

IEA Heat Pump Centre

NEWSLETTER



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Front Cover: Novel horizontal spiral ground heat exchanger developed by National Research Council Canada (see article Workshop Ground-Source Heat Pumps, pp. 16-18).

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Cooperation and Development (OECD) to implement an International Energy Programme.

A basic aim of the IEA is to foster cooperation among the 21 IEA participating countries to increase energy security through energy conservation, development of alternative energy sources and energy research, development, and demonstration (RD&D). This is achieved in part through a programme of collaborative RD&D consisting of 42 Implementing Agreements, containing a total of over 80 separate energy RD&D projects. This publication forms one element of this programme.

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Editorial

"Technical Advancements in Heat Pumps, Aiming at Improved Performance and Reliability, and New Cycles" is the topic of this Newsletter.

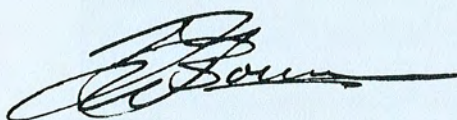
Improved performance and reliability are big challenges and very often advocated as the tools for a successful heat pump market penetration. Indeed, in many countries with a strong awareness of the strength and benefits of heat pumps these are important driving forces in heat pump programmes.

Advanced cycles that aim at higher energy efficiency and use environmentally benign refrigerants are under development.

The USA and Japan have shown good examples of performance and reliability improvements. Important steps forward have been made in recent years. Hopefully, new promising steps will be made in the near future. The Super Heat Pump Energy Accumulation System Programme in Japan, ending in 1993, could offer such an opportunity. Already, a new national programme where heat pumps will play an important role, has been launched. The use of man-made waste heat streams as well as natural heat sources and sinks for both heating and cooling will be focussed on in the new programme. The Heat Pump Technology Center Japan will play a leading role in managing the new programme; this reflects appreciation for the fruitful work of this Centre. It also is a challenging offer to the Center celebrating its 5th anniversary this year. Congratulations!

As was proved during an IEA workshop on Industrial Heat Pumps held in October, 1991, in Budapest, there is a great need to have information about advanced technologies in Eastern and Central European countries which would be part of restructuring of the energy sectors. The technologies need not necessarily be "high-tech", but be principally cost-effective, proven and reliable. The need for co-operation in various technical fields was clearly felt and the IEA is presently developing ways to implement this. The HPC can play an important role in this process.

Let me take this opportunity to wish you a very Merry Christmas and a Happy New Year on behalf of everyone at the Heat Pump Centre in Sittard.



Jos W.J. Bouma
General Manager.

High Performance Absorption Cycle with Auxiliary Refrigerant Circulation

*T. Kashiwagi

ABSTRACT

The need for the development of advanced single-effect absorption cycles with double-effect efficiency has become urgent in terms of effective energy utilization, cost and miniaturization. In this study, the author proposes a high performance absorption cooling cycle with auxiliary refrigerant circulation and predicts its expected performance with the help of a computer simulation. This paper discusses the basic principle of this cycle and presents technical criteria for the choice of auxiliary refrigerant, which plays an important role in cooling cycle efficiency. The performance of a 'real-type' experimental system in actual conditions was verified by tests conducted on an experimental apparatus. As a result of these experiments, the effect of chilled water outlet temperature and auxiliary refrigerant circulation ratio on the absorption cycle was clarified, the cycle's principle was demonstrated, and its practicability and usefulness were realized.

INTRODUCTION

Absorption cycle efficiency improvement is usually achieved by using internal heat from the condenser and absorber to heat up the low temperature generator, as in multiple-effect and generator-absorber heat exchanger (GAX) cycles. These cycles, however, are limited by heat resistance and crystallization problems resulting from the need to raise the working fluid's temperature and concentration. This heightens the importance of developing single-effect cycles that can attain double-effect cycle efficiency. Absorption cooling cycles using auxiliary refrigerant circulation loops have this capability. Being basically single-effect cycles, they can use solar and other types of unexploited low-grade waste heat as driving sources, which makes them promising in terms of energy conservation. After discussing the operating principle of absorption cycles with auxiliary refrigerant circulation this article introduces a test plant built and run as part of a joint project between the Tokyo

Gas Co. Research Institute and the Tokyo University of Agriculture & Technology, Kashiwagi Laboratory.

CYCLE OPERATING PRINCIPLE

This cycle, essentially, has the configuration of a single-effect cycle supplemented by a separator, and the circulation of an auxiliary refrigerant (insoluble in the main refrigerant and the absorbent solution) between the evaporator and absorber.

Figure 1 shows the operating principle of the absorption cooling cycle with auxiliary refrigerant circulation. As is well-known in conventional absorption cycles, thermal energy from gas combustion or the sun is used in the generator to separate the refrigerant from the absorbent solution, since it is common for both liquids to be mutually soluble. In terms of coefficient of performance (COP), this thermal energy is considered as energy input. In the present cycle, an auxiliary refrigerant, insoluble in the main

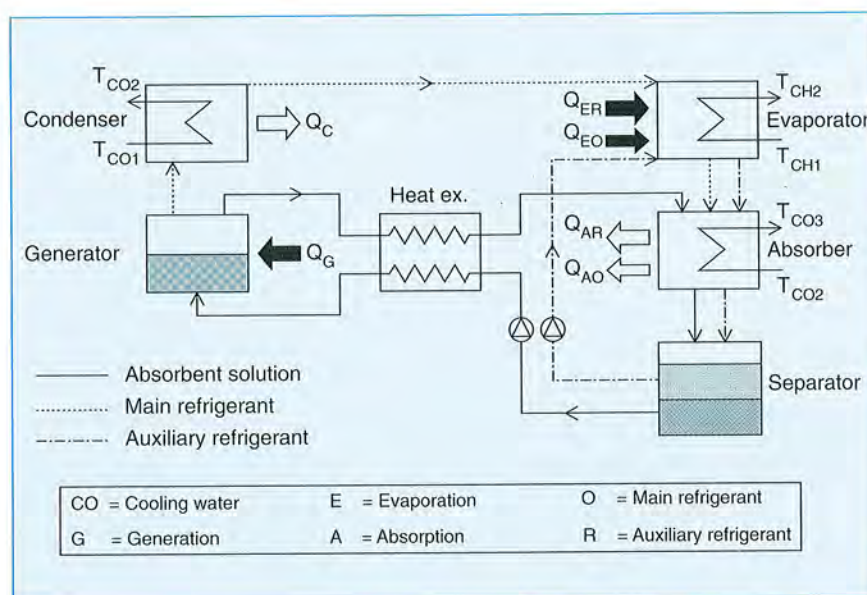


Figure 1: Cycle Operating Principle.

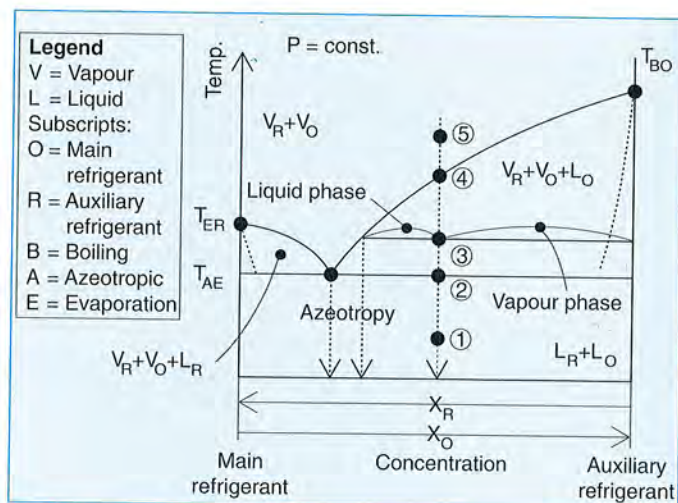


Figure 2: Evaporation With Auxiliary Refrigerant.

refrigerant, and also in the absorbent, is circulated between the evaporator and absorber, as shown by the dashed lines in the figure. In the evaporator, the main and auxiliary refrigerants are combined into an emulsion-type refrigerant mixture, and the main refrigerant's heat of vaporization (Q_{er}) is supplemented by that of the auxiliary refrigerant (Q_{eo}).

The evaporation phenomenon is described in Figure 2. The refrigerant mixture (1) starts to evaporate at an azeotropic temperature (2). When a mass flow ratio other than the azeotropic ratio is adopted, the partial pressures after complete evaporation of the refrigerant mixture are proportional to the mass flow ratio of main refrigerant to auxiliary refrigerant entering the evaporator. Therefore, the evaporation temperature rises from (3) to (5) during the evaporation process and a counter-flow type heat exchanger can be utilized. After complete evaporation, the refrigerant mixture flows into the absorber, which operates at about the same pressure as the evaporator.

Figure 3 describes the absorption process. The main refrigerant begins to be absorbed by the absorbent solution (2). A decrease in the main refrigerant's partial pressure is accompanied by an increase in the auxiliary refrigerant's partial pressure. When

the mixing ratio of the vapour phase reaches the azeotropic ratio (4), the auxiliary refrigerant starts to condense and a phenomenon of combined absorption and condensation heat transfer takes place. At the absorber's outlet, the main refrigerant is absorbed by and dissolved into the absorbent solution. However, given the auxiliary refrigerant's insolubility, it spontaneously separates itself from the diluted solution and returns directly to the evaporator without any need to enter the regenerator. In terms of cycle efficiency, the larger the heat removed by the auxiliary refrigerant Q_{eo} , the higher the cooling COP.

CRITERIA FOR AUXILIARY REFRIGERANT CHOICE

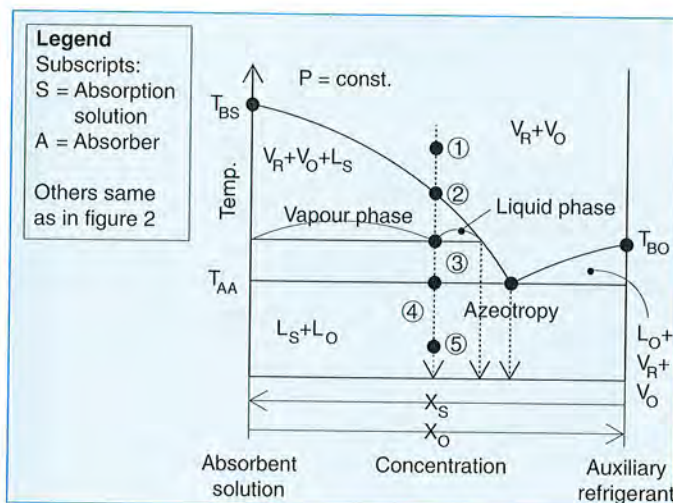
The $H_2O/LiBr$ combination, used in Japan, was selected as the main refrigerant/absorbent pair. Desirable properties for the auxiliary refrigerant include high heat of vaporization and thermal conductivity; ease of obtainment, low cost and low toxicity; cooling COP improvement; operability at designated temperature and concentration ranges; insolubility in the working fluid pair ($H_2O/LiBr$); and generation of Marangoni convection. The above-mentioned factors resulted in 2,4-Dimethyl-3-pentanol being selected as a suitable auxiliary refrigerant for use with the $H_2O/LiBr$ pair.

It must be noted that, while in the evaporator, the auxiliary refrigerant should have as high a saturated vapour pressure as possible. Its circulation ratio must be controlled to prevent its concentration from rising above the azeotropic ratio in the absorber, where excess auxiliary refrigerant may act as a non-condensable gas unless cooled down below its condensation temperature.

CYCLE SIMULATION

A computer simulation was performed to verify the influence of chilled water outlet temperature and auxiliary refrigerant circulation ratio on the cycle cooling COP, and

Figure 3: Absorption With Auxiliary Refrigerant.



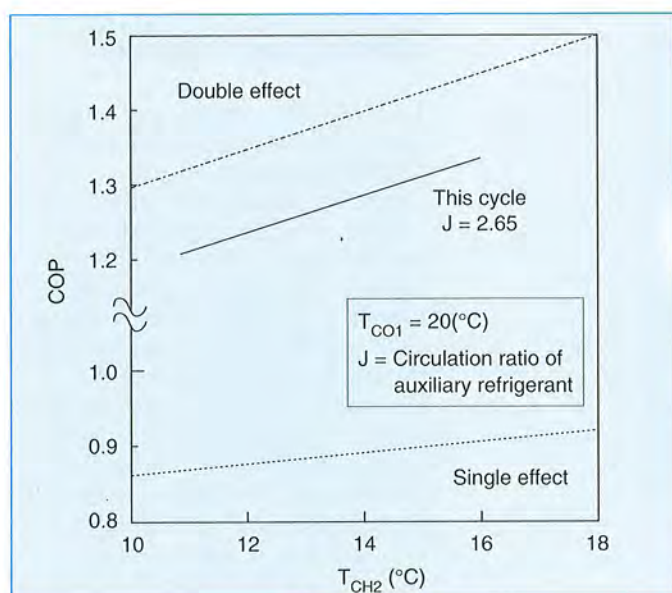


Figure 4: Cooling COP Simulation Results.

also on the absorber and evaporator heat transfer coefficients.

High circulation ratios of auxiliary refrigerant are beneficial to the cooling COP in that larger amounts of vaporization heat can be removed without the need for increased heat input to the generator. However, an upper limit to the amount of auxiliary refrigerant does exist since its concentration must not exceed the azeotropic ratio in the absorber. Figure 4 shows the predicted effect of chilled water outlet temperature on cooling COP in comparison with single-effect and double-effect cycles. Table 1 shows the heating temperatures in the generator for these three cycles. It is evident that auxiliary refrigerant circulation enables a single-effect cycle to

attain relatively high COPs without the need for a complex structure or high generation temperatures, thus broadening the possibilities of an efficient utilization of solar and waste heat for cooling.

EXPERIMENTAL

Figure 5 shows the experimental apparatus, built to visualize the operating principle and test the performance of the absorption cooling cycle with the auxiliary refrigerant in circulation. Driving heat input is done with a 3 kW electric heater; spiral-tube heat exchangers are used in the absorber, evaporator and condenser. The structure is very simple and similar to that of a single-effect cycle, with the addition of an auxiliary refrigerant pump and a separator for spontaneous separation of the condensed auxiliary refrigerant from the diluted solution.

The existence of an upper limit to the auxiliary refrigerant circulation ratio can be detected from the absorbent heat transfer coefficient data in Figure 6, where the heat transfer coefficient in the absorber decreases with higher circulation ratios of auxiliary refrigerant. If the amount is too excessive, then the

Figure 5: Experimental Apparatus.

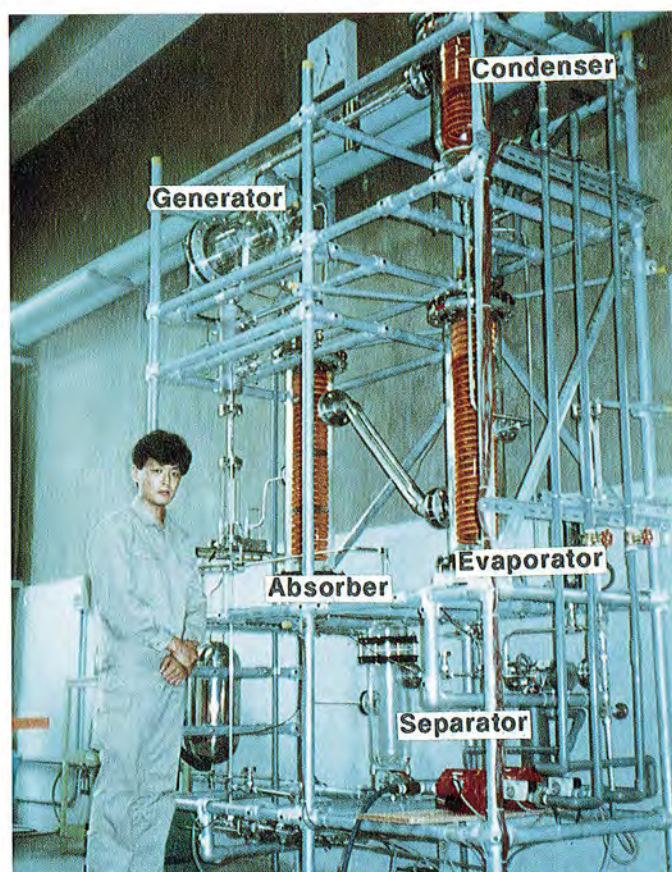


Table 1: Generator Heating Temperatures.

Cycle type	Heating temp. (°C)
Single-effect	50
Double-effect	86
This cycle	48

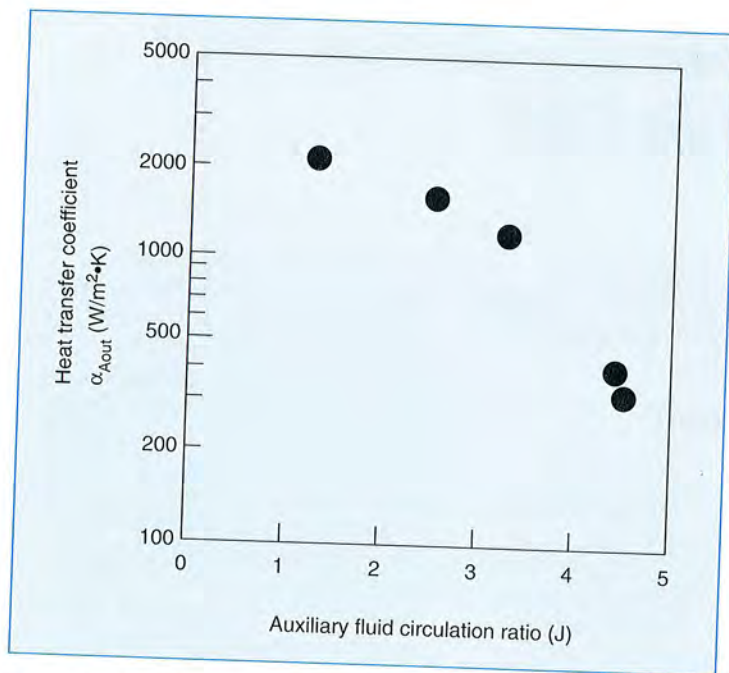


Figure 6: Auxiliary Fluid Circulation Rate (J) and Heat Transfer Coefficient (experimental).

auxiliary refrigerant may actually hinder the heat transfer in the evaporator and absorber. This can also cause sensible heat losses in the evaporator by increasing the amount of non-evaporated auxiliary refrigerant.

Figure 7 shows COP trends for different chilled water outlet temperatures. Experimental and simulation results for the single-effect cycle are comparable. However, the experimental results for the cycle with auxiliary refrigerant circulation show a decrease in COP with higher chilled water outlet temperatures. This tendency, contrary to that shown by the simulation results, is attributable to decreased wettability of the heat exchangers when larger amounts of auxiliary refrigerant are present at higher evaporation temperatures.

CONCLUSIONS

This article summarized simulation and experimental results which clarified the operating principle and demonstrated the viability of a new high performance absorption cooling cycle ($H_2O/LiBr$) with auxiliary refrigerant circulation ('2,4-Dimethyl-3-pentanol'). The

circulation ratio of auxiliary refrigerant (vs. main refrigerant) was introduced as a parameter, the increase of which theoretically tends to augment the cycle cooling COP, but in practice has an upper limit due to the azeotropic ratio in the absorber. Cycle viability was demonstrated, and the auxiliary

refrigerant showed improved cycle efficiency up to 20%. The main advantage of this cycle is the combination of the structural simplicity of a single-effect cycle with the high efficiency of a double-effect cycle, plus a suitability to low temperature driving heat sources.

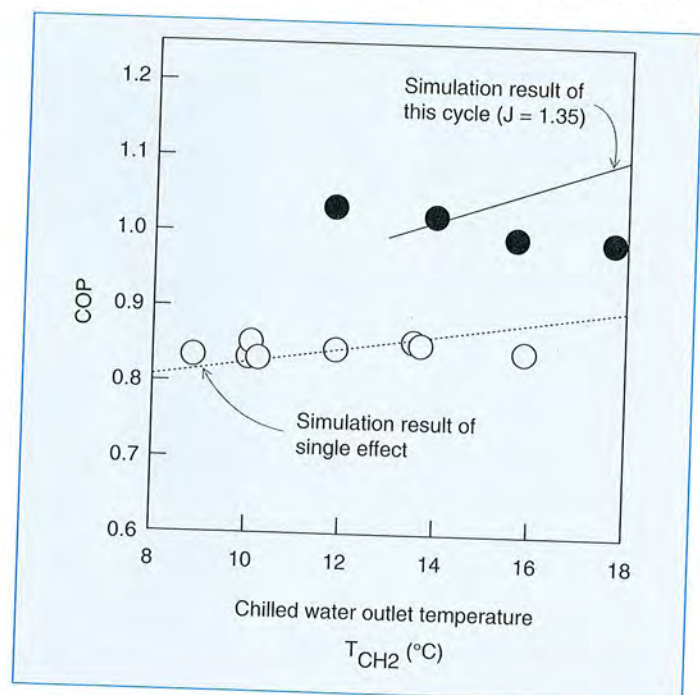
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- [2] *ibid.* — 2nd report, Experiments and Characteristic Analysis (pp. 198-205).

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Figure 7: Calculated and Experimental COPs.



A Novel Type of Electrically-Driven Heat Pump with a Very High COP

*F.H. Theijse

ABSTRACT

Using R-152a as a medium, a small electrically-driven compression heat pump is under development. Its main characteristics are that it is operated within the co-existence region of the medium and, using a turbo compressor and a turbine as expander, is continuously variable in speed. Using compact heat exchangers as evaporator and condenser, at a condensation temperature (T_c) = 50°C, and an evaporation temperature (T_{evap}) = 10°C, a Coefficient of Performance (COP) of > 6 is achieved. When used for space conditioning, the most interesting market, the annual average COP will be appreciably higher, dependent on the actual climatic conditions prevailing, due to the continuously variable speed. Available in the power range 2-50 kW, it will mainly cover all heating/cooling demands in Western Europe whilst saving considerably on energy. Moreover, having an Ozone Depletion Potential (ODP) = 0 and a Greenhouse Warming Potential

(GWP) of about half that of a condensing high-efficiency heating boiler, it is an environmentally benign machine.

INTRODUCTION

Of all primary energy used in Western Europe, 40% is used for space heating and/or air-conditioning. Market studies have shown that the number of space heating systems in Europe is far larger than the number of air-conditioning systems used. The capacities of space heating systems vary from a few kW to about 50 kW heating capacity, not accounting for industrial (space) heating. Basically, a market with an annual turnover of millions exists for devices for use in retrofit or new buildings. It is, indeed, a market of interesting size once one can find a means of mobilizing it.

ENVIRONMENTAL BENEFITS

To assess the environmental benefits of an electric heat pump using R-152a as the medium, let us

consider a reasonably modern house with an installed heating capacity of 10 kW. Under average conditions this means about 1,400 hours of heating at full load; a demand of 14,000 kWh/a. When this heat is supplied by a modern high-efficiency condensing heating boiler, having an efficiency of 90% in relation to the upper heating value and fuelled by (Dutch) natural gas, an amount of 1,391 Nm³/a of CO₂ is produced. At the same time, an environmental thermal load of 15,556 kWh is set free as low temperature heat. With this heating demand supplied by the electrical high-speed heat pump introduced in this article, using the very conservative annual average COP of 6, the following picture emerges for the worst case from a CO₂ point of view, that is, electricity being supplied by a coal-fired power station.

The electric generating efficiency at the user's-end will be approximately 33%, accounting for the distribution losses. Hence, the Primary Energy Ratio (PER) of this

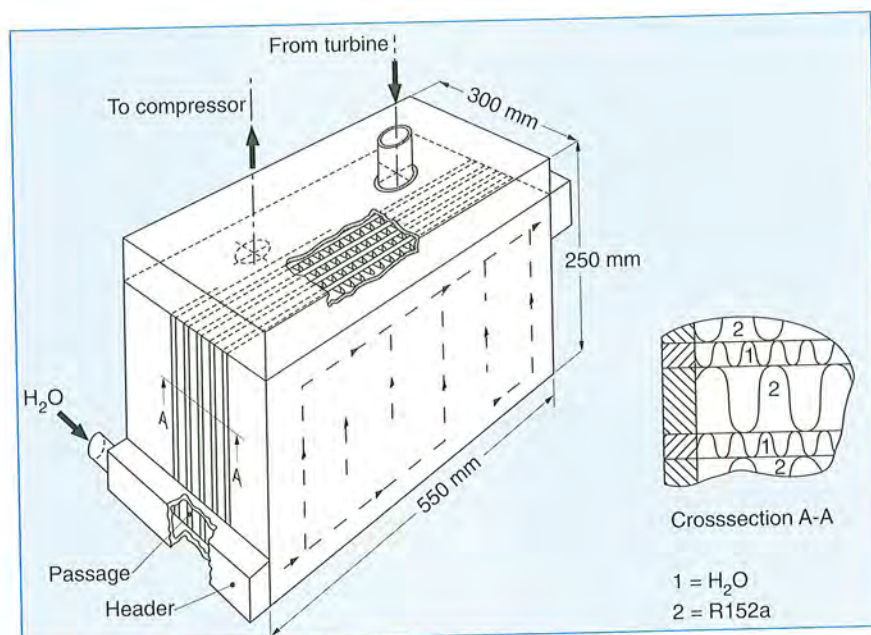


Figure 1: Compact Heat Exchanger for 36 kW Heat Pump.

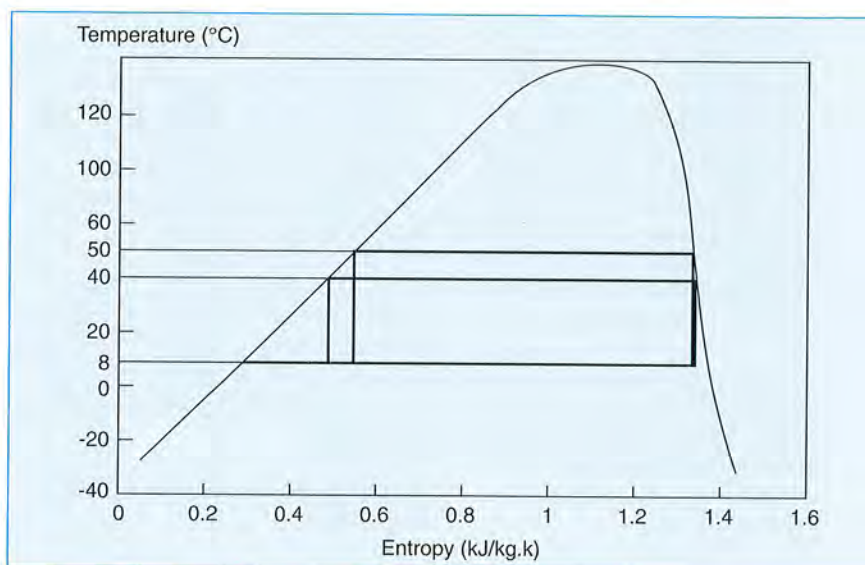


Figure 2: P/H diagram for R-152a with the heat pump cycle.

heat pump will amount to 2. The annual CO_2 production for heating the house is only $1,121 \text{ Nm}^3$. Moreover, the environmental thermal pollution is reduced to $7,000 \text{ kWh/a}$. Of course, all heat pumps that would be used for this application will be hermetically sealed against the atmosphere. Still, due to malfunctions, leaks, damage and other causes, 2.5% of the heat pumps will fail per annum, losing their medium-content to the atmosphere. This figure is based on extensive experience with comparable high quality systems. A 10 kW heat pump will contain up to 10 kg of its medium. If R-152a is used, not only is the $\text{ODP}=0$, but the GWP is less than 0.8%, of that of R-12. Its annual average leakage rate would only amount to the equivalent of $11.32 \text{ Nm}^3 \text{ CO}_2/\text{a}$ which is insignificant. From both the ODP and GWP points of view, the medium R-152a (CH_3CHF_2), is the best choice available. Moreover, it is available in commercial quantities at acceptable costs. However, it has two problems: no compatible lubricant for the compressors is known of and it is flammable.

THE HEAT PUMP

For space heating, a favourable electric heat pump should have a COP of at least 6 in its working range, the temperature between 8 and 50°C . In addition to the

environmental benefits of employing R-152a as a medium, it should have many other positive characteristics such as low noise, low investment, reliability, and be maintenance-free. This is realized as follows:

1. The working range of the basic concept should be -40°C to $+100^\circ\text{C}$ using one medium.
2. To achieve the high COP mentioned, a number of measures have to be taken. First of all, the temperature differences in the heat exchangers have to be as low as practically feasible, which calls for Compact Heat Exchangers (CHE's), as depicted in Figure 1. A very high frequency three-phase squirrel cage motor is used to drive the system. The motor and the frequency conversion system have to be of a very high efficiency type, over 90%.

To achieve the high COP , fundamental considerations are needed. It is essential that the compressor never generates pressures higher than that compatible with the temperature increase actually needed. Moreover, the efficiencies of expansion and compression should be as

close as possible to the isentropic values. To this end, compression and expansion inside the region of co-existence of the medium is of great advantage. Moreover, when operating inside this region, the criterium of variable temperature through variable pressure is easily attainable. All this means that evaporator, compressor, condenser and expander must, at all times, be filled with very wet vapour, it being essential that no de-mixing of vapour and droplets occurs. This requires very short residence times in the machines, in other words for small dimensions. Moist vapour rules out all valved or vaned machines and strongly impairs screw machines. Moreover, the expander will have to continuously cope with varying pressure ratios.

The only possible solution to these complex requirements is the use of a turbo compressor, and a turbine as expander. These have to be small, for the reason given, but should still provide sufficiently high flow velocities to raise the pressure to the levels needed. To ease control, subsonic speeds are favoured.

The solution to the engineering challenge, therefore, is a turbo compressor and a turbine of very small diameter on one shaft, driven by a very high speed electric motor with continuous variable speed. This patented principle provides excellent results, leading to a COP of $0.74 \times$ the theoretical Carnot Efficiency (8.075 at $T_c = 50^\circ\text{C}$ and $T_{\text{evap}} = 10^\circ\text{C}$) = 6 . The cycle is thus achieved, as is depicted in Figure 2.

3. As the heat pump never provides higher temperatures than needed at any one time, the annual average COP is favourably increased to over 6.
4. Slide bearings which are lubricated by the process gas, are recommended to prevent all problems concerning lubricants and materials, make a truly hermetic system possible, reduce maintenance to zero and strongly reduce noise.
5. To enable the use of the machine in buildings, the total sound pressure at 1 m distance has to be below 32 dBA. The present machine does not exceed that value.

6. The machine has to have low investment costs, so production calls for low cost techniques, and a construction material suited for shaping of components in moulds. Aluminium is mainly used as the structural material. This is also advantageous because of the small dimensions of many of the mechanical parts.
7. To prevent leakage within the system, the pressure should be kept above ambient, but to minimize pressure problems, it should not exceed 20 bars.
8. To ensure that the diameters of the compressor and the turbine are in the same range, the densities of liquid and vapour of the medium should be as close as possible.
9. To limit flow velocities, that is, rotational speeds, the vapour density has to be high.
10. The medium must be well-known, properly defined, stable, compatible with all materials used in the system, and environmentally harmless.

Points 1 to 7 inclusive are met in the way indicated. By choosing R-152a as the medium, all other

points are met as well, whilst its disadvantages are coped with. An exception might be seen in its flammability, but as the inventory in any one system is only a few kilogrammes, and kept in a hermetically sealed environment, the problem is reduced to a minimum. Of course, a malfunction of any part of the system should never cause a leak. A very small leak causes no concern because of the spontaneous slow oxidation of the medium outside the system.

DESIGN OF MACHINE AND SYSTEM

The integrated design of heat exchangers and machine is shown in Figure 3, whilst Figure 4 depicts an earlier experimental machine. The medium is admitted to the compressor through the hollow shaft of the electric motor; the turbine is integrated with the turbine shaft and carried by gas bearings. The figure shows clearly how much smaller the flow-engine is in comparison to the electric motor. The depicted assembly shows a system that delivers 36 kW of useful heat. Its compactness is noteworthy. Whereas size and costs of the heat-exchangers are directly proportional to their surface and hence to their capacities, the situation is different for the mechanical part. For both

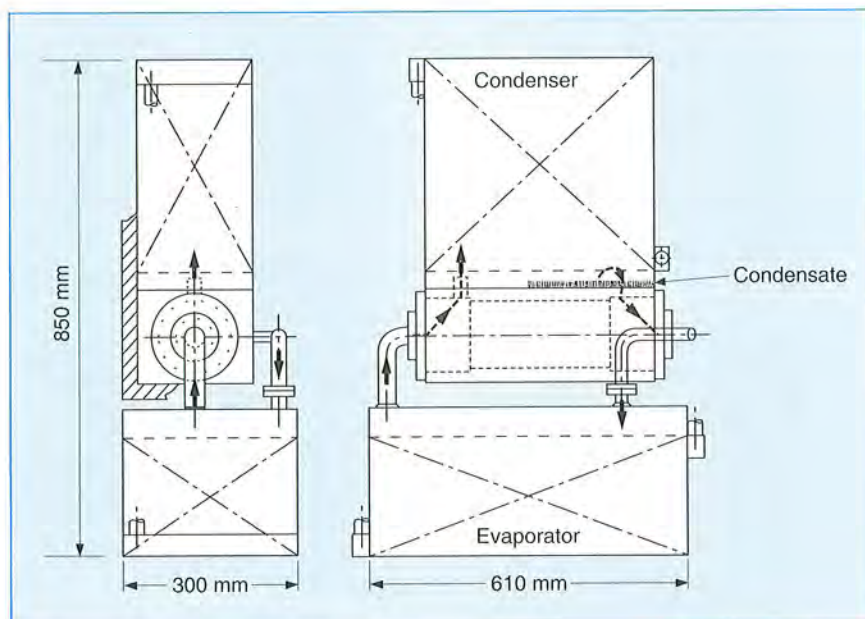


Figure 3: Layout Heat Pump of 36 kW capacity.

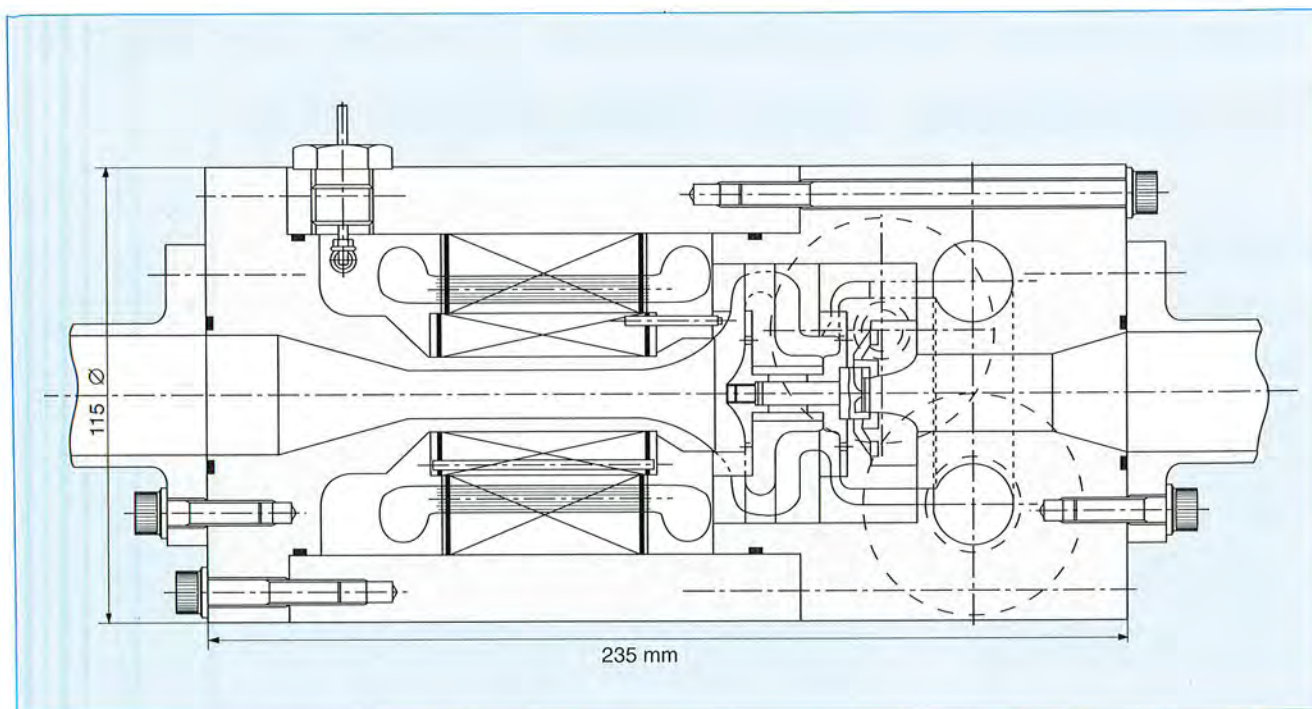


Figure 4: Experimental 36 kW Heat Pump.

compressor and turbine, the well-known power laws apply:
 $N \propto \Omega^3 \cdot D^5$. Very small changes in rotational speed (Ω) or rotor diameter (D) increase or decrease power (N) accordingly. Noting that a 40 mm rotor in the compressor produces 36 kW at a speed in excess of 60,000 rpm, it is clear that very small changes in both speed and size will vary heat pump capacity between 2 kW and 100 kW. The variations in size of the electric motor are less critical, as there the dependence is:
 $N \propto \Omega^2 \cdot D^3$. Still, only small changes are needed to cover the range indicated.

EXPERIMENTAL CONFIRMATION

Basic characteristics of the machine have been proven. The operation inside the region of co-existence was achieved, resulting in COP values in accordance with our forecasts. Moreover, the sound level proved to be inside the range of expectations. These results, however, were achieved using a different medium, of which the viscosity of the vapour was known. This is a key property in regard to its use as a lubricant in the

bearings. Currently, a programme has been started to measure this key property for R152a over the range of prevailing conditions. By the end of 1992, the first prototypes should become available.

COSTS

Appropriate research by noted marketing research organizations has shown that our expectations, in regard to market size for the machine are sound. Quite apart from the energy savings and environmental advantages, the price has to be competitive. Comparative studies by above mentioned organizations have shown that at costs in the range of heating or air-conditioning plant, the heat pump will be marketable. It was reassuring to find that even using our sophisticated heat exchangers and frequency converters, we could match the market prices for air-conditioning equipment virtually over the whole range, excluding window sets.

ALTERNATIVE APPLICATIONS

Many additional applications for the system and its components are feasible, such as freezing and cooling, low-grade heat recirculation, water chilling, textile industry, low-grade heat in the food industry and many more. However, the main target should be the space heating and air-conditioning market, since this is the largest in size.

The various components have an additional potential of their own, for instance: compact heat exchangers, high-speed motors, and frequency converters. The actual motor-compressor-turbine-unit has its own additional applications in, amongst others, vapour recompression, special compressors, etc.

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Two-phase Compression Cycles for High Temperature Heat Pump Systems

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SUMMARY

Two-phase compression cycles accompanied with liquid refrigerant injection cooling methods are the key technologies used for the high performance high temperature heat pump systems being developed within the context of the Super Heat Pump Programme.

This paper describes two-phase compression heat pump cycles that have been studied from the theoretical viewpoint of cycle analysis to clarify the optimum liquid injection conditions. This is the first step needed to develop high temperature heat pump systems, which supply 300°C thermal output from a 150°C heat source.

INTRODUCTION

High temperature process heat (300°C or more) is usually supplied by boilers. If heat pump systems that supply such high temperature output heat are developed and were to become widely used, it would have a substantial effect on energy conservation. Although, because of its thermal stability, water is suitable as the working fluid for high temperature compression heat pumps, adiabatic compression of steam from a saturated condition induces a substantial degree of discharge superheat. This does not only reduce the Coefficient of Performance (COP), but in some cases, operation of the heat pump itself becomes impossible.

Some steam recompression heat pumps [1] and a test plant for a 300°C output heat pump [2] (the latter being developed under the Japanese national R&D project "Super Heat Pump Energy Accumulation System") use water as the working fluid and control

the discharge temperature by water injection into the compression process. This two-phase compression process is a key technology in realizing high temperature compression heat pumps. Although the heat pump cycle using the two-phase compression process is not a novel idea, few systematic studies have been made to clarify its optimum conditions. The reason for this is that the two-phase compression process has no or little positive effect on conventional heat pumps which use common CFC, and HCFC refrigerants.

This paper describes a high temperature heat pump which recovers heat from a 150°C heat source (comparable to the output temperature of heat pumps being realized in the industrial fields these days) and produces 300°C heat (comparable to the boilers). The two-phase compression heat pump cycle is studied from the theoretical viewpoint of cycle analysis to clarify

its potential for increasing the COP and the optimum conditions of basic liquid injection cycles. The two-phase compression process is assumed to be quasi-static and cycle analyses are based on the thermophysical properties of real working fluids.

EFFECTIVENESS OF TWO-PHASE COMPRESSION

The most basic heat pump cycle, that of single stage two-phase compression, is shown in Figure 1 (cycle: 1-2-6-3-4-1).

The wet fluid from an intermediate point in the evaporation process is fed to the compressor and compressed adiabatically while the vapour phase and liquid phase are in equilibrium. The compressor suction wetness is represented by m_1 . The compression power per unit working fluid mass flow rate is indicated by the area:

1-2-6-3- s_3 - s_4 -4-1.

Definitions

P: Pressure,
T: Temperature,
m: Wetness,
h: Specific enthalpy,
s: Specific entropy,
x: Parameter for pressure at the injection point,
y: Injection rate (injection mass flow ratio to total mass flow rate),
Subscript ₁: Suction of compressor,
Subscript ₂: Discharge of compressor,
Subscript ₃: Outlet of condenser,
Subscript ₄: Inlet of evaporator or outlet of expansion valve,
Subscript ₅: Saturated liquid at the evaporation pressure,
Subscript ₆: Saturated vapour at the condensation pressure,
Subscript ₀₁: Saturated vapour at the evaporation pressure,
Subscript _h: Heat output side,
Subscript _L: Heat source side,
Subscript _x: Injection point.

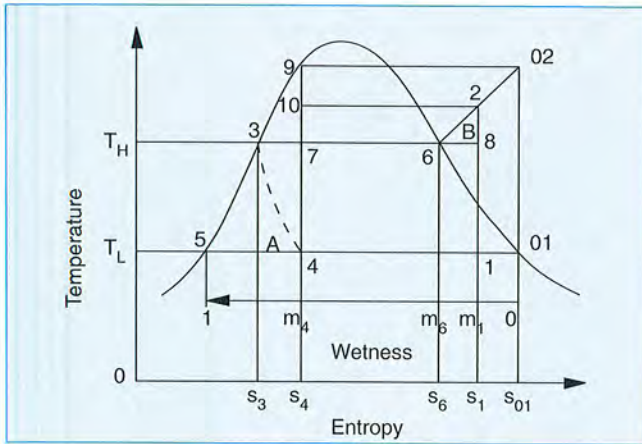


Figure 1: Two-phase Compression Heat Pump Cycle, Temperature-Entropy Diagram.

The COP of this cycle is smaller than that of the Carnot heat pump cycle for the following two reasons; firstly due to the influence of the irreversible change at the expansion valve (area A: 3-s₃-s₄-7-3), and secondly due to the superheat of the compressed fluid (area B: 6-8-2-6). As m₁ increases, the influence of A becomes relatively larger, and that of B becomes smaller. Therefore, the effect of the two-phase compression process depends on the properties of the working fluid and temperature range in which the heat pumps operate. From cycle analysis [3], the criterion of increasing the COP by two-phase compression can be obtained as equation (1)

$$\text{Area A} < \text{area: } 6-02-9-7-6, \quad (1)$$

and the COP assumes its maximum value when the m₁ satisfies equation (2)

$$\text{Area A} = \text{area: } 6-2-10-7-6 \quad (2)$$

In cases where water is used as the working fluid, the latent heat and the degree of superheat in the compression process are relatively large and the COP reaches its maximum value at an optimum value of m₁. When the heat source temperature is 150°C and the output temperature is 300°C, the optimum suction wetness is 14%, the associated improvement in COP is 5.2% and the discharge superheat is 60°C.

process following an iso-enthalpy pressure drop at the atomizer, the specific enthalpy of the moist vapour after the mixing process h₁ is written as:

$$h_1 = (1-m_4 y) h_{01} + m_4 y h_s, \quad (3)$$

Equation (3) shows that the liquid injection cycle with the injection ratio y corresponds to the cycle shown in Figure 1, m₁ of which is m₄y. Therefore, when the optimum suction wetness is m_{1opt} in the cycle shown in Figure 1, the optimum injection ratio y_{opt} is given as m_{1opt}/m₄

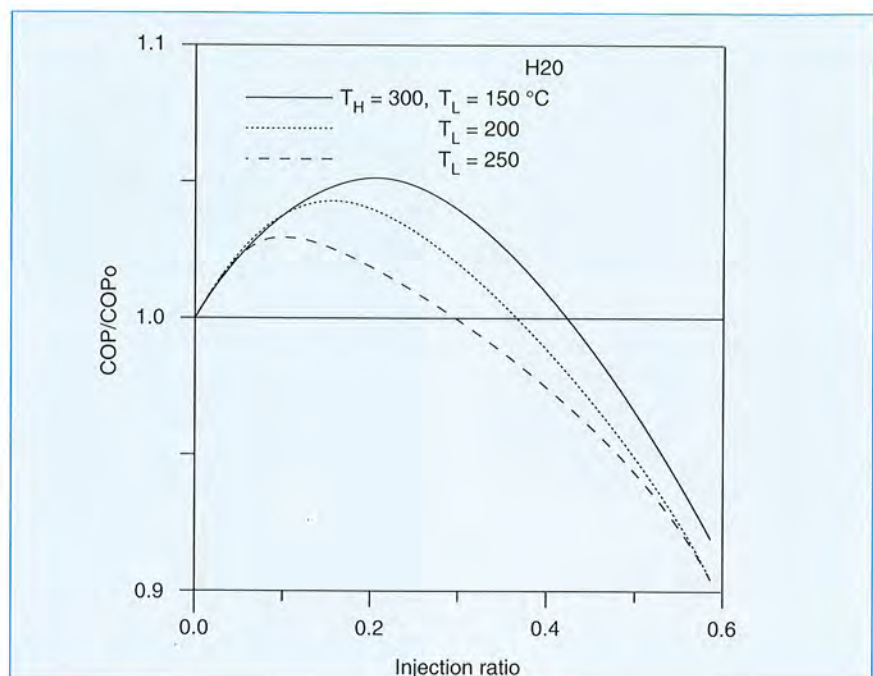
Figure 2 shows the change of COP with the injection ratio y. The COP is standardized by the COP₀, that is the COP when y=0 in each heat source temperature condition. When the heat source temperature is 150°C, the optimum injection ratio is 21%, the rate of COP improvement is 5.2%.

Injection involves mixing two phases of the same kind of fluid but of different conditions. Different mixing processes lead to different conditions of the fluid to be compressed and different effects on the COP.

LIQUID INJECTION CYCLE

The cycles that involve liquid injection into the compression process are far more practical than the previous cycles, compressing the wet fluid from an intermediate point of the middle of the evaporation process. Here we consider the cycle where the liquid from the outlet of the condenser is injected into the suction of the compressor. If the liquid phase is considered to be injected into the gas phase in an isobaric mixing

Figure 2: Change of COP with Injection Ratio, in Liquid Injection Cycle.



The power needed to mix and adiabatically compress two phases of the same kind of fluid is minimized when the phases are mixed in an isentropic process. We can theoretically consider the isentropic mixing as the most ideal case for liquid injection. In this case, the values of enthalpy and entropy of the fluid after mixing determine the compression process, and also the compression power. Isochoric (constant volume) mixing, which may be realized in ideal displacement compressors, includes intermediate degrees of irreversibility between isentropic and isobaric mixing. In this case, the values of specific volume and internal energy of the fluid after mixing are prescribed and they determine the compression process, and also the compression power.

Figure 3 shows the change of COP with the injection ratio resulting from each mixing process mentioned above.

When the injection ratio is smaller than 20%, the range in which liquid injection should be practical, isobaric mixing results in a COP improvement which is more than half that of isentropic mixing. This is noteworthy when taking into account that isobaric mixing is much easier to realize. Isochoric mixing improves the COP more than isobaric mixing. In the actual systems, when the liquid is injected at a higher pressure than the gas phase, an intermediate mixing condition between isobaric and isochoric mixing will occur and, theoretically, we can expect the COP improvement to be higher than in the case of isobaric mixing.

In the following we will consider liquid injection at an intermediate point of the compression process and an isobaric mixing process. The pressure at the injection point is P_x and it is represented by the parameter x defined by the following equation:

$$x = \ln(P_x P_L) / \ln(P_H / P_L) \quad (4)$$

The compression power is calculated by dividing the

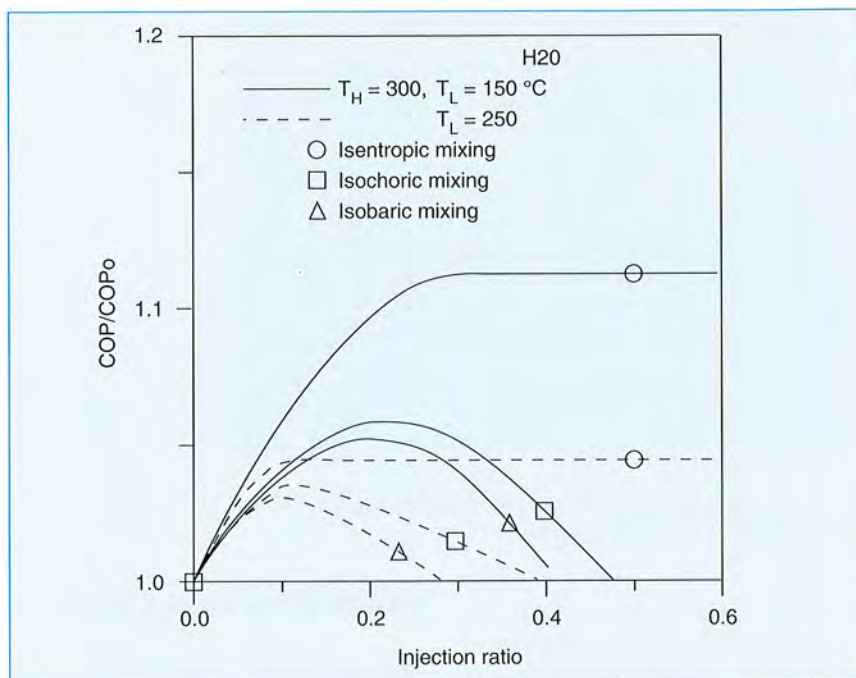
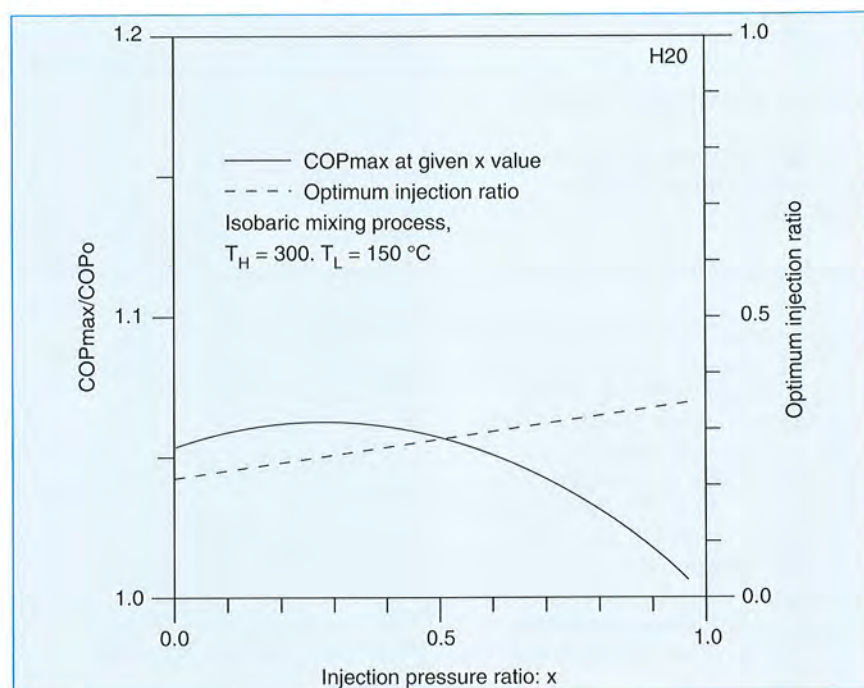


Figure 3: Change of COP Improvement for Different Mixing Processes.

compression process into two stages, i.e., the pressure ranges from P_L to P_x and from P_x to P_H , respectively. The maximum value of the COP and the optimum injection ratio depend upon the x value as shown in Figure 4.

The temperatures of the heat source and the output are 150°C and 300°C, respectively. At this temperature condition, the liquid injection cycle is optimized when the liquid phase is injected into the point where the x value is 0.30

Figure 4: Change of Maximum COP and Optimum Injection Ratio with Injection Point, in Liquid Injection Cycle.



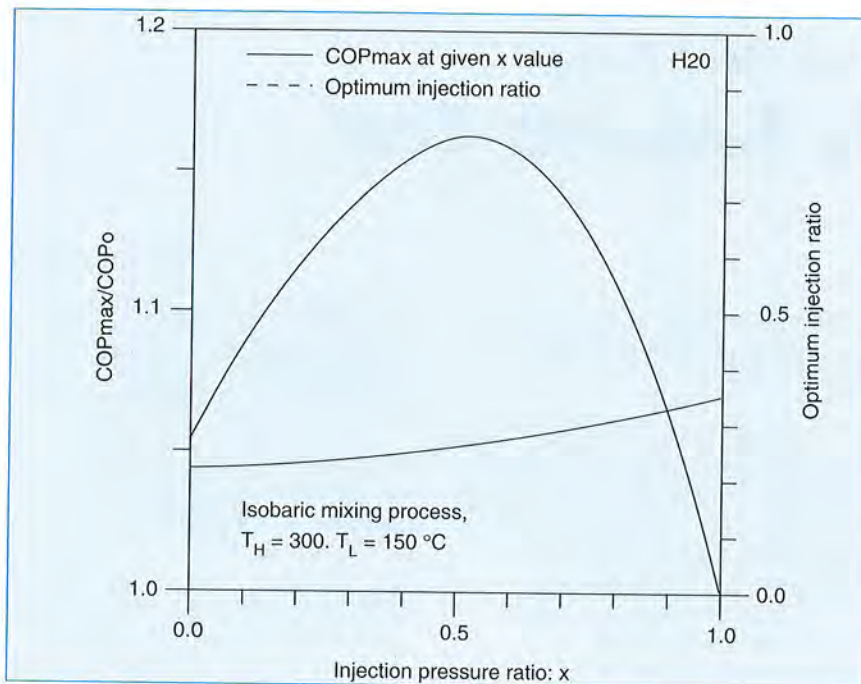


Figure 5: Change of Maximum COP and Optimum Injection Ratio with Injection Point, in Two-phase Injection cycle.

with an injection ratio of 25%. The associated improvement in COP is 6.3%. If the liquid is injected into the suction, i.e., where the x value is 0, the maximum improvement in COP is 5.2% and the optimum injection ratio is 21%. Compared with the optimum injection condition mentioned above, the COP improvement in this case is smaller by only 1.1%, whereas the optimum injection ratio is smaller by 4%. In addition, from the point of view of the available pressure drop at the atomizer, injection into an earlier stage of the compression process is advisable. Therefore, injection into the suction or at the very beginning of the compression process is the most practical in actual systems.

TWO-PHASE INJECTION CYCLE

Liquid injection cycles are effective in improving the COP and avoiding excessive discharge superheat. Gas injection cycle, i.e., economizer cycles, are very effective in improving the COP. In order to utilize the effects of both cycles, a two-phase injection cycle is considered, where the liquid phase, partially extracted from the outlet

of the condenser, and the gas phase, separated in the economizer, are injected at an intermediate point of the compression process with an isobaric mixing process. The influence of the x -value on the maximum value of the COP and the optimum injection ratio is shown in Figure 5.

The two-phase injection cycle is optimized when the liquid phase is injected at the point where the x value is 0.52, with the liquid injection ratio being 27%. The improvement in COP is then 16%; the gas injection ratio is 25%; the mass flow ratio passing through the evaporator is 48%.

CONCLUSIONS

- In the case where the working fluid is water, the COP is improved by utilizing a two-phase compression process. There exists an optimum gas-liquid ratio yielding the maximum value of the COP.
- When the heat source and output temperatures are 150°C and 300°C respectively, and

when the injection is isobaric, the most basic two-phase compression cycle is optimized when the liquid phase is injected into an intermediate point of the compression process. There, the compression ratio, standardized by the total compression ratio, is 0.30, the injection mass flow ratio is 25%. The rate of COP improvement is 6.3%.

- Two-phase injection cycle, in which the liquid, partially extracted from the outlet of the condenser, and the gas, separated in the economizer are injected somewhere in the middle of the compression process, can effectively improve the COP and reduce the discharge superheat.

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Workshop Ground-Source Heat Pumps - Advancements Towards Cost Reduction

*G.J.A.M. Meijer

The IEA Heat Pump Centre/ Canadian National Team Workshop on "Ground-Source Heat Pumps - Advancements Towards Cost Reduction" held on August 12 and 13, 1991, in Montreal, Canada, was attended by 44 participants, representing 8 countries.

Thirteen papers were presented, followed by group discussions.

Presentations

New Anti-Freeze Fluids

At the Division of Engineering Technology of Oklahoma State University, USA, potassium acetate is being subjected to laboratory and field tests to determine its corrosive effects on the various components found in typical ground-source heat pump (GSHP) systems. Previous studies have shown the material to be of low-toxicity and environmentally compatible. Preliminary tests show that commercially inhibited potassium acetate meets the necessary requirements for application in GSHP systems. (Paper J.E. Bose.)

ORTECH International, Canada, evaluated anti-freeze fluids in terms of heat transfer and pumpability, using a computer model of a typical GSHP, and cost and toxicity. ORTECH concludes that no single agent possesses superior properties in all areas. Methanol and ethylene glycol are characterized by moderate toxicity. Calcium chloride and potassium carbonate have excellent heat transfer performance properties but high pumping energy demands. Propylene glycol is practically non-toxic but is expensive and has a relatively poor heat transfer

performance. Potassium acetate provides a good compromise. (Paper P. Top.)

The Department of Applied Thermodynamics and Refrigeration of the Royal Institute of Technology in Sweden has compared thermodynamic properties of aqueous solutions and some commercial non-aqueous heat transfer liquids, such as Dowtherm J, Syltherm XLT, Paracryol, and Isopar M. The conclusion is that, from a thermodynamic point of view, aqueous solutions are to be preferred. The lower pumping power demand of these non-aqueous heat transfer liquids does not justify their use in most applications. (Paper A. Melinder, E. Granryd.)

Direct Expansion Systems

At Ontario Hydro, Canada, a direct expansion GSHP system using copper pipes, buried in three 1.8 m (6ft) deep and 30 m (100 ft) long trenches, has been tested. The unit produces about 8.5 kW (29,000 Btu/h) of heating at a COP of 3. The system, which uses 7 kg (15 lbs) of R-22, was commissioned in October 1990. The main conclusions are:

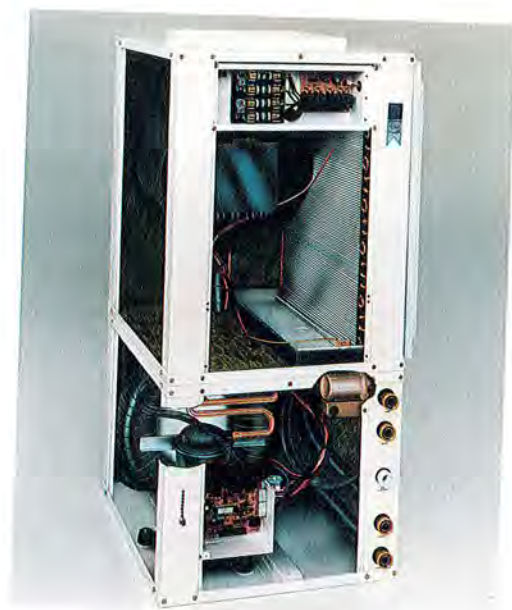
- The system operated successfully both in the heating and cooling modes. After more than 1,500 hours of operation, the performance dropped by an acceptable 21%.
- The ground coil requires about half the space needed for a conventional GSHP, making this product more acceptable for the retrofit market, where existing land may preclude a conventional GSHP.

- The smaller ground coil should reduce the overall costs and installation time by 20%. (Paper J. Lenarduzzi, P. Eng)

ECR Technologies Inc., USA, markets and installs direct expansion GSHPs with the following features:

- The in-ground coils consist of a system of concentric copper tubes. The inner and outer tubes are separated by a heat transfer barrier formed by the inner tube and an intermediate tube. These tubes are installed in the ground at an oblique angle to form a conical tap field.
- Drilling 60 ft (18.3 m) long holes in soft and medium hard soil is done by a highly manoeuvrable rig which uses a hollow bit similar to a common masonry bit.
- ECR has developed a liquid flow control to reduce sub-cooling in the condenser to near zero, and an active charge control to avoid superheating in the evaporator.

Comparative tests with conventional GSHP systems and with high efficiency air source heat pumps showed the ECR system to attain 10-30% higher efficiencies. ECR advocates adding domestic water heating to the system, resulting in a reduction of the payback period of the entire GSHP system. Refrigerant contents for a 3 ton system is approximately 7 kg. An improved coil has been developed, reducing the amount of refrigerant in the liquid/vapour space by 19% and the coil copper weight by 39%. Details will be published after patent protection has been obtained. Licensees for received patents are being sought. (Paper R.W. Cochran)



Photograph: Water Furnace Advanced GSHP with top air discharge.

bentonites, neat cement, or drill cuttings. The International Ground-Source Heat Pump Association at Oklahoma State University is developing a grouting manual. (Paper H.J. Braud)

Improvements in the Heat Pump

Water Furnace, USA, has developed and marketed a ground-source heat pump in the 3 to 14 kW range with scroll compressor, variable speed DC fan motors, and microprocessor controls (see photograph). Domestic hot water heating is optional. (Paper D. Ellis)

Advanced Heat Exchangers

Polydynamics Ltd, Switzerland, reported that in Switzerland the number of GSHPs continues to increase steadily at a rate of some 500 installations per year. Until recently, GSHPs have been used only for heating purposes in underfloor heating with delivery temperatures of 45 to 50°C. Recently, a trend has been observed towards denser under-floor coils to permit a delivery temperature of 35°C. The most popular heat exchanger has become the double (parallel) U-tube with built-in injection pipe used in holes up to 120 m deep. Monitoring campaigns have been completed over several years for single dwellings to validate numerical models, which can now be applied to studies on the effect of site properties and of GSHP design. In the paper, some conclusions concerning the dynamic behaviour of the heat exchangers, effects of the thermal conductivity of the ground, and the separation of neighbouring heat exchanger tubes upon the Seasonal Performance Factor, are presented. (Paper R.J. Hopkirk, B. Kaelin)

A paper by the Institute for Research and Construction, National Research Council, Canada, first analyzed the influence of soil freezing, on heat transfer water migration due to freezing and heat, and moisture transfer in the soil mass. In addition, it

At Oak Ridge National Laboratory (ORNL), USA, a direct expansion GSHP system of 1.76-2.64 kW (1/2-3/4 ton) is being tested in the laboratory. The tube-in-tube and U-tube ground-heat exchangers are installed in PVC columns, simulating the ground. The test results indicated that direct expansion coils have a heat transfer rate at least twice as high as that of conventional plastic coils, reducing the required coil area by a factor of two. It is concluded that, with the additional savings gained by elimination of a refrigerant fluid heat exchanger and a fluid circulation pump, the direct expansion GSHP concept is not only more efficient, it may be economically viable as well. (Paper V.C. Mei, V.D. Baxter.)

Ground-Coil Sizing, Installation Techniques, Materials, Performance

At the Department of Building Physics of the University of Lund, Sweden, personal computer implemented models for the thermal design of vertical ground-source heat exchangers have been developed over the years. In the discussion, it was advocated to use these models as an international standard and to link them to existing building heating system design models. (Paper J. Claesson.)

The Institute of Applied Geosciences, Justus-Liebig-University, Germany, discussed methods and tools for the installation of vertical heat exchangers. Good solutions have been developed for soft and hard soil. Problems may arise in unstable ground. Backfilling is a problem where an optimal solution is still to be found. Bentonite has proved to be the best compromise for deep holes in the 100 m range and above zero temperatures. Economic considerations, as well as legislative restrictions in Germany, confine the depth for vertical ground heat exchangers to approximately 100 m. In Germany, Switzerland and Austria, the costs for installed vertical heat exchangers, complete with drilling, backfilling etc., range from US\$ 45-70 per metre. (Paper B. Sanner, Prof. K. Knoblich)

Louisiana State University, Dept. of Agricultural Engineering, USA, studies and tests the influence of backfilling material on the performance of the ground-source heat exchanger. The analysis shows that dense materials are better for thermal conductivity and that a high ground-water table will improve grout conductivity. Grout tests in Louisiana, where the ground-water table is within four metres of the ground surface, revealed small differences in three

describes a spiral heat exchanger design concept and indicates the installation procedures. The spiral heat exchanger is made of 150 m (500ft) of HDPE tubing, 32 mm (1.25 inch) in diameter which is wound to a coil of 0.6 m (2 ft) in diameter with a stretched length of 20 m (70 ft) and installed in a 1.8 m (6 ft) deep ditch (see front cover of Newsletter). Sand is used for backfilling. The ground conditions must be such that the sand remains wet.

The results of GSHP systems installed in one experimental and several commercial buildings were also presented.
(Paper O.J. Svec)

Research and Development in Support of the Canadian GSHP Installation Standard

Caneta Research International, Canada, presented the topics being investigated in a co-ordinated programme:

- ice lensing experiments (Investigating organisation: Ontario Hydro);
- current backfill practice (Acres International);
- ground-coil monitoring (Caneta Research International);
- anti-freeze agents (ORTECH International, see separate paper mentioned previously);
- tremie grouting of boreholes (Golder Associates Ltd.);
- spiral heat exchangers: 59 heat exchangers will be field tested (National Research Council Canada, see separate paper).
(Paper R.L.D. Cane)

Earth Energy and Canadian Society

In Ontario, it is intended to reduce the risk of failure of heat pump purchase incentive programmes due to bad product performance, through standards, and a programme incorporating mandatory training and a

consumer protection plan. This programme is administered jointly by the Canadian Earth Energy Association (CEEAA), the industry's self-regulating body, and the utility Ontario Hydro. The programme has been implemented to protect a demand side management (DSM) strategy that provides a CN\$ 2,000 tax free incentive to residential consumers that are outside areas served by natural gas. In Ontario, the potential is 281,000 units. Two hundred and sixty contractors are accredited by the CEEA to sell and install earth energy systems. Accreditation is mandatory and requires that companies are sponsored by a distributor or manufacturer with products approved under the Energy Efficiency Act of Ontario. Company personnel must be factory trained and/or have attended CEEA's accreditation programme. Contractors are required to have a liability insurance of CN\$ 1,000,000 and approved contractor status under the EnerMark Programme of Ontario Hydro. They are also required to hold a full voting membership of the CEEA.

Under the Consumer Protection Plan, inspection of installations, maintaining a data bank about the details of each installation, delivery of warranties, acting as the final resource for consumers, and a 24-hour "Hotline" for messages or concerns are provided. The cost of the plan is CN\$ 365 per installation (CN\$ 60/kW of demand load saving) funded by Ontario Hydro.
(Paper P.J. Scott-Smith)

Group Discussions

Group discussions on three topics related to cost reduction, rating procedures and design methods of ground-source heat exchangers, and direct expansion systems were held. Results include:

- a standard nomenclature for soils and rock and a clear classification of soils are desired;
- when judging GSHPs, the large differences in design and

operation between the northern and southern parts of the USA and Europe must be taken into account;

- a drilling hand book is desired;
- US experience has shown that large projects with a large number of installations at one location lead to considerable cost reduction;
- methods for testing the thermal properties of boreholes and soil sampling should be improved;
- a heat exchanger sizing program (e.g., developed by Lund University) should be adopted as standard and linked with existing building analysis programs;
- design should not only be done on the basis of average capacity, but peaks, storage, local conditions in the borehole, latent heat, etc. should also be included;
- 'rules of thumb' should be discarded;
- direct expansion systems should be designed to minimize refrigerant inventory;
- copper tubes need cathodic corrosion protection or coatings.

Examples of R&D areas include, among others:

- expansion flow control by computer.
- new refrigerants, e.g., propane.
- evaporators for minimum refrigerant inventory.
- international demonstration projects.

The proceedings of the workshop are available to IEA Heat Pump Centre member countries. Please place all orders in writing with IEA Heat Pump Centre. Price: Dfl 60.00. Report no. WR-8.

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XVIIIth International Congress of Refrigeration: Trends in CFCs

**R.J.M. van Gerwen*

The 18th International Congress of Refrigeration, organized by the International Institute of Refrigeration, was held on August 10-17, 1991, in Montréal, Canada. More than 1,600 people from about 50 countries attended this congress. More than 500 papers were submitted. The main topic of current interest in refrigeration and heat pump technology was the CFC issue; about half of all the papers were more or less related to the CFC issue. Below is a short description of the general situation regarding possible alternatives for the most important CFCs. It is followed by a more detailed overview of the topics discussed in the papers.

Alternatives for CFC-12 and R-502

General Situation

In refrigeration and heat pump applications, CFC-12 and R-502 (an azeotropic blend of CFC-115 and HCFC-22) are the most important prohibited refrigerants while HCFC-22 is currently their most important replacement. However, according to many speakers, HCFC-22 is an interim solution because of its still significant Ozone Depletion Potential (ODP = 0.05). Other CFC-12 alternatives include HFC-134a, near azeotropic ternary blends of HCFC-22/HFC-152a/HFC-124 or flammables, such as HFC-152a, propane or Dimethylether.

Alternatives for R-502 include HFC-125, HFC-143a or near-azeotropic ternary blends of HCFC-22/HFC-290/HFC-218 and HCFC-22/HFC-290/HFC-125. The prospect for pure HFC-125 is not favourable, due to its low efficiency at high condensing temperatures. The blends are non-flammable, low-toxic, drop-in replacements and result in almost equal performances as R-502. The blends still have an ODP of 0.03 to 0.04; thus they will probably be only an intermediate solution. Many flammable refrigerants, such as HFC-143a, do have an excellent performance, but they are not yet accepted in practice.

Overview of Papers

HFC-134a

A great deal of research on HFC-134a (replacement for CFC-12) has been reported concerning thermo-

dynamics, performance, heat transfer and lubrication. However, the significant Greenhouse Warming Potential of HFC-134a was often mentioned as a potential problem for applications over a long period. Experimental investigations on motor car air-conditioning application result in an increase in cooling capacity of 5 to 10% and the same COP, compared to CFC-12. (Pettersen, Norway). Many researchers report an increase of the heat transfer coefficient of 10 to 40%, compared to CFC-12.

In general, the practical performance characteristics are equal to CFC-12. The earlier problems concerning appropriate lubricants seem to have been solved by using ester-based oils.

HCFCs

The HCFCs (22, 123, 124) have a slight Ozone Depletion Potential (ODP) of 0.02 to 0.05. Many speakers considered HCFCs, particularly HCFC-22, as an interim solution. In the future, more rigorous restrictions on the use of HCFCs, including blends containing HCFCs, should be expected.

HFC-125

The picture for HFC-125 as a proposed direct replacement for R-502 is not positive, mainly because of its unacceptable low efficiencies at high condensing temperatures (Stoecker, USA; Sundaresan, USA). Results of toxicity tests are not yet available (Forbes Pearson, UK). Production quantities are expected to be available in 1995. As a component in mixtures, it may be a promising refrigerant.

Ammonia

Ammonia obtained little attention, in spite of its excellent performance features. The main problem with ammonia is its toxicity. In some countries, ammonia is often used in large industrial plants, in combination with adequate safety measures. The thermophysical properties, found in literature, are not consistent. Additional experimental research was proposed (Keizer, the Netherlands).

Flammable Refrigerants

Many refrigerants, particularly the hydrocarbons (HCs), show very good performance characteristics,

but are flammable. HFC-152a, a promising, but flammable, replacement for CFC-12, results in a heat transfer coefficient which is 43% greater than for CFC-12 (Shi, Germany). Propane performed better than CFC-12 for refrigerated transport applications, but worse than HCFC-22. LPG performs nearly the same, but is cheaper (Boldrin, Italy). The results of flammability tests, according to the ASTM E 681 method, were reported for HFC-32, HCFC-141b, HCFC-142b, HFC-152a, HFC-143, HFC-143a and mixtures (Richard, USA). Currently, flammables are not yet accepted in practice.

(Near) Azeotrope Mixtures

Blends of HCFC-22, HFC-152a and HCFC-124 (with an ODP of 0.03) are proposed as 'drop-ins' for CFC-12, with an equal or slightly better performance (Curtis Lawson, Sundaresan, USA). A blend of HCFC-22, HC-290 (propane) and HFC-218 (ODP=0.04), as well as a blend containing HCFC-22, HC-290 and HFC-125 (ODP= 0.03), are proposed as replacements for R-502. These blends are non-flammable, low toxic, 'drop-in' replacements, with a slightly better COP and an equal refrigerating effect (Forbes Pearson, UK, Sundaresan, USA). These blends, with ODPs greater than zero, may not be considered as long-term alternatives (see also the News Column of this Newsletter).

Non-azeotrope Mixtures (NARMs)

Many papers dealt with NARMs. Evaporation and condensation take place at gliding refrigerant temperatures which, under certain conditions, may result in energy savings of 10 to 30%. Unfortunately, most of the research reported deals with NARMs based on CFC components. Some non-CFC mixtures mentioned include a HFC-143a/HCFC-124 mixture, used in a heat pump (Vineyard, USA) and a HCFC-22/HFC-23 mixture used in a low temperature refrigeration system, as a replacement for R-502 and BrFC-13B1 (Marioton, France). In the future, NARMs are expected to become more and more important as a CFC replacement as well as an energy saver.

Alternative Cycles

Theoretical and experimental research on the air cycle, using air as the refrigerant, was presented (Kauffeld, Germany). This cycle has many advantages and is one of the most promising alternatives for the conventional vapour compression cycle. Energy efficiency largely depends on the efficiency of the compressor and expander used in this cycle.

Leakage, Reclaiming, Recycling

Leakage prevention, reclaiming and recycling of refrigerants are very important contributions to the

solution of the CFC problem. However, little attention was paid to this subject. Some experience with reclaiming systems in France has been reported (Brian, France).

Conclusion

Nowadays, the CFC issue is a central theme in refrigeration and heat pump research and technology. However, the final solution is not yet available. Many measures, varying from using new refrigerants to leakage prevention, do contribute to the solution. The amount of information, obtained by world-wide research, is so overwhelming that the transfer of this information into practice is of great importance.

The complete proceedings of the Montreal Congress will be available from the end of this year at the International Institute of Refrigeration in Paris.

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Air-Conditioning Heat Pumps in Retrofit Installations

*P. Fahlén

Abstract

Installation of split-type air-conditioning heat pumps is now the most popular choice among viable energy conservation opportunities in private homes in Sweden. Some important aspects regarding the application of these heat pumps in the heating mode in retrofit installations in a cold climate must be considered. This may, for instance, concern performance under frosting conditions, auxiliary equipment to cope with low ambient temperatures and the efficiency of heat distribution when heat is supplied from a single fan-coil unit. Laboratory tests and investigations performed in actual houses to provide some background information for this type of installation, are described in this paper.

Introduction

In Sweden approximately half a million private homes are heated by electric base-board heaters and thus lack traditional heat distribution systems. In the search for energy conservation opportunities in Sweden the heat pump is, in general, the most viable proposition once any obvious flaws in the building envelope and ventilation system have been remedied. The traditional use of heat pumps in Sweden has required the existence of a heat distribution system. Retrofit installation of air ducts or hydronic heating systems will, in general, prove to be too expensive and therefore the application of split-type air-conditioning heat pumps has increased dramatically since 1988.

Air-conditioning heat pumps are reasonably priced due to the large number of units sold in Japan and the USA. Furthermore, the installation is fairly simple and inexpensive since the indoor fan-coil unit is self-contained and does not require any traditional heat distribution system. Sales figures in Sweden now range about 10,000 units per annum and this warrants taking a closer look at the possibilities of actually obtaining the savings promised by the dealers. Some important prerequisites that must be met, include:

- the amount of actual energy used in the house for heating hot water and household electricity must be known;
- the size of the heat pump must be technically and economically optimized;

- when there is a heat demand the heat supplied by the heat pump must always be usable;
- actual performance data must agree with values used in economic calculations;
- the heat pump must not negatively affect the comfort of the people, e.g., noise levels, thermal comfort, ventilation, etc.

Typical Air-Conditioning Heat Pumps

Split-type air-conditioning heat pumps operate with outdoor air as the heat source and recirculated indoor air as the heat sink. The compressor, throttling device, reversing valve, evaporator and outdoor air fan are all fitted in the outdoor unit. The indoor unit contains the condenser, indoor air fan, filter and control unit.

All heat pumps considered in the context of this paper are of the split-type and use single or variable speed rotary piston compressors with refrigerant R-22. The refrigerant charge is in the range of 0.85-1.2 kg. None of the units use any form of receiver tank and, when combined with the fairly small design capacities, this means that the systems are sensitive to the correct quantity of refrigerant charge.

Throttling devices are made up by various devices, i.e., capillary tubes, thermostatic expansion valves or a combination of these. Reversing valves are always fitted to provide the possibility of operation in both the heating and cooling modes as well as providing a means of hot gas defrosting. Defrosting is one of the most important aspects when air-source heat pumps are operated in cold climates. With regard to air-conditioning heat pumps, this is all the more important since outdoor units are, in general, designed as condensers (fin separation in the range of 1.3-2.25 mm).

Indoor units generally contain textile or synthetic air filters and, in some cases, electrostatic filters. The indoor air is circulated by 'step by step' variable speed fans providing maximal air flows between 400 and 600 m³/h. Electrical input required by the indoor fan is about 20-60 W. The outdoor units use fixed speed fans with flows of approximately 1,200-1,700 m³/h.

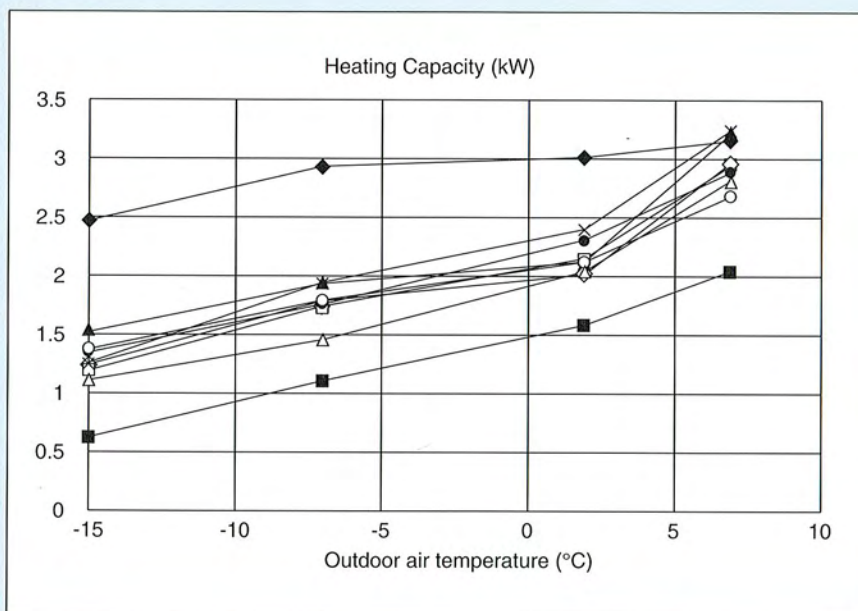


Figure 1: The heating output of nine air-conditioning heat pumps as a function of the outdoor air temperature. The flatness of the top curve is due to an inverter controlled compressor being operated at maximal capacity for all test conditions.

Laboratory Test Results

To evaluate actual performance of the air-conditioning heat pumps under consideration the Swedish Council for Building Research, in co-operation with the Consumers' and the Manufacturers' organizations, commissioned a laboratory test of nine units at the Swedish National Testing & Research Institute (SP). The test was carried out in accordance with the Swedish standard SS 2095, (equivalent to Pre European Norm, prEN 255). Extended measurements were performed to study the operation of defrosting systems at an outdoor temperature of 2°C.

Measured values of the heating output, and the coefficient of performance (COP) are presented in

Figures 1 and 2 as a function of the outdoor air temperature. The air inlet temperature to the indoor coil is 20°C in all cases.

The tests indicate that, at an outdoor temperature of 2°C, the defrosting frequency varied between once every two hours for two of the units to once every hour or more for the other units. This frequency is higher than the optimal value with regard to both the heating capacity and the COP for all of the units. Several of the heat pumps operate in practice with a fixed time-temperature control rather than according to the designed demand control strategy.

The tests that were performed with prolonged operation under frosting conditions did not indicate any gradual degradation in performance. In this test

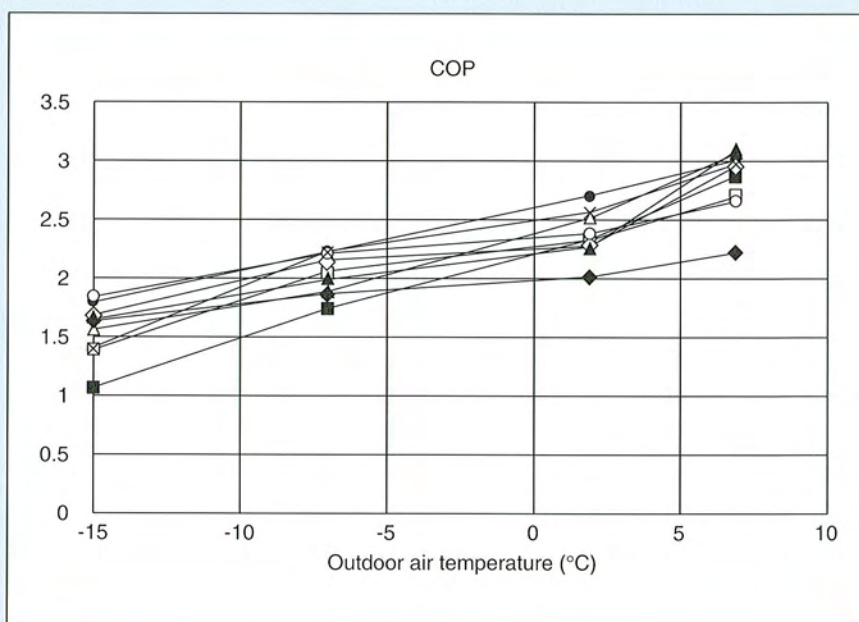


Figure 2: The coefficient of performance (COP) of nine air-conditioning heat pumps as a function of the outdoor air temperature.

instantaneous values of COP were not evaluated but previous investigations [1] showed that the average value over a full frosting/defrosting cycle was 5-15% lower than the instantaneous value with clean heat exchanger surfaces.

The emitted sound power ("noise") was determined for the indoor units of all heat pumps in accordance with ISO 3741. Results ranged between 3.2-5.0 B at the minimum fan speed (mean value = 4.4 B) and between 4.6-5.5 B at the maximum fan speed (mean value = 5.1 B). Results from measurements in the octave bands indicated that all units had a maximum value at a frequency of around 500 Hz. The emission from one of the outdoor units was measured to be LWA = 6.0 B with the maximum at a frequency of 250 Hz. The A-weighted sound power is expressed in Bels to avoid confusion with the sound pressure level which is stated in decibels (1 B = 10 dB).

Aspects of the System Design

Test results concerning heating capacity and COP, as presented in Figures 1 and 2, pertain to inlet air temperatures of 20°C to the indoor unit. If the inlet temperature should exceed 20°C then both the heating capacity and the COP would decrease due to a raised condensing temperature. The change in heating capacity ranges between -0.5 to -1.5% and the change in COP between -1.0 to -1.5% per °C change in inlet temperature [1].

The sensitivity of heat pumps to the temperature level also implies that unsuitable positioning of the indoor unit will have detrimental effects on performance for two reasons. Firstly, the operating time of the heat pump will be reduced due to inefficient and slow distribution of heat to the different parts of the house causing premature shut-off by the heat pump thermostat (which is described in more detail below). Secondly, the heating capacity and COP are reduced because of an increased return air temperature (recirculation of warm air).

Frosting will, in practice, always lead to a deterioration in the efficiency of the system, partly because of a reduced number of operating hours for heating due to the defrosting periods, and partly because of the reduced heat transfer in a frosted outdoor heat exchanger. Thus, it is vital that performance data for heat pumps include data for frosting and defrosting at outdoor temperatures at which, in practice, frosting occurs. The number of defrosting cycles will affect the heat pump both with respect to its efficiency and its life span. Each time the system reverses will impose an extra load on the compressor.

Practical experience from operation in a cold climate indicates the necessity of heating the drip tray in some way. It is also advisable to equip compressors with

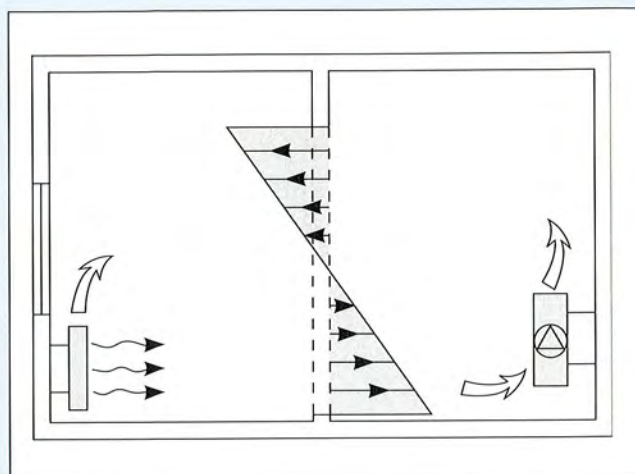


Figure 3: Air exchange through an open door due to temperature differences.

crank-case heating. In spite of the fact that all manufacturers state a minimum outdoor operating temperature (which varies between -8 to -20°C) only four of the nine tested units had an outdoor thermostat. When this thermostat is missing the owner must act as the thermostat and manually shut the heat pump off at low outdoor temperatures.

Heat Distribution

The indoor unit of the heat pump is installed in a central location to ensure the best possible distribution of heat throughout the house. Even distribution of heat is a general problem for this kind of installation. The operation of the distribution system relies on temperature gradients to transport heat to the various parts of the building. The influence of forced convection reduces quickly with increasing distance from the source. This means that the temperature close to the source must exceed the design room temperature value and that temperatures far from the source will be below this value. The magnitude of these temperature differences will depend upon the zone partitioning of the building (open spaces or closed rooms), the building envelope (insulation, windows) and the ventilation system (natural, mechanical exhaust, balanced, type and positioning of terminal devices, etc.). This dependence is complex and results in significant variations in the ability of different systems to distribute heat inside the building.

Figure 3 indicates possible air flows between two rooms. According to Norberg [2] the air exchange through a door is proportional to $(\Delta t)^{0.5}$. This may generate air flows through doors which, each in their own right, are as large as the total fan assisted air flow from the indoor unit of the heat pump (400 - 500 m³/h).

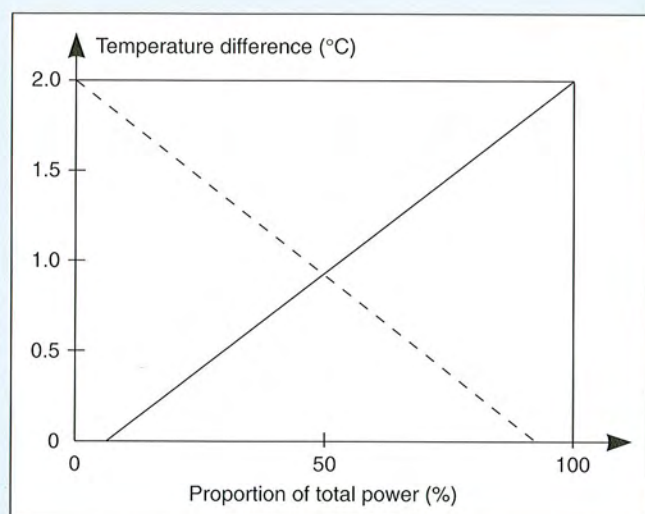


Figure 4: Temperature difference between the indoor unit of the heat pump and the room to be heated as a function of the desired proportion of heating power covered by the heat pump [2]. Note: The interrupted line indicates the corresponding heating power supplied by the electric radiators.

The dependence of heat transport on temperature differences inside the building creates the most important problem concerning the combined system of "the air-conditioning heat pump/the existing heating system/the building". When there is a need for supplementary heating from, e.g., electric base board radiators there is a risk of greatly reducing the heat supplied by the heat pump unless the control of the radiators is managed in an appropriate way. Figure 4 demonstrates the temperature difference that is required between the installation site of the indoor unit of the heat pump and a heated room in order for the heat pump to cover a specified part of the total heating power supplied to the house.

A measure of the efficiency of the heat distribution can be obtained by stating the proportion of warm air that is directly recirculated to the inlet of the fan-coil unit (see Figure 5). The heat distribution efficiency, η_{distr} , is thus given by

$$\eta_{\text{distr}} = q_{\text{room}}/q_{\text{in}} = (t_{\text{out}} - t_{\text{in}})/(t_{\text{out}} - t_{\text{room}})$$

Comfort Criteria

Other problems that may be encountered when heat is supplied from a central point source concern the thermal comfort and the air quality. Due to the operational principle of the distribution system temperature asymmetries may arise causing discomfort if they are too large. Furthermore, the portion of radiant heat will decrease when the ordinary radiators are not in operation which leads to an increase in the air temperature required to maintain thermal comfort. High air velocities in the region of the fan-coil unit adds to the requirement of increased air temperature.

The heat distribution flow is directed from the central part of the house towards the peripheral parts whereas

the supply of fresh ventilation air normally goes in the opposite direction. This means that, in most cases, the ventilation system will be affected by the installation of an air-conditioning heat pump. A central fan-coil unit also operates as a return air system and may cause pollutants from one part of the house to spread throughout the building. In practice, however, many existing houses have ventilation flow rates far below the values considered necessary to maintain an acceptable indoor air quality. In these cases the increased circulation of air between rooms, providing a larger diluting air volume, in combination with the cleaning effect of filters in the fan-coil units often result in people experiencing an improvement of air quality.

The noise level is also an important comfort criterion. The Swedish building code specifies a maximum background sound level of 30 dB in bedrooms and living rooms. Converted to an A-weighted sound power level, L_{WA} , this means that $L_{\text{WA}} < 3.4$ B complies with the requirement. The test results (4.6 - 5.5 B) indicate that none of the indoor units is quiet enough using the maximum fan speed and only one unit is sufficiently quiet with the minimum fan speed. However, even with these levels few people actually complain of noise problems with these types of installations.

Conclusion

Results from the evaluation of retrofit installations with air-conditioning heat pumps presented here, indicate that these heat pumps work reasonably well if the installation and refrigerant filling are carried out correctly. It is, however, worth noting that during the laboratory tests four of the nine units tested had to be retested due to malfunctioning or incorrect refrigerant charge. Deviations from expected performance can be detected when the heat pump is installed on a test stand. However, it remains

t = temperature
 q = heat

Sweden which hopefully save both energy and money for their owners. In most cases, the owners also seem quite pleased with their installations. Real world savings are, however, questionable. Measurements performed in commercially made installations as well as in experimental buildings indicate that true savings may vary between 40-90% of the theoretically possible savings, largely due to the inefficient distribution of heat.

References

- [1] *Fahlén, P. 1988. Laboratory Testing of Heat Pumps - Experience 1984 - 1986. (The Swedish National Testing and Research Institute). SP Report 1990:19. Borås, Sweden.*
- [2] *Norberg, C. 1990. Control of Indoor Air Temperature in Partially Converted Single Family Homes (in Swedish). (Vattenfall Utveckling AB). Report VU F 90:7. Älvkarleby, Sweden.*

Regrettably it must be noted that presentation of performance data is an area where most suppliers have to improve. It is virtually impossible for a potential buyer to compare data from different types of heat pumps. Hopefully this will improve in the near future when the European standard EN 255 is introduced. Nevertheless, there are more than 25,000 air-conditioning heat pumps in retrofit installations in

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News and Views



IEA Heat Pump Centre introduces Minie Wilpshaar

Minie Wilpshaar was born in 1946 in Schoonebeek, the Netherlands. In the 1970s she worked as a qualified Intensive Care nurse in the cardiac surgery department of the University Hospital in Leiden. During the 1980s she lived in Glasgow,

Scotland. After her return to the Netherlands she prepared for yet another profession. In 1990 she completed her secretarial education and as of August 19, 1991, she became responsible for the office management of the HPC, taking over from Roswitha Muyres who joined the publication support group.

Market News

World Demand 1988-1990 and World Supply 1988-1989 of A/Cs

In Newsletter Vol. 8, No. 3 (Sept., 1990) the estimate by the Japan Refrigeration & Air Conditioning Industry Association (JRAIA) of world demand and supply of AC/units in 1988 was quoted. JRAIA has now updated the estimate to include demand in 1989 and 1990 and production in 1989. Table 1 shows the results for world demand, Table 2 those for world production and Table 3 Japanese Exports. In Japan, "room A/Cs" are defined as window types and split types for home use, whereas "packaged A/Cs" are regarded as all other A/Cs which are manufactured and sold for commercial and industrial use.

Table 1: Estimated World Demand for A/Cs (1988 - 1990) in 1,000 units.

1. Room A/Cs			
	1988	1989	1990
World total	14,656	15,728	16,261
Japan	4,641	4,824	6,286
South-east Asia	2,755	3,074	2,876
Middle East	868	1,033	984
Europe	715	992	1,017
North America	4,452	4,805	4,060
Latin America	851	586	617
Africa	217	234	244
Oceania	157	180	177

2. Packaged A/Cs

	1988	1989	1990
World total	5,620	6,058	5,662
Japan	724	809	972
South-east Asia	262	228	315
Middle East	92	124	126
Europe	163	186	203
North America	4,278	4,600	3,920
Latin America	50	56	60
Africa	23	23	28
Oceania	28	32	38

Table 2: Estimated World Production of A/Cs (1988 - 1989), in 1,000 units.

1. Room A/Cs

	1988	1989
World total	14,656	15,728
Japan	5,862	6,031
South-east Asia	3,211	3,669
Middle East	369	464
Europe	516	603
North America	3,505	4,101
Latin America	1,033	685
Africa	125	125
Oceania	35	50

2. Packaged A/Cs

	1988	1989
World total	5,620	6,058
Japan	884	965
South-east Asia	177	178
Middle East	19	26
Europe	118	120
North America	4,377	4,726
Latin America	29	26
Africa	4	4
Oceania	12	13

Table 3: Japanese A/C Exports (1988 - 1989), in 1,000 units.

1. Room A/Cs

	1988	1989
World total	1,306	1,335
South-east Asia	642	521
Middle East	333	338
Europe	194	322
North America	52	55
Latin America	13	13
Africa	24	24
Oceania	48	62

2. Packaged A/Cs		
	1988	1989
World total	162	156
South-east Asia	62	36
Middle East	40	42
Europe	33	47
North America	9	12
Latin America	2	2
Africa	6	7
Oceania	10	10

As indicated in Table 1, world demand for room A/Cs increased by 7.3% in 1989 and 3.4% in 1990. In 1990, Japan's demand accounted for 38.7% of world demand. World demand for packaged A/Cs increased by 7.8% in 1989, but decreased by 6.5% in 1990. In 1990, Japan's demand accounted for 17.2% of world demand. In Japan as far as heat pumps are concerned, the share of room heat pump A/Cs, based on shipment figures, increased from 58.5% in ref. year 1988 to 71.5% in 1990; for packaged heat pump A/Cs from 75.8% in 1988 to 78.0% in 1990. (Source: JARN, August 25, 1991.)

Shipments

First-quarter A/C Shipments Dropped Sharply in Canada

Commercial A/C shipments - i.e., rooftops (combination heat/cool), packaged cooling, and packaged heat pumps - declined from 7,722 in the first quarter of 1990 to 4,356 in the first quarter of 1991, a decrease of 44%. Residential A/C shipments - i.e., split-system air-conditioning and heat pumps up to 5 tons - dropped from 43,319 in the first quarter of 1990 to 11,640 in the first quarter of 1991, a decrease of 73%. (Source: the NEWS, July 29, 1991.)

Skyrocketing Demand for Room A/Cs in Japan

The Japanese Refrigeration and Air Conditioning Industry Association reports that in the period from October 1990 to the end of June 1991, shipments reached 5,443,610, an increase of 32.6% compared to the same period a year ago. (Source: JARN, September 25, 1991.)

New Heat Pumps on the Market in the US

Lennox: Dual Fuel Packaged Heat Pump

The dual-fuel heat pump described in the HPC Newsletter Vol. 8, No. 1 (April, 1990) pp. 20-23 is now being produced in 7.5 and 10 ton models by Lennox Industries Inc., USA. The packaged heat pump combines a standard single package electric heat pump with a gas furnace for supplemental heating during the coldest months. Developed in co-ordination with the Electric Power Research Institute (EPRI), the "FuelMaster" is offered as an alternative to conventional packaged electric cooling/gas heating systems. The FuelMaster features two compressors in two separate refrigeration circuits, providing two stages of cooling.

During test-marketing with Pudget Power in Washington, the unit was found to provide savings that the company estimates should pay back its costs in a little over two years. Refer to the quoted Newsletter for further details.

York International: Gas-engine Powered Heat Pump

York International plans to market the gas-engine driven heat pump, which is powered by a 5 hp (3.7 kW) Briggs & Stratton Corp. engine. It is described in Newsletter Vol. 8, No. 3 (Sept., 1990) pp. 15-18. Research for developing the gas-powered heat pump was sponsored by the Gas Research Institute. York plans to begin limited marketing of 50 units in 1992, with full scale marketing in 1993. Current focus is on the residential retrofit market. Once full-scale commercialization is under way, the market for the units should be about 100,000 annually. The first units available will be 3-ton cooling capacity split-type systems, similar to the 10 prototype units that have undergone field tests for the past 18 months or more. Units with capacities of 2-1/2 and 4 tons are planned, along with single package units for commercial applications with capacities of 4 and 8 tons. The latest engineering analyses, supported by field test results, are said to indicate COPs of 1.40 in heating, and 1.26 in cooling. The service life of the engine is expected to be 40,000 hours, or about 10 years. The only regular maintenance required will be annual replacement of oil, oil filter, air filter, and spark plugs.

Hydro-Temp Corp.: Two-speed Compressor Commercial Earth Coupled Heat Pump

In Hayti, Missouri, USA, the first two-speed compressor earth-coupled heat pump in the commercial sector of the US was recently installed. Hydro-Temp Corp. has been producing two-speed earth-coupled units for use in homes since 1989.

Due to the climate in Missouri, the heating load is often twice that of the cooling load. In the past, contractors either had to size the unit for heating which implied that it was 50% over-sized for cooling, or size the unit for cooling and make up the 50% difference for heating with exquisite electric resistance heat. Using the two-speed unit, the customer can have the exact cooling and heating needed without the use of auxiliary electric heat. (Source: The A/C, Heating & Refrigeration NEWS, Sept. 16, 1991.)

HVAC Market in France is Growing

French manufacturers of A/C installations and equipment intend to increase their sales by 4 times in the coming years. In 1989, the sales had already increased by 20%. The future goal shall be reached by a better information of the potential users and a better training and education of the installers. It is anticipated that installation firms will have to undergo an accreditation procedure by the "Union Climatique de France", in which the leading firms of the branch are represented.

The Co-operative for Climatisation and Development, established by Airwell, Ciat, Carrier, Technibel, and Trane,

intends to increase the air-conditioned building surface area with at least 6.5% of the total building surface area in the sales buildings and offices sectors. They intend to prime the use of A/C in private houses as well. The electric utility "Electricité de France" (EDF) is also involved in stimulating the use of decentralized A/C. The rationale behind the activities of the A/C and utility branches is the realisation that France is under-developed in regard to the use of A/C. In France, the FF 1 billion* sales volume of A/C equipment is about 40% below that in Italy. The share of A/C equipment of the Gross National Product in France is only 60% of the share in Germany and 35% of the share in the UK. To date, the market in France concentrates mostly on public and commercial buildings which have a share of 82% of the A/C market volume. The share of small tertiary buildings (surface area < 400 m²) is only 13% and that of the residential sector 5%. Individual A/C installations make up 18% of total units sales. Of the 80,000 individual A/C installations sold in 1988 in France, about 51% were modern split-units, 33% window types, 6% portables and 10% were other although not separately recorded. Heat pump use is increasing.

Share of Floor Heating Growing in Germany

Central heating by means of hydronic systems using radiators is one of the obstacles to the diffusion of heat pumps in Western and Central Europe. A survey in Germany showed that more than 50% of all newly built one and two-family houses are provided with floor heating. Floor heating is also spreading to other sectors, including industrial buildings and sportshalls. (Source HLH, August, 1991.)

Refrigerants/CFCs

German CFC Prohibition Law in Effect as of August 1, 1991

The main requirements, concerning R-11, 12, 13, 112, 113, 114, 115, and also R-22, are:

- Containers must be marked "Contains Ozone depleting CFC";
- Within the possibilities of the existing technique, it is prohibited to release the refrigerants to the atmosphere during operation, servicing of installations or taking them out of operation;
- Servicing of installations and taking those out of operation that contain the above refrigerants, and the recovery of these refrigerants may only be done by accredited personnel;
- Type, quantity and storage location of the recovered refrigerant must be recorded. Records must be kept for a period of three years.

(Source: Die Kälte und Klimatechnik, 7/1991.)

Lowering of Acceptable Exposure Level Recommended for R-123

Scientists testing alternative refrigerants in the context of the Panel for Alternative Fluorocarbon Toxicity Testing (PAFT) recommended to their members (manufacturers of refrigerants), at the end of June, to reduce the present Acceptable Exposure Level (AEL) for R-123 from 100 to 10 ppm. The AEL is the recommended time-weighted average concentration to which workers may be exposed during an 8 hr/day, 40 hr/week without adverse effects. In Europe, that is equal to the MAC-value (maximum allowed concentration). The reason for this recommendation is said to be due to the fact that tumour formation (non-carcinogenics) was observed in rats which were tested over a long period of time. This reduction of the AEL would have important consequences for the use of R-123 as an alternative. The quoted source (Kälte und Klimatechnik) comments that it will take at least one and a half years before scientists will have determined, if at all, which trace of other materials within R-123 could be responsible for the observed test results. (Source: Die Kälte und Klimatechnik 7/1991.)

Proposed R-125/R-22/Propane mixture as Alternative to R-502 is at Best a Temporary Replacement in Germany

At the ASHRAE Congress in Indianapolis, USA, Du Pont-USA has suggested the above mixture as an alternative to R-502. The mixture contains 6% propane and is non-flammable. The capacity is slightly higher than R-502 and the COP is comparable. The source quoted below notes that, since it contains R-22 as one of the important constituents, in Germany this 'drop-in' refrigerant will, at best, be an alternative for the short time between the period when R-125 becomes available and R-22 is prohibited. (Note HPC: Isceon 69-S marketed by Rhone-Poulenc Chemicals, UK, as an alternative to R-502 also contains R-22, see Newsletter Vol. 9, No. 3.) (Source: CCI, Sept. 1991.)

ASHRAE's Position Statement about Ammonia

The following is extracted from ASHRAE's position statement: ASHRAE considers that the continued use of ammonia is necessary for food preservation and air-conditioning. ASHRAE promotes a variety of programmes to preserve the economic benefits of ammonia refrigeration while providing for management of risks. ASHRAE will:

- Promote authoritative information on ammonia by seminars and publications;
- Continue research on ammonia topics such as handling, application, operation and control of emissions;
- Maintain and develop standards and guidelines for practical and safe application of ammonia in refrigeration systems;

- Provide programmes and publications of innovative designs and applications;
- Advise governments and code officials with information regarding ammonia.

(Source: ASHRAE Journal, September, 1991.)

Total Reduction of CFCs in the Netherlands Ahead of Schedule

The first annual report on achieved CFC reduction was published May 1, 1991, and showed a reduction of 38%, in contrast to the 31% that had been foreseen in the CFC Action Programme. In the fields of foams and aerosols especially, a fast reduction was observed. Reduction in the refrigeration sector was 208 tonnes compared to reference year 1986. This reduction of about 22% relates approximately to the planned forecast. As of January 1, 1992, only accredited firms may work on CFCs and HCFCs containing refrigeration installations. (Source: Koude & Luchtbehandeling, July, 1991.)

Systems/Installations

Realization of a 1.5 MW Heat Pump with R-142b in France

An electric heat pump installation of 1.5 MW working with R-142b has been operating satisfactorily for several months in the town of Montpellier, France. The heat pump provides air-conditioning and heating for a group of buildings including the town hall, a local authority administrative services building, hotels and schools. The total office area is 800,000 m². The heat pump also supplies 80-85°C hot water from 25°C water. The COP is quoted to be above 3.4, the threshold value for economic profitability. The project is based on work by the French electric utility "Electricité de France" (EDF) which has a realized prototype installation of 100 kW. The supplier for the 1.5 MW installation is the French firm QUIRI that extrapolated the EDF study results. In total R&D efforts including solving the problem of finding an oil compatible with R-142b in such a large installation, have taken three years. A detailed technical study of the installation will appear between now and the end of the year in the quoted magazine. (Source: Revue General du Froid, July, 1991.)

EPRI/Carrier Hydrotech 2000 Heat Pump in Electric Smart House Demonstration Project

In suburban Atlanta, Georgia, USA, a 3,000 ft² (280 m²) Electric Smart House illustrates what 'whole-home' automation can offer today. About 40 utilities joined forces to fund the project. The Edison Electric Institute (EEI) managed the construction and public information aspects of

the project. The Electric Power Research Institute (EPRI) handled the technical aspects. Between January and August 1991, the house was open to the public. Until June, an average of 900 visitors per week came to tour the house. One of the highlights was an integrated variable speed heat pump developed by EPRI and the Carrier Corporation. It provided heating and cooling, as well as hot water, all year-round. Introduced to the market in 1989, it is 30% more efficient than conventional units. The variable speed compressor and indoor fan allow the system to track indoor temperatures, save energy, and provide greater comfort, and allow for dehumidification and gradual start-up, and eliminate the blast of cool air of conventional heat pumps when starting up on cold winter days. When in the cooling mode, the unit recovers waste heat rejected from the house and uses it to heat water. In the winter months, water is heated from the same heat source the HydroTech extracts from the outside air to heat the house.

The unit provides multi-zone space conditioning, has self-monitoring and diagnostic capabilities, and offers the possibility for utility load management. The outdoor noise is about one-eighth of a conventional unit. The indoor unit that contains the compressor is as quiet as a refrigerator. In the Electric Smart House, the HydroTech works together with a second variable speed pump, Carrier's Infinity 2000, a spin-off of the HydroTech that does not include the water heating feature. (Source: EPRI Journal, June 1991.)

Codes & Guidelines

New Heat Pump Brochure from the European Community

A brochure called, "Heat Pumps - an Option for an Energy Efficient and Clean Society", has been issued by the Directorate General for Science, Research and Development (DG XII) of the European Commission (EC). The EC concludes that the heat pump could satisfy up to 30% of the overall primary energy demand; mainly in buildings and industry and particularly affecting that supplied by fossil fuels. By initiating a concerted action of R&D, backed up by education and awareness programmes, and with the support of, in particular, the marketing skills of the utilities, the Commission hopes to raise the profile of the heat pump technology and application throughout Europe. The combined action will include, at an early stage, discussions with influential parties and the formation of a Working Group in which all of these parties will be represented. The brochure sets the scene for the launch of this concerted action by describing the heat pump, the contribution it might make to Europe's environmental future, and the role of those in a position to influence the direction the strategy takes. Countries participating in this co-operative action are France, Germany, Italy, the Netherlands, and the United Kingdom.

IEA and Member Countries

Governmental Funding in Some Member Countries for Heat Pump RD&D

JAPAN

Tables 1a & b show the data for Japan. In addition, the Ministry of International Trade and Industry (MITI) provides the Japanese contribution to the IEA HPC and supports the Heat Pump Technology Center of Japan (HPTCJ) for approximately US\$ 0.5 million.

Table 1a: Governmental Funding for RD&D of Heat Pumps in Japan.

Programme	Start	Term (Years)	Expected total budget (US\$ Million)	1991 Budget (US\$ Million)
Super HP, En. Accumulation System	1974	9	Approx. 70	Approx. 10
Unused Energy Utilization Tech. R&D	1991	7	Approx. 55 (50% Subsidy)	Approx. 3 (50% Subsidy)

Table 1b: Organisations Involved in Japan.

Programme	In Charge	Organizer	Participants
Super HP, En. Accumulation System	Agency of Ind Scie.& Techn. MITI	Techn. Res. Association of SHP Energy Accum.	2 Nat. Labs 2 Associations 17 Companies
Unused Energy Utilization Tech. R&D	Agency of Natural Resource & Energy, MITI	Heat Pump Techn. Center of Japan	2 Associations 16 Companies

(Source: HPTCJ.)

NETHERLANDS

RD&D funding

- In the Netherlands, the average governmental funding, by the Ministry of Economic Affairs, for heat pump RD&D is approx. US\$ 1.7 million/year, which includes a contribution to the IEA HPC and support of the National Team (approx. US\$ 0.2 million).

Grant Scheme "Demonstration Projects"

- Innovative heat pump projects can apply for a 40% demonstration investment subsidy, with a maximum of DFL 1 million (approx. US\$ 0.5 million), for projects exceeding DFL 100,000.

Grant Scheme "Investments in Energy Saving Techniques"

- Investments in heat pumps receive a grant of 25% in the context of a US\$ 5 million grant scheme for investments in energy recovery systems, automated control systems, frequency transformers, and heat pumps.

In the Netherlands, a new heat pump policy is in the process of being formulated.

NORWAY

New and existing buildings

- A 20% subsidy is granted (except for the domestic sector) for marginal investments relative to other heating systems, unless the internal rate of return is higher than 30%.
- Loans at an 11% interest rate can be obtained for a maximum of 80% of the investment for heat pump installations in municipal buildings, and at an 11.5% interest rate for installations in private industry buildings, provided the investment for the latter sector exceeds NOK 100,000 (approx. US\$ 15,000).

Prototype and demonstration plants

- Contributions are granted to cover additional risks and extra costs connected with demonstration plants. Prime candidates are plants utilizing new refrigerants.

Programme for Energy Efficiency and Environmental Sound Heating Systems

- A maximum subsidy of 50%, not exceeding NOK 100,000 is granted for all types of buildings in certain geographic areas of Norway.

Single-Family Houses

- A national subsidy programme will be implemented by January 1992. Some utilities have their own grant schemes.

(Source: Norwegian Nation Team.)

UNITED STATES

Table 2 shows the Department of Energy (DOE) funding for fiscal year 1992 (October, 1991 - September, 1992). In addition, significant non-government agency funding by, for instance, the Electric Power Research Institute and the Gas Research Institute, exists. Therefore, the DOE funding does not give a complete picture for the USA.

Table 2: US DOE budget for R&D on heat pump technologies for fiscal year 1992.

Programme Description	Funding * (US\$ Million)
Residential/Commercial	
• Gas fired heat pumps and chillers	5.1
• CFC Replacements	3.3
• Miscellaneous Heat Pump	1.0
• Domestic Refrigerators	0.6
Industrial	2.0
Total	12.0

* Note: Probable allocations; subject to change.
(Source US DOE.)

Advice from the Netherlands National Team to the Ministry of Economic Affairs

Earlier this year, the Netherlands Ministry of Economic Affairs requested the Netherlands National Team to advise the Ministry on future heat pump activities in the Netherlands.

In August 1991, the National Team advised a number of policy recommendations. According to the National Team, these recommendations will lead, in the short-term, to an accelerated, successful application of existing and proven heat pump systems, keeping expertise in the Netherlands up to par, and reinforcing it in the long-term. The latter is particularly necessary to be able to profit optimally from, or participate in, international developments.

In order to gain topical inspiration, the National Team organized a workshop called "Heat pumping or Drowning", held on 22 and 23 May, 1991 (see Newsletter Vol. 9, No. 3).

Recommendations include:

- Drafting a "Heat Pump Manual", both for the Building Environment and the Industry;
- Executing successful demonstration projects;
- Stimulating national and international co-operation;
- Establishing a knowledge platform on the integration of industrial processes and heat pumps;
- Drawing-up adequate norms and rules;
- Investigating the implications of a regulation such as the German regulation concerning the use of waste heat (Wärmenutzungsverordnung);
- Developing a heat surplus and demand map of the Netherlands;
- Stimulating integrated analyses of industrial processes;

- Creating necessary financial and economic boundary conditions;
- Promoting low-temperature heating and distribution systems;
- Promoting the integration of supply and demand of low-temperature heat;
- Further improving the efficiency of both centralized and decentralized electricity generation;
- Stimulating involvement of public utilities in the implementation of heat pumps;
- Directing short-term attention towards application aspects of heat pumps;
- Monitoring and evaluating international developments; stimulating the manufacture of foreign products under license by Dutch/European equipment builders.

The English version of the complete text of the recommendations is available from the Netherlands National Team, Mr. P. Oostendorp (see the back of the Newsletter).

Fifth Anniversary of the HPTCJ

In August 1991, the Heat Pump Technology Center of Japan (HPTCJ) entered into its 6th year. The commemorative event of its 5th anniversary was held in October 1991 in conjunction with the "Absorption Heat Pump Conference '91", in Tokyo, Japan. Prof. J. Berghmans of the Catholic University of Leuven, Belgium, gave a presentation on "The heat pump - its potential for development and application in the future". Approximately 200 people from government, universities, national laboratories, and industry, attended the anniversary celebration, including the Head of the IEA Heat Pump Centre.

Bibliography

Refrigerants/CFCs

CFC Information Pack

EPRI has developed a CFC Information Pack (CU.7001.1.91) containing the following documents which provide fundamental information about many of the issues surrounding the CFC phase-out.

Brochures:

- CFCs and Electric Chillers: Selecting Large Water Chillers as CFCs are Phased-Out;
- CFCs and Electric Utilities: Making the Transition to a Safer World;
- The Greenhouse Effect...To What Degree?
- Ozone: One Gas, Two Environmental Issues.

EPRI Journal Reprints:

- CFCs: The Challenge of Doing Without, September 1989;
- Concern over Ozone, June 1989.

For more information contact: Electric Power Research Institute, 3412 Hillview Avenue, Palo Alto, CA 94303, USA, Tel.: +1-415-855-2411, EPRI, or Mr. Greg Lamb, Tel.: +1-415-855-2449, or Mr. Dave Rigney, Tel.: +1-415-855-2419.

Alternatives to CFCs: Testing and Materials Compatibility Studies

Four articles in ASHRAE Journal, February 1991, pp. 36-42, (English).

Following an Introduction (p. 36) by Shelvin Rosen, information derived from seminar presentations at the ASHRAE Winter Meeting, January 1990, in New York is presented:

- Solubility & viscosity of R-134a refrigerant/lubrication mixtures, R.H.P Thomas et al., Fluorine Products Division of Allied-Signal, pp. 37-38. Measurements have been made on R-134a with a proprietary modified polyalkylene glycol (PAG) called BRL-300.
- HCFC-124: Applications, properties and comparison with CFC-114, P.R. Reed, DuPont Chemicals and H.O. Spauschus, Spauschus Associates Inc., pp. 40-41.

Special focus is placed on application in marine refrigeration. Alkylbenzene is the lubricant of choice with R-124.

- Durability of R-134a compressors: The role of the lubricant, K.S. Sanvordenker, Tecumseh Products, p. 42.

The paper briefly discusses the thermal instability of PAGs and Polyol Esters at temperatures of 200°C and above. It concludes that with R-134a, reaction with the lubricant is a negligible factor and that the inherent instability of the lubricant becomes the determinant.

Fluorocarbon Refrigerants and the Environment

G. Mouton, Du Pont Canada Inc., Revue Internationale du Froid, July 1990, p. 209-216 (French).

The article, presented at the XVIIIth IIR Congress in Montreal, 10-17 August, 1991, addresses CFC alternatives, the uncertainties about them, their impact on the environment and the distance to go to ensure the long-term development of the refrigeration industry on an ecologically sound basis. The article also presents the test results on the HCFCs 134a, 124, 125, 123 and 141b as of June 1990 of the Panel for Alternative Fluorocarbon Toxicity Testing (PAFT). Some of the conclusions of the paper include:

- HCFCs are an essential bridge for the success of a programme of fast elimination of CFCs;
- Important quantities of alternatives will be available between 1993 and 1995;
- The provisional results regarding the toxicity of substitutes are encouraging;
- The application of HCFCs in polyurethane insulation still presents important problems.

Test Results for 69-S

Peter Dinnage, Rhone-Poulenc UK, Refrigeration and Air Conditioning, July, 1991 p. 28-30 (English).

As reported in Newsletter Vol. 9, No. 3, Rhone-Poulenc UK has developed the 'near' azeotrope Isceon 69-S as a 'drop-in' fluid for R-502 used widely in supermarkets for the display of frozen food. The brief article provides more information on its use and test results.

Systems/Installations

Grouting Procedures for Ground-source Heat Pump Systems

Frank Eckhart, Oklahoma State University, 1991 pp. 45 (English).

This guide, validated by the International Ground-Source Heat Pump Association, serves as a reference for ground-