-0.24 mm, the lead angle is 5-25 degrees, the number of fins is 46-52, and the area expansion ratio is 1.4-2.3. For reference, the average inner diameter "d_i" of grooved tubes is defined as the inner diameter of the smooth tube that has the same channel cross-section as the actual channel cross-section of inner surface spiral grooved tube.

In the experiments, oil carried in the CO_2 stream is fully removed by the oil separator installed at the outlet of the compressor, and the heat transfer rate of almost pure CO_2 (mass fraction of 99.93% or higher) was measured at pressures of 8 MPa and 10 MPa, mass velocity of 360-690 kg/(m^2･s), and CO_2 bulk temperature of 20-75 ^{o}C. The temperature and flow rate of cooling water was regulated to allow the heat exchange rate of the test section as a whole to reach about 1000 W.

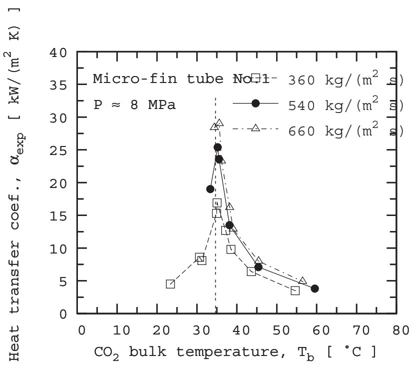
4.2 Results of experiments

Representative examples of measured values of heat transfer coefficients of the Type 1 grooved tube, at pressures of 8 MPa and 10 MPa are shown in Fig. 4 (a) and Fig. 4 (b) respectively. The horizontal axis represents the bulk temperature of CO_2, and the vertical axis represents the heat transfer coefficients, and the parameter is mass velocity. The vertical broken lines in the figures represent the pseudo-critical temperatures under the respective pressure conditions. The pseudo-critical temperature is about 34.7 at 8 MPa and about 45.0℃ at 10 MPa.

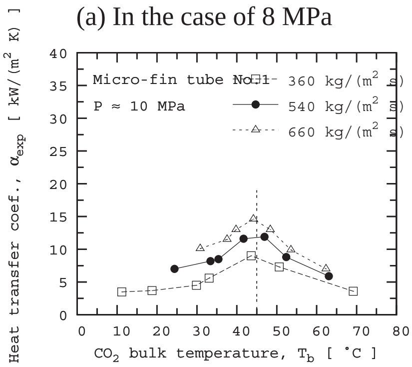
As for the measured values of heat transfer coefficients at a pressure of 8 MPa as shown in Fig. 4 (a), CO_2 bulk temperatures show the maximum values near the pseudo-critical temperature at any mass velocity. This is because the physical properties of CO_2 rapidly change around the pseudo-critical temperature, and it is considered that particularly the effect of specific heat at constant pressure has a significant impact on the heat transfer mechanism in the supercritical pressure range. The measured values of heat transfer coefficients at 10 MPa pressure, as shown in Fig. 4 (b), show the tendency of showing the maximum values near the pseudo-critical temperate at any mass velocity, as is the case at 8 MPa, which is shown in Fig. 4 (a), but the heat transfer coefficients around this temperature are lower than the 8 MPa case. This is because the specific heat at constant pressure at the pseudo-critical temperature at 10 MPa is lower than at 8 MPa.

Table 2 Specifications of experimental spiral grooved tubes and smooth tube

Type		Smooth tube	Grooved tube		
			No. 1	No. 2	No. 5
Outer diameter, d_o	[mm]	6.00	6.02	6.02	6.00
Maximum inner diameter, d_r	[mm]	4.42	4.9	4.91	5.28
Average inner diameter, d_i	[mm]	-	4.76	4.76	5.11
Height of fin, h	[mm]	-	0.15	0.18	0.24
Torsion angle, β	[deg]	-	24	5	25
Number of threads, N	[-]	-	52	46	50
Area expansion ratio, η	[-]	1	1.4	1.4	2.3



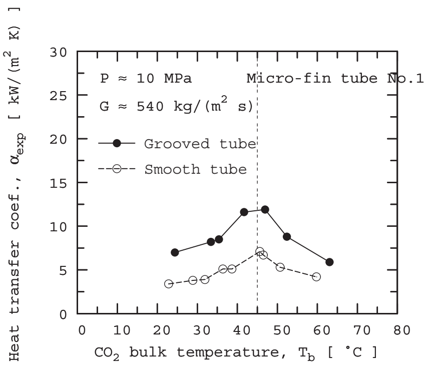
(a) In the case of 8 MPa



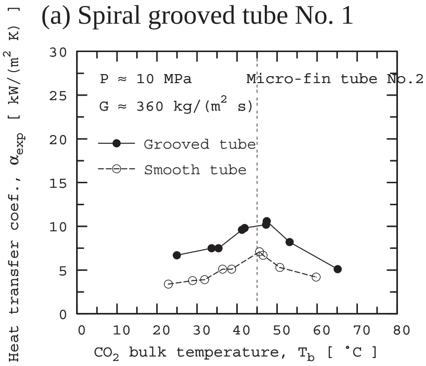
(b) In the case of 10 MPa

Fig. 4 Heat transfer characteristics of spiral grooved tube No.1

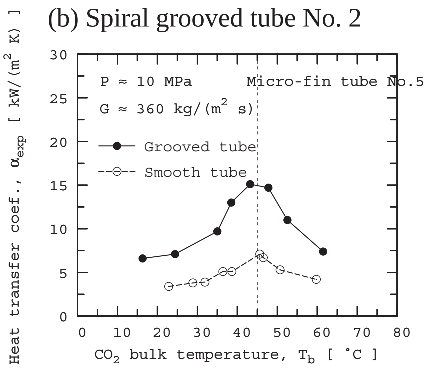
Next, in order to confirm the heat transfer enhancement effect of in-plane spiral-grooved tubes used in the experiments, the measured values of heat transfer coefficients obtained are compared with the smooth tube performance. As the representative examples of experiment results, Fig. 5 shows the measured values of heat transfer coefficients of grooved tubes and smooth tube in the case where the pressure is 10 MPa and the mass velocity is 540 kg/($m^2 \cdot s$). Figs. 5 (a), (b) and (c) show the respective results of Nos. 1, 2 and 5 spiral grooved tubes, as shown in Table 2. The heat transfer coefficient of any of the grooved tubes showed higher values than that of the heat transfer coefficient of the smooth tube over the wide range of bulk temperatures of CO_2. As compared to the smooth tube, the heat transfer enhancement rates of Nos. 1 and 2 tubes were 1.4 times, while that of No. 5 tube was twice. These results show that the heat transfer enhancement by spiral-grooved tubes is also effective in the cooling process of CO_2 in the supercritical pressure range.



(a) Spiral grooved tube No. 1



(b) Spiral grooved tube No. 2



(c) Spiral-grooved tube No. 5

Fig. 5 Comparison of heat transfer characteristics of spiral-grooved tubes and smooth tube

4.3 Correlation equation of CO_2 heat transfer in spiral-grooved tubes

Based on the results of measurement of heat transfer of CO_2 in spiral-grooved tubes in the supercritical pressure range, the following correlation equation of heat transfer was developed:

$$Nu_f = 0.036\eta^{0.58}(\cos\beta)^{-0.31} \cdot \mathrm{Re}_f^{\,0.8}\,\mathrm{Pr}_b^{\,0.6} \qquad (1)$$

where, η is the area expansion ratio and β is the lead angle of the spiral groove. Nu_f, Re_f and Pr_b are Nusselt number, Reynolds number and Prandtl number respectively, and defined as follows:

$$Nu_f = \frac{ad_i}{\lambda_f},\ \mathrm{Re}_f = \frac{Gd_i}{\mu_f},\ \mathrm{Pr}_b = \frac{\mu_b Cp_b}{\lambda_b} \qquad (2, 3, 4)$$

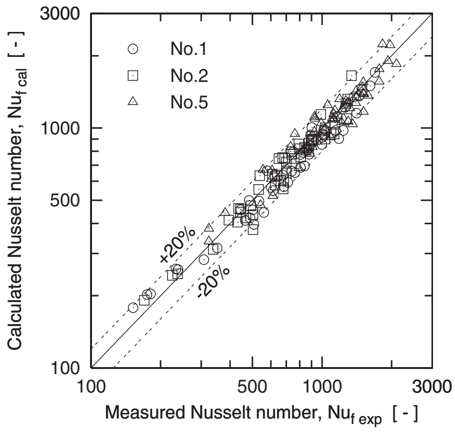


Fig. 6 Comparison of Equation (1) and experimental values

The numerical subscripts “b” and “f” respectively show that physical properties are calculated by using bulk temperatures and film temperatures shown as $(T_{wi}+T_b)/2$.

Fig. 6 shows the comparison between the predicted results using Equation (1) and experimental values. The prediction formula is consistent with most of the data of this experiment within a spread of ± 20 %.

5. Boiling and evaporating heat transfer characteristics of CO_2 in spiral-grooved tubes in the subcritical pressure range

5.1 Experimental heat exchanger tubes

Table 3 shows the specifications of five types of copper spiral-grooved tubes and one type of copper smooth tube, which were used in the boiling and evaporating heat transfer experiments of CO_2. The range of groove shape parameters of experimental spiral-grooved tubes is the same as that of the tubes used in the cooling experiment in the supercritical pressure range.

5.2 Results of experiments

Fig. 7 shows the representative examples of results of measurement of the heat transfer coefficient “α,” heat flux “q,” quality “x” in the case where the pressure in the smooth tube S1 is changed. In this figure, the horizontal axis represents quality, the vertical axis on the left represents heat transfer coefficients (represented by filled dots), the vertical axis on the right represents heat fluxes (represented by open rings). In the smooth tube, it can be known that the higher the pressure increases, the higher the heat

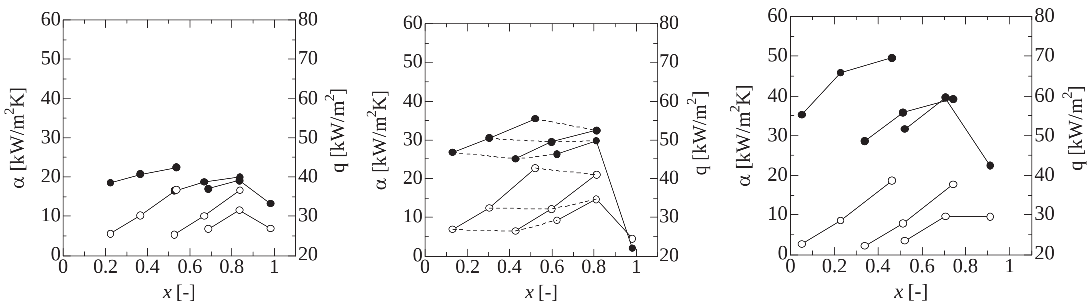
Table 3 Specifications of experimental spiral grooved tubes and smooth tube

Type		Smooth tube	Grooved tube				
			G1	G2	G3	G4	G5
Outer diameter, d_o	[mm]	6.00	6.02	6.02	6.02	6.02	6.00
Maximum inner diameter, d_r	[mm]	4.42	4.9	4.91	5	5.3	5.28
Average inner diameter, d_i	[mm]	-	4.76	4.76	4.87	5.12	5.11
Height of fin, h	[mm]	-	0.15	0.18	0.18	0.24	0.24
Torsion angle, β	[deg]	-	24	5	24	10	25
Number of threads, N	[-]	-	52	46	35	55	50
Area expansion ratio, η	[-]	1	1.4	1.4	1.4	2.3	2.3

transfer coefficient, and that when heat fluxes increase, heat transfer coefficients also increase at any pressure. As a result, it can be seen that the nucleate boiling has a strong impact on the heat transfer characteristics of CO_2. Moreover, in the range where the quality is 0.8 or higher, heat fluxes and heat transfer coefficients rapidly decrease, which is considered attributable to dryout. Though it is not represented graphically, if mass velocity increases, this phenomenon becomes obvious, and the quality that causes dryout decreases.

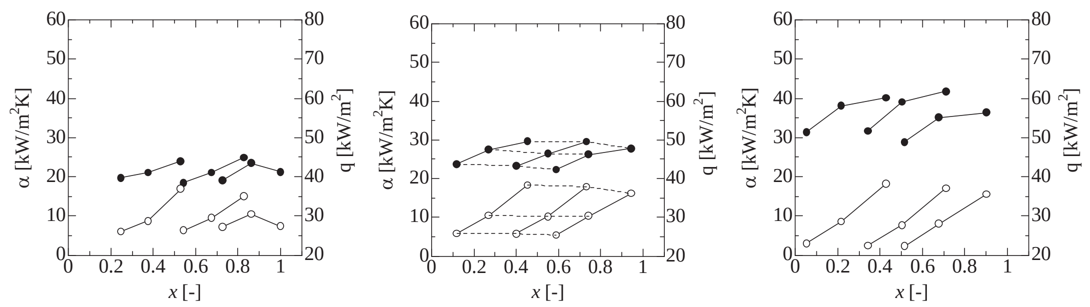
Fig. 8 shows the representative examples of results of measurement of the heat transfer coefficient“α,” heat flux “q,” quality “x” in the case where the pressure in the grooved tube G1 is changed. The symbols in the figure are the same as those in Fig. 7. Here, the heat transfer coefficients of grooved tubes are defined as the in-plane area criterion of smooth tube that has the average diameter as its inner diameter. From this figure, it can be seen that the higher the pressure increases and the more the heat fluxes increase, the more the value of heat transfer coefficient increases in the grooved tubes. This tendency is the same as in the case of the smooth tube; it shows the significant effect of nucleate boiling on heat transfer characteristics. On the other hand, the decrease in heat transfer coefficients and heat fluxes due to dryout do not clearly manifest themselves in the grooved tubes. This is because it is considered that the value of quality that causes dryout due to the effect of spiral grooves becomes higher than that of the smooth tube.

Fig. 9 shows the comparison of heat transfer coefficients between the grooved tube G1 (the height of the fin is 0.15 mm and the number of fins is 52) and the grooved tube G3 (the height of the fin is 0.18 mm and the number of fins is 35), of which the area expansion ratios and torsion angles are almost the same. At any pressure, the heat transfer coefficient of the grooved tube G1 is less than that of the grooved tube G3. Moreover, the lower the pressure decrease, the greater the difference in heat transfer coefficients between the G1 grooved tube and the G3 grooved tube becomes. This is because it is considered that if the pressure decreases, the vapour velocity increases and the liquid films formed in the grooves become thinner, and that if the fin becomes higher, the liquid films formed in the grooves become thinner. Moreover, as the number of fins of the grooved tube G1 is larger than that



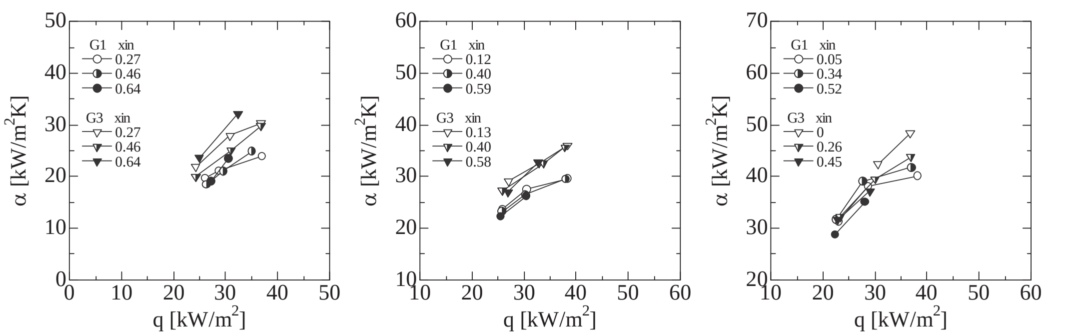
(a) P = 3 MPa, G = 363 kg/m^{2}s (b) P = 4MPa, G = 362 kg/m^{2}s (c) P = 5 MPa, G = 362 kg/m^{2}s

Fig. 7 Results of measurement of heat transfer coefficients of smooth tube S1



(a) P = 3 MPa, G = 364 kg/m^{2}s (b) P = 4 MPa, G = 363kg/m^{2}s (c) P = 5 MPa, G = 366 kg/m^{2}s

Fig. 8 Results of measurement of heat transfer coefficients of grooved tube G1



(a) P = 3 MPa, G = 360 kg/m^{2}s (b) P = 4 MPa, G = 360 kg/m^{2}s (c) P = 5 MPa, G = 360 kg/m^{2}s

Fig. 9 Relations between heat transfer coefficients and heat fluxes
(comparison between G1 and G3)

of the G3 grooved tube, it can be said that the effect of the height of fin on heat transfer coefficients is greater than that of the number of fins, if the area expansion ratios and torsion angles are the same.

Though they are not represented graphically, the effects of torsion angles and area expansion ratios were also investigated. According to the summary of the results, in the process of boiling and evaporation of CO_2, the increases in the area expansion ratio, the number of fins and the height of fin enhance heat transfer in the boiling and evaporation process, and it can be known that the angle of torsion does not have much effect on enhancement of heat transfer.

5.3 Correlation equation of CO_2 heat transfer in spiral grooved tubes

An attempt was made to develop the correlation equation of the boiling and evaporative heat transfer coefficients in spiral-grooved tubes with reference to the correlation equation of Takamatsu *et al.*[2] of the boiling heat transfer of CFC refrigerants in tubes and the correlation equation of Goto *et al.*[3] of the single-phase turbulent flow heat transfer coefficients in grooved tubes.

The heat transfer coefficient α_{real} based on the actual area can be expressed by the sum of the forced convection heat transfer coefficient $(\alpha_{cv})_{real}$ and the nucleate boiling heat transfer coefficient $(\alpha_{nb})_{real}$ That is:

$$\alpha_{real} = (\alpha_{cv})_{real} + (\alpha_{nb})_{real} \tag{5}$$

where $(\alpha_{cv})_{real}$ and $(\alpha_{nb})_{real}$ that we developed here are shown in Table 4. In this table, as the correlation equation of Goto et al. for single-phase turbulent flow heat transfer coefficients, which was used to calculate $(\alpha_{cv})_{real}$ odd, is correlated by the heat transfer coefficient calculated based on the maximum inner diameter, it must be converted into the heat transfer coefficient based on the actual heat transfer area, and it should be noted that the value of numerical subscript "real" is calculated based on the actual heat transfer area and that of "max" is based on the maximum inner diameter. As for constants C_1 and C_2 in the equation related to $(\alpha_{nb})_{real}$, the heat transfer characteristics of five types of grooved tubes from G1 to G5 could best be correlated with each other if $C_1=26\times10^{-5}$ and $C_2=2.7$.

For reference, with the grooved tube G1 as a representative example, the relations between the quality of the predicted values and experimental values calculated by the correlation equation of heat transfer are shown in Fig. 10. The open ring symbol in the figure represents an experimental value; the solid lines represent predicted values of heat transfer coefficients, and the broken lines represent the contributions of nuclear boiling heat transfer in the predicted values of heat transfer coefficients. As for the predicted values of heat transfer coefficients, the contribution of nuclear boiling heat transfer accounts for the most part of the region, and the contribution of forced convection heat transfer accounts for little in the low quality region. The contribution of nuclear boiling decreases in proportion to the increase in quality. At the pressure of 3 MPa, if the quality is

Table 4 Correlation of boiling and evaporative heat transfer of CO_2 in spiral grooved tubes

(a) Contribution of forced convection heat transfer: $(\alpha_{cv})_{real}$

$$(\alpha_{cv})_{real} = F(\alpha_1)_{real}$$

where,

$$F = 1 + 2.0X_{tt}^{-0.88},\ (\alpha_1)_{real} = \frac{(\alpha_1)_{eq}}{\eta_{eq}} = \frac{(\alpha_1)_{max}}{\eta_{max}} = \frac{(\alpha_1)_{max}}{\eta_{eq}\cdot(d_i/d_{max})},$$

$$(\alpha_1)_{max} = 0.031\,\mathrm{Re}_1\,\mathrm{Pr}_1^{1/3}\,\eta_{max}^{0.8}(\cos\beta)^{-1.3}\left\{1+(10p/d_{max})^{2.3}\right\}\left(\frac{\lambda_1}{d_{max}}\right),\ \mathrm{Re}_1 = \frac{G(1-x)d_{max}}{\mu_1}$$

(b) Contribution of nucleate boiling heat transfer: $(\alpha_{nb})_{real}$

$$(\alpha_{nb})_{real} = K^{0.745}\,S(\alpha_{pb})_{real}$$

where,

$$K^{0.745} = \frac{1}{1+0.768\eta+0.351\eta^2+0.347\eta^3+0.131\eta^4},\ \eta = \frac{(\alpha_{cv})_{real}}{S\cdot(\alpha_{pb})_{real}},\ S = \frac{1-e^{-\zeta}}{\zeta},\ \xi = \frac{D_b\cdot(\alpha_{cv})_{real}}{\lambda_1}$$

$$D_b = C_1\left(\frac{\rho_1}{\rho_v}\frac{Cp_1}{\Delta h}T_{sat}\right)^{1.25}\left(\frac{2\sigma}{g(\rho_1-\rho_v)}\right)^{0.5},\ (\alpha_{pb})_{real} = C_2\times207\left(\frac{\lambda_1}{D_{be}}\right)\left(\frac{q_{real}\,D_{be}}{\lambda_1 T_{sat}}\right)^{0.745}\left(\frac{\rho_v}{\rho_1}\right)^{0.581}\mathrm{Pr}_1^{0.533}$$

$$d_{be} = 0.51\left\{\frac{2\sigma}{g(\rho_1-\rho_v)}\right\}^{0.5}$$

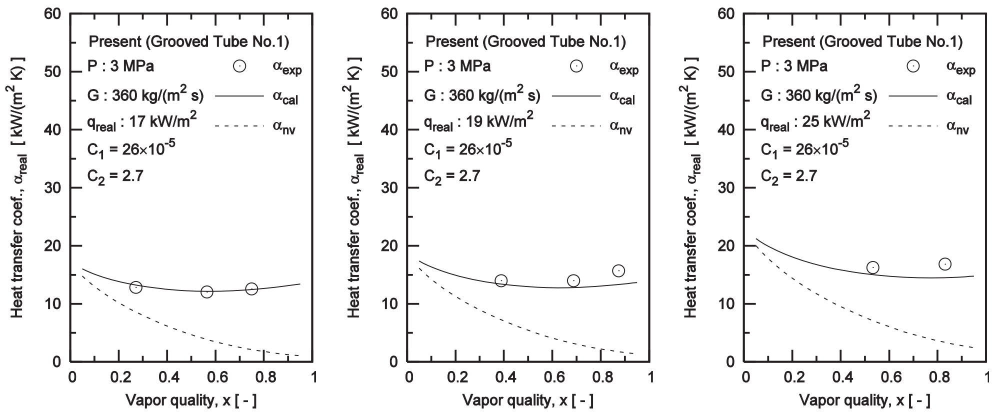


Fig. 10 Relations between heat transfer coefficients and quality (comparison between experimental values and correlation equation of heat transfer)

only about 0.8, the contribution of nuclear boiling is very small.

6. Conclusions

This paper describes our group's experimental research concerning the forced convection heat transfer of CO_2 in spiral-grooved tubes in the supercritical pressure range, and the boiling and evaporative heat transfer of CO_2 in spiral-grooved tubes in the subcritical pressure range. For thermal design of CO_2 heat exchangers, it is also necessary to understand the condensation heat transfer characteristics in the subcritical pressure range, and it is also necessary to understand the pressure drop characteristics in the respective forms of heat transfer. It is also desirable to determine the void fraction characteristics in the gas-liquid two-phase range that involves boiling, evaporation and condensation. In the session, these points are also introduced.

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Exergy Analysis: Guidance to Efficient Heating Systems using Air/Water Heat Pumps

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This report describes investigation of the behaviour of air/water heat pumps over a wide range of operating conditions, using the exergy method. An analytical model based on the relevant thermophysical properties has been developed to simulate the system at steady state conditions and to evaluate the exergetic efficiency. The exergy losses of the main process components and the heating delivery and distribution system are discussed with mathematical equations demonstrating the influence of important variables. Theoretical and experimental analysis indicate that the exergetic efficiency of common heat pumps falls as the ambient temperature increases. The reason for this is the heat pump characteristic: the less heat that is required, the more it delivers. In common heat pumps this is compensated by On/Off operation. This means that a substantial increase in efficiency requires continuous operation with power control of the compressor and, where necessary, of the fan. The generated heat flow of the heat pump must be continuously matched to the required heat flow.

Introduction

The use of air/water heat pumps as heating systems for residential buildings has grown rapidly over the last decades. Air/water heat pumps require only low investments, are easy to install and are reliable in their operation. They are not subject to official approval and in the future will run with less noise as a result of improved flow geometries and optimised fans. However, one of the most important prerequisites for increased use of heat pumps lies in the substantial increase in their efficiency.
Energy balances are necessary for the assessment of the heat pump process (and many other thermal processes, too), but alone are not sufficient. The second law of thermodynamics describes the quality of thermodynamic processes. It states that all technical processes are irreversible. In thermodynamics, we cannot speak of energy losses (in accordance with the 1st law), but may speak of irreversible increases in entropy. Instead of evaluating the quality of thermodynamic processes in terms of abstract entropy balances, exergy analysis is suitable for our purposes. A systematic representation of the thermodynamics of heating has been supplied by Baehr [1] in two contributions. He shows that exergy is an extremely practical and descriptive way of evaluating technical processes involving thermodynamic cycles and heat transfer. Its results are easy to follow and immediately demonstrate possibilities for improvement in thermodynamic efficiency because, using exergy analysis, exergy losses can be determined for both the complete system and for its subsystems. For heating and cooling systems, exergy losses always mean that additional mechanical work is necessary in comparison with ideal process in which no exergy losses occur.

The WEXA [2] study was started as a result of the extensive LOREF [3] research project, in which it was increasingly realised that, in order to achieve a significant increase in the efficiency of air/water heat pumps, all sub-processes and the heating system must be included in the analysis.

In this study, exergy balances will be calculated for the whole heat pump process and all sub-processes. In the order to do this, we aim clearly to show the exergy losses mathematically in relation to the relevant process variables and their effect on other sub-processes. For this purpose, suitable approximations will be made, e.g. where permissible by linearisation or by series expansion.

The relevant factors of the exergy losses in heat pumps

The basics factors influencing the exergy losses of the heat pumps are the temperature gradients for heat transfer in the evaporator ΔT_E and condenser ΔT_C as well as the isentropic compressor efficiency η_s. These affect each other mutually and so reduce the exergetic efficiency of the heat pump. Moreover, all exergy losses are dependent on the temperature lift ΔT_{Lift}.

For air/water heat pumps with On/Off control, the intermittently generated heating (water) temperature T_H, at ambient temperatures above the design point, is around ΔT_H higher than the heating temperature T_H^* that is continuously required by the building. The heat at temperature T_R supplied to the room is provided by the heat delivery system from a continuously required heating temperature T_H^* with a temperature gradient ΔT_R. The temperature lift ΔT_{Lift}, defined as the difference be-

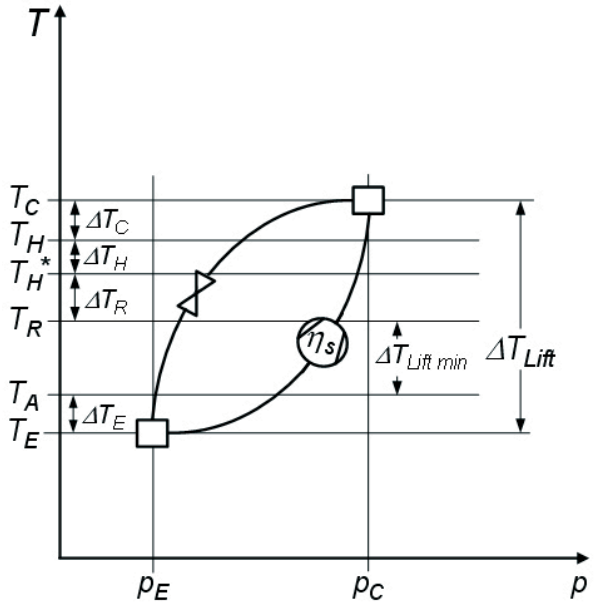

Figure 1: Temperatures, temperature gradients and pressures in the heat pump process

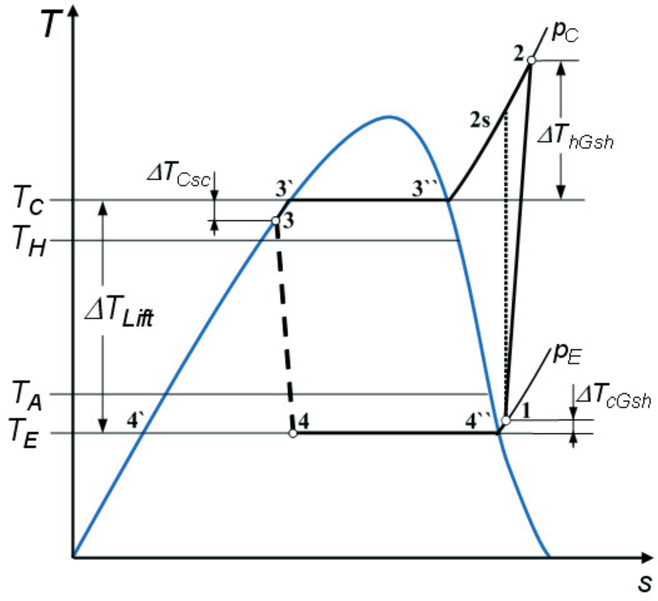

Figure 2: T,s diagram with state points noted (cGsh: cold gas superheating; hGsh: hot gas superheating; Csc: condensate subcooling)

tween the condensation temperature T_C and the evaporation temperature T_E, is greater in comparison with the minimum (ideal) temperature lift $\Delta T_{Lift\ min}$ by the temperature gradients ΔT_E, ΔT_C and ΔT_R as well as by the discrepancy ΔT_H between required and generated heating temperature.

In Figure 2 shows the heat pump process in the form of a detailed representation of the individual state points of the heat pump's working fluid in the T,s diagram. Evaporation and condensation are taken as being isobar-isotherm. Vapour superheating ΔT_{cGsh} occurs In the evaporator. In the compressor, vapour superheating ΔT_{hGsh} (with reference to T_C) occurs along with condensate subcooling ΔT_{Csc}.

Exergy losses of the heating system with heat pump

a) Exergy losses of the sub-processes of the heat pump working fluid

The exergy losses in a simple, single-stage compression heat pump are: $\dot{E}_{LCp}$ in the compressor, $\dot{E}_{LC}$ in the condenser, $\dot{E}_{LEx}$ in the expansion valve and $\dot{E}_{LE}$ in the evaporator. The exergy losses in the compressor and the expansion valve are caused by dissipative pressure drops. On the other hand, the exergy losses during heat transfer in the evaporator and condenser do not originate in the working fluid but on account of the temperature gradients ΔT_E and ΔT_C required for heat transfer. Pressure losses on the working fluid side due to flow in piping and fittings as well as in the evaporator and condenser are negligibly small and are not considered, and nor are the drive losses of the compressor or the fan power considered here.

Physical equations (cf. Table 1) were developed in order to show the quantitative effect of the relevant factors on exergy losses and exergetic efficiency. By making approximations, relatively clear equations can be established that can easily be discussed without accuracy being affected. These results are useful to the development engineer working to improve future heat pumps. The equations are not derived in detail here; they will, however, be discussed in each case (the detailed derivations can be found in WEXA [2]).

We refer to an adiabatic, single stage compressor. The isentropic efficiency η_s is particularly important for exergy loss in the compressor. The exergy loss is approximately proportional to the specific enthalpy of evaporation r of the working fluid and to the temperature lift ΔT_{Lift}, in which ΔT_E, ΔT_C, ΔT_R and ΔT_H are included. Also, the effect of η_s is immediately clear. The exergy loss in the expansion valve is proportional to c_{pl} (specific heat capacity of the boiling liquid working fluid) and to the square of the temperature lift. It can also be reduced by increasing the condensate subcooling ΔT_{Csc} [2].

The exergy loss in the evaporator is proportional to the heat flow $\dot{Q}_E$ transferred, to the temperature gradient ΔT_E as well as being inversely proportional to the square of the average evaporation temperature $\overline{T}_E$. The exergy loss in the condenser is proportional to the generated heating capacity $\dot{Q}_H$, the temperature gradient ΔT_C for heat transfer as well as being inversely proportional to the square of the generated heating temperature T_H.

Table 1: Equations for the evaluation of the exergy losses of the sub-processes of the heat pump working fluid

Exergy loss	Equation
Compressor	$\dot{E}_{LCp} = \dot{m}_r \cdot T_A \cdot r \cdot \frac{\Delta T_{Lift}}{T_E \cdot (T_E + \Delta T_{Lift})} \cdot \left[\frac{1}{\eta_s} - 1\right] \approx \dot{m}_r \cdot r \cdot \frac{1}{COP_{rev\,i}} \cdot \left[\frac{1}{\eta_s} - 1\right]$
Expansion valve	$\dot{E}_{LEx} \approx \dot{m}_r \cdot T_A \cdot c_{pl} \cdot \frac{1}{2} \cdot \left(\frac{\Delta T_{Lift}^2}{T_E^2 + T_E \cdot \Delta T_{Lift}}\right) \approx \dot{m}_r \cdot c_{pl} \cdot \frac{\Delta T_{Lift}}{2 \cdot COP_{rev\,i}}$
Evaporator	$\dot{E}_{LE} = \dot{Q}_E \cdot T_A \cdot \frac{\Delta T_E}{T_E \cdot (T_E + \Delta T_E)} \approx \dot{Q}_E \cdot T_A \cdot \frac{\Delta T_E}{T_E^2} \approx \dot{Q}_E \cdot \frac{\Delta T_E}{T_A}$
Condenser	$\dot{E}_{LC} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H \cdot T_C} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H \cdot (T_H + \Delta T_C)} \approx \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H^2}$

b) Exergy losses in the heat distribution and delivery system

If the heating water circuit (heat distribution and delivery system) is also considered, two further exergy losses occur: in the heat distribution system $\dot{E}_{LHS}$ and in the heat delivery system $\dot{E}_{LR}$.

The exergy loss in the heat distribution system results from the discrepancy caused by the generated heat flow being higher than required and, correspondingly, the generated heating temperature T_H being higher than the required T_H^*. This exergy loss is, for a generated heating capacity of $\dot{Q}_H$, directly proportional to the discrepancy between the generated and the required heating temperatures ΔT_H (cf. Table 2). The cause of this exergy loss lies in the unfavourable operating characteristic of an air/water heat pump with On/Off control. For the purpose of thermodynamic evaluation, this exergy loss must therefore be attributed to the air/water heat pump. On the other hand, the heat pump must not be charged with causing the exergy loss $\dot{E}_{LR}$ in the heat delivery system; the heating delivery system itself must take responsibility for this loss. $\dot{E}_{LR}$ is proportional to the temperature gradient ΔT_R for heat transfer to the room (cf. Table 2). The temperature gradient ΔT_R, and therefore the exergy loss $\dot{E}_{LR}$, can be influenced during the planning and design phase of the heat delivery system. The aim here is to minimise the temperature gradient by using low required heating temperatures and thus, additionally, reducing the temperature lift.

Table 2: Equations for the evaluation of the exergy losses in the heat distribution and delivery system

Exergy loss	Equation
Heat distribution system	$\dot{E}_{LHS} = \dot{Q}_H \cdot T_A \cdot \frac{T_H - T_H^*}{T_H \cdot T_H^*} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_H}{T_H \cdot T_H^*}$
Heat delivery system	$\dot{E}_{LR} = \dot{Q}_H \cdot T_A \cdot \frac{T_H^* - T_R}{T_H^* \cdot T_R} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_R}{T_H^* \cdot T_R}$

Operating characteristic of air/water heat pumps with on/off control

The operating characteristics, the exergy loss ratios and the exergetic efficiencies presented in figures 3, 4 and 5 are valid for an air/water heat pump with on/off control with 5.4 kW nominal heating capacity at -10 °C (detailed specifications can be found in WEXA [2]). The simulations were carried out, however, in great detail using a simulation program for air/water heat pumps which was developed and documented in LOREF [3].

With increasing ambient temperature, the heating capacity required by the building decreases. However, the behaviour of common heat pumps, whose compressor is operated at constant rotational speed, is exactly the opposite: The lower the heating capacity and heating temperature required by the building, the higher the heating capacity and heating temperature actually generated (cf. Figure 3 and 4).

This behaviour results in common heat pumps working in intermittent mode (On/Off control). Moreover, the temperature gradients for heat transfer in the evaporator and condenser increase; also the discrepancy between required and generated heating temperature in the heat distribution system increases with increasing ambient temperature, thus leading to bad part-load efficiencies, lower seasonal performance factors and lower annual average exergetic efficiencies.

Definition of exergetic efficiency

In order to evaluate the quality of a heat pump thermodynamically correctly, we introduce the exergetic efficiency η_{ex} : the exergy flow of heat for heating purposes, which can be calculated as a function of the temperature of a given heat flow, and is related to the internal compressor power P_i . (The mechanical and electrical efficiencies of the compressor are not considered here, and nor is the fan power.) The different exergetic efficiencies are summarised in Table 3.

The internal exergetic efficiency is used to judge the thermal efficiency of the heat pump's working fluid

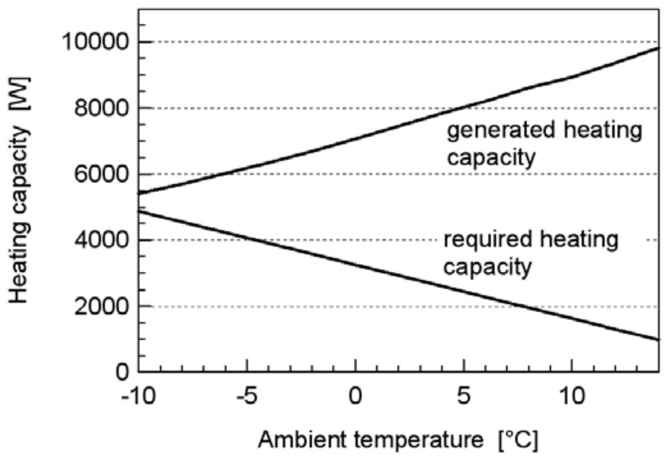

Figure 3: Generated and required heating capacities

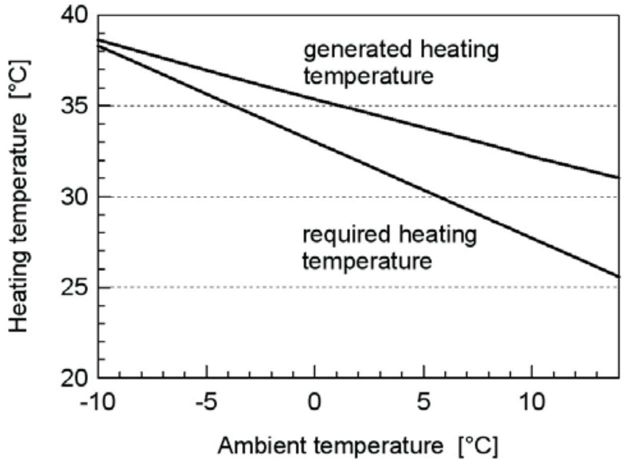

Figure 4: Generated and required heating temperature

circuit with fluid temperatures T_E in the evaporator and T_C in the condenser. It considers the total temperature lift $\Delta T_{Lift} = T_C - T_E$ and therefore includes the temperature gradients in the evaporator and condenser for heat transfer as well as the discrepancy between the required and generated heating temperature but not, however, their exergy losses. Consequently, it is influenced by the exergy losses in the expansion valve and compressor only in accordance with the pressure ratio corresponding to the total temperature lift.

For the external exergetic efficiency of the generated heating temperature T_H, the exergy of the generated heating capacity $\dot{Q}_H$ is referred to the generated temperature of the heating water T_H in the condenser. In addition to the internal exergetic efficiency, it considers the exergy losses caused by the temperature gradients for heat transfer in the evaporator and condenser.

The external exergetic efficiency of the required heating temperature serves as a third means of judgement: in this case, the exergy of the generated heating capacity is related to the heating temperature T_H^* continuously required by the building . In addition to the external exergetic efficiency of the generated heating temperature, it considers the exergy losses caused by the discrepancy between the required and generated heating temperature (exergy losses in the heat distribution system).

A fourth evaluation of the external exergetic efficiency of a heating system with a heat pump (exergetic efficiency of the heated building) is that referred to the ambient temperature T_A along with the room temperature T_R. In addition to external exergetic efficiency of the required heating temperature, this also takes the exergy losses occurring in the heat delivery system (e.g. underfloor heating) into consideration. These losses are not to be attributed to the heat pump.

Results of the exergy analysis for air/water heat pumps with On/Off control

Figure 5 shows the energy/exergy flow diagrams, including the exergy loss ratios for the entire heating system, with an air/water heat pump with On/Off control. Three different operating conditions (ambient temperatures) are illustrated. The exergy loss ratios (exergy loss flow with reference to the internal compressor power) show directly the subtractive effect of the exergy losses on the exergetic efficiency (cf. Table 3).

Table 3: Equations for the evaluation of exergetic efficiencies

Exergetic efficiency	Equation
Internal exergetic efficieny	$\eta_{ex\,i} = \frac{\dot{Q}_H}{P_i}\cdot\frac{T_C - T_E}{T_C} = COP\cdot\eta_{C\,i} = 1 - \frac{\dot{E}_{LCp}}{P_i} - \frac{\dot{E}_{LEx}}{P_i}$
External exergetic efficiency of the generated heating temperature	$\eta_{ex\,e} = \frac{\dot{Q}_H}{P_i}\cdot\frac{T_H - T_A}{T_H} = COP\cdot\eta_{C\,e} = 1 - \frac{\dot{E}_{LCp}}{P_i} - \frac{\dot{E}_{LEx}}{P_i} - \frac{\dot{E}_{LE}}{P_i} - \frac{\dot{E}_{LC}}{P_i}$
External exergetic efficiency of the required heating temperature	$\eta_{ex\,e}^* = \frac{\dot{Q}_H}{P_i}\cdot\frac{T_H^* - T_A}{T_H^*} = COP\cdot\eta_{C\,e}^* = 1 - \frac{\dot{E}_{LCp}}{P_i} - \frac{\dot{E}_{LEx}}{P_i} - \frac{\dot{E}_{LE}}{P_i} - \frac{\dot{E}_{LC}}{P_i} - \frac{\dot{E}_{LHS}}{P_i}$
External exergetic efficiency of heating system with a heat pump	$\eta_{ex\,HS} = \frac{\dot{Q}_H}{P_i}\cdot\frac{T_R - T_A}{T_R} = COP\cdot\eta_{C\,HS} = 1 - \frac{\dot{E}_{LCp}}{P_i} - \frac{\dot{E}_{LEx}}{P_i} - \frac{\dot{E}_{LE}}{P_i} - \frac{\dot{E}_{LC}}{P_i} - \frac{\dot{E}_{LHS}}{P_i} - \frac{\dot{E}_{LR}}{P_i}$

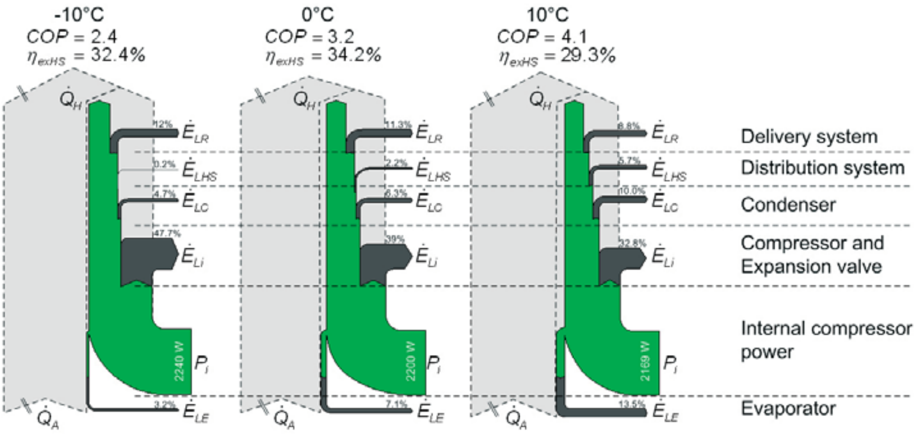

Figure 5: Energy/exergy flow diagram for an air/water heat pump with on/off control (the energy flow diagrams are not to scale)

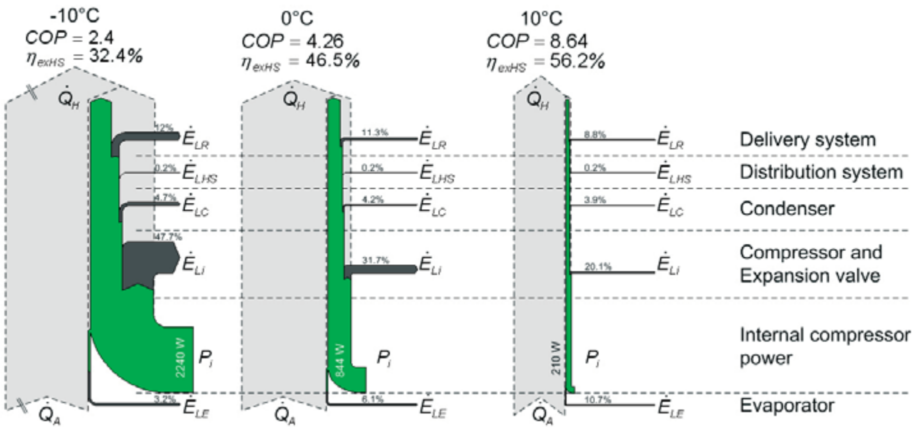

Figure 6: Energy/exergy flow diagram for an air/water heat pump with continuous capacity control (the energy flow diagram for -10°C is not to scale)

The source of the unimpressive efficiency of common air/water heat pumps with On/Off control is their unfavourable operating characteristic, which results from the characteristics of compressors running at constant speed. Admittedly, with increasing ambient temperature, the coefficient of performance (COP) of such heat pumps does increase, but the exergetic efficiency, as a thermodynamically correct evaluation factor, decreases. The thermodynamics of heating would, on the other hand, make an increase of exergetic efficiency possible. With increasing ambient temperature, the required heating capacity for a building decreases. The behaviour of a heat pump with On/Off control is the opposite. The lower the heating capacity and the heating temperature required by the building (i.e. with increasing ambient temperatures), the higher the heating capacity and temperature generated. This behaviour has the effect of causing temperature gradients for heat transfer in the evaporator and condenser to increase with increasing ambient temperature. This leads to a clear discrepancy between required and generated heating temperature. The actual temperature lift generated decreases less strongly in comparison with the ideal temperature lift, and the exergetic efficiency itself is then considerably reduced.

Air/water heat pump with continuous capacity control

It has been stated above that the discrepancy between required and generated heating capacity and temperature allows only a moderately high efficiency to be achieved for heat pumps with On/Off control. In order to allow temperature gradients for heat transfer in the evaporator and condenser to be reduced, instead of increasing, in the case of partial

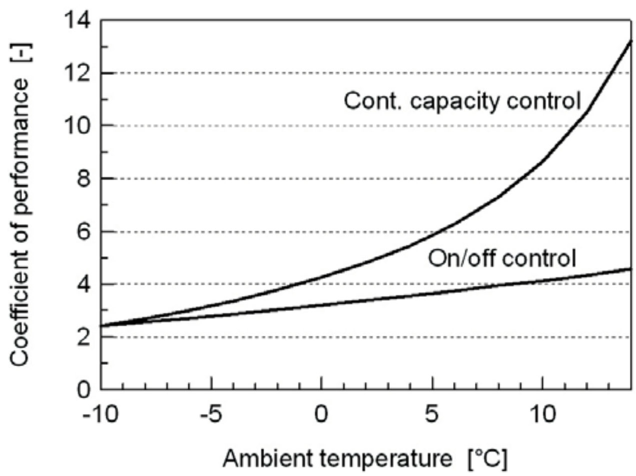

Figure 7: Coefficient of performance with and without continuous power control of the compressor and the fan

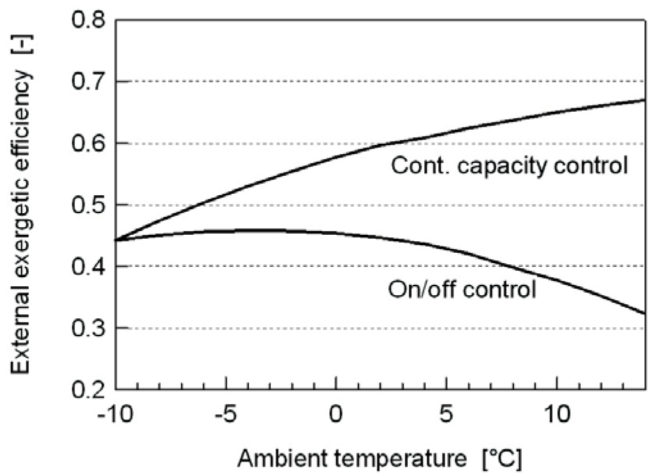

Figure 8: External exergetic efficiency of the required heating temperature with and without continuous power control of the compressor and the fan

load, i.e. for increasing ambient temperature, any discrepancy between required and generated heating capacity and heating temperature must be avoided. Consequently, the air/water heat pump should not be operated in On/Off mode, but be operated continuously - except during any defrosting processes necessary. Different strategies are available for matching the generated heating capacity to that required: the compressor can, for example, be equipped with speed control. Even just by using continuous power control of the compressor, much better coefficients of performance and higher exergetic efficiency are achieved in comparison with the On/Off control strategy. The best efficiencies can be achieved by using continuous power control of the compressor and the fan (cf. Figure 6, 7 and 8). Calculations in the WEXA study [2] show that, when using this control strategy, the seasonal performance factor can be approximately doubled in comparison with On/Off control. The electrical and mechanical efficiencies of the compressor and the fan are not considered here. Detailed findings under consideration of the mechanical efficiencies of the compressor and the fan are shown in WEXA [2].

Conclusions

Exergy is an extremely practical and descriptive way of evaluating technical processes involving thermodynamic cycles and heat transfer. Using exergy analysis, technical and economic evaluations and optimisations can be carried out in the best possible way. In this study we have concentrated on the heat pump and on the heating system. In order to attain high efficiencies, integral and optimised solutions are needed. A prerequisite for this is, in addition to highly efficient heat pumps and heating systems, well thought out architectural designs and consideration of the various physical qualities of buildings. In addition, the building, its heating system and the heat pump employed must be matched to each other in the best possible way. This calls for the co-operation of architects and building services engineers as well as heat pump manufacturers. Wellig et al. [4] present the same findings for building cooling under consistent appliance of exergy analysis.

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2009

Purdue International Compressor Conference
14 - 18 July, W. Lafayette, IN, USA

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

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

DENEX® 2009
2 - 3 March
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

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

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

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

Ground Source Live!
12 March
Birmingham, UK
www.groundsourcelive.com/Home.asp

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

Heating, Air-Conditioning, Refrigeration and Fluid Exhibition
18 - 21 March
Koyang City, Korea
www.harfko.com

ProHeatPump Conference on heat pumps
26 March
Varna, Bulgaria
Contact: Mrs. Elena Simeonova
E-mail: energy @ ubbsla.org
www.proheatpump.eu

EnReg Energia Regenerabila® 2009
26 - 28 March
International trade fair and conference for renewable energies and energy efficient building and renovation.
Arad, Romania
E-mail: redaktion @ energie-server.de

Countdown to a Sustainable Energy Future...Net-Zero and Beyond
ASHRAE Conference
29 - 31 March
San Fransisco, USA
www.ashrae.org/events/page/2094

RENEXPO® Central Europe 2009
16 - 18 April
International trade fair and conference for renewable energies and energy efficient building and renovation.
Budapest, Hungary
E-mail: redaktion @ energie-server.de
www.renexpo-budapest.com

1st IIR Cold Chain Conference
27 - 29 April
Singapore, Singapore
IIR Commissions C2, D1, D2
Contact: brendan @ airah.org.au
http://www.airah.org.au/iir-coldchain09.asp

Ammonia Refrigeration Technology
7 - 9 May
IIR Commissions B2 with B1, D1
Ohrid, Macedonia
www.mf.edu.mk/web_ohrid2009/ohrid-2009.html

3rd International Conference on Magnetic Refrigeration at Room Temperature
12 - 15 May
IIR Commissions A1, B2, E2
Des Moines, United States
Contact: vitkp @ ameslab.gov
www.mf.edu.mk/web_ohrid2009/ohrid-2009.html

Geo-Hp 2009
17 - 20 May
Europe & USA/Canada Joint Conference on Recent Advances in Geo-Source and Heat Pumps Technologies
Budapest, Hungary
www.bmf.hu/conferences/Geohp2009

5th International Workshop on Energy and Environment of Residential Buildings and 3rd International Conference on Built Environment and Public Health (EERB-BEPH 2009)
27 - 29 May
Guilin, Guangxi Province, China
www.chinahvacr.com/eerb/

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International Energy Agency

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

IEA Heat Pump Programme

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

Vision

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

The Programme conducts high value international collaborative activities to improve energy 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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Institute of Sweden

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Fax: +46 33 13 19 79
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heat pump
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