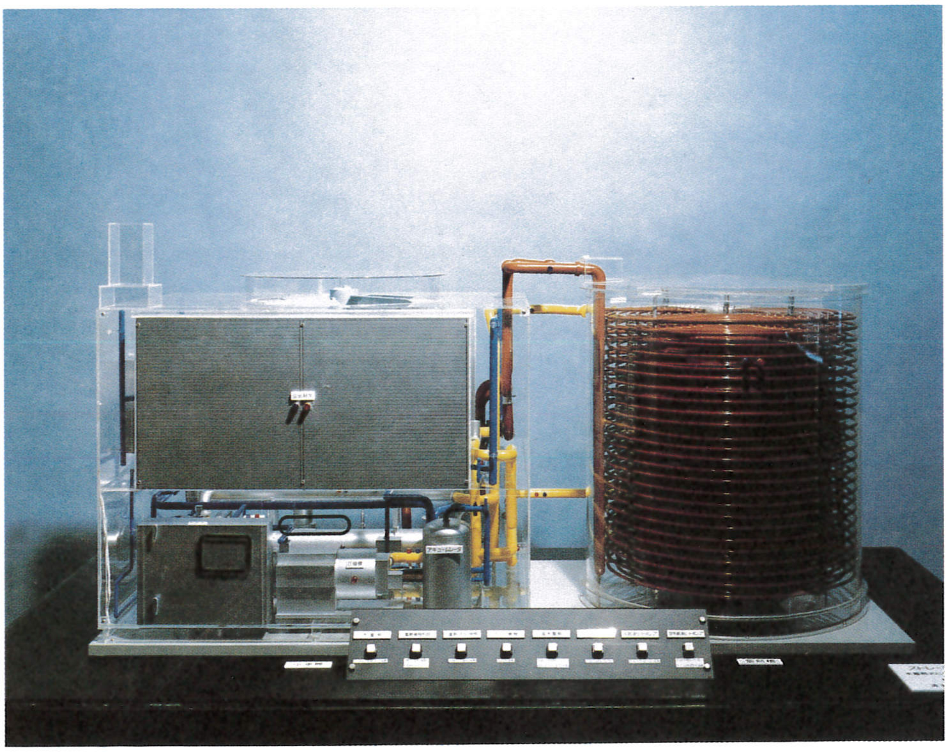
ISSN 0724-7028

LETTER

IEA heat pump center

PERIODICAL OF THE
IEA HEAT PUMP CENTER

Vol 6, No 1, Mar '88



Model of a 30 HP air-source heat pump (see page 11)

This issue: Air-source heat pumps

Contents | Page

Editorial*

Readers may notice that this issue of the Newsletter has a slightly different layout than previous issues. We hope you will continue to find the Newsletter informative and attractive, while we make an effort to reduce production costs and improve timeliness. Our goal is to actively support the development and application of heat pumps in the current market which, though uncertain in some respects, offers enormous possibilities.

The aim of the Newsletter is to increase the flow of high-quality information. Its content ranges from experience of operating plants to research results and proposed theories. It also reports on market trends. We want to support and encourage international cooperation for the penetration of heat pumps on the market by:

-- Promulgating information about the state of the art
-- Showing trends
-- Proposing research and development goals

We believe in this technology because the increased use of heat pumps saves end-use energy, helps to economize other energy sources, improves the operating flexibility of combined systems, and reduces the environmental impact. These benefits stem from the decreased use of fossil fuels in favor of previously unused energy sources, such as air, groundwater, and heat loss from industrial processes.

Evidence of the perceived applicability of heat pumps to today's energy problem is found in the increasing expenditure of manpower and money on heat pump projects in private companies, research institutions, and international organizations. This should be looked at as an evolutionary process. Heat pumps need to find their form and place in the energy systems of the world organism; however, this is a time-consuming process requiring considerable thought and effort. And this work must carry on in a world where technical, economic, and political conditions and obstacles are continually changing. Goals for these activities may be short range, such as seeking improvements based on experiences, or long range, such as research and development of new products and systems.

The topic of this issue, Volume 6, Number 1, is air-source heat pumps. This topic is now, and probably always will be, of high interest since air is an ever present, free heat source. There are, however, technical and economic problems related to using air as a heat source. But there is a trend developing as new components and systems are slowly being introduced in the market. Air-source heat pumps are primarily small units used for heating apartments and single-family houses. The large markets are in Japan and the USA. Activity today is concentrated on these small units, but some new techniques will no doubt be applicable to large air-source heat pumps for block centrals and district heating in the future.

In this issue, several articles address the pros and cons of air-source heat pumps. For example, the well-known problem of decreased performance at part-load conditions is discussed in the article *Dynamic Testing of Heat Pumps*, which explains the disadvantages of running a heat pump on-off. The article *Inverter-Aided Heat Pump Models for Light Commercial Applications* talks about one way to solve this problem. In *Development of a Simulation Program for an Air-Source Heat Pump with Latent Heat Storage*, a program is described which allows the designer to determine the effect of a heat pump on the electric supply network. It also shows that special units and systems will be developed in different markets, due to their special characteristics. This trend is also underlined in the articles *Heat Pump Room Air Conditioner Using Thermal Storage* and *Small Gas Engine Heat Pump for Commercial Use in Japan*. Applied standardization appears to be a trend.

Improved performance using advanced systems is another constant pursuit, dealt with here in the article *Heating Tower Heat Pump System* . Direct vs. indirect systems, an old subject for discussion among refrigeration specialists, is now seriously discussed among heat pump specialists. The article *Experience from Air-to-Water Heat Pump Plants for Building Heating* deals primarily with the use of indirect systems, based on experience from comparatively large air-source heat pump plants in operation. There are some strong arguments for using indirect systems. One argument is the decreased risk for refrigerant leakage, a serious issue today. For example, in Sweden there is already proposed legislation to dictate the use of indirect systems under specific circumstances. This subject was discussed in the article *Experience with 250kW Air/Water Heat Pump in a Swedish Group Central*, published in Newsletter Vol 5, No 2.

Different methods of defrosting have also been a subject of discussion among heat pump specialists. The article *Controlling the Defrosting of Air Evaporators for Heat Pumps* is a summary of a report from tests conducted on operating heat pump plants. It also contains, as a conclusion, a proposal for future action. This is the right way to evaluate and learn from experience.

The price of electric energy is likely to increase in the future, which will make absorption heat pumps more economically feasible. Research and development in the area of achieving higher outlet temperatures to meet space conditioning and industrial application demands is going on now. The article *Future Possibilities of AHP Technology for Refrigeration, Air Conditioning, and Heating Systems* presents one such system and its application possibilities based on an economic analysis. Residential application of heat pumps in Europe versus Japan and the United States is discussed in the article *Air-to-Air Heat Pumps for Residential Application in Europe*. In Europe, the heat pump market is still dominated by hydronic systems, but the author suggests that improvements to air-to-air heat pumps will allow them to penetrate specific market segments.

Reports from meetings and announcements of new publications demonstrate international cooperation. International cooperation is, of course, very important in order to save time and decrease costs, even if the efficiency of this approach does not seem apparent at first glance.

We hope it will be a pleasure for Newsletter readers to follow the evolution of heat pump technology and get involved in the process. We would be happy to know that, over the long term, the work done by all involved has promoted the penetration of heat pumps in the market, which is our main goal. In the interest of reducing production costs, many of the articles have been condensed and summarized. If you are interested in receiving a complete copy of any particular article, please write to the HPC.

**Editorial by IEA Heat Pump Center Staff, Karlsruhe, Federal Republic of Germany*

A. Bergman and E. Granryd*

Dynamic Testing of Heat Pumps: An Investigation of How On-Off Operation Influences Performance

Capacity modulation of small heat pumps is usually performed by on-off operation. The influence of this intermittent type of operation has been studied. One purpose of the tests has also been to explain differences between practical results in the field and results recorded in the laboratory under continuous operation. Results of the tests show that the differences between continuous and intermittent type of operation can be explained by taking into account the electric idling power demand and the heat losses during "off" periods. If these losses are known, the differences between results from stationary and dynamic testing can be correlated as a function of the relative operating time provided that the comparison is done at equal mean temperature of the heat source and heat sink. The influence of the start frequency seems to be very small. This project was sponsored by the Swedish Council of Building Research.

Introduction

Performance data of heat pumps are traditionally based on tests under stationary conditions. The heating capacity and the electric power demand are, in such tests, recorded during operation with different (standard) temperature levels of interest on the heat source and heat sink side.

In a real installation, however, the heat pump operates at part-load during most of the year. The majority of installations are operated according to demand by on-off control. In order to make the standard laboratory tests reveal the performance of the real heat pump system in field service, the tests should also take into consideration all the factors influencing performance under on-off type of operation. Such factors include processes in the heat pump unit as well as idling losses, control strategy used for the unit, and the thermal behavior of the heating system in the house.

In a project at the Department of Applied Thermodynamics and Refrigeration, The Royal Institute of Technology, Stockholm, a test rig was built in order to study the influence of such factors. Tests were performed on two heat pumps of different design and with different types of control strategy. Parallel to the laboratory tests a computer model has been designed to simulate temperature variations in different sections of a heating system and to study the influence of different control strategies. This article presents sample results from the complete report.

Scheme of test equipment

The test setup was designed with the intention that it would permit tests to be performed under continuous operation as well as at conditions simulating actual conditions for a heat pump installation. In the latter case, the heat pump operation would be controlled (on-off) by the control system in the heat pump unit. Water volumes were used to simulate the thermal masses of different portions of the heating system, however, somewhat shorter time constants were accepted for the model in order to avoid excessively large storages of water. A schematic of the test rig is shown in Figure 1.

Heat pump units

During the tests, two different commercially available heat pumps were used, here referred to as heat pumps "A" and

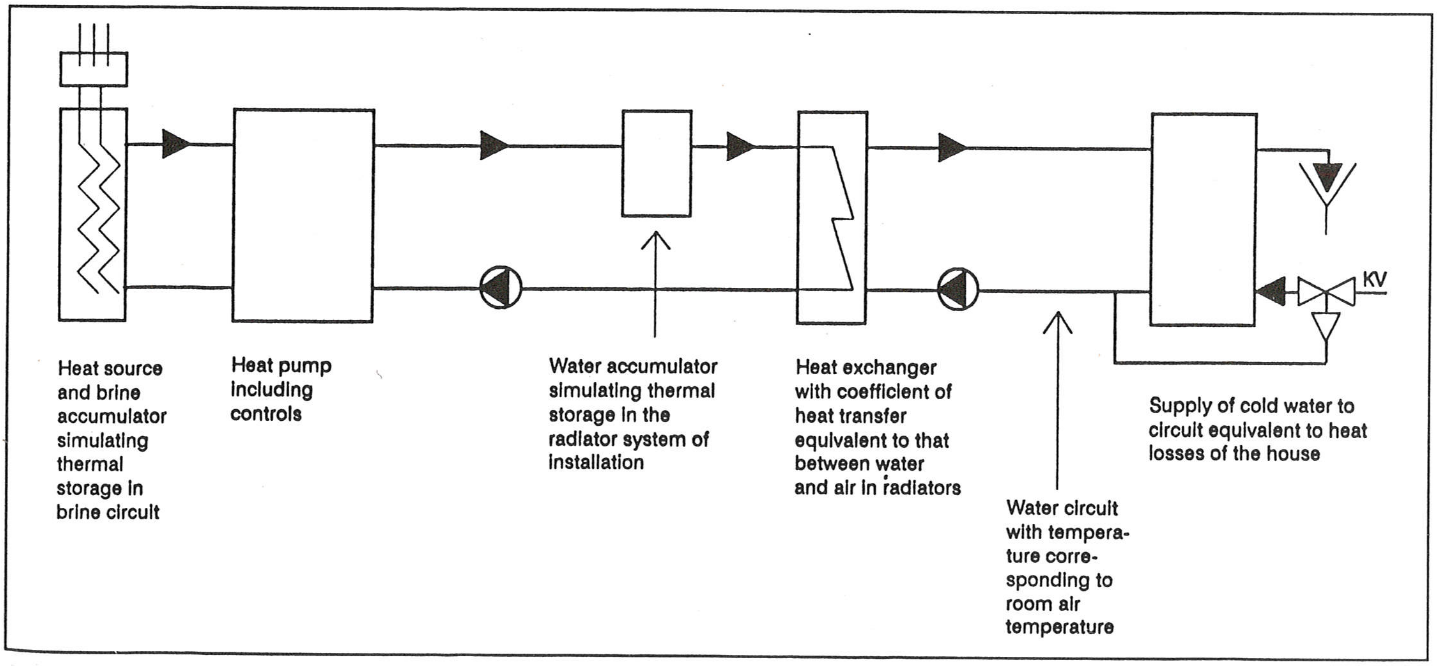


Figure 1. Schematic of test rig

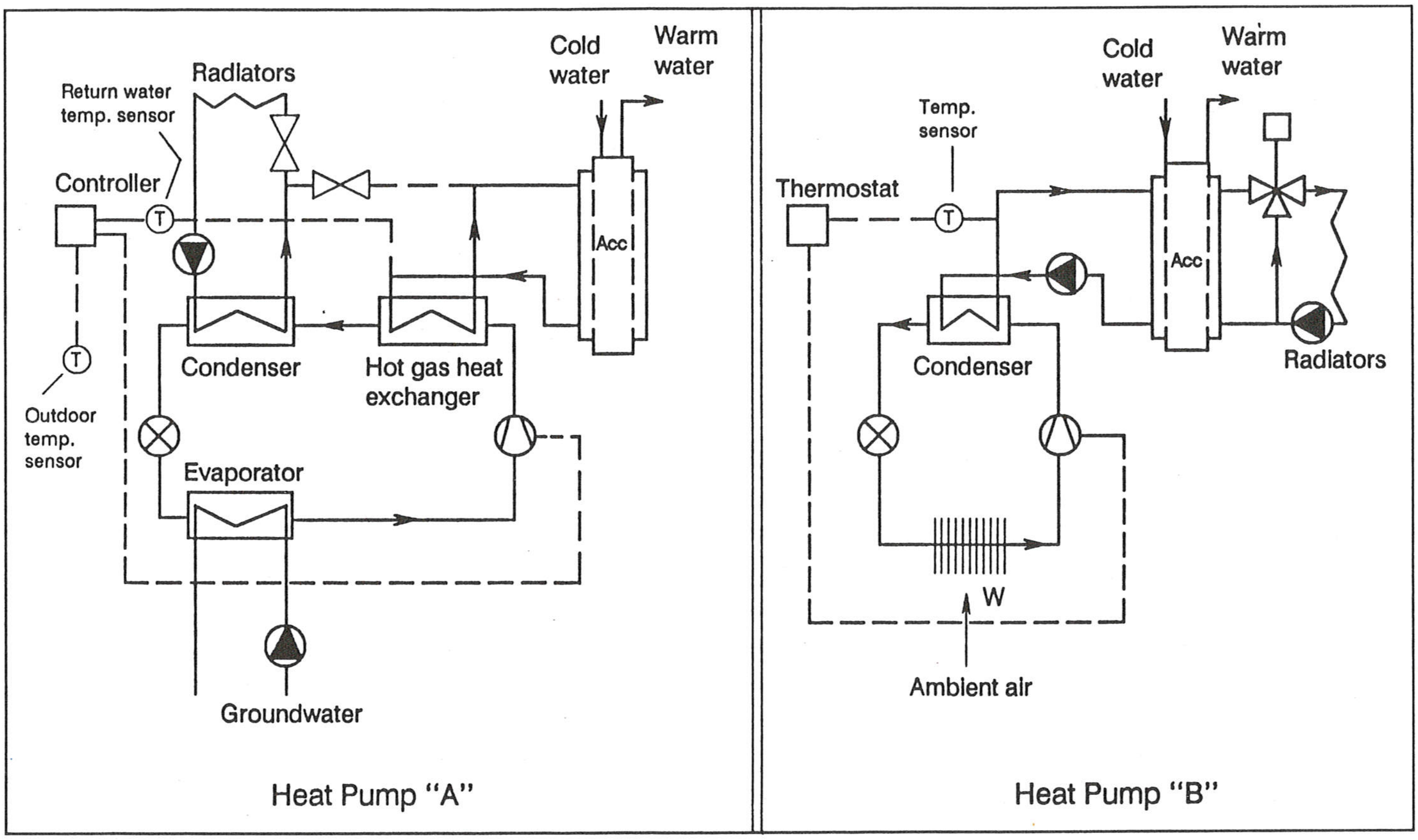


Figure 2. Schematic of heat pumps A and B

"B." Unit "A" is primarily a ground-source heat pump, while unit "B" uses ambient air as its heat source. A schematic layout of the two heat pumps is shown in Figure 2. As can be seen, the two designs are different in many respects.

Unit "A" is a liquid/water ground-source heat pump unit to be installed inside the house. A hot gas heat exchanger is used for heating the domestic water in the hot water accumulator to high temperatures. The heating of the house is accomplished through a hydronic heating system where the temperature is controlled according to the demand in the house using a sensor in the ambient air. By the control system in the heat pump, the return water temperature is automatically adjusted according to set values determined as a function of the ambient air temperature.

Unit "B" uses ambient air as a heat source and is positioned outside the house. The heat pump heats water to temperatures around 50-55°C year-round in order to satisfy the temperature demand of domestic hot water. The heating of the house is controlled by a mixing valve which distributes water of proper temperature from the jacket around the hot water accumulator.

Tests

Tests were conducted first at stationary conditions with externally fixed temperatures on the warm and cold side of the heat pump, and secondly at part-load when the operation (on-off) of the heat pump was controlled by the heat pump's own control system. In this latter mode of operation, the heat source temperature, as well as the ambient air temperature, were set in order to simulate different climates over the year. The temperature on the heat sink side was set "free" and was controlled by the heat pump control system to temperature levels according to the prevailing ambient temperature.

For unit "A" a ground-source coil temperature was simulated as a function of the ambient temperature as shown in Figure 3a. As input to the heat pump control, a set value was used corresponding to temperatures in the radiator system of 55/45°C in supply/return temperatures at a design outside temperature of -18°C. With this set value into the heat pump control, the temperature on the warm side was adjusted by the control system of the heat pump. Temperature of the return water as a function of ambient air temperature is shown in Figure 3b.

For unit "B" with air as the heat source, the heat pump was positioned in a climate chamber with temperatures and humidity simulating the ambient. The stationary tests were performed without defrosting (dry air in test chamber).

During the tests, relevant temperatures of the system were recorded. The heating capacity was determined as well as the operating electric power demand (including supply to all auxiliary such as pumps, fans, controls, and, when necessary, defrosting). Under dynamic conditions (on-off operation), the test period was extended to several full on-off cycles.

The heating coefficient of performance (COP1) was, for instance, determined from the test results as:

$$\mathrm{COP1} = \frac{Q1}{E} = \frac{\text{heating energy delivered over time } (\tau)}{\text{total operating electric energy over time } (\tau)}$$

where the time (τ) during the "dynamic

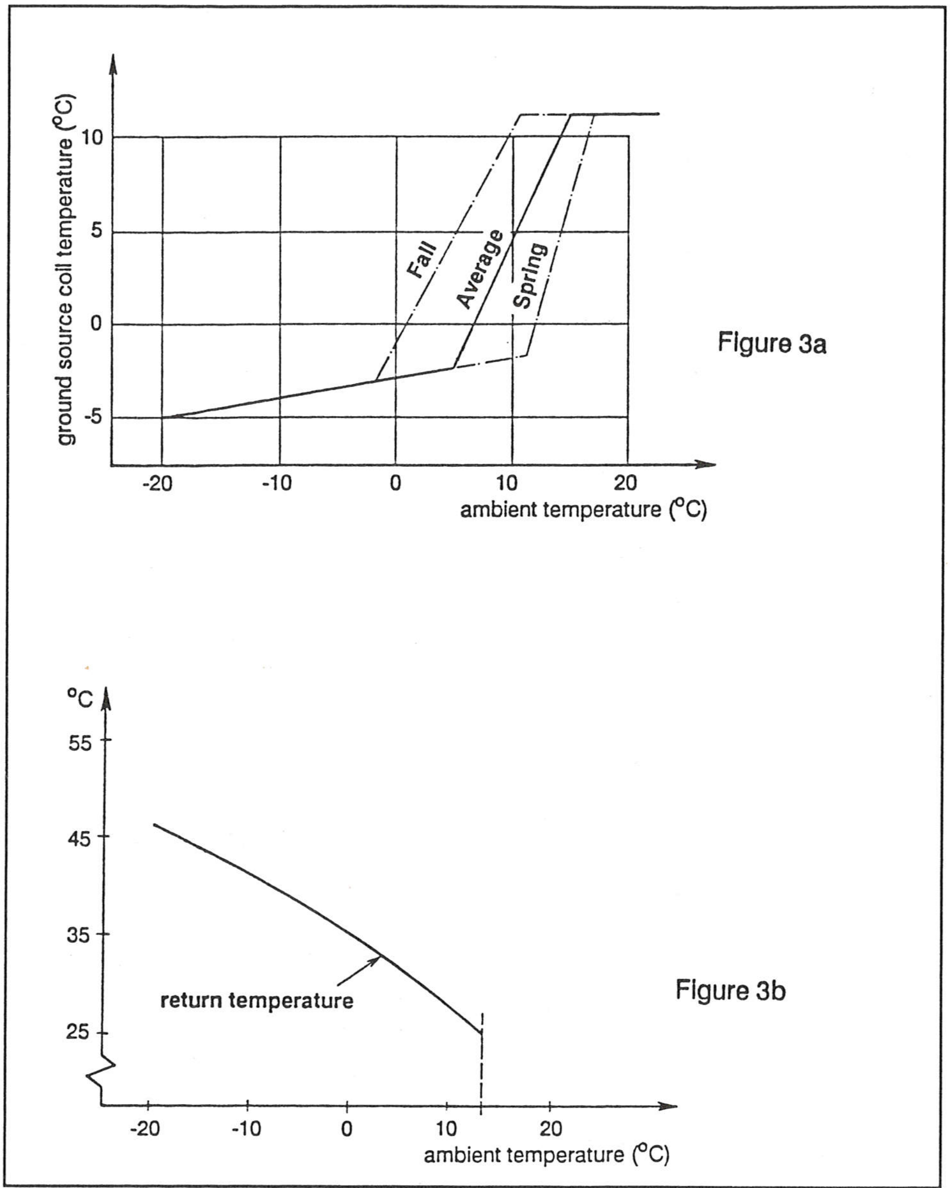


Figure 3. Ground temperatures and heating system return temperatures for Unit A

testing" was for a full number of on-off cycles.

For the tests to be shown here, the connecting lines to the hot water tank in heat pump "A" were closed (to avoid the influence of thermal storage in the accumulator). For heat pump "B" the heat delivered was determined for the heat pump unit before the hot water accumulator (see Figure 2). Hence, in both cases, results shown apply without consideration of heat losses in the hot water accumulator.

Test results

A few test results regarding the coefficient of performance are shown in Figure 4. As is seen, there is a considerable difference in the COP between the "A" and "B" units.

The difference between the two is naturally influenced by the different heat sources (ground source for unit "A" and ambient air for unit "B"), especially above 0°C ambient temperatures. However, another important reason for the difference is the different schemes of the two units (see Figure 2). Heat pump "B" is forced to operate with high temperature on the warm side at all instances in order to satisfy the temperature demand of the domestic hot water. Heat pump "A," on the other hand, operates most of the time with much lower condensing temperatures determined by the necessary temperatures of the hydronic heating system (see Figure 3b). The high temperature of the hot water is usually furnished by the hot gas heat exchanger.

In Figure 4, the solid lines result from continuous testing, while the symbols indicate results from dynamic tests. As shown, the on-off type of operation decreases the COP, especially at high ambient temperatures and short running periods.

Tests were also performed with different lengths of the operating cycles. This was accomplished by using different volumes of the warm side storage accumulator shown in Figure 1. A few of these data are shown in Figure 4 (see symbols for heat pump "B"). The tests at ambient temperature +5°C may require a special explanation. With relatively long operating cycles (about 15 minutes "on" time), the "off" time was long enough to accomplish natural, passive, defrosting. Using shorter cycles, the defrosting had to be accomplished actively. This is the main reason for the difference in COP between the tests shown in Figure 4 with 3 and 15 minutes "on" time at +5°C. In other respects, there were no negative consequences recorded for short cycles (provided that the "on" time was greater than 3 minutes; shorter cycles have not been tested).

Figure 5 is given in order to demonstrate the influences of a few phenomenon during start and stop. The diagram shows schematically the heating capacity, Q_1, and operating power, E, as a function of time. The peak of the heating capacity during the first minute after start is caused by the favorable operating conditions for the compressor (with high inlet and low outlet pressure) directly after start due to temperature equalization during the off-period. However, the influence of this increased capacity on the performance of a full cycle is very small (less than 0.4%).

As illustrated, the "energy loss" during the first few seconds after start (Area A) seems to be recovered directly after stopping the heat pump (Area B). An analysis of test results indicates that the difference in these two areas is small. For the tested units, the difference can be disregarded in comparison with the total heating energy delivered over a full cycle. During the tests it was not possible to record any significant influence of the start and stop phenomena on COP, at least not as long as the "on" periods exceeded about 3 minutes.

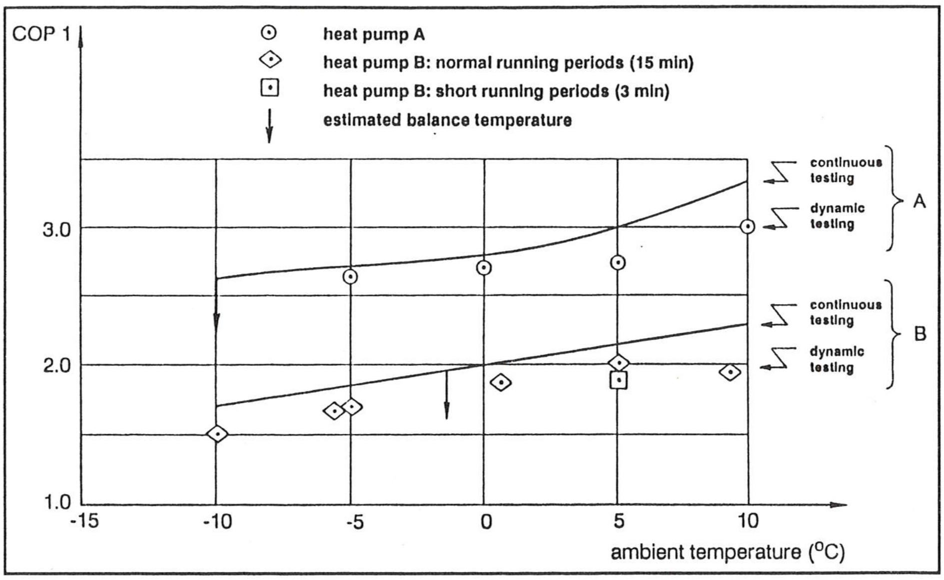


Figure 4. Comparison of COP's during continuous and dynamic operation

On the other hand, the relative operating time shows a significant influence. This is illustrated in Figures 6 and 7, where T_{rel} = relative operating time = time in operation/total time; and dynamic efficiency = COP1 in dynamic operation/COP1 in continuous operation. The mean temperature levels (especially on the warm side of the heat pump) are equal for both types of operation.

As is seen, the dynamic efficiency for both heat pumps decreases with decreasing relative operating time. The reason for this is probably the influence of the losses occurring during "off" time. This is partially due to the small electric power demand during off periods for auxiliaries (such as circulating pump for radiator water, controls, crankcase heater in the compressor, etc.) and partially due to heat losses from the warm side of the heat pump.

Figure 8 is given to illustrate the influence of the circulating pump. The values are based on the same data as in Figures 6 and 7, but compensated as if the circulation pump was stopped during off periods.

Calculation of the dynamic efficiency

By using the hypothesis as discussed (that idling losses are the main factors influencing the dynamic efficiency), the following equation was formulated for calculation of the dynamic efficiency (DE):

$$DE = \frac{1 - X \cdot \Delta \dot{Q}_1 / \dot{Q}_{1s}}{1 + X \cdot \Delta \dot{E}_o / \dot{E}_s}$$

Where:

$\dot{Q}_{1s}$ = heating capacity for unit in continuous operation

$\dot{E}_s$ = electric power demand for unit in continuous operation

$\Delta \dot{Q}_1$ = heat losses in idling

$\Delta \dot{E}_o$ = electric power demand in idling

$X = \frac{\tau_{off}}{\tau_{on}}$

τ_{on} = time in operation

τ_{off} = idling time

Measurements were performed on heat pump "A" in order to determine the idling heat losses and the idling electric power demand.

The idling heat losses can be distinguished as internal losses from the warm to the cold side in the heat pump and external losses from the heat pump to the surrounding. Two different methods were used to record these losses and it was shown that the heat loss could be correlated as a function of the temperatures on the warm and cold side of the heat pump and the temperature of the air surrounding the unit. The total idling heat loss for unit "A" varied from 100 W at relatively low radiator water temperature up to about 200 W at larger temperature differences (not including the hot water accumulator). These losses are to be related to the stationary heating capacity of the heat pump, about 12 kW.

The electric power demand during idling is easier to determine than the heat losses. For heat pump unit "A" it was 137 W, including the circulating pump of the radiator system (using about 125 W). This idling electric power demand is to be compared to the total power demand of 4 kW.

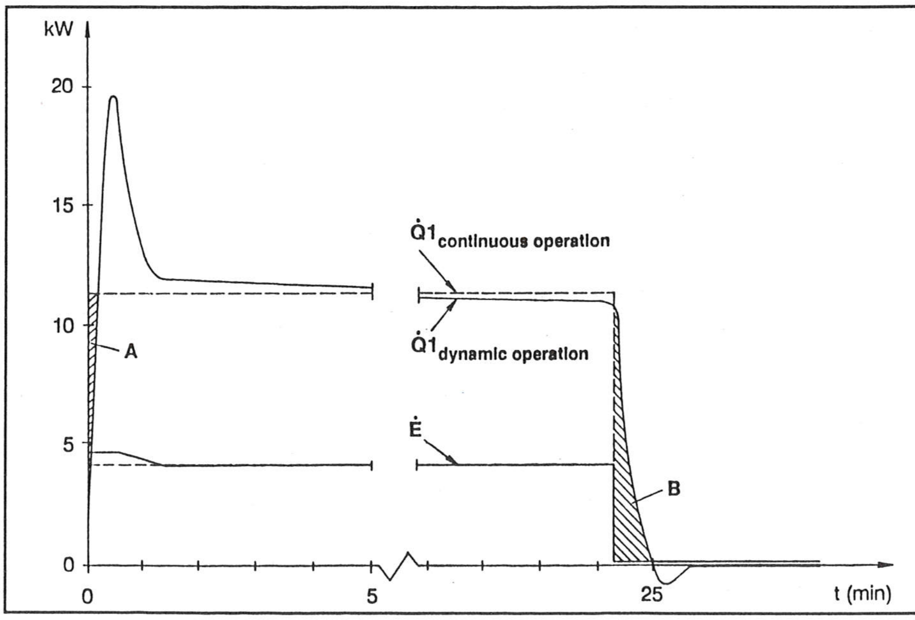


Figure 5. Conditions during starting and stopping

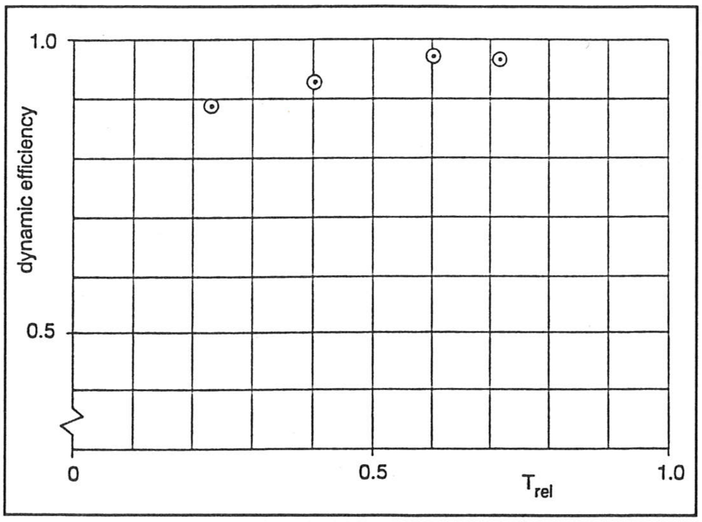


Figure 6. Heat pump A: Influence of relative operating time on dynamic efficiency

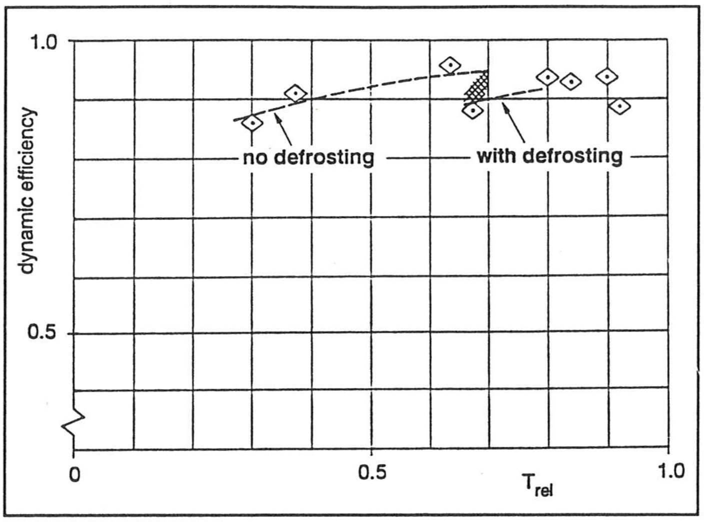


Figure 7. Heat pump B: Influence of relative operating time on dynamic efficiency

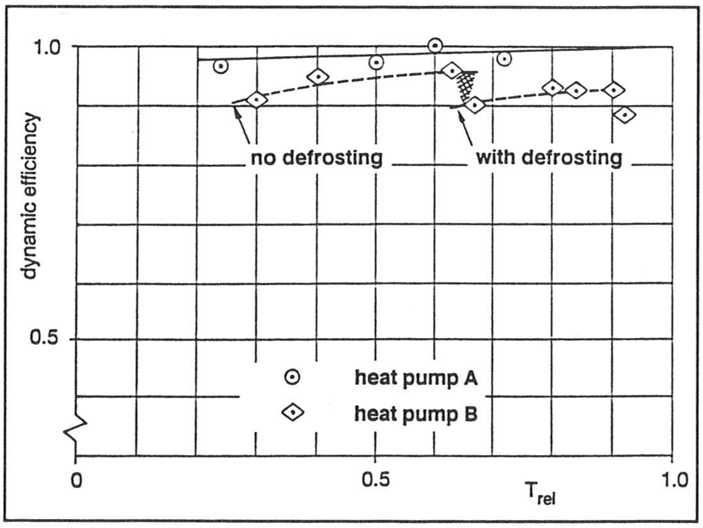


Figure 8. Dynamic efficiency neglecting the effect of the circulating pump

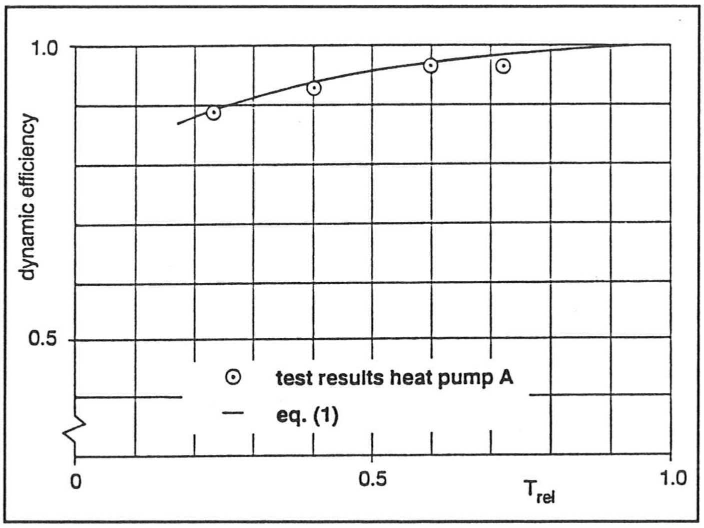


Figure 9. Comparison of test results with equation 1

Figure 9 compares the results from equation 1 (the solid line) with the test results. As is seen, the correlation is quite good. The diagram includes the operating power of the circulating pump in the radiator system.

This pump has been assumed to be in operation also under off periods. It can be questioned whether the operating power for this pump should be included in the COP of the heat pump system or not. In discussing the efficiency of a furnace, the electric power demand of the circulating pump is very seldom considered. Compare also results of Figure 9 with those shown in Figure 8, where the pump has been disregarded.

Comparison with field tests

Comparisons have also been made with experience from heat pump installations. For an installation in a house located in Skutskaer, about 200 km north of Stockholm, the annual heating demand was recorded to 24,200 kWh. A seasonal COP of COP_{year} = 2.05 was recorded. The heat pump was of the same type as heat pump "A" in this investigation. However, ambient air was used as heat source in an indirect system using a brine as secondary refrigerant. By using results from our laboratory measurements and by further using the ambient air temperature according to statistics for the year in question at the place of installation, a seasonal COP could be calculated. This gave as a result COP_{year} = 2.13 (calculated), to be compared with the result of 2.05 recorded in the field. If the idling losses had not been taken into account, the calculations would have given 2.45 as a resulting COP_{year}.

Conclusion

Good agreement between laboratory and field measurements of COP is achieved if idling heat loss and electric power demand are taken into account.

Anders Bergman and Eric Granryd, Dept. of Applied Thermodynamics and Refrigeration, Royal Institute of Technology, Stockholm, Sweden

K. Nagae*

Inverter-Aided Heat Pump Models for Light Commercial Applications

A new series of inverter-aided air-source heat pumps has been developed in Japan for light commercial applications. These particular units have compressors which are rated at 90 Hz rather than the usual 60 Hz. Their operating range has also been increased, allowing them to operate at 35 Hz to 105 Hz. This improves their seasonal heating and cooling performance compared to a conventional unit with the same rated capacity. Calculations of the SEER (Seasonal Energy Efficiency Ratio) show that the difference between the SEER of the conventional heat pump and the inverter-aided heat pump are strongly influenced by the particular design conditions such as indoor and outdoor temperature.

Introduction

Air-source heat pumps in Japan's domestic air conditioner market account for about 60% of the residential market and 70% of the light commercial market. The increasing demand for inverter-aided type air conditioning systems is especially noteworthy. In the market, 0.75 kW units for home use and 2.2 kW units for light commercial applications enjoy the highest popularity of all inverter-aided heat pumps. Manufacturers are aiming for wide capacity control ranges and improvements in heating capacity.

Sanyo Electric Co., Ltd. has developed 2.2, 3.0 and 3.7 kW inverter-aided heat pumps for light commercial applications which have been improved in heating capacity at low ambient temperature by raising the compressor's maximum operating frequency to 105 Hz. The compressors have been made more compact by using 90 Hz as the rating frequency rather than 60 Hz as in conventional units. Table 1 summarizes the frequency and capacity ranges of heat pumps available in Japan for home and light commercial applications.

Refrigeration cycle

The problems to be solved for the development of this heat pump for light commercial applications were how to make it possible to provide a wide range of capacity control (35 to 105 Hz) and still ensure stable operations at high frequencies. To solve this, it is necessary to maintain proper refrigerant circulation rates, proper refrigerant oil temperature, and stabilization of oil delivery. In these units, proper refrigeration circulation rates are achieved by the use of capillary tubes, thermostatic expansion valves, and a refrigerant modulator. Proper refrigerant oil temperature is maintained by liquid injection into the unit's rotary compressor and modifications to the compressor were made to improve oil separation and return rates at high operating speeds.

Frequency control

For an inverter-aided heat pump, proper frequency control is also important. The frequency in these units is controlled by the difference between the room temperature and the setting temperature, as well as specific protective controls to limit discharge temperature and electric current demand. Particularly in the case of the inverter-aided system, flexible protective control, which makes the best use of a wide capacity control range, is possible allowing smooth operation with less frequent on/off cycling.

Improved heating capacity

In order to solve the problem of less heating capacity at low ambient temperatures, auxiliary heaters are typically used in air-source heat pumps. In this design, the auxiliary heater was eliminated by increasing the rated frequency of the compressor to 90 Hz and its maximum frequency to 105 Hz. Due to these factors, the pull-up performance of this unit during the heating mode is improved. For example, at 0°C outdoor ambient temperature, conventional models require 30 minutes to raise the mean room temperature 15°C, whereas the inverter-aided model requires only about 18 minutes.

Figure 1 shows the heating performance characteristics at ambient temperatures of 7°C and 0°C. The inverter-aided model using a maximum frequency of 105 Hz is able to maintain the

	Mfr	Compressor	Frequency Range	Capacity Range (kW)	
Home use	A	Rotary	30 Hz - 120 Hz	(Cooling)	1570 - 3360
				(Heating)	1570 - 5100
	B	Rotary	30 Hz - 150 Hz	(Cooling)	1740 - 3250
				(Heating)	1570 - 4990
	C	Rotary	25 Hz - 180 Hz	(Cooling)	930 - 3420
				(Heating)	810 - 5220
Light commercial applications	A	Reciprocating	30 Hz - 90 Hz	(Cooling)	4120 - 8240
				(Heating)	4120 - 8930
	B	Scroll	30 Hz - 115 Hz	(Cooling)	4640 - 9280
				(Heating)	4640 - 10900
	C*	Rotary	35 Hz - 105 Hz	(Cooling)	3940 - 8820
				(Heating)	3940 - 11370

(*Heat pump air conditioner currently being developed)

Table 1. Frequency and capacity specifications

same capacity at 0°C as at 7°C. Temperatures in the range of 0°C to 7°C during the heating season in Japan have a frequency of occurrence of 42%. When compared to conventional models which have a rated capacity equivalent to the inverter-aided models, the quantity of heat supplied during the heating season by the inverter-aided model is greater.

Evaluation of unit's performance

Performance at rated frequency

When comparing performance of the inverter-aided models with that of the conventional ones in terms of rated frequency, the inverter-aided model has a lower performance; this is due to the heat loss generated by the solid-state components, such as power transistors, in the inverter circuit. This heat loss is about 5% of the primary input current and directly reduces the COP.

Cooling performance

The seasonal efficiency is described in detail as seasonal energy efficiency ratio (SEER) and heating seasonal performance factor (HSPF) in the ARI standards. In order to evaluate the energy-saving characteristics, we have computed SEER on a trial basis by using the ARI standards. Figure 2 shows the results of a trial computation and the conditions of machine operation. The building load (in kW) and the number of hours of occurrence of each temperature during the cooling season are shown. Capacities of the heat pump at inverter frequencies of 35, 90 and 105 Hz are indicated. The line for 90 Hz operation is also the capacity of the conventional heat pump unit operating at 60 Hz. Balance points for each of these frequencies are indicated by the intersection of the capacity and building load curves. At the bottom of the figure, the operating modes of the inverter-aided and conventional heat pump are shown as well as their SEERs. For the operating mode, the relative time in each mode is indicated by the size of the field. The difference in SEER between the conventional model and the inverter-aided model is governed by the following factors:

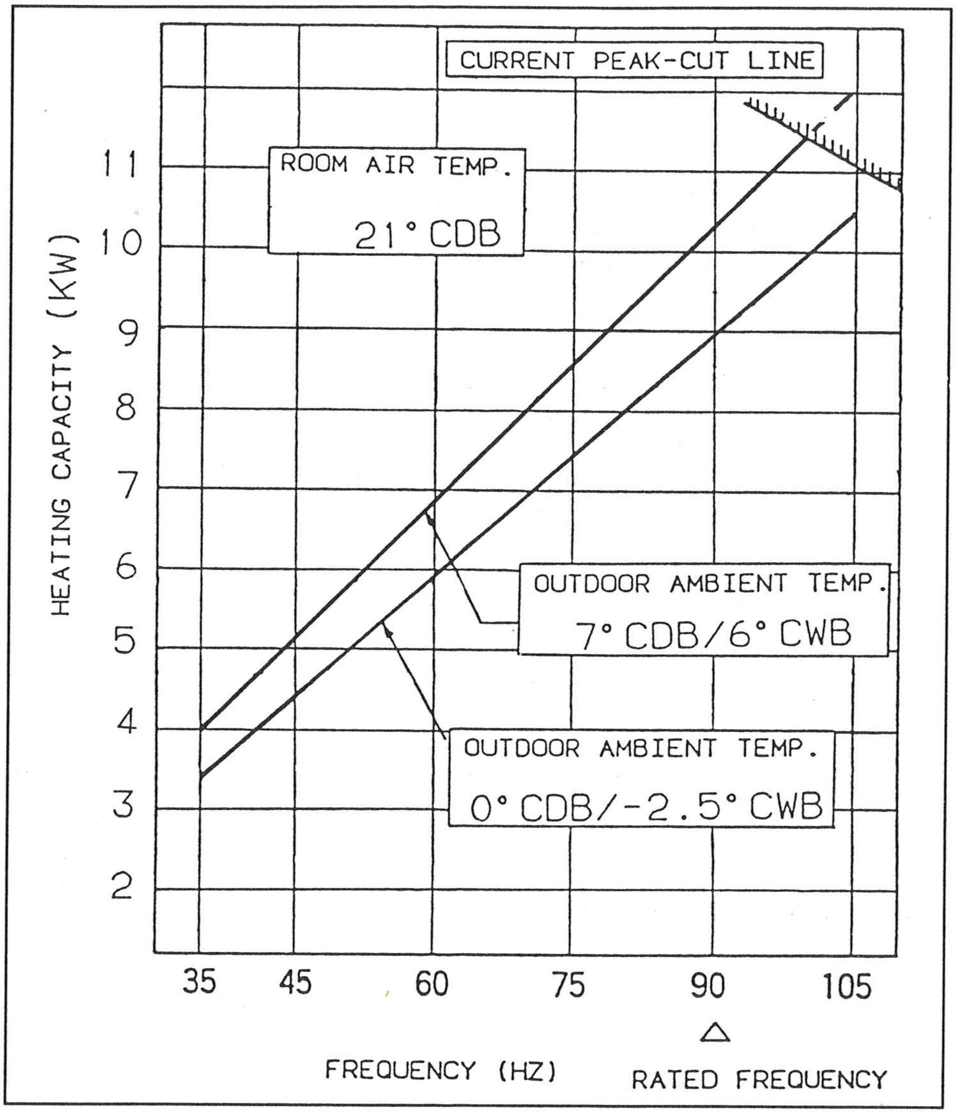


Figure 1. Heating performance

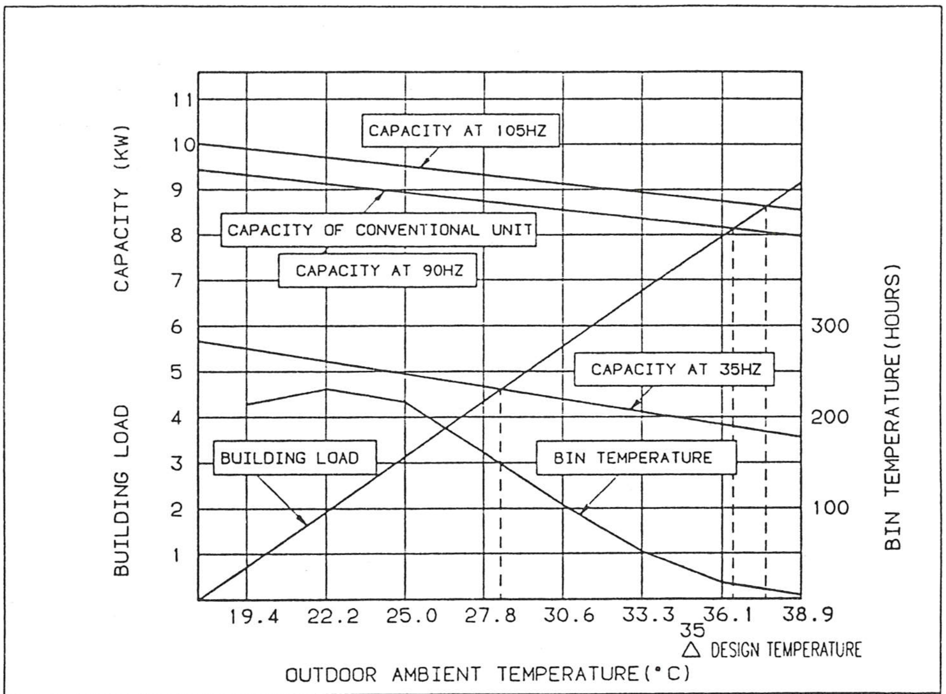
1. In the conventional model, the on/off area is so wide that the energy loss accompanied with repeated balance and separation of high and low pressure is much higher (degradation coefficient = 0.25).

2. The efficiency of the inverter-aided model is high at low frequencies. Since it can be operated at low frequencies during intermediate loads, its heat exchanging capacity increases relative to the compressor capacity and, as a result, its efficiency is improved.

The seasonal efficiency should be evaluated by using the time of occurrence of bin temperatures and the cooling load in conventional buildings, which correspond to the area and purpose of application.

Although the standards for the seasonal efficiency for unitary air conditioners in Japan have not yet been established and are not clearly defined, the result of computation of SEER, based on Japanese conditions, is shown in Figure 3. The difference in SEER is smaller. This is because the design outdoor ambient temperature was taken as 33°C and the balance point as 24°C, instead of 35°C and 18°C, respectively. This caused the insufficient capacity area to increase which required that the inverter-aided compressor run at its maximum rated frequency for a longer period of time.

It is seen from the above that the seasonal efficiency is governed greatly by the data (bin temperature) and the con-



UNIT	OPERATION MODE			SEER $\frac{(KW)}{(KW)}$
INVERTER-AIDED UNIT	UNIT CYCLING ON AND OFF AT 35HZ	CONTINUOUS OPERATION AT VARIABLE SPEED	*A	3.12
CONVENTIONAL UNIT	UNIT CYCLING ON AND OFF		*B	2.38

*A: CONTINUOUS OPERATION AT 105HZ
*B: CONTINUOUS OPERATION AT 60HZ

Figure 2. Comparison of SEER (based on ARI Standard)

3. SEER is improved by about 30% over conventional heat pumps.

4. Performance for temperature pull-up during space heating is enhanced and heating capacity at low ambient temperatures is higher.

The results shown in Figures 2 and 3 demonstrate that the local climate (bin temperature), balance point temperature, and design temperature have a strong effect on the SEER. Similar results are expected when calculating the HSPF.

K. Nagae, Sanyo Electric Co., Ltd., Oizumi-machi, Ora-gun, Gunma, Japan

cepts which form the basis for computation.

Heating seasonal performance factor (HSPF)

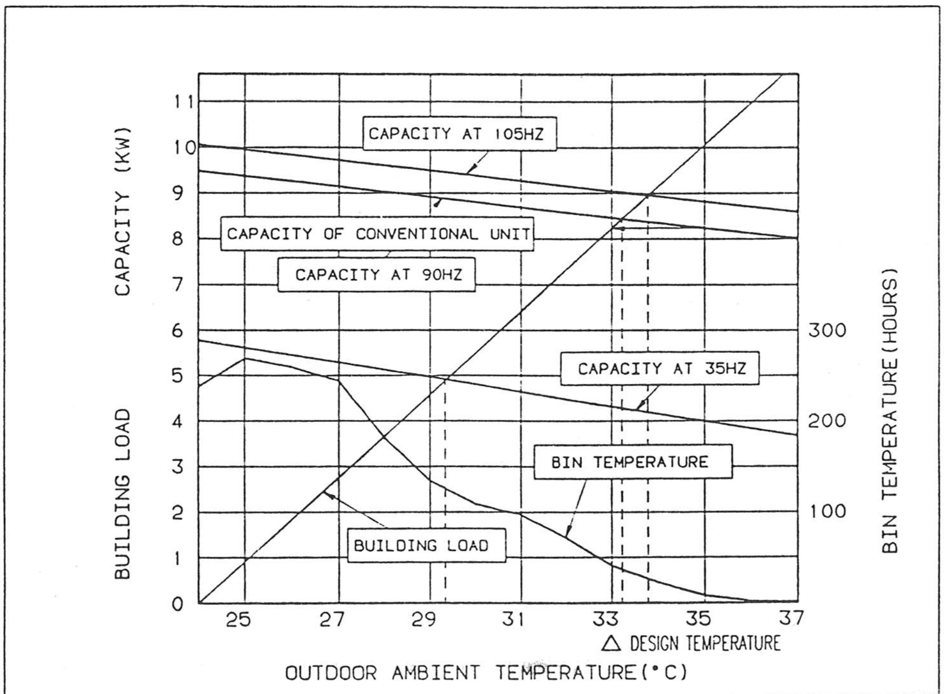
There were some unclear points in the concepts of the heating capacity during defrosting, auxiliary heaters, etc., so that the HSPF was not evaluated in the test. How to evaluate performance capabilities, such as hot gas defrosting, will need to be examined in the future.

Conclusions

The advantages of the inverter-aided heat pump are as follows:

1. By increasing the rated frequency, compressors can be made small in size and lightweight.

2. By appropriate control, stabilized room temperature and smoother machine operation can be accomplished.



UNIT	OPERATION MODE			SEER $\frac{(KW)}{(KW)}$
INVERTER-AIDED UNIT	UNIT CYCLING ON AND OFF AT 35HZ	CONTINUOUS OPERATION AT VARIABLE SPEED	*A	2.83
CONVENTIONAL UNIT	UNIT CYCLING ON AND OFF		*B	2.31

*A: CONTINUOUS OPERATION AT 105HZ
*B: CONTINUOUS OPERATION AT 60HZ

Figure 3. Comparison of SEER (based on Japan Expl.)

S. Yoshikawa, S. Igarashi, M. Goh, S. Kamiyama*

Development of a Simulation Program for an Air-Source Heat Pump with Latent Heat Storage

A computerized simulation program which combines a computer-aided design (CAD) program with an existing building thermal load program (Heating Air Conditioning and Sanitary Engineering Program) is described. The program can model equipment under part-load conditions and can also handle systems using thermal storage. Results of a simulation can be displayed visually allowing the designer to easily see the effects on energy consumption of equipment and/or operation mode changes. An example of applying the system to a 30 HP air-source heat pump with thermal storage is given.

Introduction

When designing a building heating/cooling system in Japan, the building's thermal load is typically calculated using the Heating Air Conditioning and Sanitary Engineering Program (HASP). This program uses statistical weather data compiled for various Japanese cities. Based on the results of such a simulation, a system is selected.

The program we have recently developed enables us to calculate the energy consumption based on the actual load and the system's performance in meeting this load. It takes the performance and operating characteristics of the heating and cooling equipment, which is stored in a computerized database, and matches these to variable and fixed factors. Variable factors are those that change according to the climate. These include building thermal load and heat output and input of the equipment. Fixed factors are items such as operating times, installed capacity, and number of units installed. The program consists of three subsystems called "building heat load simulation," "heat source equipment simulations," and "consumption calculation broken down into energy types."

By combining a CAD program which we have developed over many years with a revised HSAP program, and by using the latest computer hardware, the program is able to simulate and display results in easy-to-understand images. Information displayed for the user includes the building heat load on a daily or hourly basis, consumption of stored heat or ice, and the operating mode. Furthermore, the designer can change the fixed factors easily and thereby determine the needed equipment capacity and performance. The system also allows the operating results of the optimum system to be documented quickly and in an understandable manner.

The following example shows how this program is applied to an air-source heat pump which utilizes thermal storage (see cover photo). This is a complex system which could not be modelled using previous programs. A brief description of the air-source heat pump system and a comparison of actual equipment operation and simulation program functions is given in this example.

Operating characteristics

The system is designed to use ice or hot water thermal storage depending upon the time of year. Storage is used to take advantage of Japan's low nighttime electricity rates. This helps to achieve demand load leveling for the electric utility. The unit is compact enough to allow roof-top installation of the assembly. Control of the system is done by microcomputers so that optimum performance is obtained. The unit can utilize thermal storage tank water or outdoor air as the heat source during heating operations. Figure 1 shows a schematic of the 30 HP unit. For larger loads, units are combined.

Outline of simulation program modelling of equipment functions

In the cooling mode of operation, the equipment is allowed to operate from 10 p.m to 8 a.m. to form ice in the thermal storage tank. The program calculates the energy consumed in producing the required quantity of ice under the actual climatic conditions and equipment performance. During the daytime the equipment operates so that electricity use is minimized. Ice inventory is kept track of and predictions based on ice already consumed determine to what extent cooling operation of the heat pumps is required. The simulation program models daytime operation by comparing the day's cool-

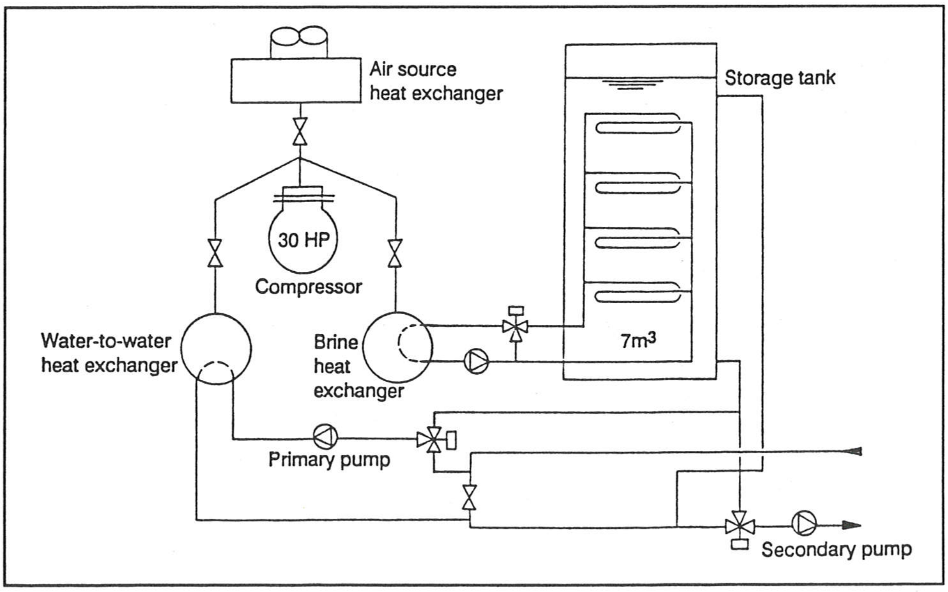


Figure 1. Schematic of 30 HP unit

ing load to the amount of ice stored and uses this to determine the number of units which need to operate during the day. If excess ice remains at the end of the day, nighttime operation is reduced accordingly.

During the heating mode, the equipment operates at night from 10 p.m. to 8 a.m. until the thermal storage has reached a temperature of 50°C. The simulation program calculates the energy needed to operate the air-source heat pump based on the initial conditions of the water in the tank and the equipment efficiency. During the day, the heating load is largest in the morning and decreases in the afternoon. Three operating modes are used: direct use of the hot water from the storage tank unit when its temperature drops down to 40°C, use of the heat pump to extract heat from the storage water down to 10°C, and using the heat pump to extract heat from the air. Use of outdoor air as the heat source has precedence when its temperature is 8°C or higher. The program models daytime operation by calculating the hourly loads and determining how many units must operate during each hour. If the load is very low, part-load performance is modelled. Similarly, the consumption of the equipment is calculated based on the air or water temperatures. Defrosting losses and thermal storage losses are also accounted for in the simulation.

Conclusion

This program allows the energy consumption of heating/cooling equipment to be modelled accurately by taking into account system part-load performance and actual climatic data. Furthermore, thermal storage systems and their effect on the electric utility load can also be simulated. Through the use of a CAD system, interactive design can take place allowing the selection of an optimum system through repeated application of the simulation. We have introduced this simulation program in the hope that it will serve as a useful tool in determining heating/cooling system energy consumption and its effect on electric utility loads.

**S. Yoshikawa, S. Igarashi, M. Goh, and S. Kamiyama, Design Department, SHIMIZU Corporation, Japan*

K. Morita and T. Ishida*

Heat Pump Room Air Conditioner Using Thermal Storage

Inverter-aided heat pump air conditioners have become quite popular in Japan. However, there are several problems when these systems are applied in cold climates such as the Hokkaido area. The first is the long time period required to heat a cold room to the desired temperature. The second is the large temperature drop which occurs in the room during the unit's defrosting operation. The former problem is due to the low capacity of the heat pump at low outdoor temperatures. The second problem exists because conventional heat pumps do not provide space heating during defrosting. In an effort to solve these problems, Toshiba, in cooperation with the Hokkaido electric power company, began development of a heat pump air conditioner using thermal storage. The product was put on the market in 1987.

System configuration

The system is divided into an indoor and outdoor unit and uses air as the heat source and heat sink medium. The outdoor unit houses a thermal storage tank. The thermal storage contains 15kg of paraffin as the storage medium. The paraffin has a melting point of 45°C and is used to provide sensible and latent heat over the temperature range 55 to 15°C. About 1.163 kW can be stored in the tank under these conditions. Complete system specifications are shown in Table 1.

Figure 1 shows the system's refrigeration cycle. The major components are the compressor, indoor heat exchanger, outdoor heat exchanger, 4-way valve, thermal storage tank, and expansion valve. Depending on the operating mode, the refrigerant flow is changed by switching the 4-way valve and opening or closing the appropriate solenoid valves. The operating modes include heating, heat storage, initial heating, defrosting, and cooling.

During the heating mode, the refrigerant exits the compressor and then flows through the heat storage heat exchanger, indoor heat exchanger, expansion valve, and outdoor heat exchanger before returning to the compressor. A portion of the heat from the high pressure refrigerant gas leaving the compressor is stored in the thermal storage tank for use during the defrosting operation.

The heat storage mode is used at night to maintain the thermal store at the desired temperature for use as a heat source in the morning during the initial heating mode. The fan of the indoor heat exchanger does not operate while the store is being recharged.

In the initial heating mode, the unit is used to bring the living space up to the required temperature. Figure 1 shows the refrigerant flow path during this mode. By using the thermal store as a heat source, the evaporation temperature of the refrigerant and the suction pressure of the compressor increase.

Model		RAS-352SVD(W)/SAVD	
Power supply		Single phase, 200V, 50Hz	
Heating performance	Temperature condition Nominal capacity Power input COP	Indoor 21°C Outdoor 7°C 5.0 kW 1.72 kW 3.0	Indoor 21°C Outdoor -10°C 3.5 kW (max. 4 kW) 1.74 kW 2.0
Maximum heating capacity using thermal storage as heat source		7 kW (15 min)	
Cooling performance	Temperature condition Nominal capacity Power input COP	Indoor 27°C Outdoor 35°C 4.0 kW 1.87 kW 2.2	
Compressor	Type Rated power	Rotary (inverter drive) 1.1 kW	
Frequency range of inverter drive		36~ 102 Hz	

Table 1. Principal specifications

This raises the capacity of the unit allowing it to rapidly heat the cold room. A conventional heat pump would take 36 minutes while the heat pump using thermal storage requires about 14 minutes.

During the defrosting mode of this system, room heating continues. Figure 1 shows the system configuration during this mode. The refrigerant which has given up its heat to the indoor air is expanded and then heated by passing through the heat-absorbing heat exchanger. The vaporized high temperature refrigerant transfers its heat to the outdoor heat exchanger thereby melting the accumulated frost. In this mode, the fan of the outdoor unit is off while the indoor fan continues operating. With a conventional system not using thermal storage, a temperature drop of 3 to 4°C would occur.

The cooling mode of this system is as with conventional units. The heat storage tank is not used at this time.

Conclusion

A heat pump air conditioner which uses thermal storage has been developed having improved operating characteristics for use in cold climates. In particular, the problems of long initial heat-up periods and room temperature fluctuations during defrosting have been solved. We hope that this development will contribute to the expansion of heat pump use in cold climates.

Keiichi Morita, Development Engineering Department, Fuji Works, Toshiba Corporation, Shizuoka, Japan; Toshiyuki Ishida, General Laboratory, Hokkaido Electric Power Co., Inc., Hokkaido, Sapporo, Japan

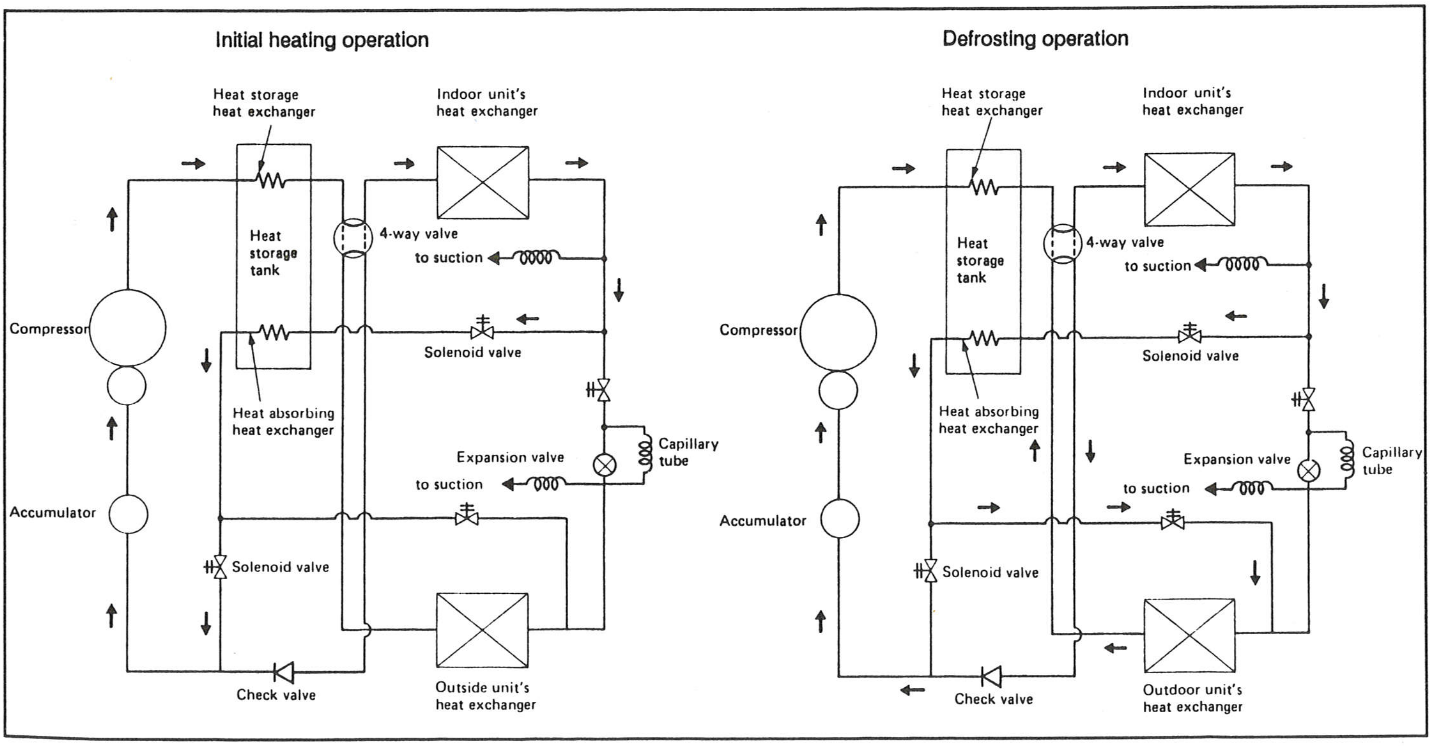


Figure 1. Refrigeration cycle (utilizing heat storage)

T. Yokoyama*

Small Gas Engine Heat Pump (GHP) for Commercial Use in Japan

Since September 1, 1987, three of the leading gas utilities in Japan have started marketing three types of small gas engine heat pumps (GHP) with capacities of 1 to 5RT. These gas companies have worked their way into the air conditioning system market, which up to now had been a monopolized market for electric air conditioners. This is expected to boost gas sales substantially throughout the year.

Introduction

Electric air conditioners, including heat pumps for residential use, are presently gaining popular acceptance in Japan with annual sales of systems for residential use of about 4,000,000 units. The electric air conditioners, including heat pumps for commercial use, have an annual sales volume of about 600,000 units. This suggests that the electric air conditioners and heat pumps are gaining a larger market share by biting into the markets appropriated by gas or oil-fired systems.

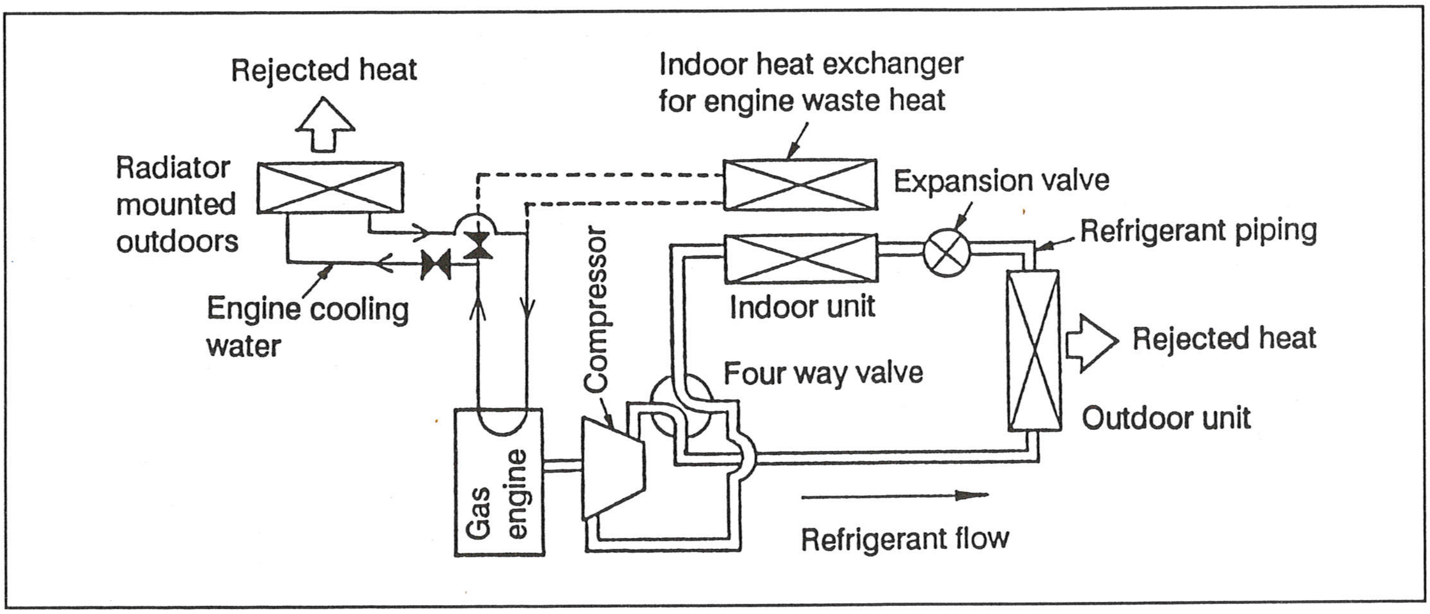


Figure 1. Cooling operation

	1 RT	4 RT	5 RT
Dimensions (m) (Height x width x depth)	1.7 x 0.6 x 0.38	1.78 x 1.0 x 0.5	1.84 x 0.97 x 0.70
Weight (kg)	148	290	450
Cooling capacity range (kW)	2.97-4.65	9.3 - 13.8	13.0 - 17.4
Heating capacity range (kW)	4.54 - 7.56	10.9 - 21.7	14 - 20.9
Gas engine type (4 cycle, water cooled)	Horizontal	3 cylinder, vertical	2 cylinder, vertical
Speed range (RPM)	1100 - 2000	1000 - 2200	1600 - 2500
Gas consumption cooling/heating (m^3/hr)	0.5/0.44	1.34/1.22	1.5/1.42
Compressor type/exhaust capacity (cc/rev)	Rotary/40	Rotary/172	Scroll/120
Auxiliary electric supply voltage/consumption (kW)	100/0.17	200/0.42	200/0.47
Power transmission system	Belt drive	Belt drive with magnetic clutch	Belt drive with magnetic clutch
Noise level at a distance of 1m (dBA)	50	60	62

Table 1. Comparison of outdoor unit specifications

Accordingly, the major task facing gas utilities in shaping their future market viability lies in developing gas-fired air conditioners designed with a heat pump cycle. This action has the support of the Ministry of International Trade and Industry (MITI), since any switchover to gas cooling will reduce the summer peak electricity demand, and hence reduce the amount of oil used for the generation of electricity. Consequently, in 1981, three gas utilities and twelve equipment manufacturing companies, with the support of MITI, established the "Research Association for Gas Engine Heat Pump Systems" in an attempt to develop a small gas engine heat pump within three years. Product development began in 1984 based on results obtained by the Research Association, with the aim to commercializing products in 1987. Various strenuous efforts were made in regards to efficiency, durability, cost reduction, noise maintenance intervals and so on. Thus, three commercial

	1 RT	4 RT	5 RT
Dimensions (m) (Height x width x depth)	0.22 x 1.03 x 0.68	0.228 x 1.49 x 0.65	0.24 x 1.29 x 0.65
Weight (kg)	32	46	45
Auxiliary electric supply voltage/consumption (kW)	100/0.08	200/0.15	200/0.11
Noise level at a distance of 1m (dBA)	45 - 33	32 - 30	23 - 16

Table 2. Comparison of indoor unit specifications (ceiling suspended type)

	Yamaha 1 RT	Yanmar 4 RT	Aishin 5 RT
Engine oil	2-3 years	1 year	1 year
Oil filter	2-3 years	2-3 years	2-3 years
Air filter	Inspection only	1 year	2-3 years
Ignition plugs	1 year	2-3 years	2-3 years
Cooling water	2-3 years	2-3 years	2-3 years
V-belt	5 years	2-3 years	2-3 years
Battery	N/A	N/A	2-3 years

Table 3. Replacement intervals for maintenance items

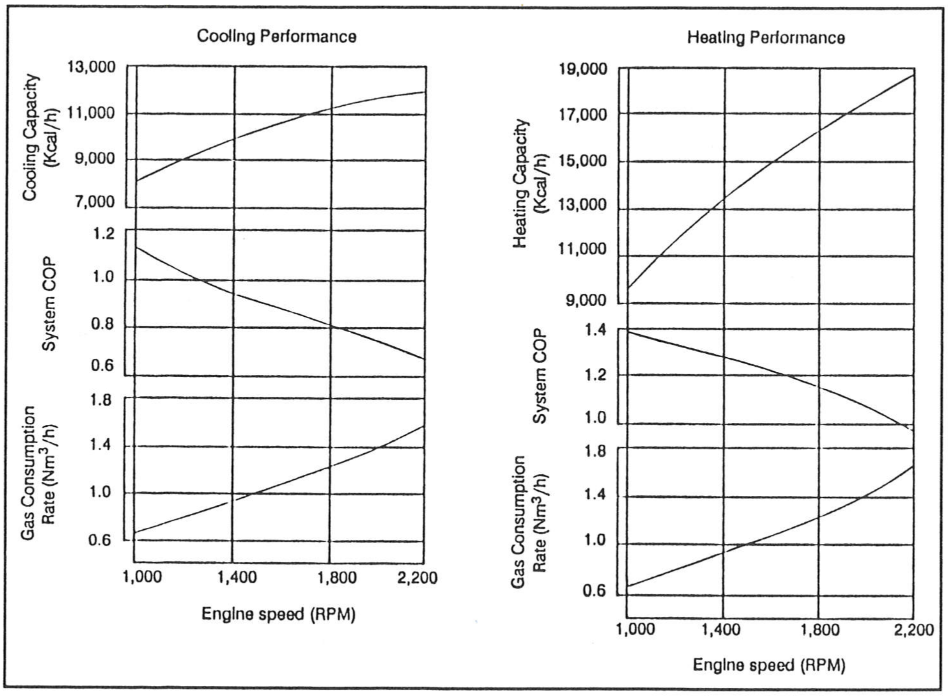


Figure 2. Cooling and heating performance of 4RT unit

GHP's were put on the market on September 1, 1987.

Principles of operation

The three systems use a standard vapor compression cycle with R22 as the refrigerant. Air is used as the heat source and sink for the evaporator and condenser, respectively. A schematic of the system during cooling operation is shown in Figure 1. To allow heating, a four-way valve changes the flow of refrigerant causing the indoor unit to become the condenser and the outdoor unit to act as the evaporator. During the cooling operation the engine's exhaust heat is used to supply domestic hot water. During heating operation, the exhaust heat is used to produce hot water as well as room heating via an indoor heat exchanger. During defrosting operations the heat output of the engine is used to continue heating. For the 4RT unit the exhaust heat is also used to supplement the defrost cycle, while for the 5RT unit it is used to provide the complete defrosting. Both the 1 and 4RT unit can utilize only one indoor unit whereas the 5RT machine can cool or heat two rooms simultaneously by using two independently controlled indoor units.

Specifications

Tables 1 and 2 compare the outdoor and indoor unit specifications. For the outdoor units the target noise levels were set at those of similar capacity electric heat pump air conditioners. This was necessary due to the noise problem in many metropolitan areas, especially Tokyo. These noise levels are shown at the bottom of Table 1. Emissions of CO and NO_x from the engines have been reduced by burning the gas with excess air. Ignition system and combustion chamber improvements have also been introduced to reduce emission problems.

The maintenance requirements of these units are shown in Table 3. For installations in Tokyo, the Tokyo Gas Heat Pump Service Center will perform regular maintenance.

Performance

The performance characteristics of the 4RT unit are shown in Figure 2 for cooling and heating. Conditions are as follows for the data shown:

Cooling:

outdoor temp. = 35°C / 27°C WB
indoor temp. = 27°C / 19.5°C WB

Heating:

outdoor temp. = 7°C / 6°C WB
indoor temp. = 21°C

Energy content of fuel: 11000 kcal/Nm^3

Conclusion

The small gas heat pumps described in this article represent the world's first attempt at mass production of equipment of this type. Sales of these units are expected to be good. Further work by the manufacturers and gas utilities will concentrate on reducing initial costs and maintenance requirements.

**T. Yokoyama, Development Center, Tokyo Gas Co., Ltd., Tokyo, Japan*

M. Moto, I. Toda, and M. Kojima*

Heating Tower Heat Pump System

Because of its design temperature condition, the head required by the compressor of a universal type air-source heat pump in summer differs greatly from that in winter. Since such differing heads can be suitably dealt with by a single screw compressor, an air-source heat pump is general employed. In these systems, however, the screw compressor is typically designed for winter operating conditions when the compression head is high, meaning that it must be operated with less efficiency for cooling in summer. Furthermore, the heating capacity drops in the coldest season because of frosting of the air heat exchanger. In order to solve this problem, we have developed an improved heat pump system and have constructed a test unit which is described below.

System configuration

A flow diagram for the heating tower heat pump system is shown in Figure 1. The system is composed of a heat pump unit consisting of two centrifugal compressors, a cooling/heating tower, a thermal storage tank, a brine (anti-freeze solution) tank, a brine pump (which acts as a cooling water pump during cooling operation), and a control device for concentration of anti-freeze solution.

The cooling/heating tower is used as a cooling tower in summer and as a heating tower during the winter. The operating conditions of the cooling tower and heating tower can be determined from a psychometric chart. In general, when the temperature falls, the gradient of saturated air line becomes gentle; therefore, for a heating tower, a larger quantity of air is required than for a cooling tower. Thus the performance capabilities of a cooling/heating tower are determined by the specifications for the heating tower.

During heating operation in winter, an antifreeze solution which has been cooled by the heat pump is sprayed in the heating tower so as to absorb heat from the warmer outside air. The transfer of heat at the design point is as follows: brine, at 11°C, flows into the heating tower, and by coming into contact with the outside air (whose wet-bulb temperature is -1.5°C), is heated to a temperature of -7°C. This brine flows into the evaporator of the heat pump and gives up heat to a still colder refrigerant (R11), which evaporates at a temperature of -16°C. The brine is cooled in the evaporator until its temperature becomes -11°C and then returns to the heating tower. At the same time, the evaporated refrigerant is carried by the two compressors to the condenser which produces 52°C hot water.

Since the cooling tower used for the test unit was an open type, it was modified so as not to allow any rainwater to mix directly with the brine in order to prevent the solution from diluting. In addition, it was equipped with a special device to minimize the loss due to carryover.

Another problem which arises is that the concentration of the brine changes. For example, when the relative humidity is high, the moisture from the outside air mixes with the brine, diluting it. On the other hand, when the relative humidity is low, the moisture of the brine evaporates, concentrating the solution. For prevention of brine freezing caused by dilution and for adjustment of the liquid level which varies according to the dilution and concentration of the brine within the heating tower, a concentration control unit is provided to adjust the liquid level in the heating tower.

Features

Compared with the universal-type air-source heat pump system, the heating tower heat pump system has the following features (refer to Table 1).

		Improved Type (Heating Tower Heat Pump)		Universal Type (R22 Screw Type Air-Source)	
		Cooling	Heating	Cooling	Heating
Performance	Cooling/heating capacity (kW)	2460	1766	2422	2488
	Chilled/hot water output temp.	5°C	52°C	5°C	52°C
	Open air condition	27°C WB	3°C WB	32°C DB 27°C WB	7°C DB 3°C WB
	Input (kW)	530	580	789	800
	COP of heat pump itself	4.64	3.04	3.07	2.72
	Auxiliary power:				
	Outdoor unit's fan (kW)	5.5 x 4 units 22	22	11 x 10 units 110	110
	Pumps (kW)	Cooling water 55	Brine 45	Not required	Not required
	Overall COP (incl. aux. units)	4.05	2.61	2.69	2.39
Installation space (m^2)	Heat pump	19.2		33.5	
	Heating tower/air heat exchanger	59.9		105.5	
	Cooling water/brine pump	2		--	
	Brine tank	15		--	
	Total installation space	96.1		139	

Table 1. Comparison of improved type and universal type

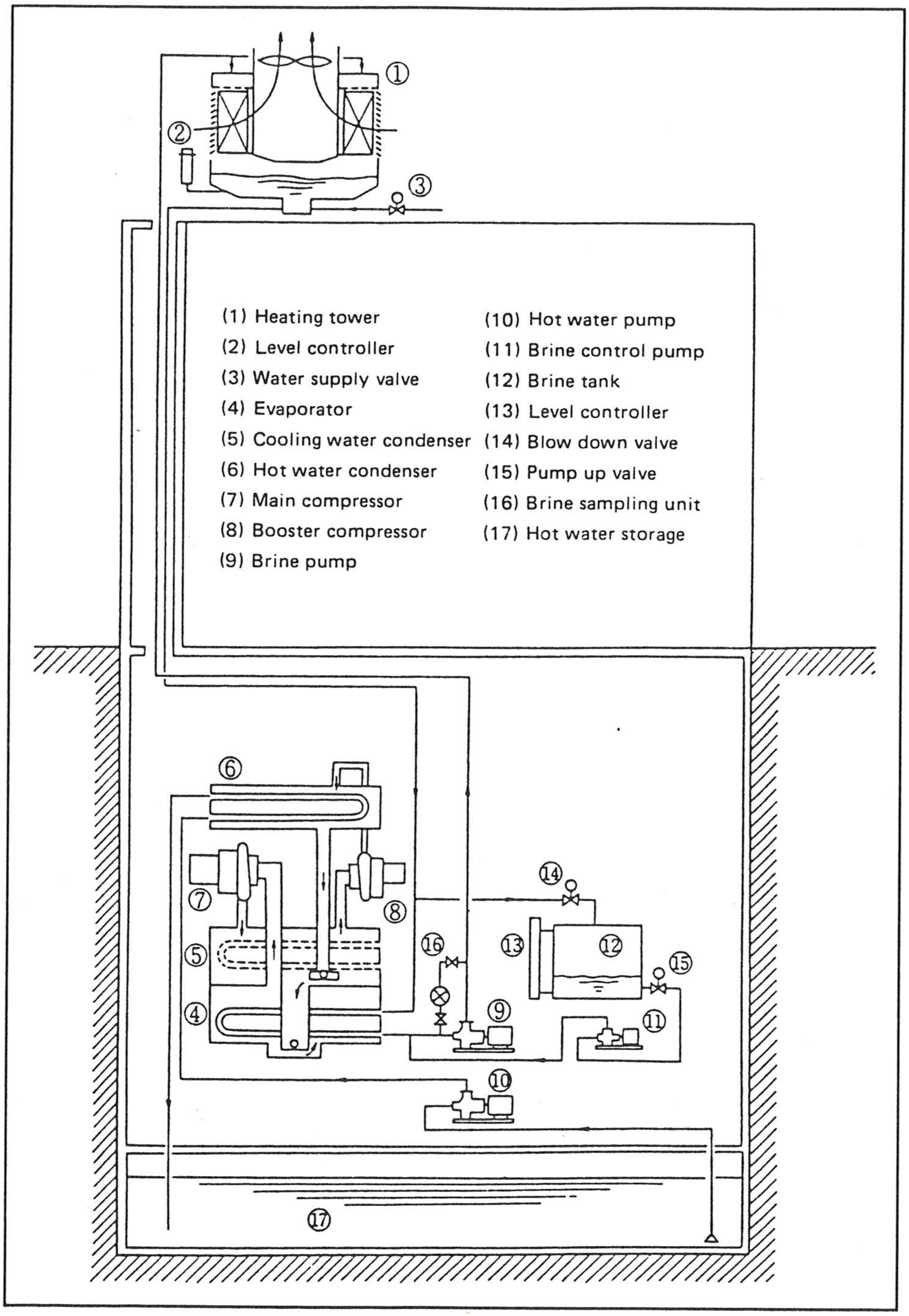


Figure 1. Heating cycle flow

Excellent year-round system COP

The difference in compression head between summer and winter operation is handled by dividing the centrifugal compressor unit into a main compressor and a booster compressor. During the cooling season, only the main compressor is operated; during the heating season, series operation of the two compressors is used so that the insufficient capacity of the main compressor is augmented by the booster compressor. This operational mode, along with an economizer cycle utilizing the intermediate pressure of the two compressors, helps improve the system's COP.

Further COP improvements are achieved since the heating/cooling tower's yearly fan power requirements are 1/4 of that required by an equal capacity finned-coil heat exchanger. The use of a heating/cooling tower also improves the heat transfer efficiency and eliminates the frosting problem associated with finned-coil heat exchangers.

Noise reduction

Since the tower fan power of the outdoor unit decreases to about a quarter of its original value, the noise produced by it decreases accordingly.

Reduced installation space

If installation space is limited, this type of heat exchanger can be advantageous since the floor area required by the heating tower is less than that required by a universal type air-heat exchanger (about 60% less).

Feasibility of manufacturing large-capacity machines

Since this system uses centrifugal compressors, a single heat pump unit with a capacity that can be expanded to about 5.3 MW can easily be manufactured.

Control of brine concentration

The mean winter weather conditions in Tokyo are as follows: during the coldest months of January and February when the heat load is large, the relative humidity is low; during the off-season months of November, December, March, and April when the heating load is small, the relative humidity is high. This is a desirable phenomenon for a heating tower heat pump system. During the days of the coldest season when the heat pump system is operated for many hours, the brine tends to concentrate and there are not many time periods when the brine tends to dilute. Also, since the outside air temperature is usually high during the off season months, there is little danger that the brine will freeze, even if it is diluted, since the heating load is small and the number of operating hours of the heat pump is small.

Diluted brine can be concentrated by circulating the brine through the tower to allow the excess water to evaporate when the heat pump is not in use. Therefore, it is a point of machine operational control to concentrate the brine during periods when the heat pump is at rest.

In general, the typical heating load in Tokyo can be met in a satisfactory manner if the brine tank is utilized as a cushion, the thermal storage tank is used properly, the heat pump is operated properly, and the tower fan performs the brine concentrating operation according to the load and weather.

In the prototype system, an automatic control panel which has a built-in microcomputer is used to monitor the concentration of the brine and the liquid level of the heating tower and brine tank. In order to achieve an optimum concentration control system, the present control system will be improved by utilizing data collected under various operating conditions.

When the brine tends to dilute, the following actions may be taken:

- Since the liquid level of the heating tower rises, recover the amount of brine corresponding to the rise in liquid level.
- Detect the concentration of the brine when the concentration has become smaller than the preset concentration for control, then change the machine operation over to the demand operation mode to prevent the brine temperature from falling.

When the brine becomes more concentrated, the following actions may be taken:

- Since the liquid level of the heating tower decreases, control the liquid level of the heating tower by adding brine from the brine tank.
- When the brine has been concentrated to the point that the brine tank has become empty and the liquid level of the heating tower has decreased below the lowest allowable level, water is added to the brine.

Brine

An ethylene glycol water solution, used for general anti-freezing purposes, is employed for this system. Results from investigations and examinations of this substance indicate that the health effects on humans are negligible. Drainage of the substance into the public sewage system is limited to concentrations of less than 160 ppm by the Water Pollution Prevention Act since its presence increases the biological and chemical oxygen demand. The results of measurements on its carryover into the atmosphere show that its maximum value at the heating tower outlet is 0.04 ppm. It is unlikely that this can have an influence on organisms.

Brine replacement

It is estimated from measurements that the consumption of the brine due to carryover and adherence to piping during changeover from heating to cooling and vice versa during one heating season is about 1 m^3. Therefore, the solution has to be replenished once a year before starting the heating operation.

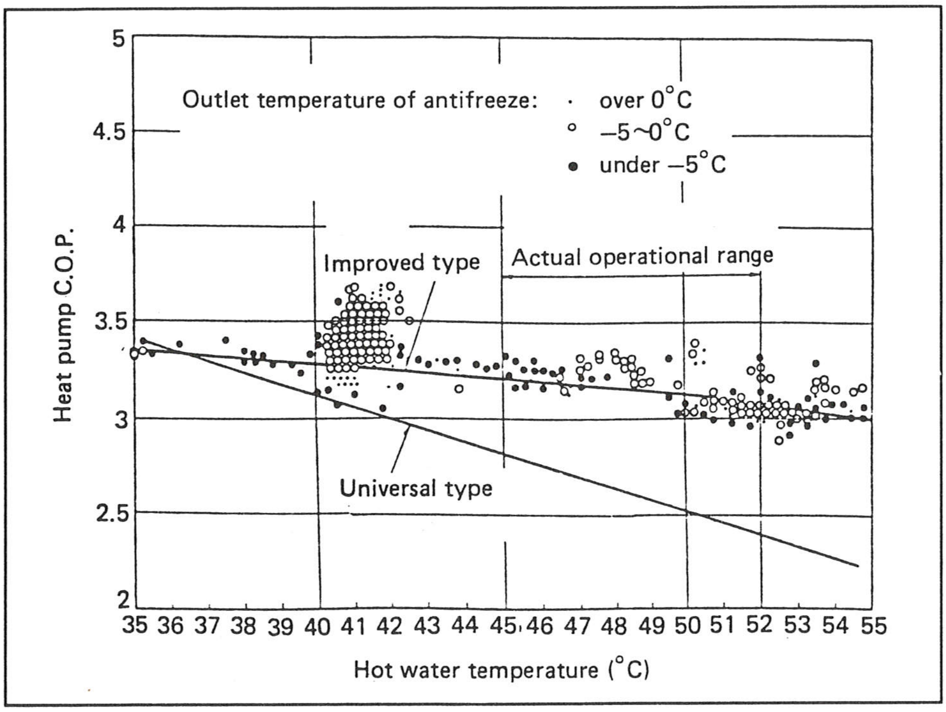


Figure 2. Heat pump COP

Deterioration

According to the experience collected over a period of 26 years by the Hiroshima Broadcasting Station, no sign of denaturation of the brine was ever detected so that the brine was not replaced but only replenished each year. For this reason, deterioration of the brine is not expected. Nevertheless, the brine will be analyzed each year to check for deterioration.

Heat pump operating characteristics -- COP during the heating season

The hot water temperature to be supplied for heating purposes by the demonstration unit was taken as 52°C. This accounted for heat loss, etc., in supply piping when the machine is used for DHC. In a general type system which

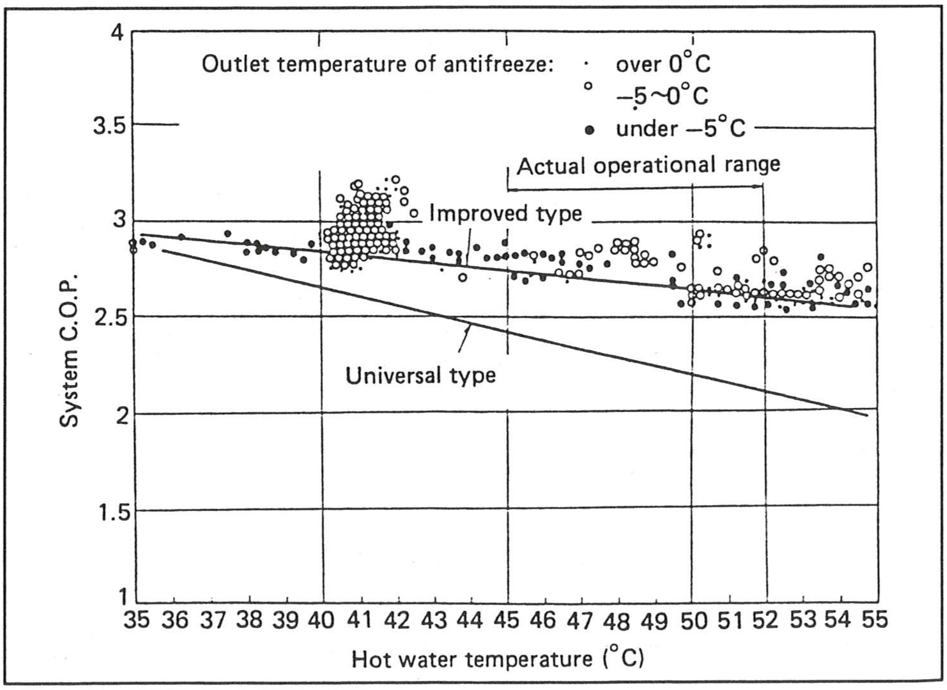


Figure 3. System COP

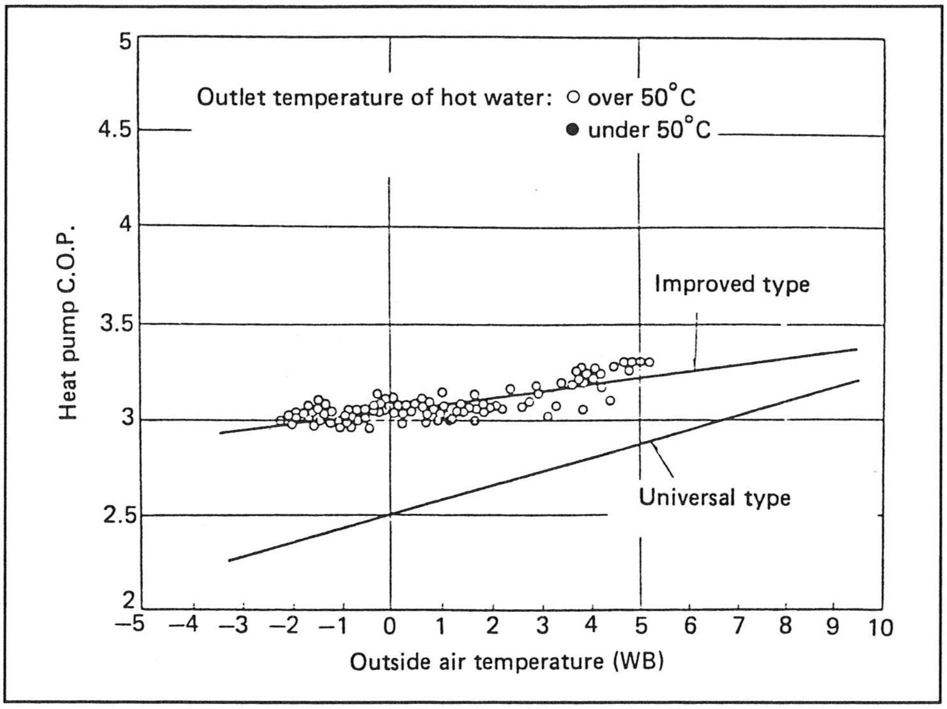


Figure 4. Heat pump COP

serves a single building, a hot water temperature from 45 to 48°C is generally used. Figures 2 and 3 show the COP of the heat pump and the system, respectively, by using the temperature of the hot water outlet as a parameter. The system COP includes the auxiliary power (tower fan and brine pump) when the wet-bulb temperature of the outside air is -1.5°C. It is seen from these figures that the COP of the demonstration unit over the actual working hot water temperature range is better than that of the universal-type machine.

Figures 4 and 5 illustrate the COP of the heat pump and the system, respectively, by using the wet-bulb temperature of the outside air as a parameter when the warm water temperature is 52°C. A heat pump COP of greater than 3 was achieved, confirming the energy-saving characteristic in comparison to the universal-type heat pump.

Heating operation during off-season

During the off-season, the outside temperature is comparatively high and, accordingly, the heating load is very small. If during this season the brine in its circuit is replaced with water and the heating tower system is operated with a water circuit, a higher COP can be expected. At the same time, an increment or decrement of the water held in the circuit can be dealt with by either draining water from the overflow pipe or adding water. (The heat pump operation is usually total-load operation, but in this case, cold water outlet temperature control also must be used in order to prevent freezing.) The heating tower heat pump system operation using water instead of brine will be executed by March 1988 to determine the operating range which the system can handle.

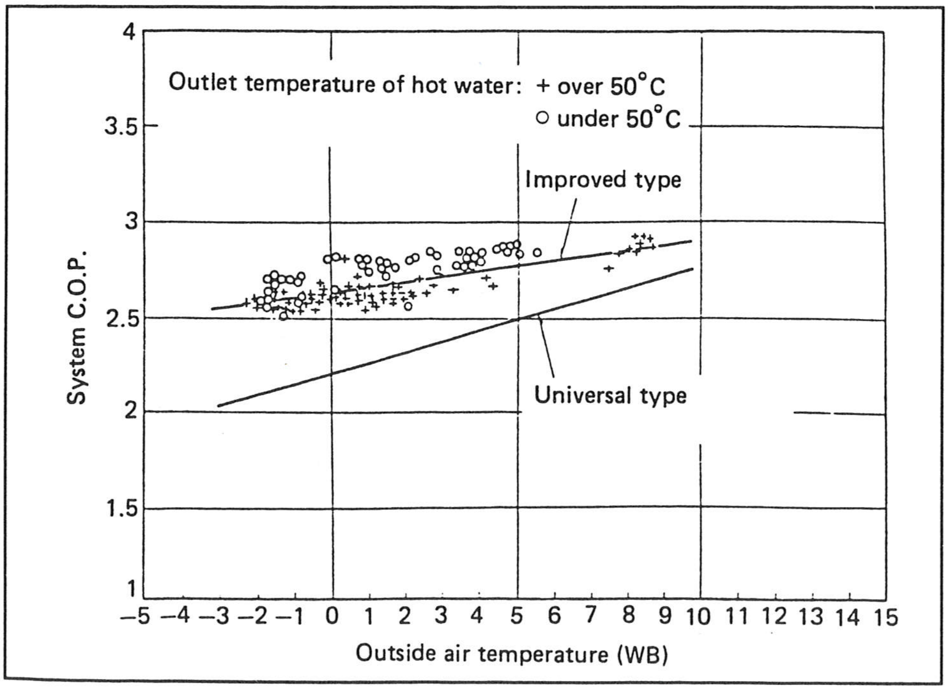


Figure 5. System COP

Economy -- initial cost

For large capacity (1.8 - 5.6 MW) units, the goal of reducing the initial cost of the system to that of the universal-type air-source heat pump system has been achieved. Further cost reductions of 10 to 20% are being sought.

Medium capacity units (under 1.4 MW) are more costly and units under 1 MW are considered uneconomical.

Running cost

Compared to the universal-type air-source heat pump system, the system has an excellent system COP all year-round which means that energy consumption is greatly reduced.

Conclusion

The use of a heating tower heat pump system has a number of advantages over the universal-type air-source heat pump system. For large capacity units, therefore, it is expected that these systems will be widely used. Future work will be to monitor property changes of the brine, pursue optimization of the concentration control system, and to simplify the system, making it more compact and efficient. Medium capacity units will also be examined from all angles in order to make them as competitive as possible with the universal-type air-source heat pump system.

*M. Moto, EBARA Corporation; I. Toda, Tokyo Electric Power Company; and M. Kojima, Tokyo Electric Power Real Estate Maint. Co., Inc.; Tokyo, Japan

L.-O. Glas*

Experiences from Air-to-Water Heat Pump Plants for Building Heating

The following is a translated summary of the paper "Drifterfarenheter fraan luft-vattenvaermepumpar foer bostadsvaermning" written by Mr. Lars-Olof Glas, Skandinavisk Termoekonomi AB, Stockholm, in conjunction with the Heat Pump Group of the Swedish Council for Building Research.

Introduction

Adjustments can never be avoided on a new installation. They usually have to be done even if the project has been thoroughly planned. But by experience it is known that adjustments and modifications are inversely proportional to the care taken in designing and installing the plant. In the cases described in this paper, the author has been involved in all matters concerning the installation of the plants. Special attention has been given to optimizing the capacity of pumps, fans and heat exchangers.

All the plants are equipped with electric motor-driven units and are installed in the Stockholm area. Design outside air temperature is -18°C and the mean temperature during the heating season September 20-May 20 is around +3°C. Plant numbers 1 through 5 use outside air as a heat source and numbers 6 and 7 use waste air from the building.

In all the installations except number 7, the compressors cycle on-off at part-load. Comparative tests have been made on Plants 2 and 6 which showed a 20-30% higher COP with on-off operation, in spite of lower evaporation and higher condensation temperatures, than with the compressors operating at part load. Table 1 gives a short description of the plants.

Data and performance

Table 2 shows main design and performance data. Refrigerant R12 has been used in some cases instead of R22, so that higher temperatures, needed for the production of domestic hot water, can be obtained. Temperatures indicated are typical operating values.

In Table 3, the values from Table 2 have been converted into characteristic relative values. The cooling load of the surface of the air coolers is of great importance for the length of the intervals between defrosting, which is also indicated in the table. The higher the specific load, the shorter the defrosting intervals.

The table also shows that the specific amount of refrigerant can be drastically decreased when using standardized factory-built water-to-brine units and indirect cooling systems instead of site-built installations with direct cooling systems. Compare, for instance, Plants 2 and 5. Other advantages are:

-- Simple installation and adjustments

-- Less risk of dirt in the system with consequent operating interruptions and breakdowns

-- Fewer potential leakage spots

-- Lower installation costs due to standardized units (compare Plants 2 and 5).

However, there is the essential advantage with direct cooling of being able to operate at lower outside air temperatures. For instance, Plants 1 and 2 have

Plant Number	Application	Year Installed	Type
1	Single-family house	1981	1 hermetic reciprocating compressor Direct expansion evaporator
2	800 apartments and a shopping center	1984	2 screw compressors Flooded evaporator w/refrigerant pump circulation
3	10 one-family houses	1984	1 semi-hermetic piston compressor Indirect cooling using a 24% $CaCl_2$ brine
4	460 apartments	1987	4 screw compressor units Indirect cooling using a 24% $CaCl_2$ brine
5	2430 apartments, 1 shopping center, 2 schools	1987	2 screw compressor units Indirect cooling using a 24% $CaCl_2$ brine
6	120 apartments	1985	1 semi-hermetic reciprocating compressor unit Indirect cooling using 30% ethylene glycol
7	2500 apartments	1986	2 screw compressor units Indirect cooling using 18% ethyl alcohol

Table 1. Plant descriptions

Plant number	1	2	3	4	5	6	7
Refrigerant	R502	R12	R22	R12	R12 and R22	R22	R12
Cooling coil surface (m^2)	53	10000	368	8400	34000	680	13060
Air flow ($1000 m^3/h$)	1.5	540	12.8	210	840	23	350
Heating capacity (kW)	7.5	2400	58	950	3450	185	3450
Hot water temperature (°C)	37	50	40	60	50	45	70
Air temperature in/out (°C)	0/-7	0/-6	0/-6	0/-5	0/-5	20/5	20/5
Cooling capacity (kW)	5.2	1600	40	590	2400	135	2100

Table 2. Design and performance data

Plant number	1	2	3	4	5	6	7
Balance temperature (°C) [1]	-3	-2/+3	+4	0/+5	+5/+10	+5	+10/+15
Cooling capacity per cooling coil surface (W/m^2)	100	160	110	70	70	200	16
Frosting time (min)[2]	120	90	120	180	180	-	-
Refrigerant charge per heating capacity (kg/kW)	0.8	3.3	0.5	1.0	0.5	0.3	0.5

[1] Lowest outside air temperature at which the heat pump capacity equal to heat capacity needed. Where two values are shown, this is dependent on domestic hot water consumption as no accumulator is installed.

[2] Time for frosting to zero gap between the fins of the cooling coils at air intake when outside air 0°C and 95% humidity.

Table 3. Characteristic relative values

Plant number	1	2	3	4	5	6	7
Operating time per compressor (h)	4254	5660 6270	5038	3684 4100 2735 2493	3079 2238	6531	6855 8340
Heat from heat pump Q (kWh)	25.9	12560	283	2905	7054	1170	23181
Electric energy consumption ET (kWh)	10.4	5676	125	1468	3068	412	10365
COP total	2.49	2.21	2.26	1.98	2.3	2.84	2.22

Table 4. Measured values for a one-year period

Plant number	1	2	3	4	5	6	7
Total investment I (thousand SEK)	50	13000	500	5000	12000	1100	14000

Table 5. Installation costs

Plant		1985	1987
2	Heat energy costs to be replaced (SEK/kWh)	0.24	0.15
	Electric energy costs (SEK/kWh)	0.26 (mean)	0.24 (low load)
	Energy cost saving (SEK/yr)	1,560,000	450,000
	Total cost saving (SEK/yr)	1,070,000	-940,000
5	Heat energy costs to be replaced (SEK/kWh)	0.24	0.15
	Electric energy costs (SEK/kWh)	0.24	0.29
	Energy cost saving (SEK/yr)	3,120,000	690,000
	Total cost saving (SEK/yr)	180,000	-870,000
7	Heat energy costs to be replaced (SEK/kWh)	0.24	0.15
	Electric energy costs (SEK/kWh)	0.24	0.26
	Energy cost saving (SEK/yr)	3,050,000	1,500,000
	Total cost saving (SEK/yr)	1,510,000	-1,050,000

--Q and Et according to Table 4
--I according to Table 5
--ES for heavy oil for all plants
--Plant 2:
1985 u = 0, a = 0.038
1987 u = 0.02, a = 0.08
Q = 9700 MWh/year
Et = 4200 MWh/year due to 1500 h/year stop when
EH = 0.44 SEK/kWh
--Plant 5:
Q=22,000 MWh/year
Et = 9000 MWh/year, when adjusted
a = 0.11
1985 u = 0
1987 u = 0.02
--Plant 7:
a = 0.11
1985 u = 0
1987 u = 0.02

Table 6. Calculated profitability of selected plants

operated without any problems at outside air temperatures down to -26°C with a COP of 1.8 and 1.4, respectively. Compare this to the fact that today there are no cold-carrying fluids available which can operate economically at outside air temperatures lower than -20°C if the performance at temperatures of -10°C and higher are profitable.

In Table 4, experience values from operation over a one-year period are shown. The period for Plant 4 is 24 August 1987-2 March 1988, and the period for Plant 5 is 22 October 1987-2 March 1988. As all installations are thoroughly designed, adjusted and maintained by skilled personnel, the performance attained can be said to be on the highest level possible with the actual heat sources and heat sinks. However, Plants 4 and 5 are still being adjusted. Their capacity should be increased by 10% and the COP by at least 5%.

Economy

All the plants have been financed by special government energy loans which were available at the time of installation. Table 5 shows the total installation costs, including necessary new buildings.

The plants were planned and designed during a period with considerable oil price increases and relatively slow increases in the price of electricity. Swedish hydro- and nuclear electric energy production is cheap and normally covers the total demand in the country. A consequence of the recent decision to shut down all nuclear power stations is, however, that the price for electricity has started to increase, partly to decrease the consumption and partly to collect the money needed to build the replacement power stations needed. Since 1986 oil prices have decreased considerably. These circumstances have drastically decreased the profitability of the plants.

Energy cost saving (SE) can be calculated as:

$$SE = Q \times ES - Et \times EH$$

and the total cost saving (ST) calculated from:

$$ST = SE - I \times (a + u)$$

where:

Q = heat pump heat energy prod.
ES = price of replaced heat (oil heat)
Et = heat pump energy consumption (electricity)
EH = price of electricity
I = total investment (see Table 5)
a = annuity factor, amortization plus interest
u = service and maintenance cost factor (= 2%; during guarantee period = 0%)

Relevant data and calculated values related to Plants 2, 5, and 7 are presented in Table 6.

As can be seen, the plants were profitable with the energy prices valid at the time of design and installation but are not profitable with the 1987 electric-to-fuel price ratio. Still, it is profitable to operate the plants as they are saving energy costs and the capital costs are independent of operation time.

Further information

The complete report also contains a short presentation of the presumptions for the installations, the changes made in the heating systems, the plant design and some of the difficulties experienced during the initial operation period. It also contains a chapter concerning calculation of frosting and defrosting of cooling coils. The theories have been tested with very good agreement with the experience from the plants studied.

**Lars-Olof Glas, Skandinavisk Termoekonomi AB, Stockholm, Sweden. Translated from Swedish into English by Åke Bratt, IEA Heat Pump Center.*

Å. Bratt

Controlling the Defrosting of Air Evaporators for Heat Pumps

The following is a summary of the Swedish Council for Building Research (BFR) Report R52:1987 entitled "Defrosting of Air Evaporators in Heat Pumps: Evaluation of Defrosting Methods" (Avfrostning av luftberoerda foeraangare i vaermepumpar: Utvaerdering av avfrostningsmetoder) by Ulf Bergstroem and Reinhold Larsson. This study is of the control and performance of defrosting systems for seven air heat pumps in the capacity range of 10 to 3000 kW.

Formation of frost

Frost is formed on the surface of heat pump air evaporators when water condenses and the temperature of the surface is lower than the freezing point of the water. In most cases, frost is formed when the outside air temperature is below 5-8°C. Normally, frosting is most rapid at outside air temperatures around 2°C.

Factors which affect the formation of frost on air evaporators are primarily the following:

-- Outside air temperature
-- Air humidity
-- Evaporator design
-- Cooling capacity
-- Airflow

Under the climatic conditions in Sweden, frosting can be calculated to occur during about 4,000 hours per year.

Frost is most commonly formed at the entrance of the air evaporator, but there are a number of parameters influencing where the frost is formed. This has been studied by, among others, A. Maelhammar in his doctoral dissertation at the Royal Institute of Technology in Stockholm.

Air evaporators must be defrosted

The latent heat of vaporization and fusion contributes to the heat capacity when water condenses and freezes in an air evaporator. But the consequence of the formation of frost is mainly that the flow of air is decreased when the frost thickness closes the space between the cooling coil fins considerably. This means that the performance of the heat pump decreases with increasing frost formation.

The overall heat transfer coefficient is not influenced very much by the fact that the frost layer increases the turbulence. As the performance of the heat pump decreases with increasing frost, the cooling coil must be defrosted to maintain reasonable performance and operating costs.

How to defrost

Defrosting is usually achieved by adding heat to melt the frost. There are different ways to do this. The most common ways are:

-- Hot refrigerant gas defrosting

-- Hot refrigerant liquid defrosting

-- Outside air defrosting (when the air temperature is higher than 0°C)

-- Hot water defrosting

-- Electric defrosting

-- Hot brine defrosting (indirect systems)

Mechanical defrosting is also used. A study of the different defrosting methods is presented in BFR Report R39:1986 entitled "Heat Pumps Using Air as Heat Source - Defrosting Methods." Most of the methods have been shown to be suitable. However, it is difficult to judge which method is most economic.

When using heat for defrosting, the theoretical amount of heat energy needed is equal to the heat energy for melting the frost. In reality, there are heat losses connected to the defrosting procedure such as:

-- Decrease in running time

-- Heat convection and radiation losses

-- Water evaporation losses

-- Losses due to heating of material

These losses can be considerable, especially in the case of flooded evaporators containing large amounts of refrigerant or when indirect systems with brine are used. The method of initiating the defrosting sequence also has a large influence upon the losses.

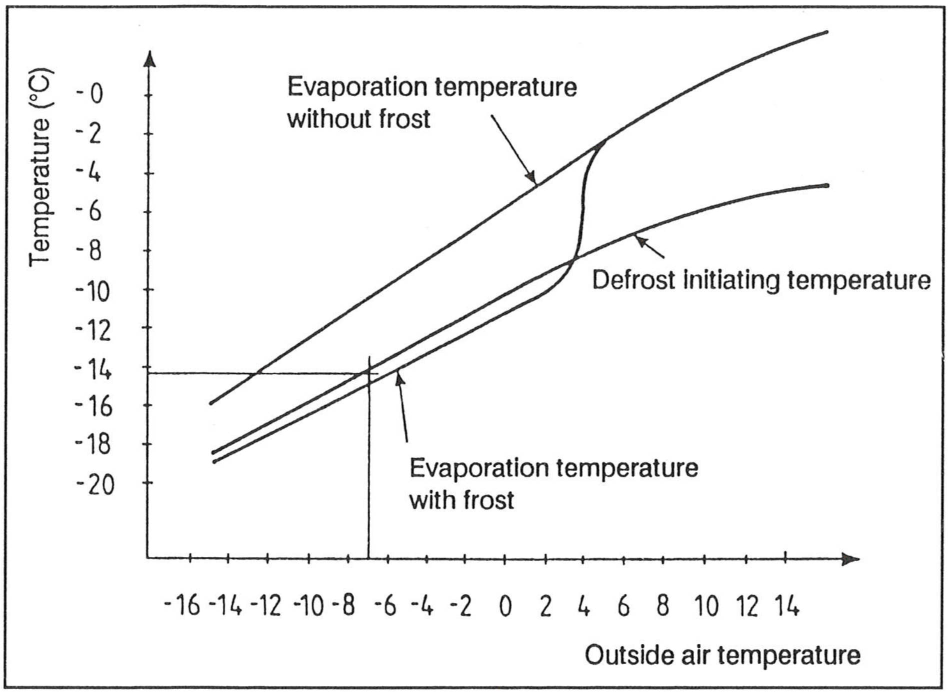


Figure 1. Evaporation temperature versus outside air temperature

Control of the defrosting sequence

Starting and stopping the defrosting sequence can be made in two different ways: initiation by time or initiation by need.

Initiation by time means that the defrosting sequence is controlled by some kind of timer with an adjustable, preset defrosting program. Although this method is simple and reliable, it is not very economic.

Initiation by need means that the defrosting sequence is controlled by some measured values which indicate that defrosting is needed or finished, respectively. This method is more complicated and needs thorough adjustment and careful attention to be reliable, but can be very economic.

The tested value to initiate defrosting is usually either the difference between outside air temperature and evaporation temperature, or the pressure drop on the air side of the evaporator. The temperature-difference method is based on the fact that, when frost is formed on the evaporator surface, the evaporation temperature is decreased due to the decrease in heat transfer. This principle is illustrated in Figure 1.

The tested value to stop defrosting is usually the temperature of the surface of the evaporator.

How much can be saved by a well-designed defrosting system?

When designing the defrosting control system for an air heat pump, it is important to calculate how much extra energy can be saved by using the sophisticated control equipment, and if this will pay for the extra installation costs.

In small installations for single-family houses, it is difficult to justify a complicated defrosting control system from an economic point of view. For this type of installation, a simple and reliable system is more important. In high capacity installations, however, there is a great potential for savings by using a well-designed and adjustable sophisticated defrosting control system.

Tests have also shown that defrosting control systems are often not functioning properly. They need adjustment and maintenance, and should, therefore, be used only on those large installations where qualified operators are available.

Proposal for a defrosting control system

The most common automatic control system for defrosting, initiated by need, is based on the temperature comparison method. This method is based on knowledge of how the evaporation temperature changes with, primarily, the temperature of the outside air. When the evaporator is frosted, heat absorption from the air decreases and the evaporation temperature decreases. By testing the temperature of the outside air and the evaporation temperature, it is possible to decide if the evaporator is frosted.

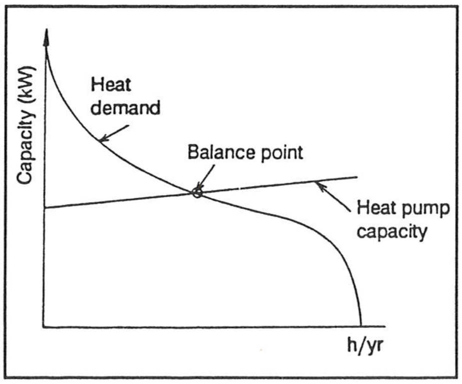


Figure 2. Capacity duration (definition of balance point)

It is now necessary to know how much the evaporation temperature must deviate from the normal value when initiating defrosting in order to get the highest cost saving. It should also be kept in mind that condensation temperature and air humidity have some influence on the evaporation or brine temperature, which means that the final adjustment must be made with the installation in operation.

Before going further, it is necessary to define the "balance" point for a heat pump in a bivalent system. This is shown in the capacity duration diagram for the installation in Figure 2.

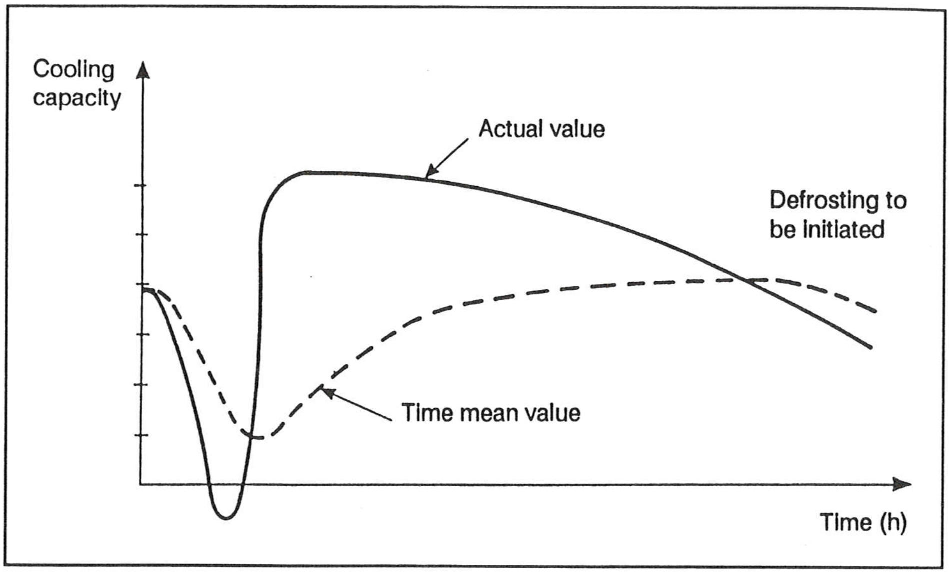


Figure 3. Cooling capacity versus time

Theoretically, optimal intervals between defrosting can now be decided upon under the following conditions.

Under bivalent operating conditions (to the left of the balance point), the capacity of the heat pump is less than the heat capacity needed. Optimal savings are achieved when the time mean value of the cooling capacity of the heat pump is maximum. To determine initiation of defrosting, one obviously must know the time mean value of the cooling capacity after the start of the last defrosting sequence as well as the actual value of the cooling capacity. When the actual cooling capacity is less than the time mean value, further operation will result in less cost savings and a new defrosting sequence should be initiated. This is illustrated in Figure 3.

Under heat pump only operating conditions (to the right side of the balance point), the capacity of the heat pump is higher than the capacity needed and the heat pump operates at part-load. Optimal savings are achieved when the addition of driving energy to the heat pump is minimum, which means that the time mean value of the coefficient of performance (COP) should be maximum. A new defrosting sequence should then be initiated when the actual value of the COP is less than the time mean value. This is illustrated in Figure 4.

On a large plant, the heat and driving capacities should always be tested. Since the cooling capacity is the difference and the COP is the ratio between these two values, it should be possible to introduce this defrost initiation strategy without excessive control equipment costs.

Summary

The defrosting system on seven air heat pumps in operation has been studied and it was found that, in many cases, defrosting is initiated without there being any real need for it. Even in installations with need-initiated defrosting control equipment, improper installation and adjustment causes this problem. Expensive defrosting control systems are normally not justifiable from an economic point of view in small capacity plants. Here, simplicity and safe operation are most important. In high capacity installations, need-initiated control systems have a high cost-saving potential. In Report R52:1987, a proposal for a need-initiated defrosting control system strategy is presented.

Å. Bratt, IEA Heat Pump Center, Karlsruhe, Federal Republic of Germany

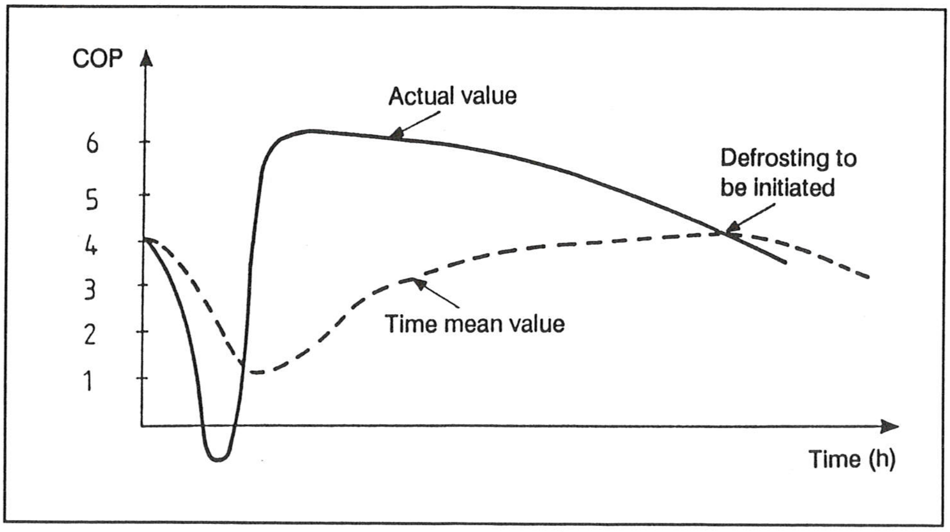


Figure 4. COP versus time

P. Vinz*

Future Possibilities of AHP Technology for Refrigeration, Air Conditioning, and Heating Systems

On the basis of an improved ammonia-water absorption heat pump (AHP) cycle, the potential of two promising AHP systems is analyzed and the market potential for cooling, air conditioning and heating applications is discussed. AHP-A, using air as the heat source, is most appropriate as a year-round heat generator for use in high temperature hydronic heating systems. AHP-B, on the other hand, is therefore most applicable to cooling/heating and for low temperature hydronic heating applications. Both AHP-A and -B are compared to electric heat pumps and modern boilers as part of an economic analysis which accounts for the energy costs, interest rates, inflation, and tax advantages currently available in Germany.

Technical improvements

Two absorption heat pump designs (referred to as AHP-A and AHP-B) are proposed based on the familiar single stage NH_3-H_2O absorption refrigeration cycle. Both designs are suitable for construction as small or large capacity systems. These systems will not only allow conservation of fossil fuels but also provide an alternative to systems using CFCs.

Both AHP designs use the following patented improvements to increase the cycle's efficiency:

- an energy-saving rectification method for dehydration of the high pressure refrigerant
- an energy-saving concentration stabilization method for the low pressure refrigerant mixture
- a simple method of circulating the refrigerant and absorbent which allows capacity control and minimizes pump energy input
- a method whereby the absorbent stream is divided into two streams and each stream is enriched to a different concentration with the ammonia refrigerant.

The optimization goals of both designs are different. For AHP-A the improvements are used to optimize a heat pump able to provide a high temperature lift (water supply temperature of 60-80°C) at the lowest possible refrigerant evaporation temperatures (down to -30°C). For AHP-B operating at lower heat rejection temperatures (35 to 55°C) and higher refrigerant evaporation temperatures (-5 to 15°C) the improvements are used to increase the internal recovery of waste heat. In the latter case, a significant fraction of the absorption waste heat is reused in the cycle, reducing the amount of external heat input required. Both designs can easily be combined into one unit.

Application possibilities

With the above proposed system, a complete spectrum of application possibilities exists, ranging from refrigeration to high temperature space heating systems. In all cases the efficiency of the proposed AHP system is greater than that of existing standard compression systems under the same operating conditions.

Because of the low evaporation temperatures it can achieve, AHP-A can use air as a year-round heat source. Thus it can provide the total heating requirements year-round without the need for auxiliary boilers or thermal storage systems. In addition, the high investment required for securing year-round heat sources, for example, well drilling for groundwater use, are eliminated. AHP system A can also be used to provide refrigeration and air conditioning at high outdoor temperatures, making it an especially attractive system for use in hot regions with limited water supplies. In these regions solar energy could be used as the heat input.

Systems based on AHP-B are especially useful for air conditioning and low temperature hydronic heating system applications. They are less suitable for refrigeration use. By using groundwater as a heat source AHP-B can be used year-round, whereas with air as the heat source, an additional boiler would be necessary to meet the heating load. The most economic performance of AHP-B is obtained in applications requiring simultaneous heating and cooling or winter heating and summer cooling.

The combined AHP (A + B) is universally applicable for simultaneous or separate refrigeration, air conditioning, and low or high temperature hydronic heating. A system of this type always operates at optimum performance depending upon the load and the corresponding heat sink and source temperatures. As soon as an operating mode has reached its limit, the system switches to the other operating mode, thereby maximizing efficiency.

Performance of the proposed AHP systems

The performance of the AHP-A and AHP-B is shown in Table 1 for various application conditions encountered in air conditioning and high and low temperature hydronic heating systems. All values shown take into account a 5-10K temperature difference in the heat exchanger. The extensive temperature ranges covered for calculation of the efficiencies clearly indicates the performance of these systems. The last column in Table 1, Actual COP/Carnot COP, shows that the proposed AHP comes much closer to achieving the full theoretical Carnot performance than any currently available alternative systems operating even under optimized conditions.

For all cases, air was the heat source since it requires the lowest investment in peripheral equipment. By using AHP-

System Type	Outdoor Air Temp (°C)	Refrigerant NH_3				Heating Water (Heat Sink)		Cooling Efficiency	COP Actual	
		Evaporator		Condenser		T_{inlet}	T_{outlet}	$\xi = \frac{q_o}{q_H/\eta_{generator}}$	$HZ = \frac{q_o + q_H + \Delta q_{AG}}{q_H/\eta_{generator}}$	COP Actual / Carnot-COP
		T(°C)	P(bar)	T(°C)	P(bar)	(°C)	(°C)			
High temp AHP-A	≤ -15	-25	1.5	80	41.3	50	75	0.35 [1]-0.37 [2]	1.30 [1]-1.32 [2]	0.697[1]
	≥ -15	-20	1.9	78	39.6	49	73	0.40-0.42	1.35-1.37	0.706
	≥ -10	-15	2.4	76	38.8	48	71	0.46-0.49	1.41-1.43	0.715
	≥ -5	-10	2.9	71	33.9	47	66	0.48-0.51	1.44-1.46	0.715
	≥ 0	-5	3.5	66	30.2	46	61	0.50-0.52	1.46-1.48	0.713
	≥ +5	0	4.3	60	26.1	44	55	0.56-0.59	1.52-1.54	0.680
	≥ +10	+ 5	5.1	55	23.1	42	50	0.59-0.61	1.55-1.57	0.633
	≥ +15	+10	6.1	50	20.3	40	45	0.63-0.66	1.58-1.61	0.616
Low temp AHP-B	≥ +5	0	4.3	50	20.3	40	50	0.88-0.92	1.82-1.87	0.618
	≥ +10	+5	5.1	50	20.3	40	50	1.00-1.05	1.95-2.00	0.613
	≥ +3	-5 and +5	3.5/5.1	50	20.3	40	50	0.88-0.92	1.83-1.87	0.585
	≥ +8	0 and +10	4.3/6.1	50	20.3	40	50	1.08-1.12	2.03-2.07	0.613
	≥ +3	0 and +10	4.3/6.1	40	15	30	40	1.12-1.17	2.07-2.12	0.568

[1] Generator: η = 88% (95% use of HHV)
[2] Generator: η = 92% (combustion air preheating, 95% use of HHV)

Table 1. Monovalent performance characteristics of AHP A and B in direct-fired air conditioning and heating applications using outside air as the heat source

A as a monovalent heat source to replace a high temperature boiler, the costs usually associated with installing a heat pump, such as costs due to modifications of the existing system, are kept to a minimum.

The high temperature AHP-A uses heat from the air in the range 25 to -25°C while the low temperature AHP-B can use air as a heat source only above 0°C but at a higher efficiency. The heat supply conditions used for AHP-A are based on typical supply and return water temperatures for high temperature hydronic heating systems. Required heating capacity for a typical residence in Germany is calculated based on actual outside air temperature data (DIN 4701). For AHP-B the heating water supply temperatures were selected for use in a low temperature hydronic heating system. AHP-B, though, would be most applicable to air conditioning systems using air cooled to about 0°C.

Table 2 compares the performance possible with the proposed AHP systems when used in high and low temperature hydronic systems. Either groundwater or air are used as year-round heat sources. The corresponding supply and return water temperatures are also given as well as the SPF. The SPF's given for the bivalent systems take into account the efficiency of the boiler.

With a standardized AHP of a certain capacity, larger heating loads can be met by combining the system with a boiler. The effect of the boiler on the system's SPF in such cases was investigated in detail. The results for a system consisting of an AHP and boiler of equal capacity are shown in Table 2. These results are used in the economic analysis which follows. They also indicate that standardization, which is a necessary prerequisite for mass production of these units, is possible.

Both AHP-A and -B can operate at part loads from 25% to 100% while still delivering the required hot water temperature to the heating system. This means that the technical barriers to using heat pump units having a higher capacity than that actually required by the heating system are eliminated. This decoupling of capacity and temperature regulation is a valuable characteristic of the proposed AHP systems. Most important, though, is the fact that four standard capacities of AHP can cover the 10 kW to 2.5 MW heating capacity markets.

System Type (capacity distribution)	AHP-A 20 kW ⟶ AHP-A+Boiler (20 + 20) kW		AHP-B 20 kW ⟶ AHP-B+Boiler (20 + 20) kW	AHP(A+B) 20 kW ⟶ AHP(A+B)+Boiler (20 + 20) kW
Heat source Temp. (°C)	Outdoor air - 20 to +25	Groundwater 8 to 12	Groundwater 8 to 12	Outdoor air - 20 to +25
Heating water temp.	High temperature		Low temperature	High temperature
Inlet: °C	35 - 50		30 - 40	35 - 50
Outlet: °C	40 - 75 (80)*		35 - 50 (55)**	40 - 75
Seasonal COP	1.46 ⟶ 1.36	1.53 ⟶ 1.40	1.83 ⟶ 1.61	1.54 ⟶ 1.43

*with groundwater heat source **by use of boiler

Table 2. Performance comparison of various AHP systems applied to heating systems

High temperature System type:	Heating capacity: 20 kW Total heat load: 32,000 kWh/yr			Heating capacity: 40 kW Total heat load: 64,000 kWh/yr		
	Boiler old	Boiler new	AHP-A	Boiler old	Boiler new	AHP-A + Boiler Load fraction: 0.87 + 0.13
Operating mode	monovalent			monovalent		bivalent-parallel
Seasonal performance factor (SPF)	0.70	0.85	1.46	0.70	0.85	1.36
Life expectancy	20 yrs			20 yrs		
Investment cost (DM) (installed system)	0	5,000	12,500	0	7,500	17,500
1st year of operation						
Energy costs (DM)						
Electricity	70	70	120	100	100	170
Fuel oil	1,828	1,505	877	3,657	3,012	1,882
Maintenance (DM)	150			150		250
Total costs (DM)	2,048*	1,725	1,147	3,907	3,262	2,302
Savings (DM)*	0	323	901	0	645	1,605

*Basis values for calculation of payback period and specific operating costs (SOCs)

Table 3a. Comparison of operating costs of high temperature heating systems

Low temperature System type:	Heating capacity: 20 kW			Total heat load: 32,000 kWh/yr		
	Boiler old	Boiler new	AHP-A	AHP-B	EHP	EHP + Boiler Load Fraction 0.79 + 0.21
Heat source	-	-	Outside air	Groundwater	Groundwater	Outside air
Seasonal performance factor (SPF)	0.70	0.90	1.50	1.83	3.64	2.70/0.9
Life expectancy	20 yrs			20 yrs	15 yrs	15/20 yrs
Investment costs (DM) (installed system)	0	5,500	12,500	18,000	16,000	16,500
1st year of operation						
Energy costs (DM)						
Electricity	70	70	120	120	1,282	1,310
Fuel oil	1,828	1,422	853	700	0	300
Maintenance (DM)	150			150	200	350
Total costs (DM)	2,048*	1,642	1,123	970	1,482	1,960
Savings (DM)*	0	406	925	1,078	566	88

*Basis values for calculation of payback period and specific operating costs (SOCs)

Table 3b. Operating cost comparison of low temperature hydronic heating systems

Economic analysis and market possibilities

Heating systems for single- and multi-family houses in Germany are a major market. Currently the replacement market for old boilers is larger than the new construction market. Heating systems for the replacement market range in capacity from 10-40 kW. In this market, only houses which have been built since the beginning of the 1970's are able to use low temperature hydronic heating systems. The majority of older houses are equipped with high temperature hydronic heating systems which are costly to replace with low temperature systems.

The AHP-A using outdoor air as the heat source takes this market situation into account. It is applicable to both high and low temperature heating systems and does not require any additional costs for system integration. The capacity range up to 20 kW can be met by a single standard AHP unit. For larger capacities a boiler can be added which can effectively double the capacity while only slightly decreasing the seasonal COP of the system (see Table 2).

Thus, for high temperature hydronic

Energy costs (1987)	Fuel oil:	0.40 DM/ltr + 4% per year
	Electricity (AHP+Boiler):	0.22 DM/kWh + 4% per year
	Electricity (EHP):	0.14 DM/kWh (constant for 15 years)
Maintenance cost increases:		3% per year
Capital costs:		7.5% per year
Tax write-offs:		ESTDV 82a and 82b (German tax law)
Effective interest rate:		(Bank loan interest rate - inflation rate) = + 2% per year

82a ESTDV:	Installation of heat pumps in buildings less than 10 years old. Valid from 1985 to 1992 (10% write-off/year over a 10-year period)
82b ESTDV:	Installation of heat pumps as a method of replacing an old boiler (100% write-off of investment in the first year or 20% write-off/year for 5 years)

Table 4. Cost increases used in calculating the payback period

Low temperature hydronic heating system	Capacity: 20 kW; Total heating load: 32,000 kWh/yr Tax write-off method based on current German tax laws				
Tax bracket	0%	20%	30%	40%	50%
Boiler (old)	0 yr				
Boiler (new)	20.4 yr	14.0 yr	11.8 yr	9.9 yr	8.9 yr
AHP-A (air source)	19.2 yr	13.0 yr	11.8 yr	9.9 yr	8.2 yr
AHP-B (water source)	> 20 yr	17.6 yr	14.6 yr	12 yr	9.9 yr
EHP (water source)	> 15 yr	> 15 yr	>15 yr	> 15 yr	14.9 yr
EHP (air source + boiler new)	> 15 yr	> 15 yr	>15 yr	> 15 yr	> 15 yr
High temperature hydronic heating system	Capacity: 20 kW; Total heating load: 32,000 kWh/yr				
Tax bracket	0%	20%	30%	40%	50%
Boiler (old)	0 yr				
Boiler (new)	> 20 yr	16.2 yr	13.5 yr	11.2 yr	9.3 yr
AHP (air source)	20.1 yr	14.4 yr	12.1 yr	10.1 yr	8.4 yr

Table 5. Payback periods of various heating systems as a function of the tax bracket and associated tax write-offs

heating systems only the boiler and the air source AHP-A are competing systems. On the other hand, a number of competitive alternatives exist for application to low temperature hydronic systems. These include the new boiler, air source AHP system A, groundwater source AHP-B, groundwater source electric heat pump, and the air source electric heat pump and boiler. The system to be replaced in all cases is assumed to be an oil-fired boiler with a reduced efficiency, based on the technology of 20 years ago.

With the above considerations, separate analyses for high and low temperature heating systems were performed which compare the AHP systems to presently available alternatives. For this comparison, the SPF, investment costs, and yearly fuel, electricity, and maintenance costs were estimated and are shown in Tables 3a and 3b for the high and low temperature hydronic heating systems. Table 4 shows the yearly cost increases used for this analysis. For the electric heat pump, the electricity price is assumed to remain at its 1987 level, whereas for the AHP and the boiler, energy costs are assumed to increase 4% per year. An increase of this type over a 20-year period results in a heating oil price of 0.85 DM/liter. More rapid increases in the fuel price will make alternative systems even more attractive.

The analysis also takes into account tax write-offs allowed under current German tax laws (paragraph 82b of the Deutsche Einkommensteuer- Durchfuehrungsverordnung). These are applicable to investments made to replace existing heating systems in private homes. Furthermore, all expenditures are considered as being financed through a bank loan.

The investment costs for each alternative are for installed, ready-to-operate systems. For the vapor compression heat pump driven by an electric motor, the investment cost used for the economic analysis is taken as 1/2 of the present installation costs of such systems in Europe. This value takes into account the effect of future development work in reducing the costs of electric heat pumps. It should be mentioned that currently the total investment cost for an electric heat pump in Europe consists of 50% for the unit itself and 50% for installation, which includes

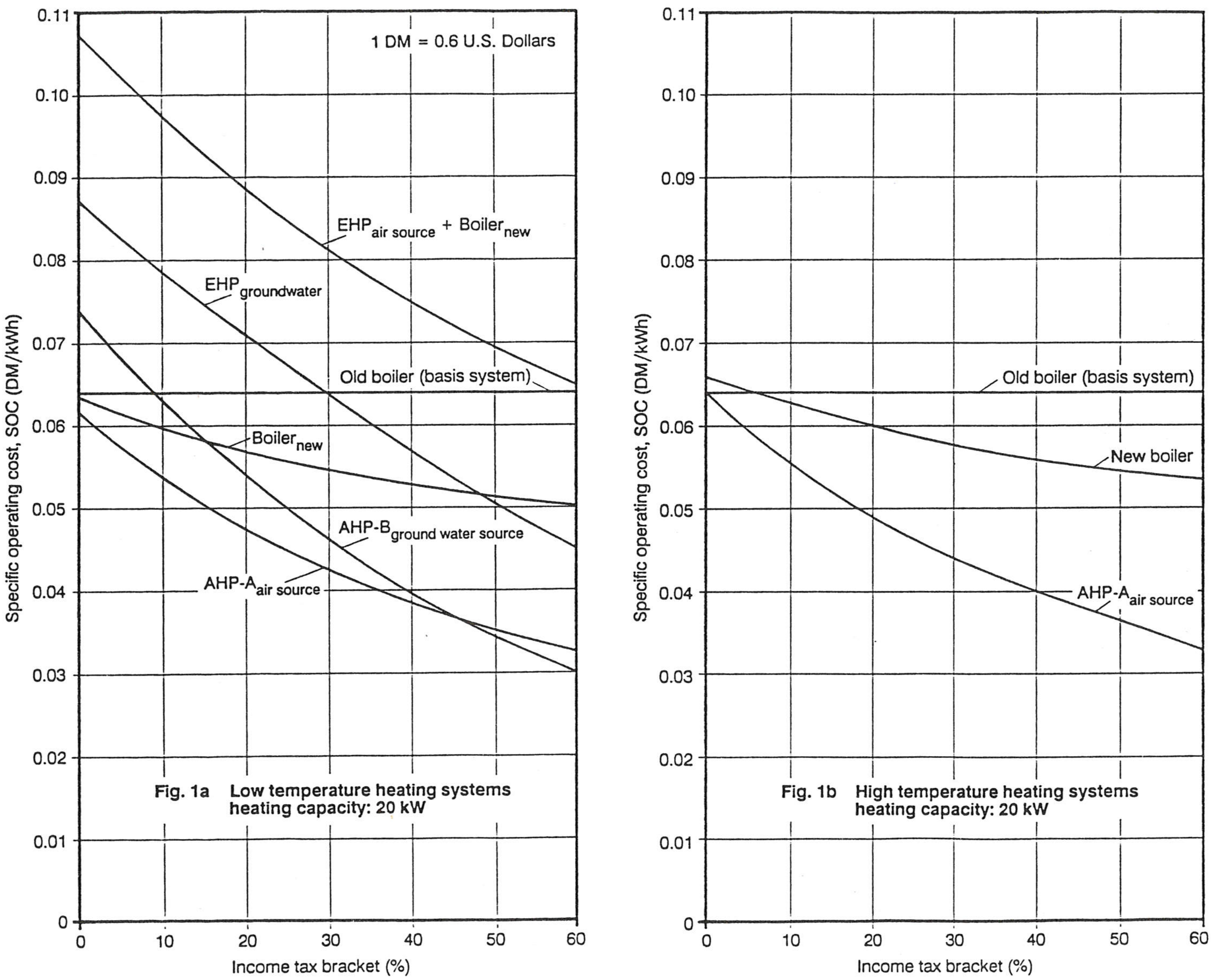


Figure 1. Influence of income tax bracket on the specific operating costs of various high- and low-temperature hydronic heating systems

heating system and control modifications. Thus, a 50% reduction in investment cost for such a replacement system is probably not achievable.

The total yearly operating cost for each of the heating systems consists of the energy and maintenance costs as well as interest rate costs. The yearly profits or losses compared to the total operating costs of the existing boiler (basis system) are used to determine the system's payback period. The specific operating cost (SOC) is calculated based on the profits or losses accumulated over the operating life of the system. This profit or loss is distributed over the total operating lifetime of the system taking into account the time value of money. For the first year's operation, this amount is subtracted from the total operating costs of the basis system, which represents the system to which the alternative system is compared, to determine the first year's operating costs for the alternative system. The SOC for the alternative system is then obtained by dividing the first year's operating costs by the heat demand.

Table 5 shows the payback periods for the low and high temperature heating systems as a function of the income tax bracket of the owner. This calculation is based on a comparison of 20 kW systems having 1600 operating hours per year and a yearly heat output of 32000 kWh (see Table 3). As shown in Table 5 for the low temperature heating system at a 0% income tax level, only AHP-A (using air as a heat source) and the new boiler achieve payback periods which are less than the operating life. Based only on the payback period criteria, both alternatives are essentially the same with the AHP-A having a slight advantage. In contrast, the AHP-B and the EHP, both utilizing groundwater as the heat source, require the owner to be in the 9% and 30% tax brackets, respectively, before these systems have payback periods which are shorter than their operating life. For the EHP using air as the heat source in bivalent operation, the payback period is longer than the system's operating life, even at the highest possible tax bracket (56%). For the high temperature heating systems the situation is similar. In this case, only the AHP-A has a payback period less than its operating life when no tax writeoff is used. On the other hand, the

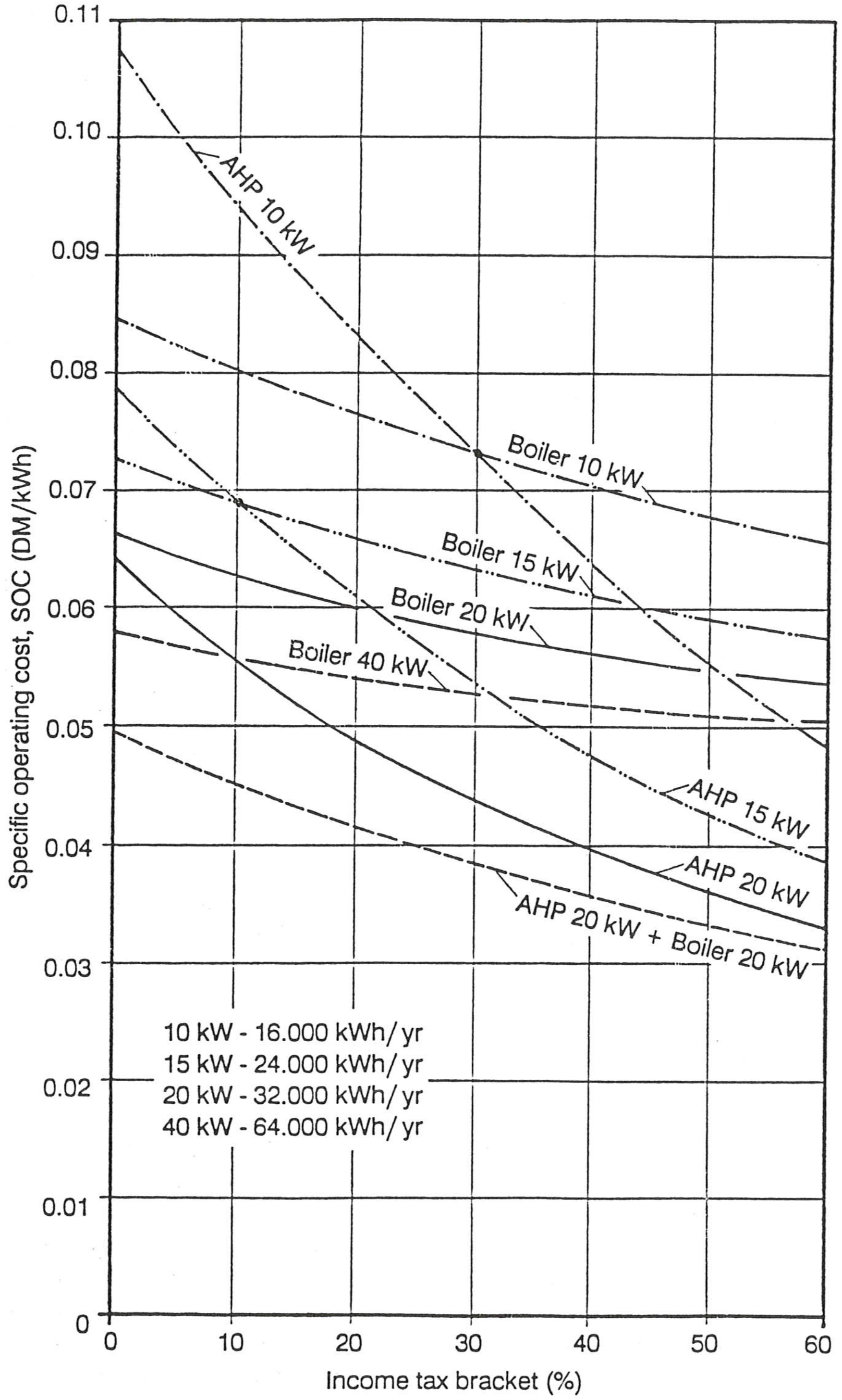


Figure 2. Effect on economics of standardizing the capacity of AHP-A air source at 20kW when applied to various capacity high-temperature hydronic heating systems in single- and multi-family houses

new boiler has a payback period less than its expected operating life only when tax breaks at income tax levels higher than 6% are accounted for. The advantage of the AHP in this case is more clear but the actual operating costs of the systems cannot be evaluated by only considering the payback period.

By considering only the payback period of the system, incorrect conclusions concerning the operating costs of a particular alternative are often made. In some cases, systems having similar payback periods will have SOC's which vary considerably for a tax-paying owner. This situation can be seen clearly in Figures 1a and 1b. In Figure 1a, the specific energy costs for the various low temperature hydronic heating systems compared to the basis system are plotted as a function of the owner's income tax bracket. Figure 1b shows the specific operating costs for the high temperature heating system alternatives. In both figures, the specific operating costs of the basis system (existing old boiler) are constant and independent of the owner's tax bracket. For the high temperature heating systems, the specific operating costs are slightly higher than for the low temperature heating systems. This is due to the lower COP's of these systems.

The SOC's for each of the alternative heating systems decrease as the income tax bracket of the owner increases. Even though the alternative systems save fossil fuel resources and have reduced emissions, they only become economical at the point at which they intersect the horizontal line representing the basis system (the old boiler). Only when the SOC is below the horizontal line is the alternative system advantageous for the owner. The intersection points with the basis system also represent the tax bracket in which an owner must be so that the particular system has a payback period equal to its operating life.

An unexpected result of these calculations is that the SOC's of the low temperature EHP are much higher than those of the AHP and the low temperature boiler, even with an assumed investment cost 50% less than present day values and a constant electricity price. For example, the specific operating costs of the bivalent EHP and boiler do not go below those of the basis system, even at the highest possible income tax bracket. Similarly, the groundwater source EHP becomes as economical as the basis system at a 30% tax bracket and at 48% compared to the new low temperature boiler. In Germany (1988) the 48% tax bracket applies to a minimum yearly income of DM 155,000. This indicates that both EHP heating systems will in the future remain uncompetitive compared to the modern low temperature boiler, even with drastic reductions in capital costs (50%). If one also takes into account that these systems typically use chlorofluorocarbons (R12 and R22) which will probably be strictly controlled in the future, and that the use of groundwater as a heat source will also become more difficult due to regulating measures, one can see that the market potential for low temperature EHP's is not good. The use of EHP's in high temperature hydronic heating systems is currently not possible because of operating limitations.

In a similar manner, the SOC of the ground-source AHP-B at the 0% tax bracket is higher than that of the old boiler. Only at a tax bracket of 10% do both systems have the same SOC. At a tax bracket of 15% AHP-B becomes competitive with the new boiler and at 46% with AHP-A. This means that only for owners with an income of DM 150,000 or more is AHP-B a money-saving alternative. A further restriction is that AHP-B using groundwater is not applicable to high temperature heating systems.

Because of its limited applications and the need for a groundwater heat source, the development of the AHP-B for use in low temperature heating systems in uneconomical. In contrast, the development of AHP-B for use in air conditioning (heating and cooling) applications is appropriate especially as an efficient replacement for compression systems which use CFCs. A large market potential for this system exists in the U.S. and Japan where it could be used as an air-to-air system for both cooling and heating.

A comparison of the SOC's of the AHP-A, basis system, and new boiler gives a more encouraging result. For AHP-A in both high and low temperature applications, the SOC is always lower than that of the basis system and the new boiler. As a replacement system the AHP-A represents the more economic alternative compared to the new boiler, even though the payback periods of both systems are equal.

Based on the results shown in Figure 1, AHP-A is an economical heating system for both high and low temperature hydronic heating applications. Its largest market potential is in Europe where Germany, with its supportive tax laws, England, the Benelux countries, Austria, Switzerland, and Scandinavia would have applications for this system. Figure 2 is of interest in this case and shows the SOC of a standardized AHP-A with a 20 kW capacity applied to various high temperature heating systems. Parameters in this graph are the required heating capacity in kW and the yearly heat load in kWh/yr for the two competing systems, AHP-A and the new boiler. While the investment costs for the boiler are taken to increase with increasing capacity, they are taken as constant for the standard size AHP-A. The investment cost for the bivalent 40 kW capacity system consists of the costs for the 20 kW AHP-A and a new 20 kW boiler. The basis systems used for comparison are boilers of equivalent capacity with efficiencies ranging from 0.65 for the 10 kW unit to 0.72 for the 40 kW unit. As seen in Figure 2, only the 10 and 15 kW capacity alternative systems intersect the line for the specific operating costs of the equivalent capacity boiler. The 20 and 40 kW systems, on the other hand, have specific operating costs which are always lower than that of the same size boiler. The 10 kW alternative system becomes as economical as the boiler at an income tax bracket of 30%. This means that for owners in the lower tax bracket (< 30%) the boiler is most economical, while the AHP becomes an economical alternative for those with higher income (> DM 75,000). For the 15 kW systems, the AHP-A becomes more economical than the boiler at an 11% tax bracket or an equivalent income of the owner of DM 22,000.

For the 20 and 40 kW heating systems, AHP-A system is most economical even without the benefits of a tax write-off. An initial estimate of the potential market in Germany for the alternative systems shown in Figure 2 can be based on the German electric utility association's estimates (VDEW) for replacement of old boilers with electric heat pumps. Based on this source, approximately one million single and multiple family houses could be retrofitted with heat pumps. In addition, about 100,000 systems/year could be installed in new construction. This can be assumed as an accurate estimate for AHP-A applications using air as the heat source, since, as was shown previously, the EHP is not a competitive system for high temperature hydronic heating applications. In the case of new construction, AHP-A represents a more economic alternative than the modern low temperature boiler for approximately 50% of the new houses built in Germany.

Taking into account the various tax systems, energy price levels, and interest rates, the market for AHP-A for the remainder of Europe is approximately equal to the market in Germany. This means that the total replacement market for old boilers is approximately 2 million units with a yearly market for new construction of about 100,000 units. This estimate neglects applications of the AHP in heating and cooling systems. This would represent an additional market for this system, especially when CFC use becomes more restricted.

Conclusions

The proposed AHP must be mass produced using large volume manufacturing methods similar to those used in the automobile industry so that purchasing costs are minimized. By limiting production to four standard capacities and combining these with modern boilers, the 10 kW to 2.5 MW heating capacity market can be covered. Under such conditions, mass production manufacturing methods are justified. Furthermore, only maintenance free, mature AHP systems will in the future be able to penetrate the sensitive European heating system market. For this part of the world, the modern low temperature boiler represents the system with which an alternative heating system must be able to compete.

**P. Vinz, IEA Heat Pump Center, Karlsruhe, Federal Republic of Germany. Translated from German into English by Andre Milbitz, IEA Heat Pump Center.*

H. Halozan*

Air-to-Air Heat Pumps for Residential Application in Europe

Air-to-air heat pumps for residential application are not very common in Europe; heat pumps are mostly integrated into hydronic systems. In the case of small high-temperature hydronic systems as well as dwellings with single room heating devices, new air-to-air split heat pumps become more and more interesting. Relatively inexpensive units with integrated room temperature control, simple installation, and additional functions like cooling and dehumidifying are very strong arguments for introduction in Europe. Improvements of indoor units to meet the requirements of the European customer should be possible.

Introduction

In 1985, 3.8 million heat pump units were installed worldwide, and this quantity is still increasing; the majority of these units are used for residential space heating and cooling.[1] There are significant differences between the markets in European countries and in Japan and the US, primarily in the sales figures (millions in the US and Japan vs. thousands in Europe), and even increasing markets in the US and Japan vs. a decreasing market in Europe.

In Japan and the US, dual-mode (heating and air-conditioning) air-to-air heat pumps are in use, and real heat pump markets for residential application only exist in regions with the need for both heating and cooling. The Japanese prefer single-room heat pump air conditioners, i.e., small units consisting of one outdoor unit combined with one or more indoor units; room temperature control is part of the indoor unit. In the US, one central unit is integrated into an air-duct distribution system; room temperature control is carried out by one central room thermostat.

Heat pumps in Europe used for residential application are usually hydronic heat pumps, utilizing various heat sources beside outside air. Further applications are exhaust-air heat pumps, either used for preheating fresh air or delivering the heat to a hydronic system. Air-to-air heat pumps are not very common for residential application in Europe.

Residential heat pump application in Europe

The European heat pump presently used for residential application is a heating-only device, most commonly integrated into a hydronic system.[2] Heating-only means a limited number of operating hours during the heating season only. A hydronic system means specified supply temperatures that must be maintained by the heat pump. Therefore, integration must be done very carefully to meet the requirements of both the hydronic system and the heat pump unit.

Operation modes are monovalent, bivalent-parallel, and bivalent-alternative. In the case of bivalent operation, an auxiliary heating system must be available. The sizes of hydronic heat pumps used in Europe range from 6 kW (for bivalent operation in single family houses) to 45 MW (integrated into a district heating network) heating capacity.

In the case of new buildings, the desirable solution is the monovalent heat pump integrated into a low temperature system. Heat sources can be either groundwater or the soil. But even with outside air, monovalent operation is possible, however, a solid fuel boiler as backup system or a tiled stove as auxiliary heating system is often added.[3] The advantage of monovalent operation is that many conventional components can be omitted, therefore, these systems can compete with conventional systems.

Bivalent operation means that the investment for the heat pump, including the unit itself, control and installation, must be amortized by heating cost savings. The costs for control and installation are high, one to two times the cost of the unit itself.[4] Considering the present cost ratio of electricity to oil, in most of the European countries this is hardly possible.[5] It is possible that the situation can be changed by improved heat pump units, which are not yet available on the market.[6]

One exception are the large heat pump units integrated into district heating networks in Sweden. The running time of these units is extremely high due to long heating seasons and the hot water demand during summertime.

Another interesting application is the exhaust-air heat pump used mainly in Northern Europe and in France. One concept uses air-to-air heat pumps, commonly arranged in series with a conventional air-to-air heat exchanger for heat recovery from the exhaust air of forced ventilation systems. The running time of this heat pump is limited to the heating season.

The more interesting solution is the utilization of exhaust air with an air-water heat pump connected to a hydronic system. During the heating season this unit is used in bivalent mode for heating purposes, during summertime it is used for hot water production. The result is a very long running time, and in well-insulated houses in Sweden more than 60% of the total heat consumption for heating and hot water can be covered by this heat pump.

Incentive for air-to-air heat pumps

The incentive for air-to-air heat pumps in Europe is the result of increased prosperity of the population and, therefore,

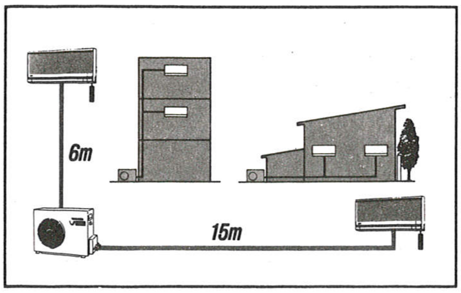


Figure 1. Installation of a split heat pump air conditioner (Toshiba Corporation)

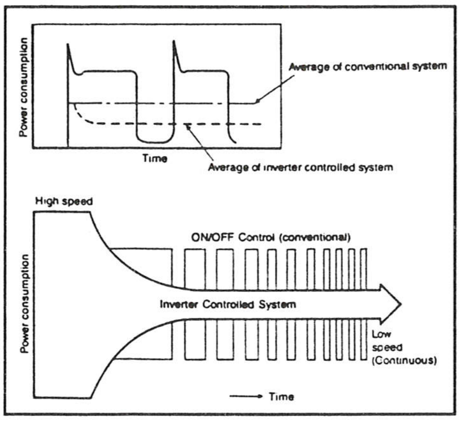


Figure 2. Comparison of power consumption of an inverter-controlled system with a conventional system (Toshiba Corp.)

the desire for more comfort. The consumer is willing to make investments to reach this goal, if the investment is not too high in relation to his income and if running costs can be reduced. And there are many potential customers for air-to-air heat pumps.

A remarkable percentage of dwellings in Europe are heated by single room heating devices like stoves fired with oil or solid fuels like coal, coke, and wood. Operation of these stoves, especially in case of solid fuels, is not very comfortable; room temperature control is relatively poor and the efficiency of these stoves is low. However, heating costs are low because single rooms are commonly heated intermittently.

Contrary to these systems, direct-electric resistance heating especially with heat transfer by means of natural convection with large heat transfer surfaces combines high comfort with excellent room temperature control. The main disadvantage of these systems is the high electricity cost in most European countries. As mentioned above, retrofit of high-temperature hydronic systems with heat pumps is very expensive and presently, considering the increased cost ratio of electricity to oil, not very economic.

These groups are the potential customers of air-to-air heat pumps even if the existing heating systems they use and their financial background are very different. With an air-to-air heat pump they get a very comfortable automatic heating device for moderate outside temperatures, which is relatively inexpensive, easily installed, and which allows cooling and dehumidifying operation during summertime. Even if cooling is necessary only for a short time it means an increase of comfort and raises the prestige.

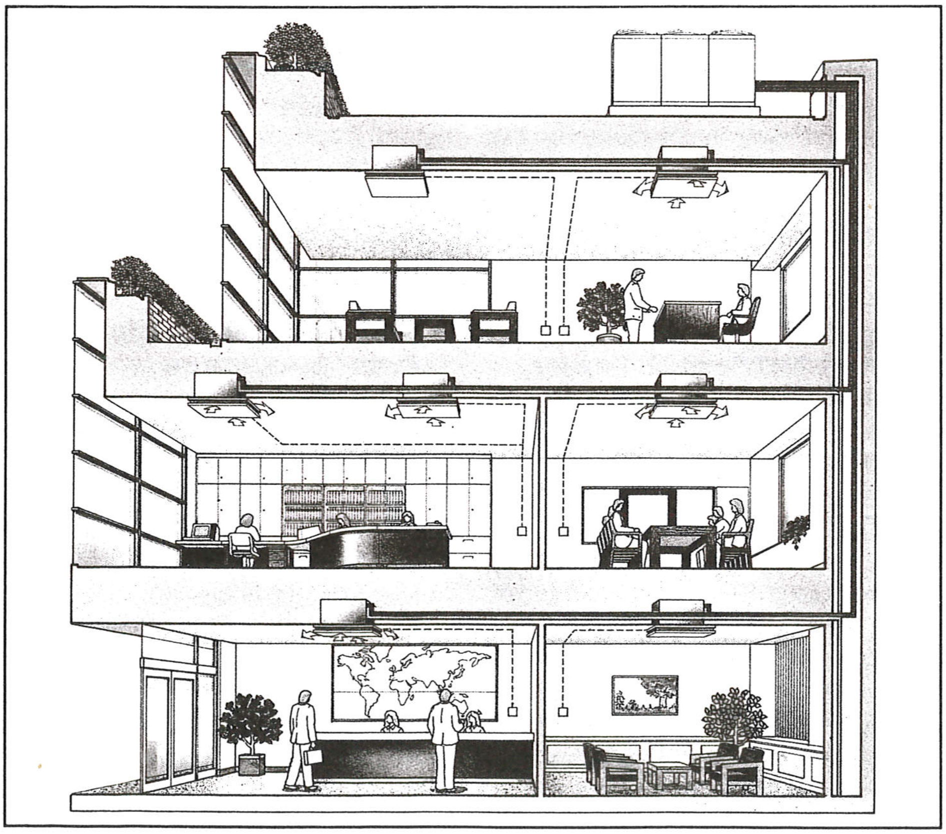


Figure 3. Large split multi-system (Daikin Industries)

Available new technologies

Investigations to utilize air-to-air heat pumps for the reduction of heating costs of electrically heated dwellings in Europe started in the 1960's, when small heat pump air conditioners available on the market were of the window type. The heating COP of these units was low, heat pump heating operation limited to +7°C outside temperature because defrosting control was not provided. The main problem, however, was the noise level of these units, which was not accepted by the customer. The only field of application was cooling for small offices and shops.

In the meantime many improvements were introduced by the heat pump industry. The window-type was replaced by the split-type heat pump air conditioner (see Figure 1). The indoor unit consists only of the condenser, the indoor fan, and the room temperature control, the outdoor unit contains the compressor, the evaporator and the outdoor fan. In cooling mode the functions of condenser and evaporator are changed. Installation is very simple, indoor and outdoor unit are connected by two refrigerant pipes, usually prefabricated and filled with refrigerant. Permitted pipe lengths are up to 6m vertical and 15m horizontal length. The indoor unit can be connected to a common plug, an additional cable is installed between indoor and outdoor unit. For the outdoor unit, a drain must be provided, also for cooling operation for the indoor unit.

Indoor units are available in different designs for various applications. The most common indoor unit is the wall-mounting type, further designs are the floor type, the ceiling type, the cassette type, and even a duct type is available.

Considering technical improvements, the reciprocating compressor was replaced by the rotary compressor and the scroll compressor. The heating SPF was improved remarkably, the operating limit decreased to about -10°C by providing a defrosting control.[7,8]

The most recent developments are inverter-driven heat pump air conditioners, which provide fast heat-up and cooling, respectively, and afterwards continuous operation without on/off cycles. The increase in comfort is combined with reduced power consumption because of reduced cycling and lower condensing temperatures (see Figure 2). Using rotary compressors up to 1 HP, frequency ranges from 30 to 180 Hz can be realized. Compressor speed control, actuation of the electronic expansion device, heat pump monitoring, defrosting and room temperature control by measuring indoor air temperature as well as the radiation from the walls is carried out by a microcomputer. Defrosting was changed from reverse-cycle operation to compressor bypass operation to avoid indoor cooling during defrosting.[9]

A further step was the development of multi-split units, i.e., one outdoor unit supplies two indoor units which can be operated independently. Using change-over valves, up to four indoor units can be supplied by one outdoor unit, however, simultaneous operation is possible with two units only. In addition to these small systems, multi-split systems for even larger buildings are available on the market (see Figure 3).

Future developments

The split heat pump air conditioners mentioned above have been developed for regions with demand for heating and cooling, i.e., a more moderate climate than in Northern and Central Europe. Therefore, the limit for heating operation is -7 to -10°C outdoor temperature.

As long as bivalent operation is the objective in introducing such units in Europe, this limit is adequate. For safety reasons, this means that in order to avoid frost damage within dwellings during absence of the inhabitants, this limit must be decreased to about -20°C. First experiments were recently conducted in the cold regions in Northern Japan. Measures for improvement are newly designed evaporators and the utilization of refrigerant mixtures to increase the heating capacity at low outdoor temperatures.

The ideal indoor unit for residential heating applications in Europe is the floor-standing unit, which results in the same room air flow characteristic as a hydronic system with radiators mounted below the window. But there are some desirable developments for the future:

-- In the case of retrofit of hydronic systems, a coil for the hydronic system should be added so that the radiator may be removed.

-- In every case, a duct for forced ventilation including heat recovery should be included.

Both measures are only completions of existing indoor units with existing components. Even single room controlled ventilation systems with heat recovery for retrofit are available on the market.

Conclusions

Hydronic heat pumps sized for monovalent operation as well as exhaust air heat pumps delivering heat to a hydronic system will keep their market. Air-to-air split heat pump air conditioners can become serious candidates for retrofitting high-temperature hydronic systems as well as dwellings with individual room heating.

Relatively inexpensive units with integrated room temperature control, simple installation, and additional functions such as cooling and dehumidifying are attractive reasons for introduction in Europe. Operation limits matching the climatic situation in Northern and Central Europe, an additional coil for connection to a hydronic system, and an integrated single room ventilation system with heat recovery are requirements which can be realized in the near future.

References

1. Calm, J.M. "Estimating International Heat Pump Use." *Proceedings of the IEA Heat Pump Conference on Prospects in Heat Pump Technology and Marketing*. Orlando, FL, 1987, pp. 553-566.

2. Gilli, P.V., R. Brandner, and H. Halozan. "Guideline on Representative Market Segments." *Proceedings of the Workshop on Electric Heat Pumps for Retrofit in Small Residential Buildings (Hydronic Systems) 1985*. HPC-ACG-R3, IEA Heat Pump Center Karlsruhe, FRG, 1987.

3. Halozan, H. "Energiesparhaus Graz: Integration of Heat Pumps into Heating Systems." *Proceedings of the IEA Heat Pump Conference on Current Situation and Future Prospects*. Graz, Austria, May 1984, pp. 149-161.

4. Halozan, H. "Background Paper A: Retrofit with Air-Water Heat Pumps." *Proceedings of the Workshop on Electric Heat Pumps for Retrofit in Small Residential Buildings (Hydronic Systems) 1985*. HPC-ACG-R6, IEA Heat Pump Center Karlsruhe, FRG, 1987.

5. Gilli, P.V., R. Brandner, and H. Halozan. "Parametric Study on the Competitiveness of Electric Heat Pumps." *Proceedings of the Workshop on Electric Heat Pumps for Retrofit in Small Residential Buildings (Hydronic Systems) 1985.* HPC-ACG-R7, IEA Heat Pump Center Karlsruhe, FRG, 1987.

6. Halozan, H. "Current and Future Advances in Small Heating-Only Heat Pump Systems." *Proceedings of the IEA Heat Pump Conference on Prospects in Heat Pump Technology and Marketing*. Orlando, FL, April 1987.

7. Matsuo, K. "How Can Heat Pumps Be Made More Cost-Competitive? Improvements of Heating Seasonal Performance Factor, HSPF." *Proceedings of the IEA Heat Pump Conference on Current Situation and Future Prospects*. Graz, Austria, May 1984, pp. 87-96.

8. Mitsubara, K., S. Kazutaka, and H. Kuno. "The Latest Compressor Technologies for Heat Pumps in Japan." *Proceedings of the IEA Heat Pump Conference on Prospects in Heat Pump Technology and Marketing*. Orlando, FL, April 1987, pp. 131-142.

9. Suma, S. "Technologies, Economics and Future Potential of Variable-Capacity Heat Pump Room Air Conditioners." *Proceedings of the IEA Heat Pump Conference on Prospects in Heat Pump Technology and Marketing*. Orlando, FL, April 1987, pp. 101-114.

**Hermann Halozan, Institute of Thermal Engineering, Graz University of Technology, Austria.*

Bibliographic Review

Listed below are excerpts from the results of a search in the ENERGY database of STN International. The search was limited to entries made in the database since the beginning of 1987. "Air-source heat pumps" was used as the search criteria. Individuals from member countries may contact the HPC to obtain the complete results of this search, or to request a search on a specific heat pump topic.

STN International, the Scientific & Technical Information Network, provides an on-line computer database service. This service is operated cooperatively by FIZ, Karlsruhe, Federal Republic of Germany; the American Chemical Society, Columbus, Ohio, USA; and the Japan Information Center of Science and Technology, Tokyo, Japan. STN offers over 30 separate databases on various subjects. The ENERGY database contains references of worldwide literature on energy research and technology for all kinds of energy sources. It contains more than 1.5 million citations from journals, reports, books, patents, and conference proceedings, and is updated biweekly. The database can be searched on subject terms, titles, authors' names, and other bibliographic information.

Hydronic heat pumps for residential and light commercial applications; volume 2, appendices: final report. Fischer, D., D. Ball, S. Talbert, J. Broehl, A. Buhr, and T. Martineau. Report number EPRI-EM-5113-Vol. 2, Electric Power Research Institute, Palo Alto, CA, USA, March 1987 (English).

The technical and economic potential of hydronic heat pump heating systems was investigated. The work included review of the status of current hydronic heat pump technology in the U.S. and Europe, development of candidate design concepts and performance parameters for air-to-water, water-to-water, and backup systems, development of control strategies to minimize costs and electrical demand, analysis of transient and seasonal performance for a wide range of applications, and preliminary investigation of the financial attractiveness and market potential of hydronic heat pump systems. The status review uncovered only a few suppliers of hydronic heat pumps in the United States. In Europe, hydronic heat pumps are technologically mature, but the market expansion is impeded by high initial and installation costs. Steady-state and transient performance of a candidate hydronic heat pump was computed using available models adapted for the hydronic application. Design and application features explored included refrigerant type, source temperature, convector sizing, compressor type, condenser and evaporator sizing, optimum charge, frosting/defrosting losses, control strategy, effect of day and night rates, thermostat setback and electrical demand, and thermal energy storage. Based on life-cycle cost savings, a number of hydronic heat pump systems appear financially attractive in selective applications. Consumer awareness and education are considered essential to market growth.

Heat pump house heating system with phase change heat storage in the place of auxiliary heating. S. Krause. *Sol. Energy* (1987) v. 39(1), p. 65-72 (English).

A house heating system consisting of an energy roof (unglazed collector), a heat pump, and medium-term phase change heat storage in the place of auxiliary heating was studied using a simple computer model. The least required storage volume is calculated and discussed as a function of the significant parameters. The choice of water for the storage medium is justified by showing the closeness of its phase change temperature to the economic optimum. The system appears technically feasible and permits minimum fossil fuel consumption together with the associated environmental benefits. Its economic position will depend on future price developments.

The energy storage in a solar house. Charlier, I.M., and P. Wauters. Proceedings of the 21st Intersociety Energy Conversion Engineering Conference, San Diego, California, USA, 25 August 1986 (English).

A good alternative solution for the problem of building heating in Belgium is the coupling of a heat pump with solar collectors. This was realized in a house near the University of Louvain-la-Neuve, Belgium. Its heating installation is composed of the following elements: $63 m^2$ of solar collectors, a $7 m^3$ water tank, and a water-air heat pump. The first step of the study consisted of measuring of temperatures, flows and electric consumptions. Next, the authors developed a simulation computer program for this system. The simulation has determined the influence of each part of the installation (collectors, storage tank and heat pump) on its efficiency. The program has shown that the capacity of the storage cylinder was a very important parameters; these results were further confirmed by measurements of the installation without heat storage during the 1985-86 winter season. The recommended improvements to the original heating system resulted in an average saving of 20% of electric consumption.

Combined solar and air source collector evaporators. Krakow, K.I., and S. Lin. ASHRAE Trans. (1986) v.92(1A), p. 474-485, ASHRAE semi-annual meeting, San Francisco, California, USA, January 1986 (English).

A model for combined solar and air-source collector evaporators has been formulated based on experimental investigations and analytical considerations. This model is suitable for incorporation in computer programs used for the determination of performance characteristics of solar and air-source heat pump systems. The performance characteristics of an experimental heat pump having solar source, air source, and combined solar-air source modes of operation are compared.

News Briefs

CFC Meeting

An informal meeting of the IEA Implementing Agreement for a Program of R&D on Advanced Heat Pump Systems was held in Karlsruhe, Fed. Republic of Germany, on February 9, 1988. The purpose of this meeting was to discuss the CFC problem. Professor Thore Berntsson from Chalmers University of Technology in Sweden organized and chaired the meeting. At an Executive Committee meeting held in Paris, it was decided that Sweden should take the lead in the initial work needed to establish international collaboration on the CFC problem. This most recent meeting was part of this effort. The specific purpose of the Karlsruhe meeting was to review some of the long- and short-term measures to deal with the CFC problem which have been proposed in different countries, to get an update of the various national activities, and, above all, to discuss proposals from Sweden on further activities. A Swedish proposal for a questionnaire on CFCs was also presented and discussed.

The measures proposed to date can be divided into three groups. The first group is applicable to existing plants, the second to new plants built within the next five years, and the third to plants which will be built in 5-10 years. Measures in the first group include improved maintenance procedures and exchange of R12 (for example with R500, R152a, hydrocarbons, or nonazeotropic mixtures using R22 as the main component). In new plants built within the next five years, emphasis will be to improve designs to use less refrigerant and allow less leakage, design plants for higher operating pressures so that R22 can be used, and design for operation with refrigerants such as R152a, hydrocarbons, or nonazeotropic mixtures. For plants built in 5 to 10 years, the possible measures include the use of new CFC and FC fluids as well as azeotropic and nonazeotropic mixtures. Other options are the use of new compression cycles and a return to ammonia as a working fluid. Since new fluids will require extensive toxicity testing and changes in the manufacturer's production line, at least five years will be required before they are introduced on the market.

A review and update of activities occurring in Norway, Germany, Austria, the Netherlands, Canada, Switzerland, U.S.A., Japan, and Sweden revealed that activities are concentrated mainly on implementing national laws to limit CFC usage. Presently most research work for CFC alternatives is being carried out by industry although the research being sponsored by governments is expected to increase. A few countries have already decided upon major research programs, notably Sweden.

In order to gather information on previous and planned work dealing with the CFC problem and to obtain opinions on future measures to deal with it, a proposal for a questionnaire was presented and discussed at the meeting. It was decided that this IEA questionnaire will be sent only to government authorities and agencies as well as research institutes. It was also mentioned that the International Institute of Refrigeration (IIR) has prepared a questionnaire dealing mainly with refrigerant use. It was pointed out by Professor Berntsson that the IIR work is aimed more at manufacturers and users of refrigerants and will not duplicate the results of the IEA questionnaire. The results of the questionnaire will be used as the basis for a seminar with invited specialists to be held in May, 1988. The results of the questionnaire and the discussion will be used to formulate a proposal for research cooperation among IEA member countries.

1988 ASHRAE Winter Meeting

More than 2,850 people attended the 1988 Winter Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) held in Dallas, Texas, January 30 to February 3, 1988. 15,200 people also visited the 1988 Southwestern Air-Conditioning, Heating, Refrigerating Exposition, held February 1-3. The meeting and the exposition were jointly sponsored by ASHRAE and the Air-Conditioning and Refrigeration Institute (ARI).

One of the topics addressed at this meeting was the CFC problem. ASHRAE President H.E. "Barney" Burroughs reported that "ASHRAE is taking steps to find solutions and increase understanding of CFCs and their potential global impact."

In a position statement on the possible environmental effects of chlorofluorocarbons approved by the ASHRAE Board of Directors on February 4, ASHRAE pledged to develop guidelines to reduce CFC losses, to encourage research and allocated funds to discovering alternatives to CFCs, and to educate and cooperate internationally. The statement also supports the following positions:

-- The CFC issue is an international issue. ASHRAE fully supports the United Nations Environment Programme (UNEP) protocol of September 1987 as the basis for controlling worldwide CFC emissions.

-- CFC-22, other CFCs which are not fully halogenated, and CFCs which are reclaimed should be exempt from regulation.

-- CFCs must remain available for continued use of existing equipment designed for fully halogenated CFCs.

-- Restrictions must take into account the time necessary for industry to implement changes in a safe and energy-efficient manner.

Burroughs announced at the meeting his intent to gather industry leaders internationally to plan a coordinated program to find solutions to current CFC refrigerant problems. ASHRAE's CFC Roundtable will be held in April.

It was also announced that ASHRAE's guideline for reducing emissions of CFC refrigerants, which was approved for development by ASHRAE's Board of Directors in October, will cover all fully halogenated chlorofluorocarbon refrigerants and will include all uses such as building equipment, appliances, and automobiles.

1988 ASHRAE Energy Awards

Winners of the 1988 international award for design of energy-efficient buildings were recognized by ASHRAE at its 1988 Winter Meeting held in Dallas, Texas, January 30 to February 3. The prestigious awards, presented by ASHRAE President H.E. "Barney" Burroughs, are given annually to ASHRAE members who have achieved substantial, measured energy cost savings through the design and installation of energy-saving features in new or existing structures.

The first-place winners in the category "Existing Commercial, Institutional, Public Assembly Buildings" were Walter I. Crutchfield, III, and Ronald L. Wash, of Cii Engineered Systems, Inc., Richmond, Virginia, for Cave Spring High School in Roanoke, Virginia.

In theory, the central heat pump plant originally represented an efficient means to meet the HVAC demands of the facility, but the aged equipment was operating at considerably lower efficiency than expected. A new forced draft gas-fired boiler was added to supply hot water to the heating loop and provide domestic hot water needs. The boiler was adequately sized to handle the building's heating needs with no additional capacity.

A new air-cooled chiller was installed to meet the chilled water requirements of the facility, with two individual refrigerant circuits to operate efficiently at part-load conditions.

A microprocessor-based facility management system now controls all major energy consuming HVAC equipment.

The second place award was presented to James Partridge, P.E., of James Partridge Associates, Inc., Birmingham, Michigan, for Lumen Christi High School in Jackson, Michigan.

The one-story school was built in 1967 with a gross building area of 155,000 square feet. It was heated and air conditioned with multiple rooftop units. A closed-loop hydronic heat pump replacement system was installed in the classroom, administration, and cafeteria areas. The gymnasium and locker room heating and ventilation systems were replaced to eliminate 90% of the outside air. Ventilation air supplied to the gymnasium was introduced to the locker rooms with transfer fans. Supplemental hot water heating coils in the transfer ductwork maintained the locker room temperatures.

Replacement of HVAC systems and the implementation of selected energy cost avoidance measures resulted in a 44% energy consumption reduction.

ASHRAE, founded in 1894, is an international organization of 50,000 persons, with headquarters in Atlanta, Georgia. Its sole objective is to advance through research, standards writing, and continuing education the arts and sciences of and relating to heating, ventilation, air conditioning, and refrigeration for the public's benefit.

Newsletter Article Clarification

The flow diagram and heat pump system described in the article entitled *Seawater Source Heat Pump Used for Live Lobster Storage*, published in Newsletter Volume 4, Number 4, December 1986, was designed by A. Orlic and Associates, Halifax, Nova Scotia, Canada. Mr. J. W. Linton, co-author of the article, was an employee of that company during the design period.

Subscription Notice

The IEA Heat Pump Center is currently reviewing its subscriber database. Please take a moment to check the address on the copy of the Newsletter that you are reading. If it is addressed to an individual who is no longer with your organization, please let us know the correct person to whom the Newsletter should be addressed. In an effort to reduce the number of Newsletters sent to incorrect addresses, we will be requesting new subscription cards from all readers in late 1988. Watch future issues for details.

Schedule of Conferences and Trade Fairs

April 17-20, 1988

Washington, D.C. (USA); **2nd DOE/ORNL Heat Pump Conference**. Sponsored by the U.S. Department of Energy and Oak Ridge National Laboratory (DOE/ORNL). Contact: Pamela J. Lewis, Oak Ridge National Laboratory, P.O. Box Y, Building 9102-1, Oak Ridge, Tennessee 37831, USA, telephone 01-615-574-2012.

June 5-10, 1988

Tiberias, on the shores of the Sea of Galilee (Israel); **The 2nd International Congress on Energy**. Sponsored by the Israel Ministry of Energy & Infrastructure. Contact: Congress Secretariat, c/o International Ltd., P.O. Box 29313, 65121 Tel Aviv, Israel, telephone 03-654541, telex 33554.

June 13-17, 1988

Joenkoeping (Sweden); **6th International Trade Fair on Minimum-Waste Technology, Waste Recycling, Energy Recovery, and Waste Collection and Management**. Sponsored by Elmia AB. Contact: Elmia AB, Box 6066, S-550 06 Joenkoeping, Sweden.

June 13-17, 1988

Saarbruecken (Fed. Rep. of Germany); **International Trade Fair of Energy Technology, Energy Savings, Environmental Protection, and Construction in Harmony With the Environment** (Internationale Messe fuer Energietechnologie Energieeinsparung, Umweltschutztechnik und Umweltfreundliches Bauen). Contact: Saarmesse GmbH, Messegelaende, D-6600 Saarbruecken, Fed. Rep. of Germany.

June 27-29, 1988

Ottawa (Canada); **1988 ASHRAE Annual Meeting**. Sponsored by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE). Contact: Judy Marshall or Judith Breese, ASHRAE International Headquarters, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA, telephone 01-404-636-8400, telex 705343.

July 18-21, 1988

West Lafayette, Indiana (USA); **1988 International Compressor Engineering Conference - at Purdue**. Sponsored by the Ray W. Herrick Laboratories, School of Mechanical Engineering, Purdue University, the International Institute of Refrigeration (IIR), and other cooperating groups. Contact: Ms. Vicki Delaney, Conference Secretary, Ray W. Herrick Laboratories, Purdue University, West Lafayette, Indiana 47907, USA, telephone 01-317-494-2132, telex 4930593 PHERLUI.

September 14-17, 1988

Wilmington, Delaware (USA); **International Symposium on Energy Options for the Year 2000: Contemporary Concepts in Technology and Policy**. Sponsored by the U.S. Department of Energy, Argonne National Laboratory, and the Government of Finland. Contact: University of Delaware, Center for Energy and Urban Policy Research, Attention: Dr. J. Byrne, Newark, Delaware 19716, USA.

September 26-29, 1988

Graz (Austria); **2nd Workshop on Research Activities on Advanced Heat Pumps**. (Part of the Austrian-Italian-Yugoslav Chemical Engineering Conference.) Sponsored by the University of Graz. Contact: Technische Universitaet Wien, Inst. fuer Verfahrenstechnik, z.Hd. Dr. H. Schnitzer, Inffeldgasse 25, A-8010, Graz, Austria.

October 3-7, 1988

Madrid (Spain); **11th International Congress of Electrical Heating**. Sponsored by the Spanish Committee of the International Union of Electrical Heating (U.I.E.). Contact: UIE Kongress 1988, c/o Comité Español de Electrotermia, Francisco Gervas, 3, E-28020 Madrid, Spain, or Viajes Iberia, San Bernardo, 20-3.º Dcha., E-28015 Madrid, Spain.

October 17-20, 1988

Versailles (France); **JIGASTOCK 1988 Thermal Exploitation of Underground Resources and Storage**. Sponsored by the European Community Commission (ECC) and the International Council for Thermal Energy Storage (ICTES). Contact: JIGASTOCK 88 Office, c/o Agence Française pour la Maîtrise de l'Energie (AFME), Madame M. Leblanc, 27, rue Louis-Vicat, 75737 Paris Cedex 15, France, telephone 33-(1)47652182.

January 28-February 1, 1989

Chicago, Illinois (USA); **1989 ASHRAE Winter Meeting**. Sponsored by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE). Contact: Judy Marshall or Judith Breese, ASHRAE International Headquarters, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA, telephone 01-404-636-8400, telex 705343.

Products, Services, & Ordering Information

Inquiries

Have a specific question about heat pumps?

Call, write, or telex the Heat Pump Center directly with your questions about heat pump technology, marketing, economics, etc. HPC staff members will do their best to answer directly or point you to the right expert.

Telephone:	49-(0)7247-82-4541
Write:	IEA Heat Pump Center, c/o FIZ Karlsruhe D-7514 Eggenstein-Leopoldshafen 2, F.R. Germany
Telex:	17724710+
Teletex:	724710=FIZKA
Telefax:	49-(0)7247-2968

Reports

The following reports are available from the HPC:

Report No.	Report Title	
HPC-WR2*	Proceedings of the Workshop on Ground-Source Heat Pumps (1987), 245 pages	DM 50,--/U.S. $30
HPC-WR3	National Reports on the Status of Heat Pumps (1987), 105 pages	DM 40,--/U.S. $25
HPC-R2-1*	Heat Pump RD&D Projects Summary Report, Edition 2 (Dec. 1986), 514 pages	DM 75,--/U.S. $45
HPC-LR-3*	HPC Bibliography - Sorption Heat Pump Systems (Oct. 1986), 372 pages	DM 40,--/U.S. $25
HPC-LR-2*	HPC Bibliography - Industrial Heat Pumps (July 1986), 378 pages	DM 40,--/U.S. $25
HPC-LR-1*	HPC Bibliography - Environmental Aspects of Heat Pump Applications (May 1986), 105 pages	DM 25,--/U.S. $15
HPC-WR-1/1-12*	Workshop: Electric Heat Pumps for Retrofit in Existing Small Residential Buildings (1985), 13 separate reports, 506 pages	DM 120,--/U.S. $70

**Sale of this report is restricted to IEA Heat Pump Center member countries only.*
Write to the HPC for a complete listing of HPC publications. All prices include postage. VAT additional in F.R. Germany.

NORTH AMERICAN ORDERS: You may order in the U.S. by sending your order form to: Mr. Fred Creswick, Oak Ridge National Laboratory, P.O. Box Y, Oak Ridge, Tennessee 37831, USA. If enclosing a check, please make it payable to **Heat Pump Center** (U.S. dollars only).

The IEA Heat Pump Center Newsletter is published quarterly.

Subscribers from countries participating in the IEA Heat Pump Center (Austria, Belgium, Canada, Fed. Rep. of Germany, Finland, Italy, Japan, the Netherlands, Norway, Sweden, and the USA) will receive the newsletter free of charge.

Subscribers from other countries will be billed DM 80,-- per year.

Subscription cancellation is possible with one month's notice to the end of the calendar year.

ORDER FORM AND SUBSCRIPTION CARD

NEWSLETTER SUBSCRIPTION:

Member Countries:
- ☐ start my subscription to the HPC Newsletter (no charge)

Non-member Countries:
- ☐ start my subscription in 1988 (DM 80/year)
- ☐ also send me all back issues through the end of 1987 (DM 80 additional)

PLEASE SEND ME THE FOLLOWING:

Report No.	Title
______	______
______	______
______	______

PAYMENT:

☐ Check enclosed. ☐ Please bill me later.

Date ______ Signature ______

Please fill in address on reverse side

03/88

Newsletter of the IEA Heat Pump Center

Future Issues:

Vol/No	Topic	Deadline for Contributions
6/2	Working Fluids	April 29, 1988
6/3	10th Anniversary of the IEA Implementing Agreements	July 8, 1988
6/4	Absorption Heat Pumps	October 7, 1988

If you would like to contribute an article on any of these topics, please send it to the Heat Pump Center.

Our regular features (bibliographic review, news updates, and schedule of conferences and trade fairs) will also be included.

IEA Heat Pump Center Newsletter (ISSN 0724-7028).
Edited by: IEA Heat Pump Center
Editors: Andre Milbitz/Lisa White
IEA Heat Pump Center
c/o Fachinformationszentrum
Energie, Physik, Mathematik GmbH
D-7514 Eggenstein-Leopoldshafen 2
Telephone: 49-(0)7247-824541
Teletex: (17)724710=FIZKA

Published quarterly. Annual subscription DM 80,-- including postage (V.A.T. not included). Free of charge for subscribers from HPC Member Countries.

Produced in West Germany.