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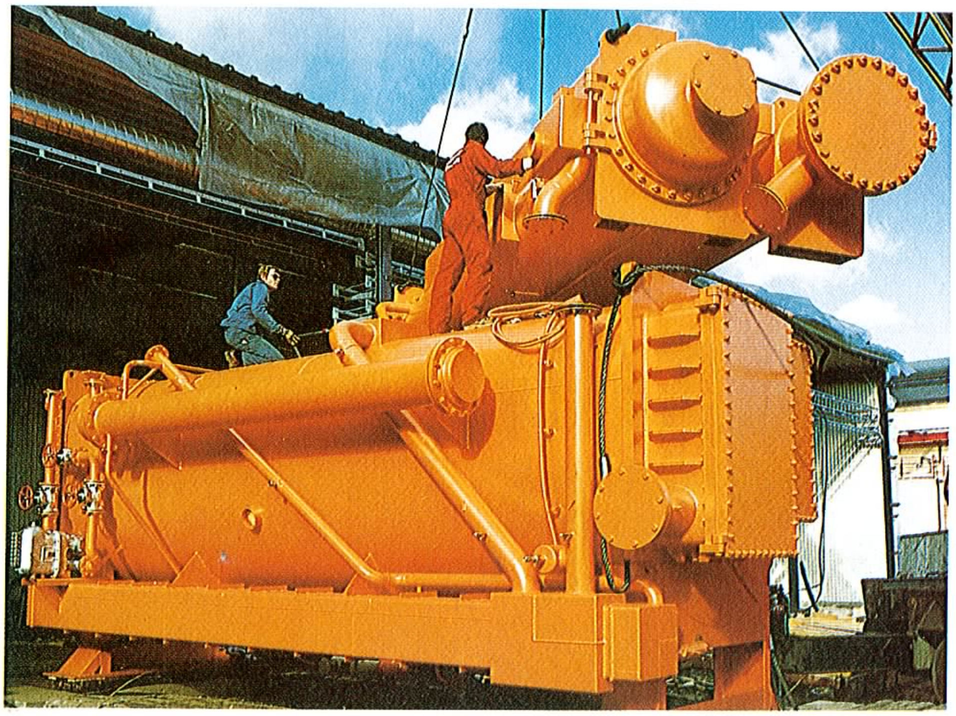

Installation of the world's largest absorption heat pump in Trollhättan, Sweden (see page 17)

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D. A. Didion*
J. Calm*

Advances in Non-Azeotropic Mixture Refrigerants: Symposia and Workshop Summary

Various benefits of refrigerant mixtures have been recognized for more than a century, yet their application remains quite limited. Using the rate of publication on the subject as a measure, interest in non-azeotropic mixtures has dramatically decreased in recent years. Within the last decade, several efforts have been undertaken to develop analytical models for research and development of systems using such refrigerants. Moreover, a number of researchers have begun to measure and develop techniques to predict essential thermodynamic and thermophysical data. Premature efforts to commercialize several specific mixtures have been attempted without success, leading a few people to question whether non-azeotropic mixtures are a solution looking for a problem. The majority view remains considerably more optimistic regarding the future potential for such mixtures.

The term "non-azeotropic mixture refrigerants" refers to fluids consisting of multiple components of different volatilities that, when used in refrigeration cycles, change composition as they evaporate (boil) or condense. The description "non-azeotropic" is derived from the Greek words "zeo" (to boil) and "tropos" (to change) preceded by two cancelling negation prefixes, the Latin antecedent, "a", and the English, "non". The double negative results from interest following the chronological acceptance of azeotropic mixtures (refrigerants compositions that act as single components often with properties more favorable, for specific applications, than the separate constituents). Purists sometimes revert to "zeotropic" as a synonym to "non-azeotropic". Not surprisingly, some researchers have adopted acronyms (e.g. NARBs for Non-Azeotropic Refrigerant Blends) or simpler descriptions (e.g. "blends to connote the deliberate mixing of fluids to obtain desired properties).

Whatever they are called, experts agree that non-azeotropic mixtures offer unique characteristics as refrigerants. These features have been investigated in hopes of achieving improvements in efficiency, capacity, and - by composition management - capacity modulation in heat pump and refrigeration systems. Additional benefits investigated include system optimization, when desired refrigerant properties fall between discrete choices afforded by single-component and azeotropic refrigerants, and split-temperature evaporator or condenser functions (e.g. refrigerator/freezer or space/water heating applications).

An update on research and development of non-azeotropic mixture refrigerants was included in two symposia at the June 1985 Annual Meeting of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), held in Honolulu. These sessions, comprising eight papers, were cosponsored by ASHRAE Technical Committees 3.1, Refrigerants and Brines; 7.6, Unitary Air Conditioners and Heat Pumps; and 9.4, Applied Heat Pump and Heat Recovery Systems. These papers include:

- "The ABCs of NARBs (Non-Azeotropic Refrigerants Blends)" by T. Atwood,
- "The Characteristics of Fluid Mixtures and Their Utilization in Vapor Compression Refrigeration Systems" by U.W. Schulz,
- "Two Refrigerant Mixtures and the Hard Sphere Fluid" by G. Morrison and M. McLinden,
- "Prediction of Refrigerant Ternary Mixture Properties Using the Redlich-Kwong-Soave Equation of State" by E.G. Wright,
- "Heat Transfer of Non-Azeotropic Mixtures in a Falling Film Evaporator" by T. Berntsson, K.M. Berntsson, and H. Panholzer,
- "Condensing Coefficients when Using Mixtures" by W.F. Stoecker and E. Kornota,
- "Simulation of a Heat Pump Operating with a Non-Azeotropic Mixture" by P.A. Domanski and D.A. Didion, and
- "Theoretical and Experimental Investigations of Advantageous Refrigerant Mixture Applications, by H. Kruse, M. Küver, U. Quast, M. Schröder, and B. Upmeier.

A workshop on research and development of non-azeotropic mixture refrigerants for heat pumps, conducted after the ASHRAE meeting, was organized and sponsored by the Electric Power Research Institute with the cooperation of the Hawaiian Electric Company and the National Bureau of Standards. The intent ot this workshop was to:

- Assess the status and potential of non-azeotropic mixture refrigerants,
- Identify research and development needs associated with such refrigerants and the equipment in which they would be used, and
- Encourage international cooperation and coordination in the conduct of such research.

Participation in the workshop was by invitations extended to the authors of the symposia papers, internationally recognized researchers in non-azeotropic mixtures, representatives of both refrigerant producers and heat pump manufacturers, utility representatives, and others selected from research organizations. The workshop was held on June 27, 1985, at the Hawaiian Electric Company in Honolulu.

The following specific research and development topics were proposed for the near term:

- Completion and validation of current efforts to develop equations of state for binary and ternary mixtures.
- Integration of these equations of state into simulation models to enable development studies.
- Theoretical cycle analyses, including hybrid absorption-compression cycles, to identify opportunities.
- Screening of non-azeotropic mixture candidate fluids suitable for vapor-compression cycles.
- Collection of property data as needed for the above; sensitivity analyses based on these data to determine precision requirements for future data collection.
- Laboratory tests of new concepts and fluids as they emerge (considerable attention has already focused on R22/R114, R12/R114 and R13B1/152a mixtures; R23/R12, R23/R22 are possible candidates warranting evaluation for increased capacity for low evaporator temperature applications).
- Impact study identifying likely applications and quantifying the increase in performance as well as the return to the manufacturer and consumer.

Several conclusions were derived from the workshop discussions:

- Non-azeotropic mixture refrigerants are not ready for widespread commercialization. Demonstration of their practical utility will require at least several more years.
- Research, both basic and applied, is needed to identify and establish the viability of specific non-azeotropic mixtures and associated cycle modifications. This research must develop the necessary data, design tools, and thermodynamic impact of selected cycles for specific applications.
- Because of opportunities to exploit the advantages of gliding temperature evaporation and/or condensation (e.g., a Lorenz rather than a Carnot cycle), non-azeotropic mixture refrigerants may initially be found more attractive in heat pumps and refrigeration systems requiring high temperature glide. Such systems may include heat pump water heaters, community heat pump systems, industrial heat pumps, and multistage refrigeration systems.
- Although nearly 100 years have passed since the use of refrigerant mixtures was first proposed, the full potential of non-azeotropic mixtures in refrigeration systems is still relatively unexplored. Opportunities for identification and development of appropriate mixtures and associated cycles exist, and more intensive research and development are needed, and believed justified, to evaluate their ultimate potential.
- To date, test results with mixtures have shown only modest improvements over single component refrigerants. It is felt that this has been due to inadequate design changes to the hardware systems to fully capitalize on mixture attributes.
- Commercialization of mixtures introduces a number of complexities in all stages of industry practice - from equipmnet design to field services - presenting additional cost elements beyond those directly attributable to the refrigerants themselves.
- This workshop provided a useful forum for examination of non-azeotropic mixture refrigerant status. Moreover, this meeting afforded international leaders in

this field an opportunity to meet and establish a basis for further communication and cooperation.

The eight ASHRAE symposia papers, resulting discussion, and a summary of the workshop are available in a bound volume entitled "Advances in Non-Azeotropic Mixture Refrigerants for Heat Pumps" (TDB-54) from ASHRAE, 1791 Tullie Circle NE, Atlanta, GA 30329, USA. The cost is US $ 15 for ASHRAE members and US $ 30 for others.

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Research on Residential Air-Source Heat Pump Dynamic Losses at ORNL

Developers and manufacturers of air-source heat pumps have long understood that heat pumps in service do not perform as efficiently as heat pumps operating under steady conditions in laboratory tests because of losses due to frosting and defrosting of the outdoor coil and due to on-off cycling under part load. Yet there is still a scarcity of high quality information on the magnitude of these losses and an incomplete understanding of the design measures that can be taken to reduce these losses. For the past several years, researchers at ORNL have been conducting both laboratory and field experiments to characterize these losses and gain insights on how they can be reduced. This article summarizes some of that work, which has been led by William A. Miller and V. D. Baxter of the ORNL staff.

Frosting Experiments

Our frosting experiments were conducted in a laboratory environment with a heat pump having a single-row spine-fin heat exchanger in the outdoor unit. Tests were run at several levels of air temperature and humidity. Fig. 1 is an example of the results of these tests, showing the effect of humidity on the coefficient of performance (COP) at an air temperature slightly above freezing (4°C). At 50% and 60% relative humidities, no frosting was encountered. At higher humidity levels, the initial COP was higher because condensation on the coil raised the heat flow to the evaporator. However, once freezing began, the COP dropped rapidly as the frost accumulated and the evaporator airflow was choked. At still higher humidity, there is less delay in the onset of the frosting process.

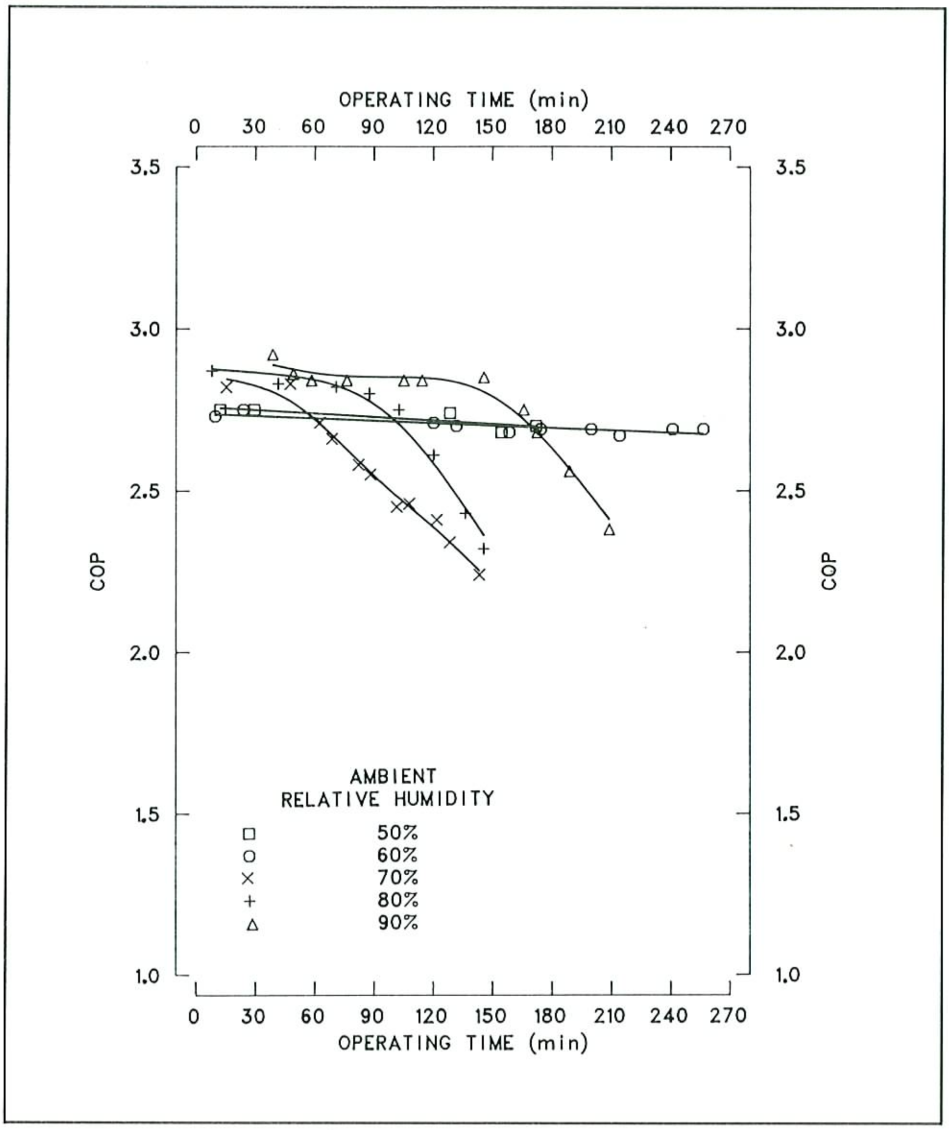

Fig. 1. Typical results of laboratory frosting tests

At lower air temperatures, there was no frosting delay, and the rate of COP degradation increased with increasing humidity levels. This study produced similar charts on capacity, evaporator wall temperature, and airflow. When the results of these experiments were used in an analytical simulation of the seasonal performance of a heat pump with a time-temperature defrost initiation control, a heating season efficiency loss due to frosting of about 5% was predicted for various climate regions of the United States.

One means of reducing frosting losses is fairly obvious: increase the outdoor coil size. This increases the coil temperature, which reduces frosting tendency. Another is increasing the evaporator airflow, which reduces the amount that the air is cooled. Coil geometry is another factor, but to our knowledge, this has not been thoroughly investigated. In our frosting tests with both spine-fin and plate-fin evaporators, we found that the spine-fin coil frosted more rapidly. However, since there were other differences in the tests besides geometry, we cannot say for sure that the spine-fin has the greater frosting tendency. Certainly plate spacing is a factor with conventional heat exchanger construction.

Cycling Loss Laboratory Tests

Our cycling loss laboratory tests were conducted in an environmental chamber facility. A unique feature of these tests was that the outdoor unit was suspended on a balance scale to measure the migration of refrigerant during startup and shutdown transient operating periods. This arrange-

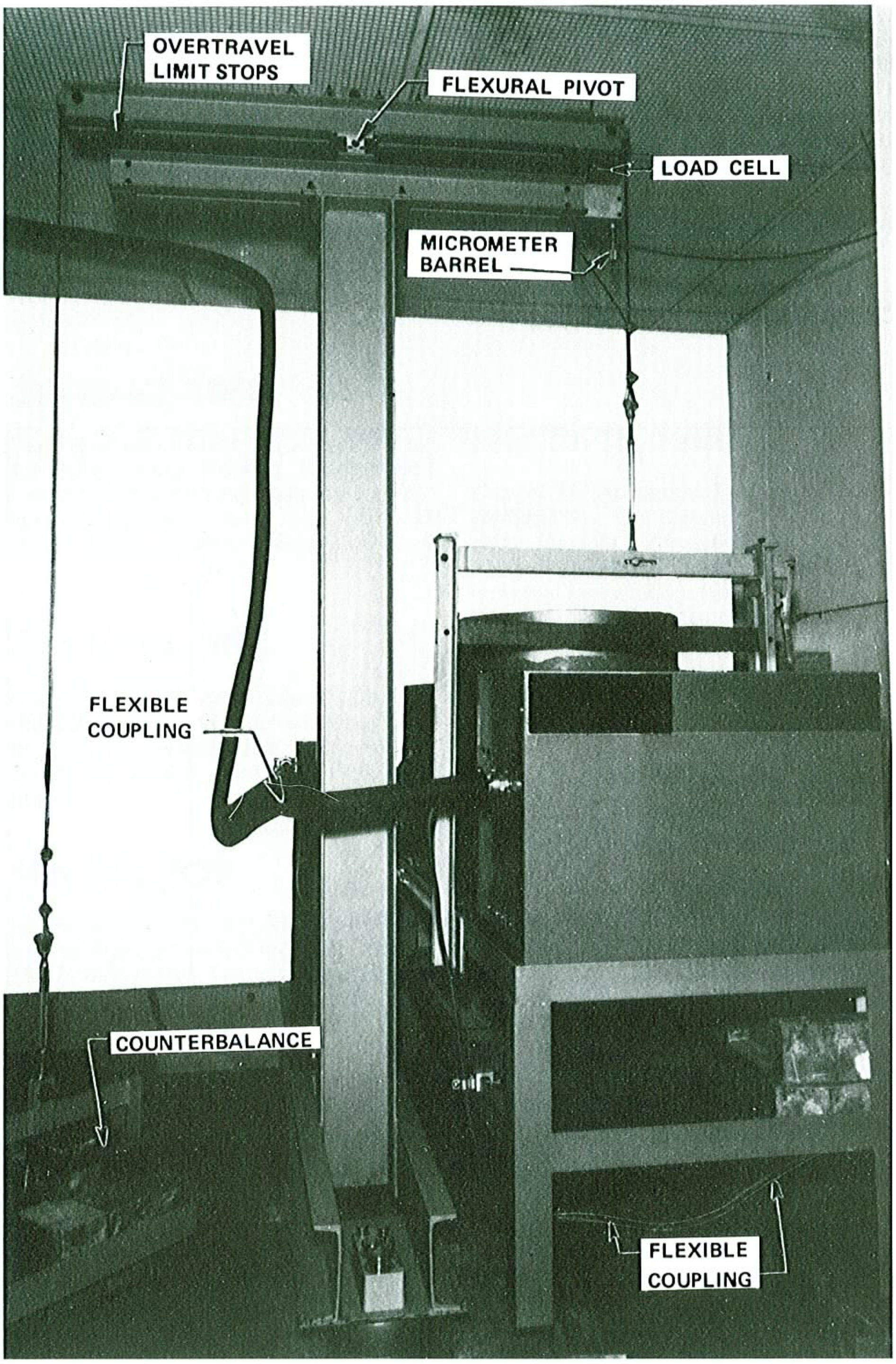

Fig. 2. Weighing system for measurement of refrigerant weight in outdoor unit

ment is shown in Fig. 2.

Fig. 3 and 4 show some typical results of this study. Fig. 3 shows the rate of pressure and temperature equalization after shutdown. In these tests, while one-third to one-half of the refrigerant was in the indoor unit during steady-state operation (depending upon the source air temperature), almost all of the refrigerant migrated to the coil and suction-line accumulator of the outdoor unit after shutdown. This migration of the refrigerant charge constitutes a major portion of cycling losses.

On startup, the liquid refrigerant charge that has migrated to the low side dumps into the suction-line accumulator, from where it is released slowly back into the active refrigerant system. Fig. 4 shows the slow rate of movement of refrigerant back to the high side and the attendant loss in capacity. In effect, the system operates undercharged during the startup transient and suffers a loss in capacity and efficiency during this period. With an 8-min-on, 30-min-off cycling rate at a 10°C source air temperature, the total observed loss in COP due to cycling was 29% of the steady-state value.

We investigated the use of valves in the systems to reduce refrigerant migration during the 'off' cycle, and found that the cycling-mode loss in COP was reduced from 29% to about 18%. Running the indoor blower a short time (about 2 min.) after compressor shutdown also reduced the cycling-mode efficiency loss by a few more points. It also seems likely that reducing the volume of the refrigerant side of the indoor coil would reduce the cycling loss, but we have not investigated this effect as yet. Similarly, it seems that some means to increase the rate at which the level of liquid refrigerant in the suction-line accumulator returns to the steady-state level would help. We have conducted some preliminary experiments with adding heat to the accumulator, which showed a substantial improvement.

Field Experiments

We have conducted field experiments for several years at a test site in Knoxville, Tennessee, USA, operated jointly by Oak Ridge National Laboratory and the University of Tennessee. Initially an annual test was run with a moderate-efficiency heat pump. This was followed by a similar test with a high-efficiency heat pump. The deviation between field performance and ideal performance was similar for both units. As shown in Fig. 5, the difference between the ideal steady-state COP and the field-measured COP increased with increasing air temperature. Since there was no way to determine the individual losses from these results, a more detailed test procedure was planned in which data were recorded separately in startup, normal, and defrost modes of operation. With the new instrumentation, the subsequent heating season produced the following result.

Loss type	Percent of heating season energy consumption
Defrosting	10.1
Frosting	3.7
Startup transient	8.4
Off-cycle parasitics	4.3

The total degradation due to these losses was 26.5%. As indicated, about half of these losses were due to frosting and defrosting and half were dynamic losses which included off-cycle parasitics (principally the compressor crankcase heater).

The test unit employed a time-temperature defrost initiation control; the use of one of the several available types of demand defrost controls would no doubt have reduced this loss. Various means are

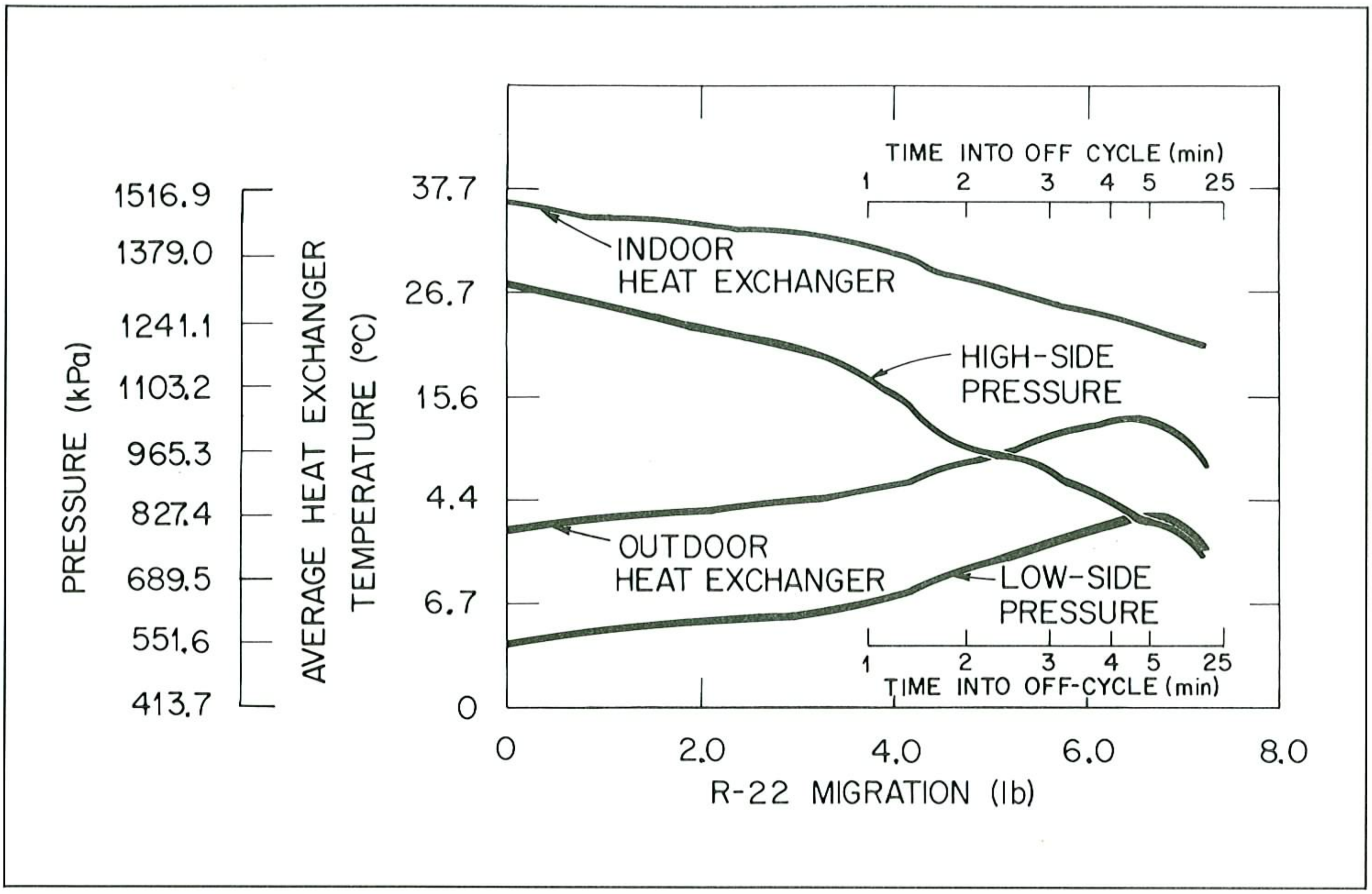

Fig. 3. 'Off'-cycle refrigerant pressures, temperatures, and migration rates during heating mode test

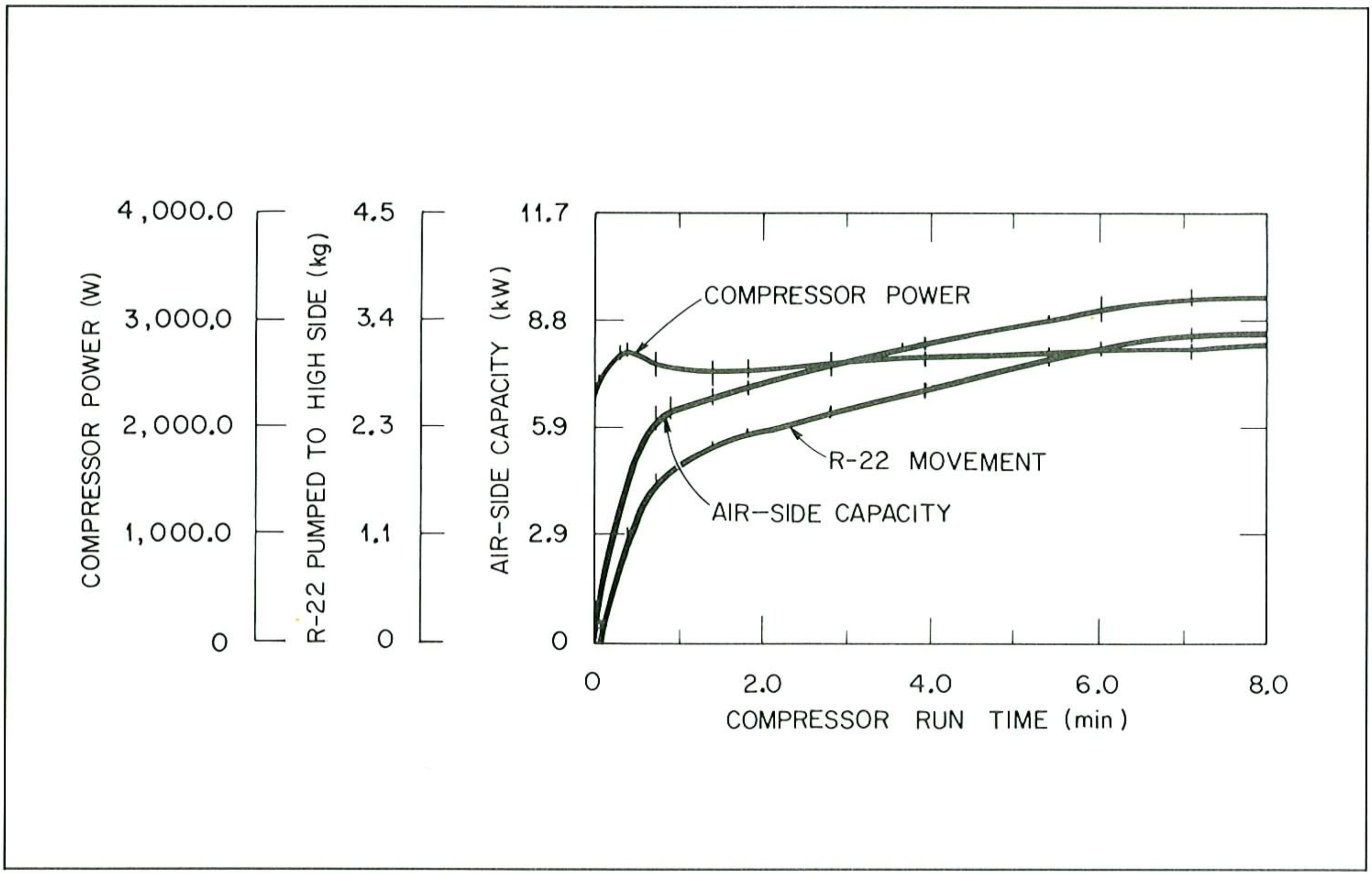

Fig. 4. System behavior during a startup transient of heating mode test

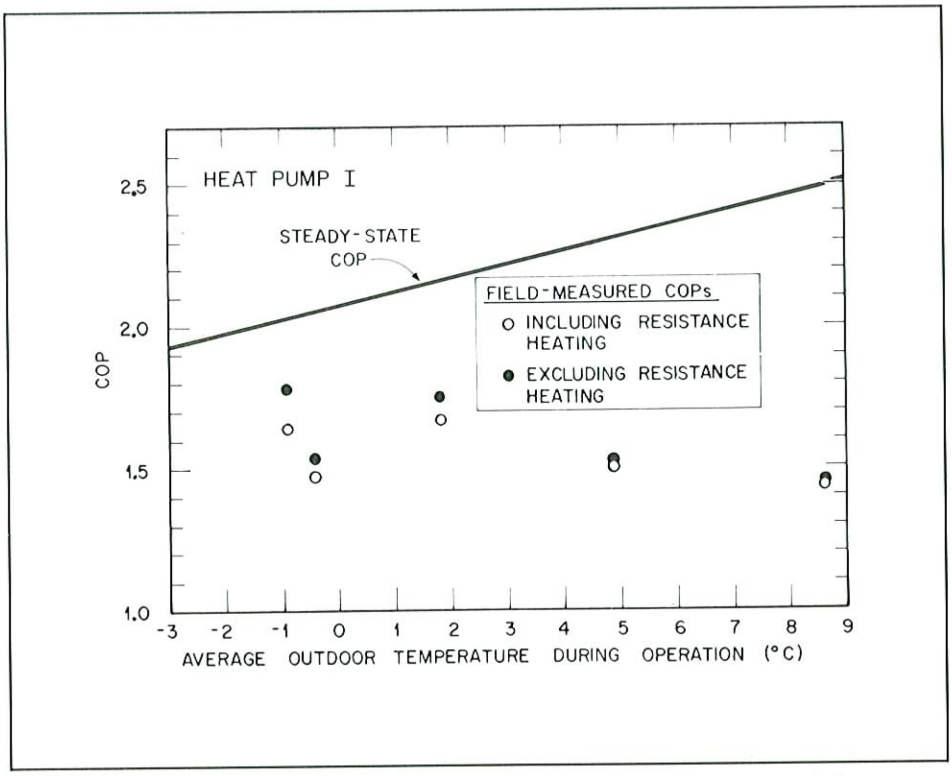

Fig. 5. Heat pump steady-state COP and field-test COPs as a function of average temperature during compressor operation

also available for reducing parasitic losses.

Our studies have shown that all of these dynamic losses can be significant; that is, they were significant in the particular systems used in our tests and presumably would be not much different in other systems of this type. While variable-capacity compressors seem to be a very promising means of reducing cycling losses and probably frosting/defrosting losses as well, our analytical studies indicate that it is important even in variable-capacity systems to include means for reducing dynamic losses.

We are now completing an annual field test of a high-efficiency two-speed heat pump at our test site. The results of this experiment are expected to be available in mid 1986.

Further information on ORNL dynamic loss experiments may be found in the following publications.

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Acknowledgement

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Sizing of Large Exhaust-Air Heat Pumps

Exhaust-air heat pumps can be said to have achieved a major breakthrough on the Swedish market. They are being installed in large numbers, both in owner-occupied homes and in multi-family dwellings, despite the fact that many installations have shown low profitability, especially when compared to other types of heat pump systems. Reasons for the popularity of exhaust-air heat pumps are relatively simple system design adapted to the needs of the market, favorable loan conditions, and, undoubtedly, the satisfying feeling of actually "recovering" heat that has already been paid for.

Reality, however, will catch up sooner or later. It is important, therefore, for exhaust-air heat pump technology to progress towards development of more cost-effective systems. One development is to increase the per-apartment capacity of the heat pump. If the temperature drop in the exhaust air stream is increased, a higher capacity per apartment will result, and the heat pump will be able to meet the major share of the total heating requirement (often two-thirds or more). Large centralized facilities where the heat pump works with a network of pipes and subcenters are also being developed to increase overall cost-effectiveness.

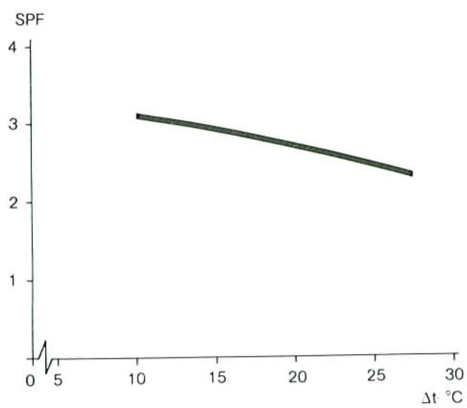

Fig. 1. Seasonal performance factor (SPF)

Temperature drop in the exhaust-air stream

The limiting factor for profitability of exhaust-air heat pumps is the availability of the heat source, the exhaust air. Exhaust-air flow is determined by building and occupant requirements, not by the heat pump. The heat extracted by the heat pump is determined, then, by the temperature drop in the exhaust-air stream. If the temperature drop is increased, so is the amount of heat extracted, and the output of the heat pump increases.

Development of exhaust-air heat pumps went from very small temperature drops in the exhaust air (5-8 K for heating tap water only), to more moderate lowerings of 10-15 K, to even larger drops of about 20 K. [1] The greater the temperature drop for which the heat pump is sized, the lower the coefficient of performance (COP). The decrease in COP is primarily due to lower evaporating temperatures as the outgoing temperature of the exhaust stream falls, but may also be due to a rise in condensing temperature. The variation in seasonal performance factor (SPF) is shown in Fig. 1.

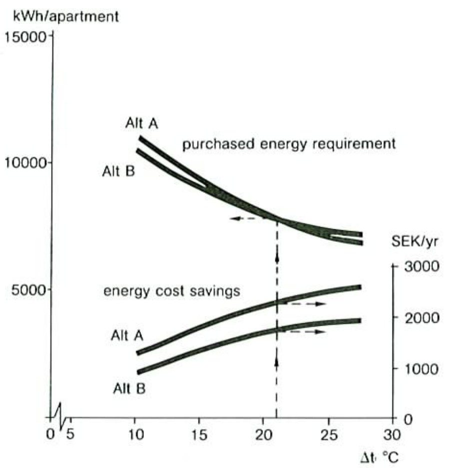

Fig. 2. Annual purchased energy requirement and energy cost savings

Fig. 2 shows the variation in the annual purchased energy requirement and the annual savings in operating costs with an increase in the maximum temperature drop in the exhaust air for a residential apartment in the Stockholm region with a bivalent exhaust-air heat pump system. Results are based on computer simulations and on the following assumptions:

- Exhaust-air flow: alternative A, 150 m^3/hr; alternative B, 100 m^3/hr;
- Annual heating requirement: alternative A, 15,000 kWh; alternative B, 13,500 kWh, of which 3,500 kWh are for domestic hot water heating;
- Temperature of the exhaust air: 20°C and a dewpoint of 5°C;
- Heating price: SEK 0.30/kWh; total electricity price: SEK 0.275/kWh.

When the temperature drop in the exhaust air is small, the purchased energy requirement is greatest for alternative A, due to a higher exhaust air flow. As the temperature drop increases, the output of the heat pump increases, and less energy from the backup heating system is required. At the same time, the difference between the energy requirement for alternatives A and B decreases, and eventually changes sign. The higher exhaust-air flow for alternative A results in more heat extraction for each additional degree of temperature drop, and therefore, a larger decrease in the purchased energy requirement. The exhaust-air heat pump becomes more and more of an outdoor heat pump the more the exhaust air is cooled. When the exhaust air is cooled below the actual outdoor air temperature, the outdoor air takes over as the most important heat source.

The economics of varying the temperature drop in the exhaust air are shown in Figs. 3 and 4. Fig. 3 shows the simple payback time and Fig. 4 shows first-year costs and savings with a capital recovery factor of 0.15 applied to the initial investment cost. The investment cost is based on a fixed cost of SEK 1,500 per apartment and a variable cost dependent on the temperature drop for which the heat pump is sized.

These figures show that for applications where the limiting factor is the flow of the heat source (in this case, exhaust air), the profitability of the heat pump system is inversely proportional to the coefficient of performance, as long as the temperature drop for which the heat pump is sized is not too large. At very large temperature drops, the degradation in the COP becomes so large that the heat pump is no longer more efficient than the backup heating system.

This result is an excellent example of the fact that the COP is only one of several factors that affect heat pump plant cost-effectiveness. At least as, if not more, important factors include: availability, tuning, monitoring of operations, heat output, loan and grant availability and conditions, cost-effectiveness criteria, and operating and maintenance costs. All these factors must be considered when making cost-effectiveness calculations.

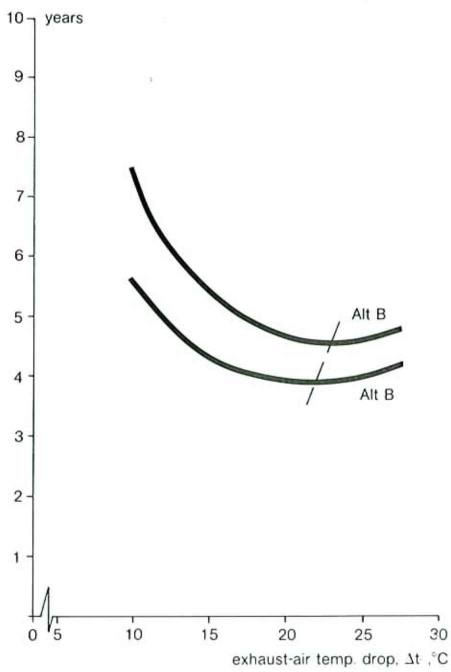

Fig. 3. Simple payback time

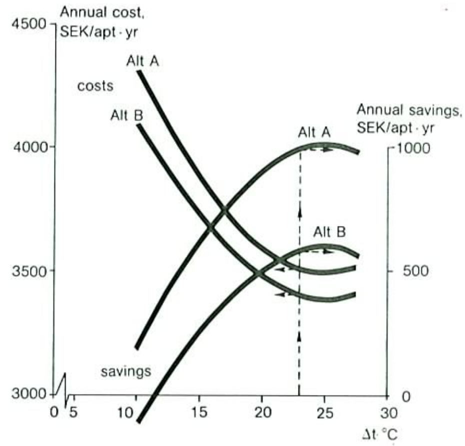

Fig. 4. First year costs and savings

Frosting and Defrosting

When the outgoing temperature of the exhaust air falls below 5°C, there is a risk of frosting on the exhaust-air heat exchanger. In some extreme cases, frosting may occur at higher outlet temperatures. [2,3] Equipment or methods for defrosting the exhaust-air heat exchanger are essential for full utilization of the heating potential of the exhaust air. Timer-controlled defrosting is the simplest, but defrosting may also be requirement-controlled to a varying extent. As a rule, defrosting should take place passively, decreasing the amount of heat extracted from the exhaust air so that the exhaust air itself thaws the frost. The following four techniques can be used:

1. The heat pump and the heat-transfer fluid (brine) pump stop.
2. The flow of heat-transfer fluid is stopped in a section or sections of the exhaust-air heat exchanger. (This technique is equivalent to (1) above if flow through all sections is stopped.)
3. Heat pump capacity is reduced so that the amount of heat extracted from the exhaust air decreases enough for defrosting.
4. Each section or group of sections can be separated from the main heat exchanger coil, and a separate pump can circulate the heat-transfer fluid through these sections. In this case, the heat-transfer fluid can be heated with, for example, electricity. This is not "passive" defrost, but the defrosting process will be faster.

Unfortunately, there is limited operating experience with exhaust-air heat pumps that require defrosting.

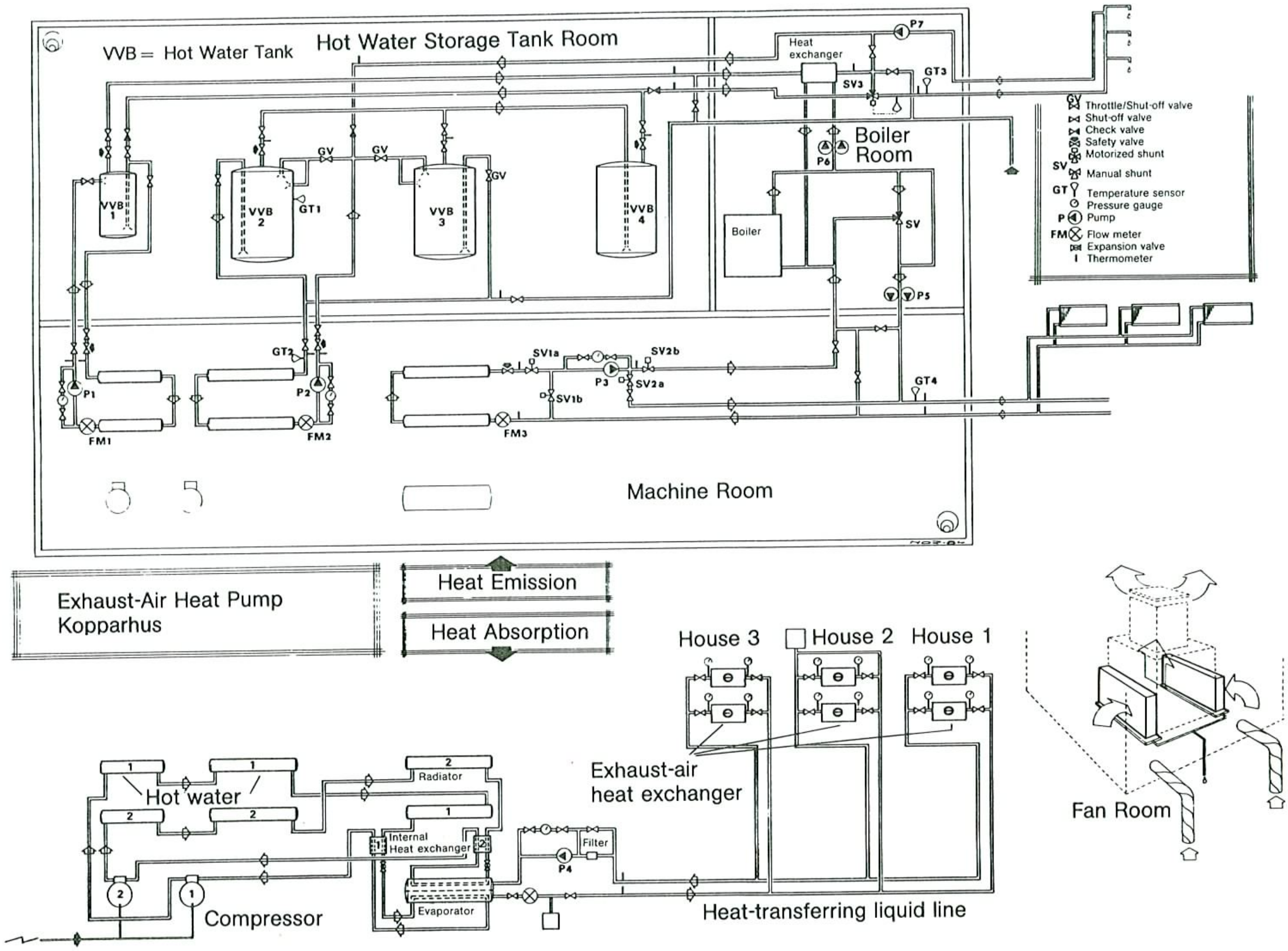

Fig. 5. System flow diagram, Kopparhusen

The Exhaust-Air Heat Pump at "Kopparhusen"

Near the center of Mörby, in Danderyd (just north of Stockholm), a heat pump supplies 150 apartments (known as "Kopparhusen," the Copper Houses) with heat and domestic hot water at outdoor temperatures above about +5°C, and has been in operation since the summer of 1984.

Exhaust-air flow rate

The plant was designed to lower the temperature of the exhaust air by a maximum of 24°C. The flow of exhaust air is slightly higher than assumed, however, so the maximum achievable temperature drop is lower. The owner proposed to reduce the flow of exhaust air to 15,000 m^3/hr, but this was not possible due to comfort requirements. Today, the flow of exhaust air is 20,000 m^3.

The flow of exhaust air has little effect on overall costs. As shown in Fig. 4, there is only a difference of SEK 100 between the annual costs for optimally-sized heat pumps for apartments with 150 m^3/hr and 100 m^3/hr exhaust-air flow rates.

Exhaust-air heat pump technology provides an excellent opportunity for raising the level of comfort for apartment occupants at a low cost, while at the same time dealing with some of the other problems with the housing stock: reducing moisture and mold, as well as radon concentrations.

Operating Experience

In general, the exhaust-air heat pump plant for the block of apartments near Mörby Center has operated as expected. There have been some operating problems, as described below.

- On three occasions, the exhaust-air heat exchangers have been clogged by frost, reducing heat pump output and causing the heat pump to stop twice. The first freeze-up occured after about two months of operation, during the period when defrosting intervals were being fine-tuned. The other two freeze-ups occurred because the exhaust-air fans failed to start after a power failure.
- Both compressors stopped on two occasions due to excessive motor temperatures. In both cases, the cause was a tiny glass fuse that was not strong enough. This was not the first time that a small, inexpensive part has caused a malfunction that could have resulted in costs that were hundreds or even thousands times the cost of the part itself.
- Leakage of refrigerant has deteriorated performance and caused one compressor to stop on two occasions.

A flow diagram of the Kopparhusen plant is shown in Fig. 5. Heat is collected in three exhaust-air fan rooms located high up in the building. The heat-transfer fluid, 35% monoethylene glycol, circulates through the exhaust-air heat exchangers, and then goes to a newly-built machine room in the garage, where all other equipment is located (apart from the storage tanks). The heat pump (STAL VMP112) is equipped with hot water condensers and hot gas heat exchangers. The hot gas heat exchangers are used for direct domestic hot water heating, so that the required DHW temperature of 50°C is easily maintained, even with refrigerant R22.

The main storage tanks (3 at 2 m^3) are heated by the hot water condensers to only 45°C. The remaining temperature rise to 50°C is achieved by mixing high-temperature water heated by the hot gas heat exchanger in a fourth storage tank. This mode of operation allows for a 5 K reduction in the condensing temperature throughout the system.

When the system is operating to heat ra-

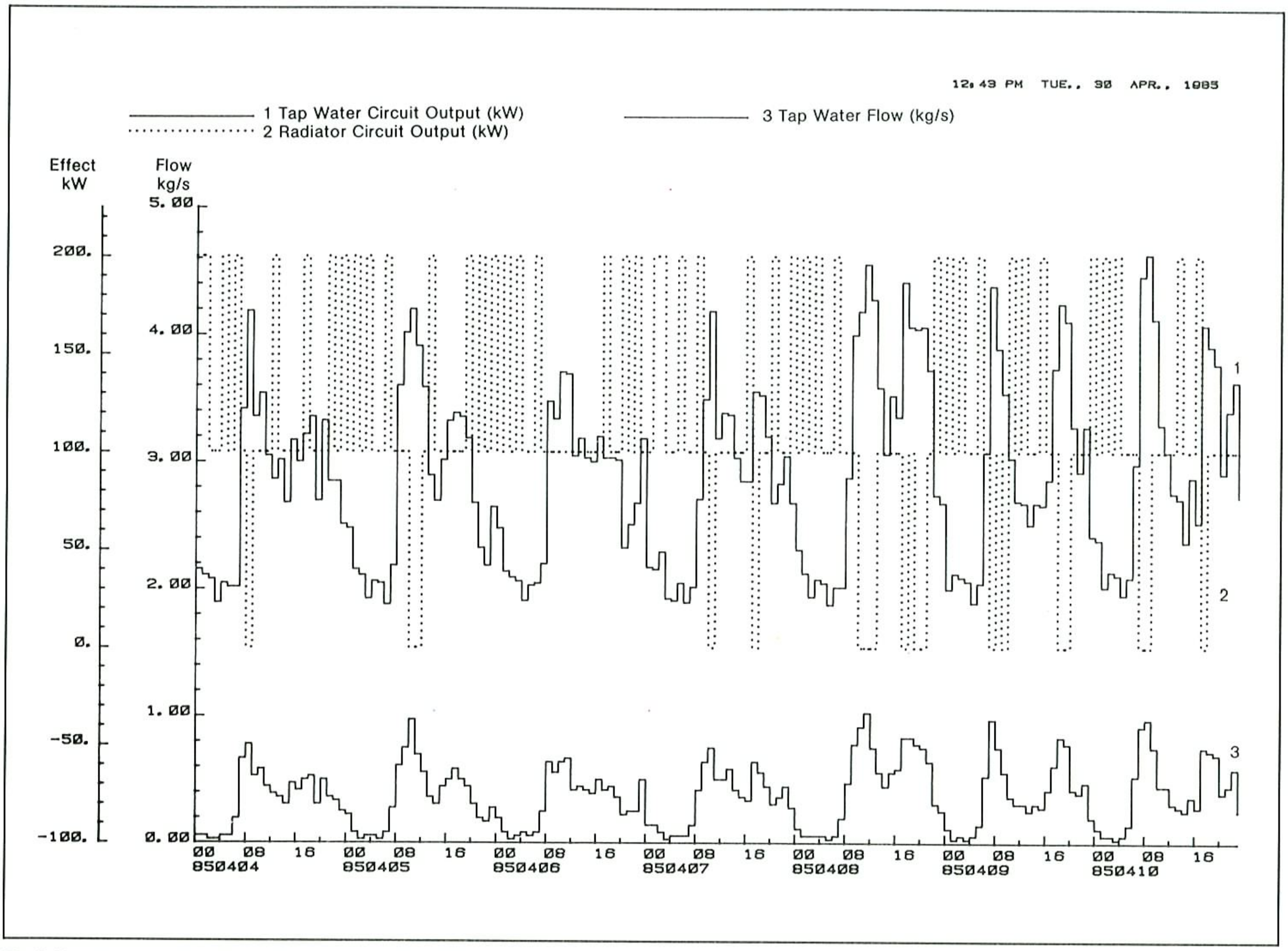

Fig. 6. Measured data from Kopparhusen, April 4-11, 1985: Domestic hot water and radiator circuit temperatures

diator water only, the heat pump is capacity controlled to maintain the required supply temperature. At outdoor temperatures below 3 to 5°C, heat pump capacity is insufficient and additional heat must be supplied by the oil-fired boilers. All system controls are done automatically.

Plant operation is monitored with the support of the Swedish Council for Building Research in a joint project with the Department of Applied Thermodynamics and Refrigeration at the Royal Institute of Technology in Stockholm, the Measurement Center for Energy Research, also at the Royal Institute of Technology, and Termoekonomi. Measurement data are collected during so-called "campaign periods" that last about one week. During these periods, many measurements are recorded on magnetic tape at 12-minute intervals.

Results from the first measurement period are presented in Figs. 6-9. The heat pump is controlled so that all heat is diverted to the domestic hot water heating system whenever necessary (Fig. 6). Unfortunately, the heat quantity meter for the radiator circuit had poor resolution when these data were collected. The figure does show, however, that the power curves for domestic hot water and radiator circuits are mirror images of each other.

Water in all four storage tanks is heated in preparation for the large amount of hot water that is likely to be required in the future. The three tanks that are heated by the hot water condensers only are maintained at a temperature of 47-48°C and the tank heated by the hot gas heat exchangers is maintained at a temperature of 85-90°C (Fig. 7). The heat pump runs at night to heat the radiator circuit. Pumps P1 and P2 (see Fig. 5) operate at night, so heat is also passed through the hot gas heat exchangers and the hot water condensers. At hot water condenser temperatures above 45°C, heat pump capacity is automatically adjusted downward. The reduction in capacity is clearly not desirable and the control strategy has since been modified.

The motorized mixing valve in the boiler room (SV3 in Fig. 5) functions well. As shown in Fig. 7, the temperature of the domestic hot water is kept at a constant 50°C. The heat pump is able to maintain this temperature quite easily; the temperature from the hot gas heat exchanger never falls below 65°C.

Fig. 8 shows data from only two days during the recording period. The 12-minute data show system dynamics with greater clarity than the one-hour data shown in Figs. 6 and 7. Fig. 8 also shows the temperature of the incoming radiator water. There is no difference between the incoming and outgoing radiator water temperatures when the system is operating only to heat domestic hot water.

Operation of the heat pump is only "interrupted" by variations in domestic hot water temperature and by defrosting at intervals of 70 minutes. Fig. 8 shows the defrost intervals, especially in the variation in the temperature of the outgoing heat-transfer fluid.

Fig. 9 shows the daily mean values for heat output during the April 4-11 measurement period (1985). Losses during defrosting are included, otherwise the outputs would have been 10% greater. Demand-controlled defrosting should reduce this figure to about 5%.

During this measurement period, the heat pump supplied a total of 39,070 kWh of thermal energy, and consumed 13,230 kWh of electrical energy, resulting in a to-

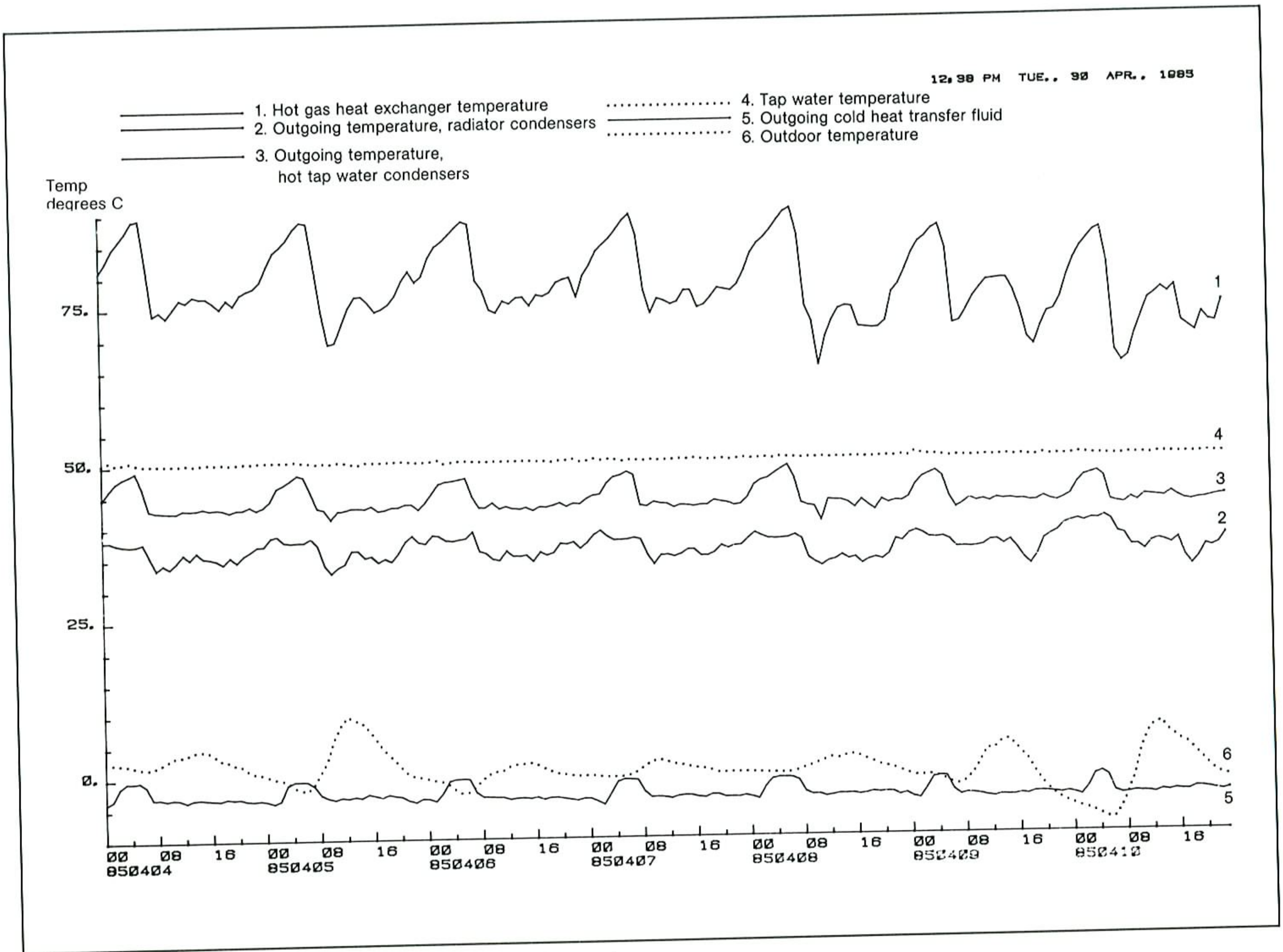

Fig. 7. Measured data from Kopparhusen, April 4-11, 1985: System temperatures

tal COP of 2.95. The COP based on compressor input only was 3.35.

Operating results so far have shown that preliminary calculations are in good agreement with actual values. At least two-thirds of the annual heat requirement is expected to be met by the heat pump. The capital cost of the plant was about SEK 1.4 million, resulting in a simple payback time of 5 years.

Malfunctions that result in a stop of plant operation are easy to identify. It is far more difficult to optimize operation to achieve the best results. A well-thought out system design and thorough monitoring and fine adjustment are essential.

Large centralized systems

Large exhaust-air heat pump systems are being developed. One or more heat pump units are centrally sited and supply a housing area through subcenters. At the surface, it may seem unnecessary to take on the extra expense of additional heat exchangers and piping for the heat-transfer fluid. Larger units and centralization, how-

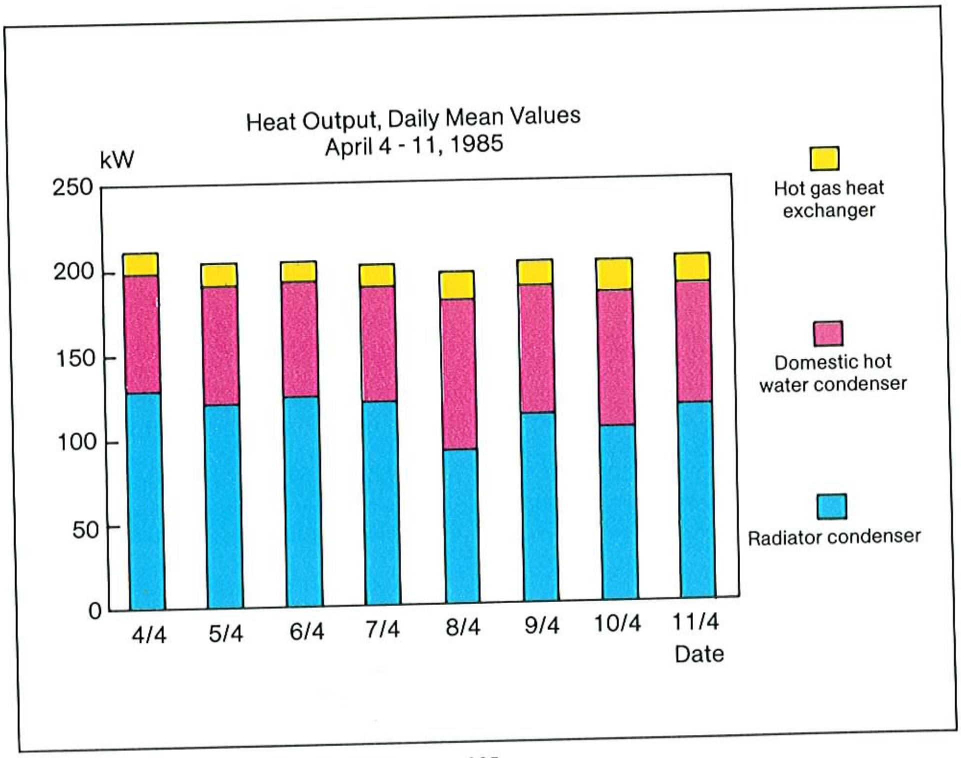

Fig. 9. Heat output, daily mean values, April 4-11, 1985

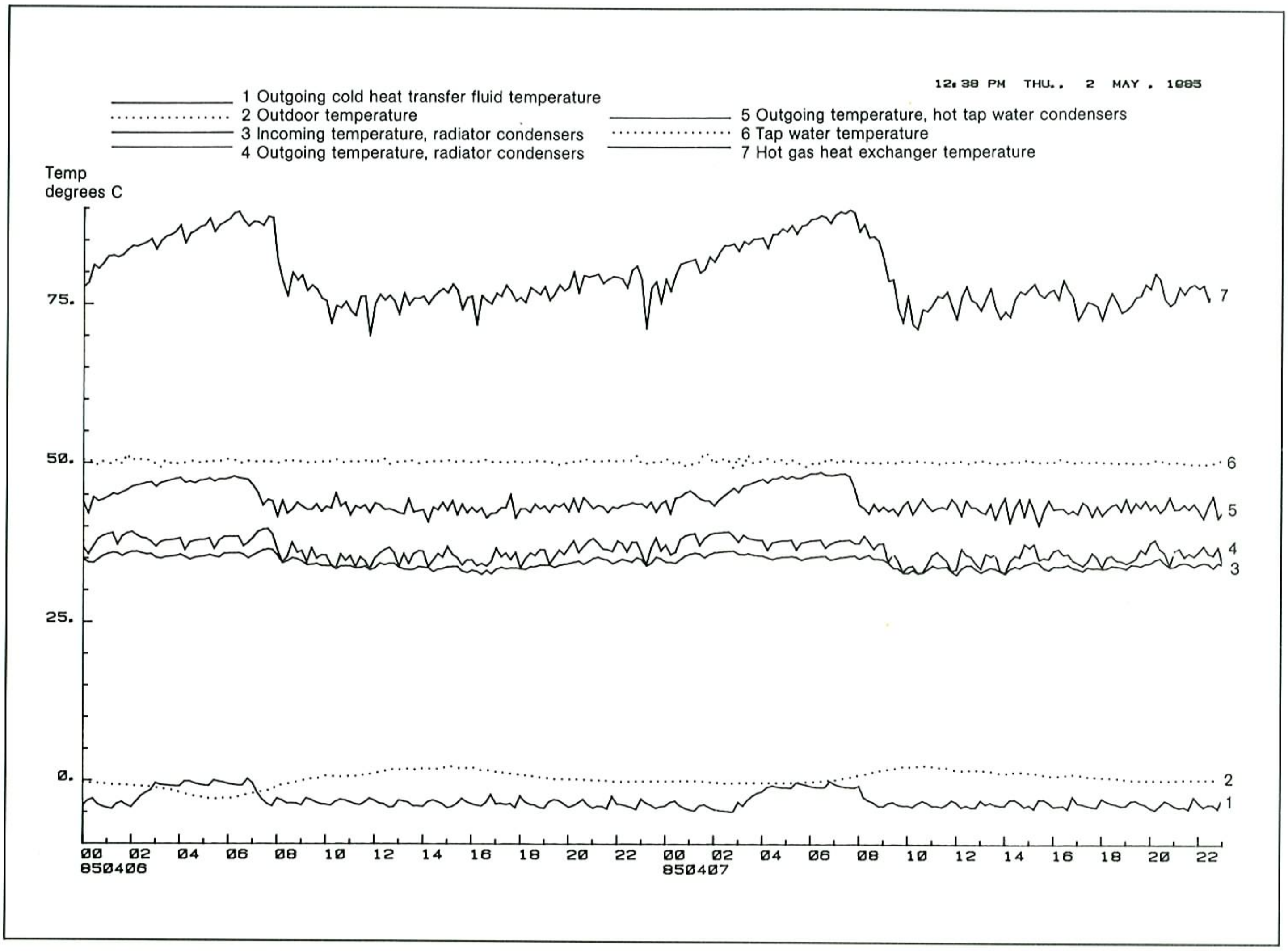

Fig. 8. Measured data from Kopparhusen, 12-minute intervals, two days during measurement period April 4-11, 1985: System temperatures

ever, mean that the overall investment cost need not be greater than the cost for a decentralized system. With the appropriate subcenter equipment, possibly combined with increased heat-transfer fluid flow rates, the supply temperature can be significantly reduced. In Fagersjö, for example, the outgoing temperature from the boiler station is about 50°C at a 0°C outdoor temperature, after installation of a large central outdoor air heat pump. [4]

To obtain the best results, however, the overall picture must be well understood. One individual or company should be responsible for the entire installation. Unfortunately, it is often difficult to gain support for the idea of extending the concept of the "system" beyond the heat pump itself.

Three large centralized exhaust-air heat pumps are under construction in the Stockholm area (Table 1). All of these plants are being equipped with equipment for automatic defrost in accordance with method (2) above. Screw compressors with refrigerant R12 are used in all plants. The investment cost is between SEK 6,000 and 9,000 per apartment. In the Jakobsberg and Täby facilities, the exhaust air is not taken from all apartments but only from about 600 and 2,000 apartments, respectively.

	(1)	(2)	(3)
Location	Jakobsberg	Täby	Skärholem
Client	Stockhomshem	HSB	Stockholmshem
Number of apartments	1,000	2,500	870
Heat output	1,300 kW	3,400 kW	1,650 kW
Auxiliary heat	District Heating	Oil/Electricity	Oil

Table 1. Three large centralized exhaust-air heat pump installations under construction in Sweden

These plants show that the technique of utilizing exhaust air as a heat source can take the step from being located in properties to being located in boiler stations. The exhaust-air heat pump is a viable alternative even in group centers. The benefits on the operating and maintenance side are clear and centralized systems should prove to be competitive with decentralized systems in many cases.

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E. Miura*

Household Energy Prices in Japan

In Japan, electric heat pumps must compete with systems fired by other fuels for heating. Once the relative efficiency of two heating systems is known, it is the relative fuel cost that determines which system is least expensive to operate. This article presents prices for kerosene, electricity, city gas and LPG in Japan, and compares costs based on heating values.

In 1981, household energy consumption in Japan was 30% kerosene, 28% electricity, 24% city gas, 17% LPG and 1% other sources. The country of Japan spreads from very cold districts in the north to warm districts in the south. Similarly, there are large regional differences in average household energy consumption and in the type of fuel used. Prices for kerosene, used mainly for heating and hot water supply, vary widely across the country. 250 companies produce city gas, and due to the wide range in company size, composition of gas produced, etc., gas rates vary considerably. In contrast, electricity prices are relatively constant across the nine power companies that produce electricity in Japan.

In this article, energy prices in the Tokyo area (the central market in Japan) are presented. Electricity and city gas prices include local taxes. Low heating values are used to calculate prices per unit of energy. At the time of printing, the Yen/Dollar exchange rate was about 200 yen/US dollar (December 1985).

1. Kerosene Prices

Kerosene is used for space heating and hot water supply. The monthly average price for kerosene in Tokyo for the period August 1983 to July 1984 was 1,634 yen for 18 liters, or 90.78 yen (U.S. $ 0.454) per liter. This figure is based on a survey conducted by the Japanese government (Statistical Bureau, Prime Minister's Office).

2. Electricity Prices

Tokyo Electric Power Company supplies electricity in the Tokyo area. Several services are available to households. Here, prices for two types of service are presented. "Residential Service B" applies to service for electric lights and appliances. "Night Only Service" (B and II) is intended for storage-type electric heaters that are designed to operate at night.

2.1 Residential Service B

Residential Service B applies to most ordinary households whose contract current is between 10 and 60 amps. Power is delivered at 100 V single phase, 50 Hz. The average household consumes 200 kWh of electricity per month. Electric rates include both a fixed charge based on contract current and a variable charge based on energy consumption (see Table 1).

2.2 Night Only Service

Night Only Service is for households with storage-type electric heaters that operate at night. Night Only Service B applies to households requiring a large volume of hot water storage, typically 360 - 380 liters with a heating power of 4.4 kW (contract power 4 kW) and an average monthly consumption of 500 kWh. Night Only Service II applies to households with a small volume of hot water storage, typically 300 liters, with a heating power of 5.4 kW (contract power 5 kW) and an average monthly consumption of 400 kWh. In general, power is delivered at 100 V single phase or 200 V three phase, 50 Hz.

Night Only Service B applies to energy consumption between 11:00 PM and 7:00 AM. Night Only Service II applies to energy consumption between 1:00 AM and 6:00

Fixed Charge	
Contract Current	Payment Rate (yen/month)
10 amp	260
15 amp	390
20 amp	520
30 amp	780
40 amp	1,040
50 amp	1,300
60 amp	1,560

Energy Charge	
Electricity Consumption (kWh/month)	Payment rate (yen/kWh)
Up to 120 kWh	20.95
Next 80 kWh	28.20
Over 200 kWh	33.25

Table 1. Residential Service B

	Fixed Charge yen/kW contract power	Energy Charge yen/kWh electricity
Night Only Service B	360	13.25
Night Only Service II	220	11.80

Table 2. Night Only Service

Fixed Charge yen/month per meter	Energy Charge yen/m^3
690	154.72

Table 3. Gas rates (based on a heating value of 11,000 kcal/m^3)

AM. Monthly electricity charges include a fixed charge based on contract power and an energy charge based on kWh consumption (see Table 2). If the monthly electricity charge exceeds 3,600 yen (US $18), a 5% tax is imposed.

3. Gas Rates

Gas is the primary fuel for hot water, space heating and cooking in Japanese households. Tokyo Gas Company supplies gas in the Tokyo area. Four kinds of gas are supplied, with heating values of 3,600, 5,000, 9,500 and 11,000 kcal/m^3. Natural gas now amounts to 70% of the total supply, so a heating value of 11,000 kcal/m^3 is used for the fuel price comparisons presented below. Gas charges include both a fixed charge and an energy charge, as shown in Table 3. When the monthly gas charge exceeds 12,000 yen, a 2% tax is imposed. In areas where 11,000 kcal/m^3 gas is delivered, households consuming less than 73 m^3 per month are tax exempt.

4. LPG Prices

LPG is used for cooking and water heating. In general, households have their own LPG tank. According to a government survey, the average price for LPG in Tokyo last year was 5,134 yen per 10 m^3, or 513.4 yen (US $ 2.57)/m^3.

5. Heating Values of Fuels

Heating values for kerosene, natural gas, LPG and electricity are given in Table 4.

6. Fuel Price Comparisons

Prices of different fuels can only be compared if they are in common units, that is, price per unit of energy (e.g. yen/MJ, $/GJ). The household fuel prices presented above are summarized in Table 5, and are converted into price per MJ in Table 6, using the heating values given in Table 4. These results show that electricity under Residential Service B is about three times as expensive as kerosene.

Fuel	High Heating Value	Low Heating Value
Kerosene	37,260 kJ/m^3	34,830 kJ/m^3
Natural Gas	46,050 kJ/m^3	41,650 kJ/m^3
LPG	101,300 kJ/m^3	93,350 kJ/m^3
Electricity	3.6 MJ/kWh	

Table 4. Heating values of fuels

	Yen	U.S. $
Kerosene	90.78/l	0.454/l
Electricity (tax incl.)		
- Residential Service B [1]	27.78/kWh	0.139/kWh
- Night Only Service B [2]	16.94/kWh	0.085/kWh
- Night Only Service II [3]	15.28/kWh	0.076/kWh
Natural Gas (11,000 kcal/m^3, tax incl.) [4]	167.20/m^3	0.836/m^3
LPG	513.40/m^3	2.570/m^3

Notes:

1. Contract current 20 amps, electricity consumption 200 kWh/month, electricity tax 5%
2. Contract power 4 kW, electricity consumption 500 kWh/month, electricity tax 5%
3. Contract power 5 kW, electricity consumption 400 kWh/month, electricity tax 5%
4. One gas meter, gas consumption 75 m^3/month, gas tax 2%

Table 5. Summary of household energy prices in Japan

	Yen/MJ	$/GJ	Ratio of Fuel Price to Kerosene
Kerosene	2.61	13.05	1.00
Natural Gas [4]	4.01	20.05	1.54
Electricity			
- Night Only Service B [3]	4.24	21.20	1.63
- Night Only Service II [2]	4.71	23.55	1.81
LPG	5.50	27.50	2.11
Electricity			
- Residential Service B [1]	7.72	38.60	2.96

See Table 4 for heating values. Notes: see Table 5.

Table 6. Relative prices of household energy, based on heating values

Table 7 shows how much more efficient an electric heat pump must be to be competitive with a heating system fired by each of the other fuels. A heat pump that operates during the day under Residential Service B, for example, will have the same operating costs as an LPG-fired heating system, if the heat pump is 1.4 times as efficient. If the heat pump operates at night under Night Only Service B, its efficiency can be 23% less than an LPG-fired system and still have the same operating costs.

*E. Miura, IEA-HPC National Team of Japan

Rate Schedule	Ratio of Electricity Price to Fuel Price		
	Kerosene	Natural Gas [4]	LPG
Residential B [1]	2.96	1.93	1.40
Night Only II [3]	1.80	1.17	0.86
Night Only B [2]	1.62	1.06	0.77
See Table 4 for heating values.		Notes: see Table 5.	

Table 7. Electricity prices relative to fuel prices

P. Hofmann*

Badenwerk Stands Behind the Electric Heat Pump for Space and Water Heating

Badenwerk's Heat Pump Promotion Program, Using Selected Heat Pump System Installers

On June 10 and 16, 1985, the electric utility Badenwerk, in cooperation with the heat pump manufacturers Siemens and Stiebel Eltron, held two informational presentations for heating system installers in Lenzkirch and Wiesloch. The primary emphasis of the Badenwerk program is promotion of a heat pump system with compact distribution system connections for retrofit of 5 to 10 year old oil-fired central heating systems in single- and two-family houses.

System Description

The Badenwerk program is unique in that a particular heat pump application is being promoted, specifically, an air-to-water heat pump operated in the bivalent-parallel mode, using the existing oil-fired boiler to cover the peak load. Depending on design, the boiler will only cover between 10% and 30% of the annual heating load, while the heat pump covers the remaining 70% to 90%. Two heat pump systems are available under the preliminary standard

I. Series integration with storage and bypass connections			
Manufacturer	Heat pump "type I"	Heat pump "type II"	
Siemens AG	LI 8 LA 11	LI 13 LA 14	
I = Indoor installation	A = Outdoor installation		
II. Parallel integration with separate storage			
Manufacturer	Heat pump "type I"	Heat Pump "type II" a)	b)
Stiebel Eltron	WPL 8 Ki WPL 8 Ka	WPL 15 Ki WPL 15 Ka	20 Ki 20 Ka
i = indoor installation	a = outdoor installation		

Table 1. Possible heat pump systems under the promotion program

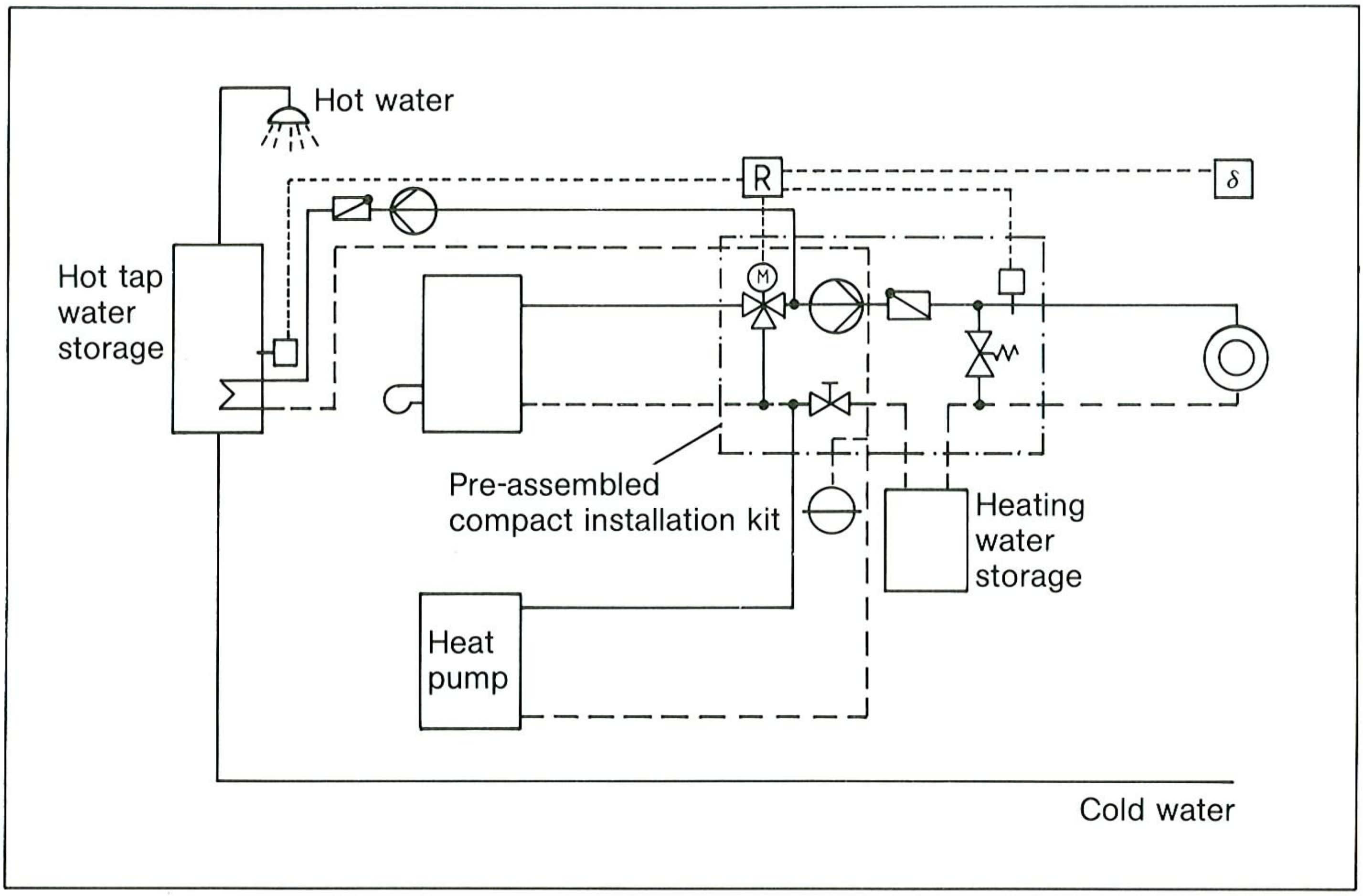

Fig. 1: Heat pump plant with series integration and bypass connections

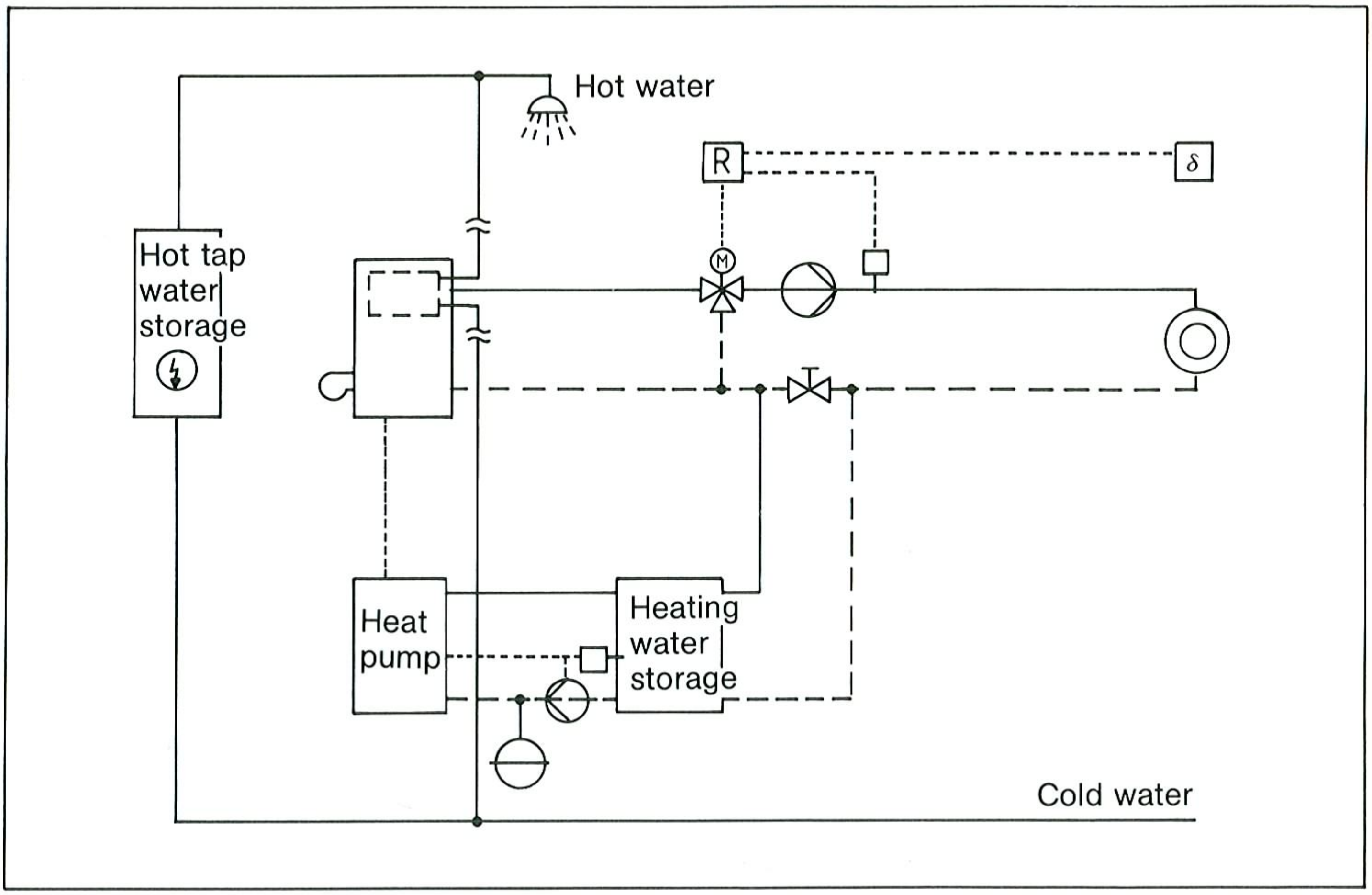

Fig. 2: Heat pump plant with parallel integration and separate storage

DIN 4759, part II: parallel integration with separate storage or a series integration with storage and bypass connections.

Both systems have been well-proven in actual practice, and prefabricated components (equipment and distribution system connections) allow for simple installation. Heat pumps of two different capacities can be selected for each system, based on either the heating demand or the desired load share between heat pump and boiler. The heat pump types for each system are listed in Table 1. The recommended heat pump types, based on heating demand, are shown in the matrix presented in Table 2. Using this design matrix, excessive oversizing is avoided, reducing first costs considerably.

Figs. 1 and 2 show integration of each heat pump in a hydronic system (in accordance with the preliminary standard DIN 4759, part II). Standardized installation, prefabricated components and extensive installation experience can keep the price of installation as low as DM 2500-3000.

Assistance to consumers

Badenwerk offers interested customers so-called "system advice," where a technical consultant from Badenwerk inspects the customer's heating system and arranges for a free estimate for installation of the desired heat pump system. Customers choose their heating system installers from a previously selected group of specially trained installers.

The heat pump plants are available to consumers at a price that includes installation and turnover tax, DM 12,600 for small systems without water heating and DM 20,575 for large systems with water heating. These prices (good until December 31, 1985) include a two-year full guarantee on system operation. Special financing arrangements with especially favorable interest rates are available from the German building and loan association Landesbausparkasse (LBS) and the savings bank of Baden.

Assistance to Installers

Badenwerk provides promotional assistance to heating system installers that participate in this promotion program, through the development of newspaper advertisements, information brochures, displays, stickers and the like. In addition, open houses are held to provide interested heating system installers with program information, and the customer service departments of Badenwerk organize "publicity weeks." Well-trained technical consultants at Badenwerk assist heating system installers with heat pump system planning and design.

The two heat pump systems will be presented to a wide range of customers at Badenwerk's exhibition stand at the regional fairs in Freiburg, Offenburg, Karlsruhe and Mannheim (Figs. 3 and 4).

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Design Heating Demand	Heat Pump Type: HP "type I"	HP "type I or II"	HP "type II"
up to 15 kW	X		
13 to 18 kW		X	
15 to 25 kW			X

Table 2. Heat pump type according to heating demand

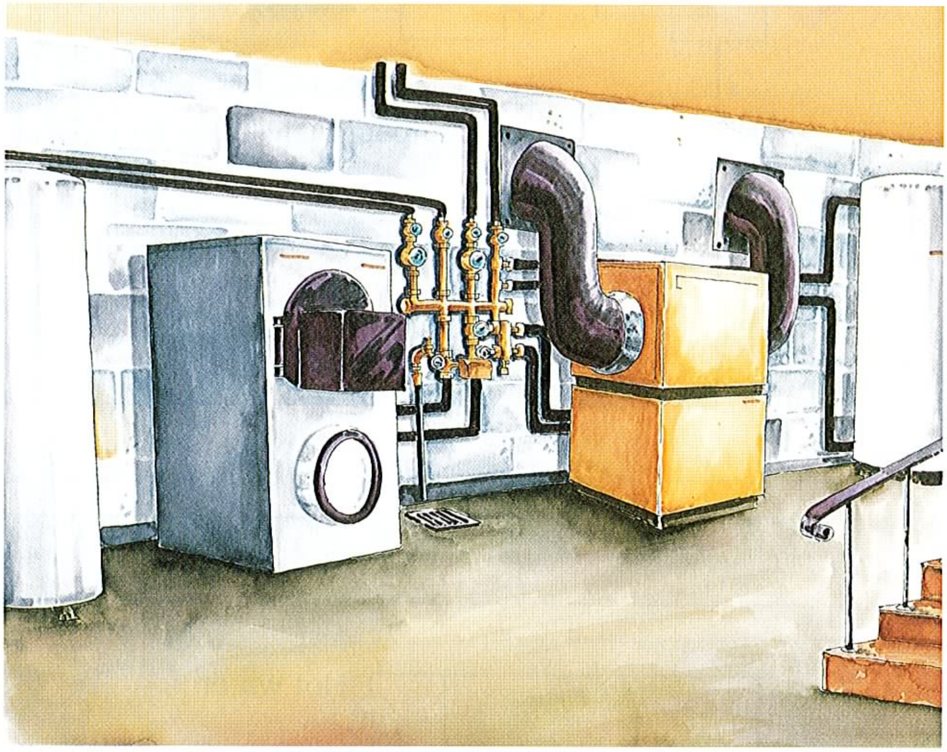

Fig. 3: The complete heat pump plant for indoor installation

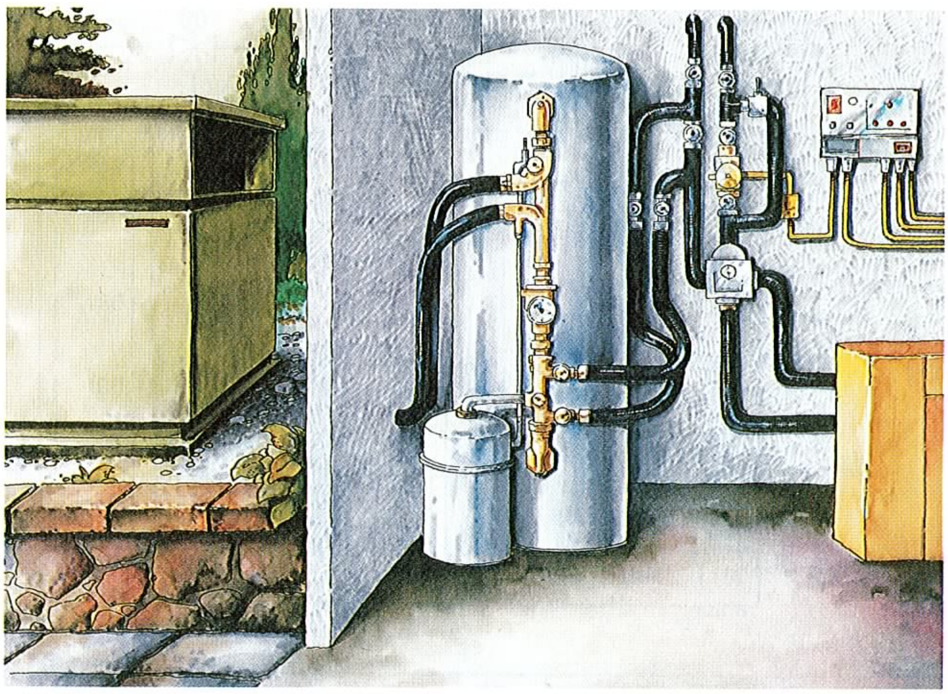

Fig. 4: The complete heat pump plant for outdoor installation

W. Raldow*

The World's Largest Absorption Heat Pump in Trollhättan, Sweden

The Swedish Council for Building Research supports many full-scale demonstration projects, to collect realistic performance data, determine economic results and provide first hand experience to consultants, building contractors, manufacturers and researchers. Development of absorption heat pump technology is considered important for Sweden because Sweden plans to phase out nuclear power plants by the year 2010. In this article, an ongoing large absorption heat pump project in the city of Trollhättan is described.

Hydrogen Instead of Electricity

The world's largest absorption heat pump began operation in Trollhättan on December 11, 1984 (see photo, page 1). The heat pump is driven by steam produced through the combustion of hydrogen gas. The hydrogen is a byproduct of the manufacture of chlorates at the nearby Kema Nord factory.

The heat pump, which has a total output of 7 MW, was constructed by the Japanese company Sanyo and is the first heat pump of its kind to be installed outside Asia. The model number is TSA-GH-900 and the working fluid pair is water and lithium bromide. The heat pump will reduce Trollhättan District Heating Company's demand for oil by 1,500 m^3 annually. The total cost of the absorption heat pump is SEK 5 million. A long-term experimental building loan of SEK 4 million has been provided by the Swedish Council for Building Research.

Joint Effort with Industry

Trollhättan District Heating Company is committed to working together with industry. Three neighboring industries, Ferrolegeringar AB, Union Carbide Norden AB, and Kema Nord, contribute thermal energy to the district heating network.

Ferrolegeringar AB is located opposite the Stallabacka heating plant. Cooling water from this plant is used to raise the temperature of the district heating water from +55°C to +80°C. Thermal energy is recovered by means of a compressor heat pump from ASEA Stal (7 MW). The total amount of energy supplied is estimated to be 35,000 MWh per year. Ferrolegeringar AB is also a customer of the Trollhättan District Heating Company.

Union Carbide's electrode manufacturing factory is located in the same industrial area. Trollhättan District Heating Company recovers heat from flue-gases produced in the manufacturing process. Flue-gas is led to an exhaust-gas boiler, and heat from the combustion process is supplied to the district heating system via a 150 mm culvert. Heat output varies between 0 and 6 MW, depending on load variations in the production process. Trollhättan District Heating Company receives about 10,000 MWh per year from this source.

In addition to the absorption heat pump and the compressor driven heat pump at the Stallabacka heating plant, there are two oil-fired boilers, one at 25 MW and the other at 5.8 MW, and a 25 MW electric boiler.

The Absorption Heat Pump is Steam Driven

The absorption heat pump is driven by steam that heats the generator. The Kema Nord Company supplies industrial cooling water with an output of about 3 MW to the heat pump, raising the temperature of the district heating system water from 40°C to 75°C (see Fig. 1). At the maximum heating load requirement the coefficient of performance (COP) is 1.67.

The heat pump was delivered in parts and assembled and tested on site; work proceeded smoothly thanks to skilled installation supervision from Japan and very precise illustrated and written instructions. Lithium bromide was delivered in 20 liter plastic containers. The capacity of the machine is 6,500 liters, making the number of containers rather impressive. In contrast to the factory's usual standard, a special pressure relief valve was installed to protect against possible overpressure.

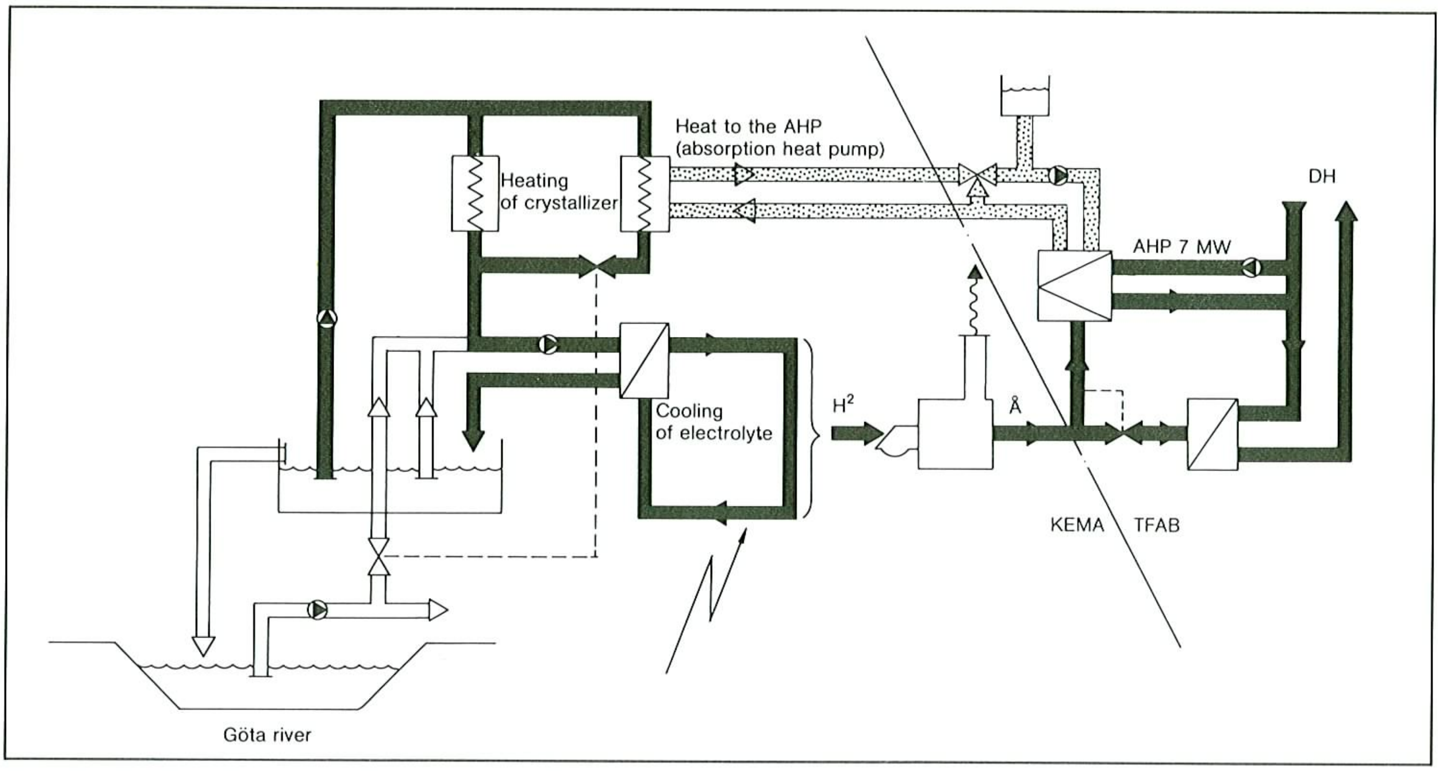

Fig. 1. The Kema Nord hydrogen and cooling water system for the absorption heat pump

This valve was required by the Swedish Plant Inspectorate.

Positive Experience

To date, operational experience has been positive. This success may be partially attributed to few moving parts and a simple control system. Two small solution pumps, a vacuum pump and a steam supply regulating valve are the only parts that have been exposed to wear. Naturally, the conventional condensate drain pipe also requires some maintenance.

Thorough Follow Up

Since the Swedish Council for Building Research has financed a good part of the construction, the operation of the Trollhättan plant will be closely monitored. The Monitoring Center for Energy Research at Chalmer's University of Technology in Gothenburg will measure and evaluate performance data over a two-year period. Flows, temperatures and pressures will be recorded continually, and heat pump performance will be evaluated from both technical and economic viewpoints.

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T. Yamazaki*
Y. Kubo*

Development of a High-Temperature Heat Pump

Japan is heavily dependent on imported fuels to ensure a stable supply of energy, and must look for new ways to use energy effectively. The industrial sector accounts for 50% of Japanese energy consumption, and produces vast amounts of waste heat at temperatures that are lower than required for most uses. A heat pump that could recover this waste heat and raise its temperature enough to produce steam could be used in several industrial applications to reduce energy consumption and costs. Tokyo Electric Power Company and Mayekawa Manufacturing Company, Ltd. have jointly developed a compres-

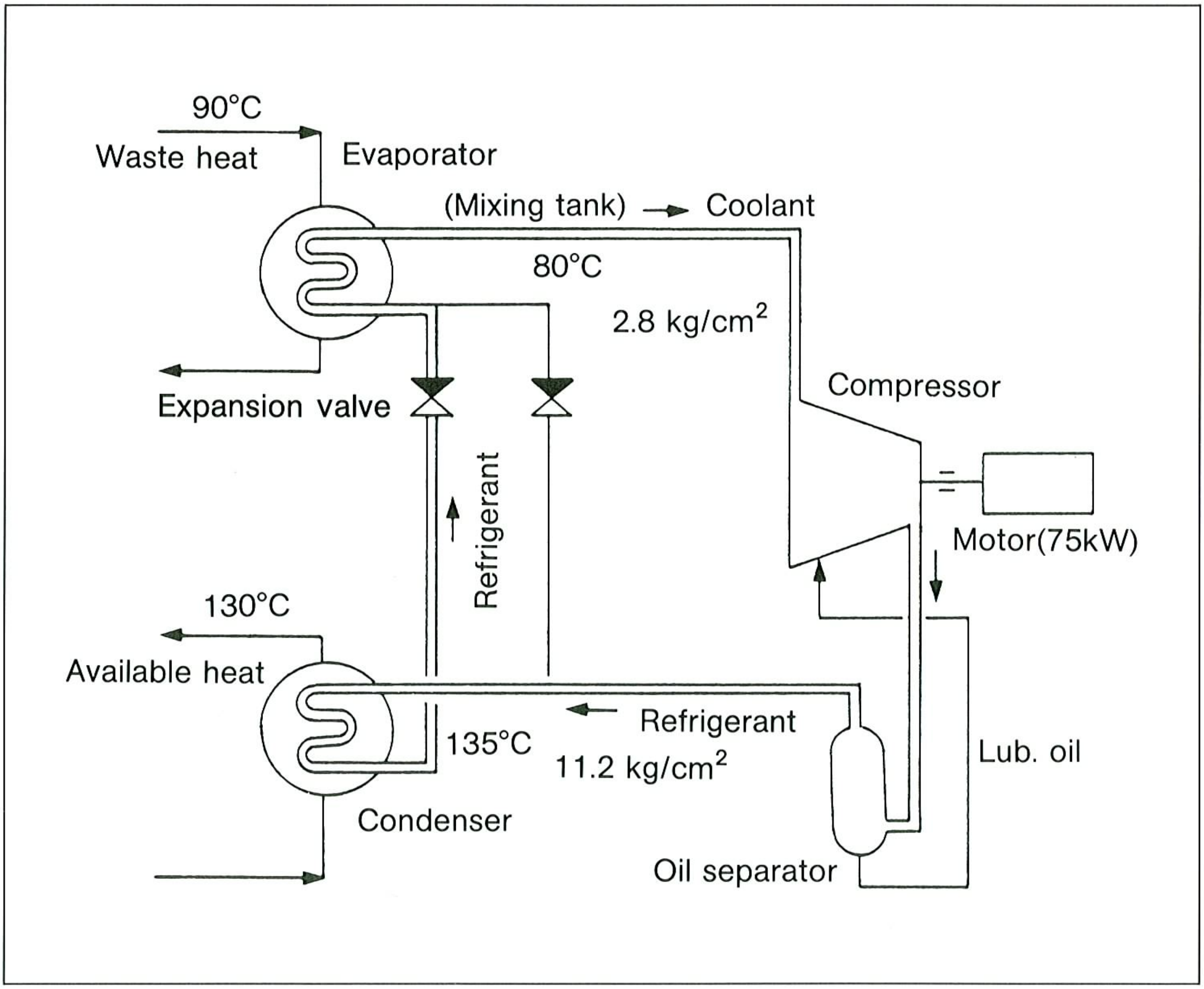

Figure 2. Flow diagram of pilot plant

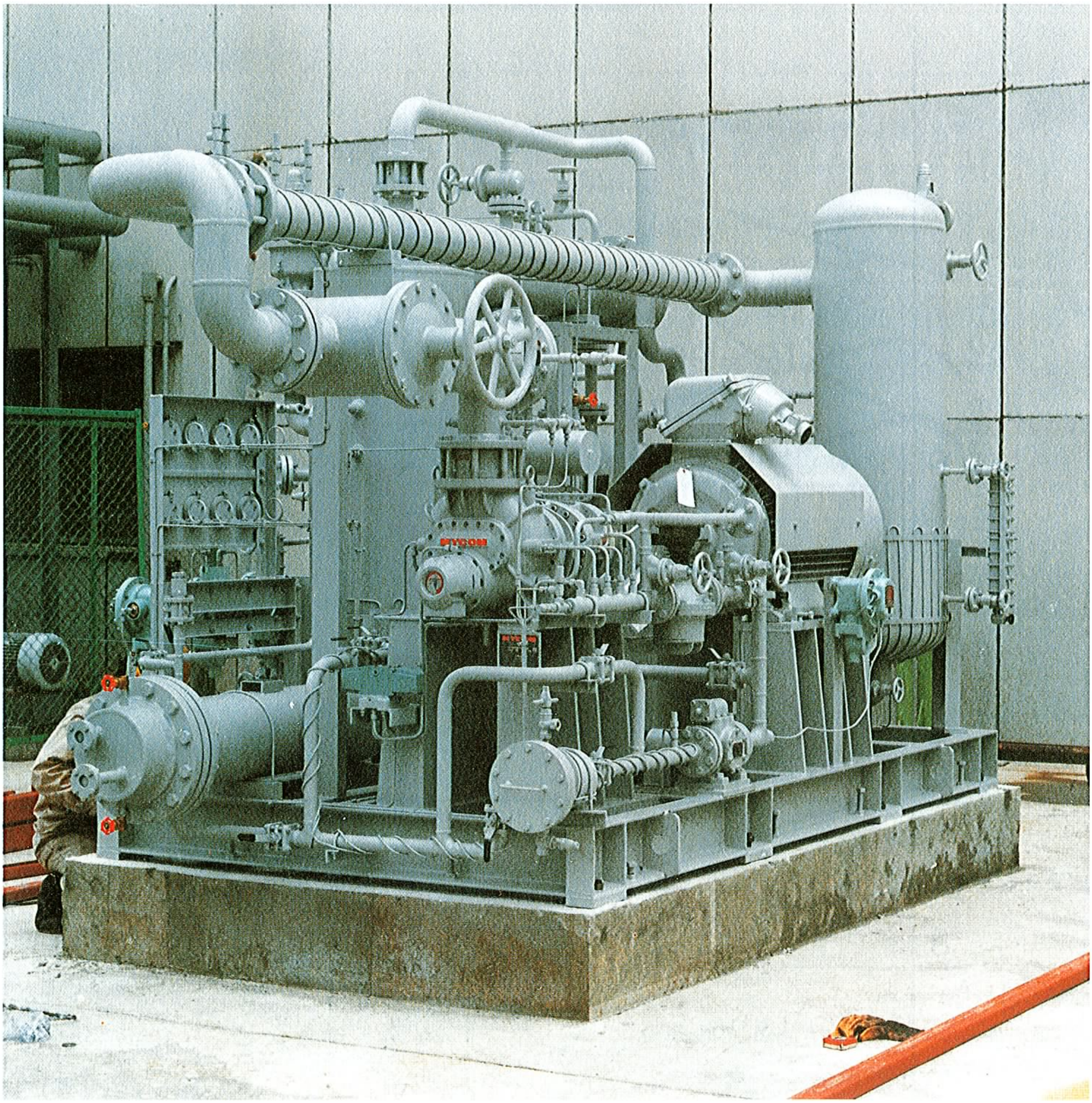

Figure 1. High-temperature heat pump

sion-type high-temperature heat pump that can supply heat at temperatures up to 130°C.

This high-temperature heat pump has several key features that allow the system to be used easily for several industrial applications:

- The maximum output temperature has been raised from the conventional 110°C to 130°C, or the equivalent of saturated steam at 2.75 kg/cm^2.
- The coefficient of performance is high, even at high output temperatures (the heat pump has a COP of 4 at a condensing temperature of 135°C and an evaporating temperature of 80°C).
- The system uses a screw compressor, ensuring high performance over a wide range of loads.
- The screw compressor is an oil-jet type, ensuring working-fluid and lubricating-oil stability at high temperatures.

We recently developed and tested a pilot plant (see Fig. 1). The specifications for this plant are listed below.

Type:
electric-motor-driven compression

Compressor: variable-capacity screw compressor

Working Fluid: pentane

Lubricating Oil: synthetic oil (polyglycol-base oil)

Maximum output temperature: 130°C

Condensing temperature: 135°C (135°C down to 95°C)

Evaporating temperature: 90°C (90°C down to 60°C)

Capacity: 75 kW (motor input)

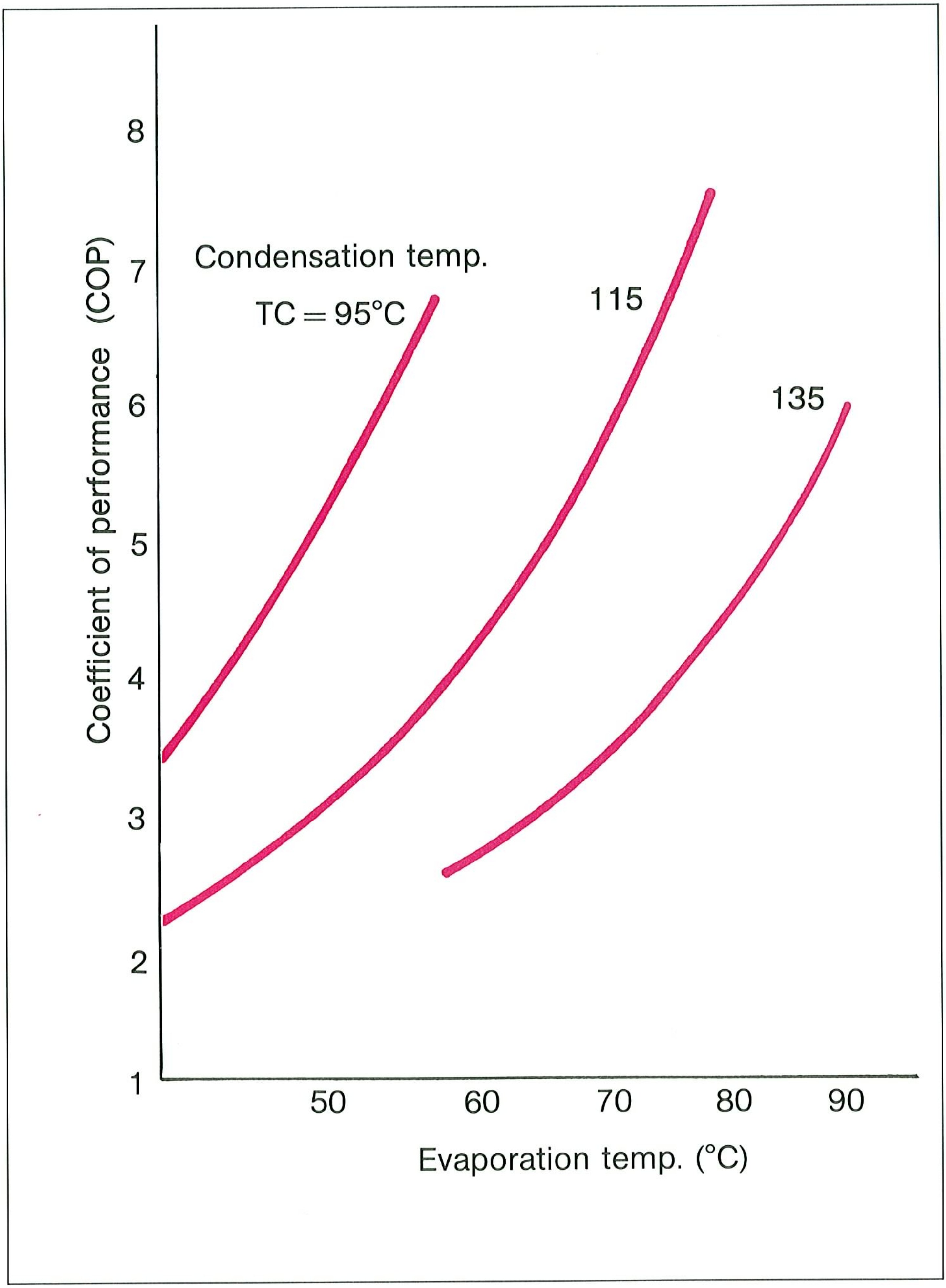

Figure 3. Performance characteristics of pilot plant

System Development

To increase the number of heat pump installations for process heating and utilization of waste heat in industrial applications, primary objectives for system development were attainment of a maximum output temperature of 130°C, and a coefficient of performance of about 4 when utilizing waste heat at 95°C. A working fluid that retains its physical properties at high temperatures had to be selected, as well as a lubricating oil that has the proper viscosity at high temperatures even when mixed with the working fluid. Further, a screw compressor that could operate reliably at high temperatures had to be developed.

Selection of Working Fluid and Lubricating Oil

Based on data from refrigerant manufacturers, we selected possible working fluids with condensing temperatures of 135°C: the freon series working fluids (pentane), and two new working fluids, Fluorinart 72 and Fluorinol 85. All of these were suitable as far as their temperature characteristics were concerned. After evaluating the thermal integrity, coefficient of performance, safety, cost, operating pressure, required compressor capacity, etc. of each of these we chose pentane, R114, Fluorinart 72 and Fluorinol 85 as the best candidates. The working fluid and the lubricating oil coexist inside the screw compressor and must behave properly at high temperatures, so each of the possible working fluids and lubricating oils were combined and placed in a sealed tube, which was kept at 175°C for 17 days. After an analysis of changes in physical properties (as well as other characteristics such as cost) pentane was selected as the best-suited working fluid for the pilot plant.

Lubricating oils used in compressors must be stable at high temperatures and be resistant to oxidations, and must have the proper viscosity over the range of operating temperatures. The oil temperature in the compressor used in the pilot plant goes above 100°C, requiring a viscosity of about 30 stokes. Based on past experience, we selected a synthetic oil that is very resistant to pyrolysis oxidation as the first candidate. We selected a polyglycol series synthetic oil as a result of the sealed-tube test.

We set up a mini heat pump with a capacity of 15 kW to assess the thermal characteristics of the selected working fluid and lubricating oil. The system was operated continuously for 1100 hours at a condensing temperature of 135°C. Neither the working fluid nor the lubricating oil showed any significant change in physical characteristics, making them a suitable combination for a high-temperature heat pump.

Operation of the Pilot Plant

The pilot plant, with a capacity of 75 kW (motor input), was designed, built and tested in order to assess the performance and reliability of the system. A flow diagram of this pilot plant is shown in Fig. 2. The plant was operated continuously for 1500 hours with the working fluid and lubricating oil described above. During operation, both the durability and performance of the compressor and the stability of the working fluid and lubricating oil were monitored. Results were satisfactory, brightening the prospects for putting the system into practical use.

Energy Savings and Economic Effects

The characteristic curve (coefficient of performance) for the 75 kW pilot plant resulting from our experiments is shown in Fig. 3. The plant has a high COP even at high output temperatures. To raise the temperature of waste heat at 90°C to 130°C, for example, the coefficient of performance is about 4 (based on a condensing temperature of 135°C and an evaporating temperature of 80°C).

A plant that requires heating steam at 1.5 M kcal/hr, for example, will need a high-temperature heat pump with an electric power input of only 450 kW, based on a 40°C temperature rise and a COP of 4. More importantly, if the boiler alternative produces steam at an efficiency of 85%, and the fuel-oil requirement for producing electricity is 2,450 kcal/hr·kW (2.85 kW_{th}/kW_e), the total energy savings will be 40%, or 500 kl of fuel oil per year (based on a heating-plant capacity factor of 80%). This reduction in fuel-oil consumption is of benefit to both the plant owner and the nation as a whole.

Except for those industrial plants that can produce low-cost steam with their own power generation facilities, industrial plants that require steam for heating purposes can expect large economic benefits from high-temperature heat pumps. Given the plant in the example above, and a steam price of 5,500 yen/ton, the initial investment in the heat pump is estimated to be recovered in about two years. Operating costs of the heat pump are low.

Efforts Towards Widespread Installation of High-Temperature Heat Pumps

Before the 75 kW pilot plant was developed, several industrial applications (e.g. chemical and food industries) were studied to ensure that installation of a high-temperature heat pump would be suitable. Two obstacles remain in the path towards widespread installation of high-temperature heat pumps: first, the initial investment is a large burden on private enterprise, and second, knowledge and information on these systems is limited. Governmental programs, such as subsidies, loans and tax breaks are being considered, but first costs may have to be reduced. At present, high-temperature heat pump installations are limited, and industries are often reluctant to disclose operating information for fear of releasing trade secrets regarding industrial processes.

Future Work

The pilot project uses combustible pentane as its working fluid, and may only be suited for chemical plants where employees are familiar with its use. To make these systems suited to a wide range of applications, we are conducting experiments using non-combustible working fluids. We are also planning to conduct field tests on a larger demonstration plant.

At the governmental level, studies are continuing on the development of a"Super Heat Pump Energy Integrating System" with higher efficiency and higher output temperatures (over 150°C). It is hoped that the recent development of a high-temperature heat pump and its success in actual use will increase the installation of heat pumps in various industrial applications.

Y. Yamazaki, Y. Kubo, Tokyo Electric Power Company, Inc.Uchisaiwai-Cho, Chiyoda-Ku, Tokyo

Heat Pump Study Trip to Japan

Visits to manufacturing facilities and installations of advanced heat pump systems in Japan

A study tour to Japan and Hong Kong from

March 8-21,1986,

is being organized by the 'Promotor Verlag,' Karlsruhe, with technical support from the IEA Heat Pump Center and the Japanese National Team.

Japanese heat pump research projects as well as advanced heat pump applications will be presented. Guided tours will be conducted at Japanese heat pump manufacturing facilities and at heat pump installations.

Participants will have the opportunity to discuss capacity control of refrigeration cycles with Japanese experts, particularly the recent Japanese success with inverter control of electric heat pumps.

For registration, please contact the

Promotor Verlag in Karlsruhe directly (Hardtstrasse 26, D-7500 Karlsruhe 21, Tel. 0721-593053).

Schedule of Conferences and Trade Fairs

Jan 19-22, 1986

San Francisco, California (USA); ASHRAE Winter Meeting; Contact: ASHRAE Meeting Dept., 1791 Tullie Cir., NE, Atlanta, GA 30329, USA

Jan 20-22, 1986

San Francisco, California (USA); Western Air Conditioning, Heating, Refrigeration Exposition; Contact: International Exposition Co., 200 Park Ave., New York, NY 10166 (USA), phone 212-986-4232

Feb 12-19, 1986

Hannover (FRG); Constructa '86; Contact: RG-Bau im RKW mit Arbeitsgruppe "Baubetriebs-beratungswesen", Düsseldorfer Straße 40, D-62366236, Fed. Rep. Germany

Mar 5-6, 1986

Los Angeles, California (USA); West Coast Energy Management Congress; Contact: Ms. A. McFarland, Association of Energy Engineers, 4025 Pleasantdale Rd., No. 340, Atlanta, Georgia, 30340 USA

Mar 10-12, 1986

Liege (Belgium); International Meeting on the Rational Use of Energy in Industry; Contact: Association des Ingenieurs Electriciens, 31 rue Saint-Gilles, B-4000 Liege, Belgium

Mar 11-15, 1986

Harumi, Tokyo (Japan); Exhibition of Refrigeration, Air Conditioning, Heating and Solar System Equipment; Contact: The Japan Refrigeration and Air Conditioning Industry Association, Kikai Shinko Bldg. 201, 5-8, Shibakoen 3-chome, Minato-ku, Tokyo 105 (Japan), phone 03-432-1671, telex 02422222 JRAIA J, telefax 03-438-0308 (Presentation of IEA Heat Pump Center)

April 3, 1986

Bristol (United Kingdom); MIREF '86 - Micros in Refrigeration; Contact: Mr. Peter Fitt, South Western Branch Institute of Refrigeration, c/o Dept. of Mech. Engineering, University of Bristol, Queen's Building, University Walk, Bristol, BS8 1TR (UK)

April 16-19, 1986

Vienna (Austria); Aquatherm 86; Contact: Wiener Internationale Messen, Messeplatz 1, A-1071 Wien

June 10-15, 1986

Guangzhou (People's Republic of China); China's 2nd International Total Energy Exposition and Conference; Contact: Ms. D.C. Rowe, International Trade and Expositions Ltd., 553/579, Harrow Rd., London W10 4RH, UK

June 22-26, 1986

Portland, Oregon (USA); 1986 ASHRAE Annual Meeting; Contact: ASHRAE Meeting Dept., 1791 Tullie Cir., NE, Atlanta, GA 30329, USA

June 23-26, 1986

Rome (Italy); Third International Stirling Engine Conference; Contact: Organizing Secretariat, Gibi studio congressi, Via Marco Besso, 40, 00191 Rome, Italy. Telephone: 3273291 or 3286897

Position Available

IEA Heat Pump Center offers a highly qualified person the position of a

Senior Engineer

At least five years experience with heat pump technology, not exclusively research, are expected. Excellent knowledge of English and German is required, as well as the capability to effectively work on the broad range of heat pump development and application tasks undertaken by the IEA Heat Pump Center.

For further information contact Dipl.-Ing. K. Holzapfel, tel. 07247-82 45 41

Written applications to be addressed to Fachinformationszentrum Energie, Physik, Mathematik GmbH, PA/Personalwesen, attn. Mr. Wuest, D-7514 Eggenstein-Leopoldshafen 2

Selected Book and Report Reviews

Advances in Non-Azeotropic Mixture Refrigerants for Heat Pumps, ASHRAE Technical Data Bulletin, Vol. 1, No. 9, 1985. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA (USA), ISSN 0884-0490 (in English)

The Technical Data Bulletin is a collection of eight papers on non-azeotropic mixture refrigerants for heat pumps that were presented at the 1985 ASHRAE Annual Meeting in Hawaii. The volume also includes a summary of a workshop on non-azeotropic mixtures that was held after the conference. For a more detailed description, see the article by J. Calm in this issue (page 1).

Heat Pump Manual, published by the Electric Power Research Institute (EPRI) and the National Rural Electric Cooperative Association (NRECA); EPRI EM-4110-SR, August 1985 (in English)

This manual is a comprehensive guide on heat pumps for residential and small commercial applications in the United States. The manual is written for utility energy management and customer service personnel, marketing specialists, and corporate planners, and the information provided is general in scope. Available systems and equipment types, their operation and maintenance, and methods of assessing their energy use and economics in comparison to conventional alternatives are summarized. Topics covered are: heat pump basics, heat pump components, air-source heat pump systems, water-source heat pump systems, performance ratings, sizing, energy estimating, heat pump economics, selection, installation, operation and maintenance, and heat pump water heaters. The manual has many illustrations, diagrams, and tables that help to clarify concepts and terminology. This manual should be useful to anyone who needs a basic summary of heat pump concepts, technology, and applications.

The manual is available through either EPRI or NRECA. Contact: Research Report Center, EPRI, P.O. Box 50490, Palo Alto, CA 94303, phone (415) 965-4081, or NRECA Research Publications, 1800 Massachusetts Avenue NW, Washington DC, 20036, phone (202) 857-9599.

Technical Papers, AIRAH Tech 85 Conference and Exhibition, Melbourne, sponsored by the Australian Institute of Refrigeration, Air Conditioning and Heating, Victoria Division (in English)

This volume is a collection of the technical papers presented at the AIRAH Tech 85 Conference, held April 15-18, 1985, in Melbourne, Australia. The papers give a good overview of Australian interests in HVAC and refrigeration developments. Although Australia is primarily a cooling market, several papers that were presented are relevant to heat pump technology, including Refrigeration and Energy Economy, Design Principles of Refrigeration Waste Energy Recovery, Development and Testing of Hermetic Systems, Screw Compressor Packages, Plate Heat Exchangers, and The Electric Expansion Valve as an Energy Conservation Device.

International Institute of Refrigeration, Commission E2, Heat Pumps and Energy Recovery, Commission Meeting on Systems and Components for Large Heat Pumps, June 19-20, 1985, Trondheim, Norway (in English)

Thirty-three papers on large heat pumps were presented at the meeting of the Heat Pumps and Energy Recovery Commission. The meeting was divided into seven sessions: system simulation, industrial heat pumps, absorption heat pump systems, heat and mass transfer in absorption systems, heat pumps for district heating, components and control for large heat pumps, and performance and operating experience.

The papers describe on-going research and results from large heat pump projects in 15 different countries. The projects range from theoretical studies on specific system components to evaluations of the field performance of several plants. There is a large emphasis on absorption heat pumps and heat pumps for district heating applications.

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In our next issue you will find contributions on the following topics:

Our next issue, the first issue of Volume 4, will focus on the environmental aspects of heat pump applications, including:

1. Environmental Effects of Electric Heat Pump Applications
2. Environmental Effects of Large Heat Pump Stations
3. Update on On-Going RD&D Projects Dealing With Environmental Questions
4. A Brief Literature Survey
5. Selected Book and Report Reviews
6. Schedule of Conferences and Trade Fairs

We would welcome any further contributions on this topic. Please send your contributions (results or progress reports from investigations into environmental questions concerning heat pumps) to the Heat Pump Center by February 15, 1986.

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