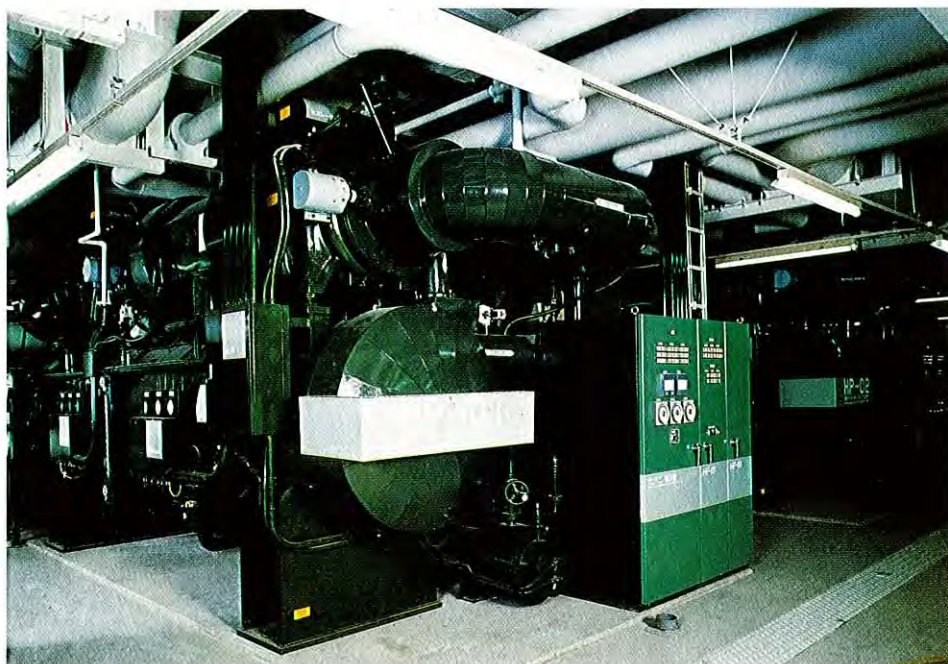


NEWS LETTER

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Air-source heat pump (see below)

T. Kuragano*

Outline of the Second Plant for District Heating and Cooling for Shibaura 4-chome Area

Recently, waterfront areas in Tokyo have been redeveloped energetically by various ministries and authorities of the Tokyo metropolitan government. This redevelopment includes ten areas of the city. In the Shibaura Konan area (about 79 hectares), various reconstruction plans are being carried out and are expected to be completed in 1995. These plans include construction of housing and an improvement of the local infrastructure. The area is composed of four main zones, among which is the Shibaura 4-chome district. This particular area once consisted mainly of factories and warehouses which are now being replaced by office buildings and condominiums.

Area supplied and heat load

The Shibaura 4-chome area, which is to be supplied with heat, covers about 15

hectares and is surrounded with canals. The expected floor space in this district is about 600,000m². The district heating and cooling (DHC) system will consist

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of two plants. The first of these is still under construction whereas the second plant was completed in April 1987. The final capacity of the combined plants will be 34MW for cooling and 27MW for heating and will cover about 70% of the area's heating and cooling demand. Table 1 shows the chilled and hot water supply conditions for the DHC plants. The second plant, which is described below, will supply 40% of the final capacity.

The location of the second plant is in the basement and penthouse of the Shibaura Shimizu building. Figure 1 shows the plant's location and the area to be supplied with heat. The heating and cooling loads are listed in Table 2. Presently, six customers will be connected to the DHC system and agreements have been reached with the owners of other buildings for future connection to the system.

Features of the system

In designing this DHC project, the most important consideration was to ensure a sufficient quantity of stored thermal energy so that peak power consumption during the summer would be low. This reduces the operating costs and the required equipment capacity.

The thermal storage system is installed under the parking garage and can store 48% and 58% of the maximum daily heating and cooling load, respectively. Three of the tanks used are stratified-type storage tanks five meters in depth, each with a capacity of 1450m³. These are used for either chilled or hot water storage and are connected in parallel, each tank being equipped with a disk-type distributor as shown in Figure 2. These distributors are designed to return and withdraw water from the tanks with a minimum of mixing, thereby helping the tanks achieve a storage efficiency of 0.9 (ratio of energy input to useful output). Testing of a one-tenth scale model of the tank was used to determine the size of the distributor disks. Figure 2 also shows the flow configurations for winter and summer operation of the tanks. Separate non-stratified tanks exclusively for chilled water (50m³ x 19 tanks = 950m³) and for hot water (50m³ x 7 tanks) have also been installed.

	Supply	Return
Chilled water temp.	7°C +/- 1°C	15°C +/- 1°C
Hot water temp.	47°C +/- 2°C	37°C +/- 2°C
Pressure	1.5 - 2.5 kg/cm ²	1.5 - 2.5 kg/cm ²
Supply period is 24 hours/day, year round		

Table 1. Chilled and hot water supply conditions

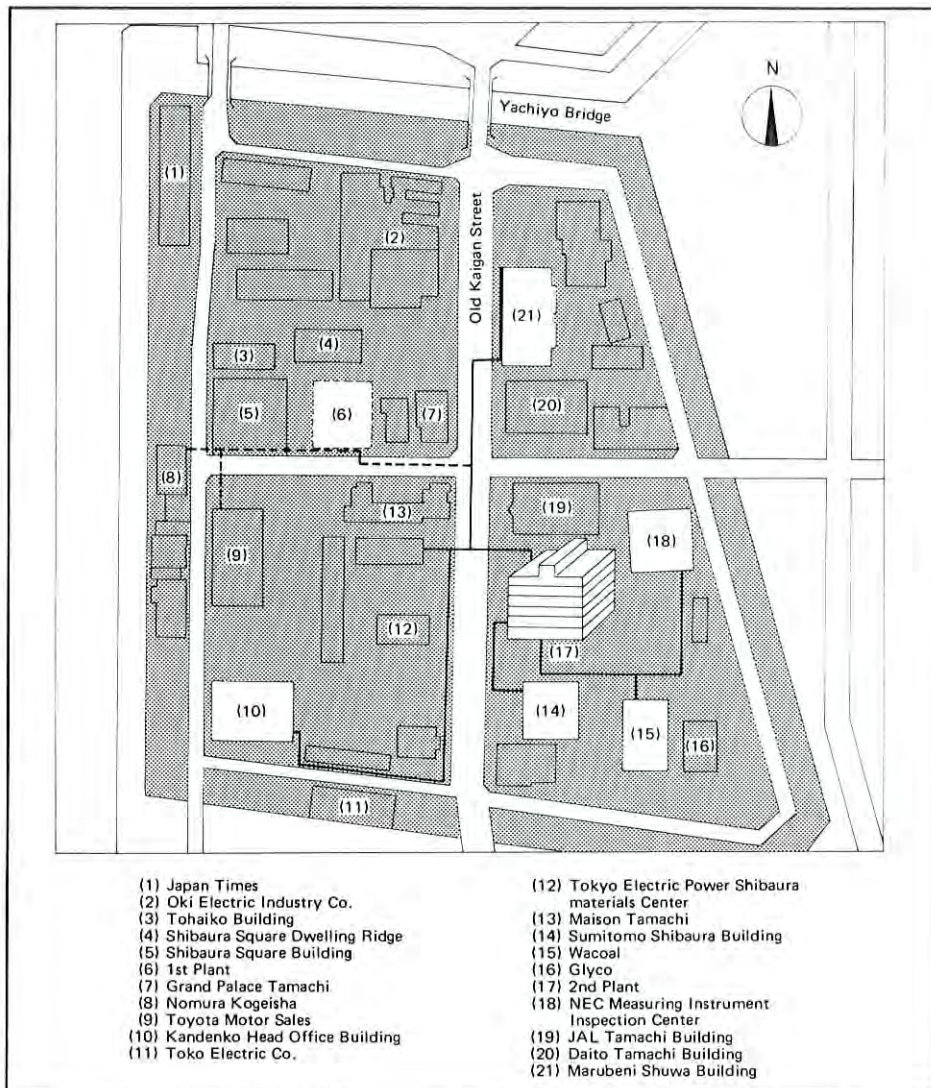


Figure 1. Areas supplied with heat

User	Total Floor Area (m ²)	Max. Heat Load (Gcal/h)	
		Cooling	Heating
Shibaura Shimizu Building	23,000	1.61	1.38
NEC Meas. Instrument Ins. Ctr	13,000	0.84	0.71
MS Building	29,000	2.17	2.03
Wacoal Building	17,000	0.94	0.85
Sumitomo Shibaura Building	21,000	1.47	1.15
Kandenkoh Headquarters Building	24,000	1.68	1.32
Total	127,000	8.71 (10.1 MW)	7.44 (8.64 MW)

Table 2. Heat loads required at present

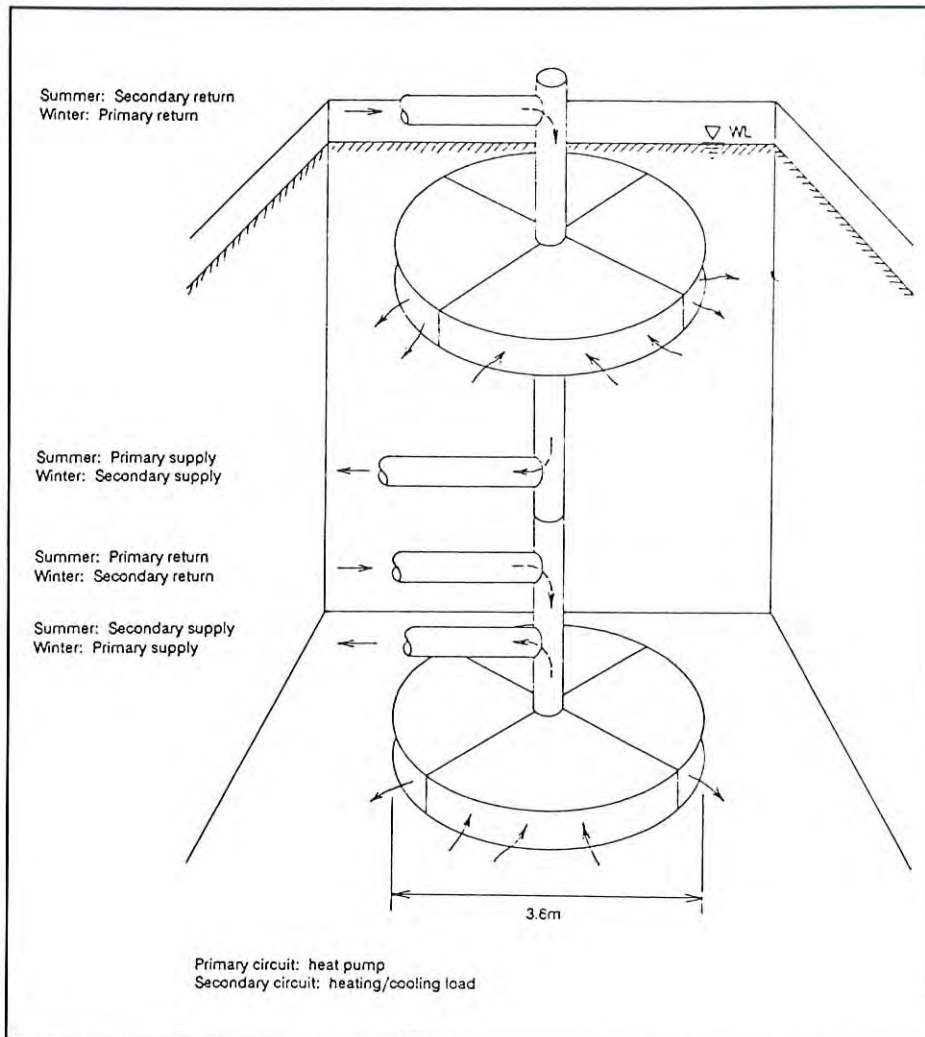


Figure 2. Disk-type distributor



Figure 4. Chilled/hot water secondary supply header

The selection of refrigerating machines for the DHC plant was done as follows:

1. The stored heat capacity is subtracted from the maximum daily heating load in winter.
2. The value obtained in 1. is divided by ten daytime machine operating hours to determine the required machine capacity.

The required cooling capacity is determined in a similar manner. In this case, the heat pump's chilled water capacity is used to further reduce the required capacity of the refrigerating machine. The machines selected are listed in Table 3.

The turbo compressors of the heat pump used in this plant are composed of one compressor exclusively for cooling and one for heating, allowing optimum operation year round. Each compressor is a two-stage unit equipped with an intermediate cooler and is also designed to allow continuous control of the inlet vane rotation speed resulting in highly efficient and stable operation over a wide range of output. Each heat pump is equipped with an auxiliary heat exchanger consisting of an evaporator and a condenser. During the winter the evaporator is used to provide chilled water, thereby providing a heat source in addition to the outdoor air. In summer, the condenser of the auxiliary heat exchanger is used to supply hot water at 50°C without lowering the heat pump's COP during cooling (the air source heat pump is shown in the cover photo).

Figure 3 shows the configuration of the plant during summer operation and Figure 4 shows the chilled and hot water secondary supply header. The connecting piping between the stratified-type heat storage tank and the flat-type tank is located 1m above the water level in the stratified tank in order to accommodate an overflow of the flat-type tank. The supply pumps for the system are controlled according to the heating/cooling required. For small loads, as in the case of hot water supply in summer and chilled water supply in winter, the supply pumps can be adjusted to the proper capacity. This is possible since one of the pumps in each set is equipped with an inverter.

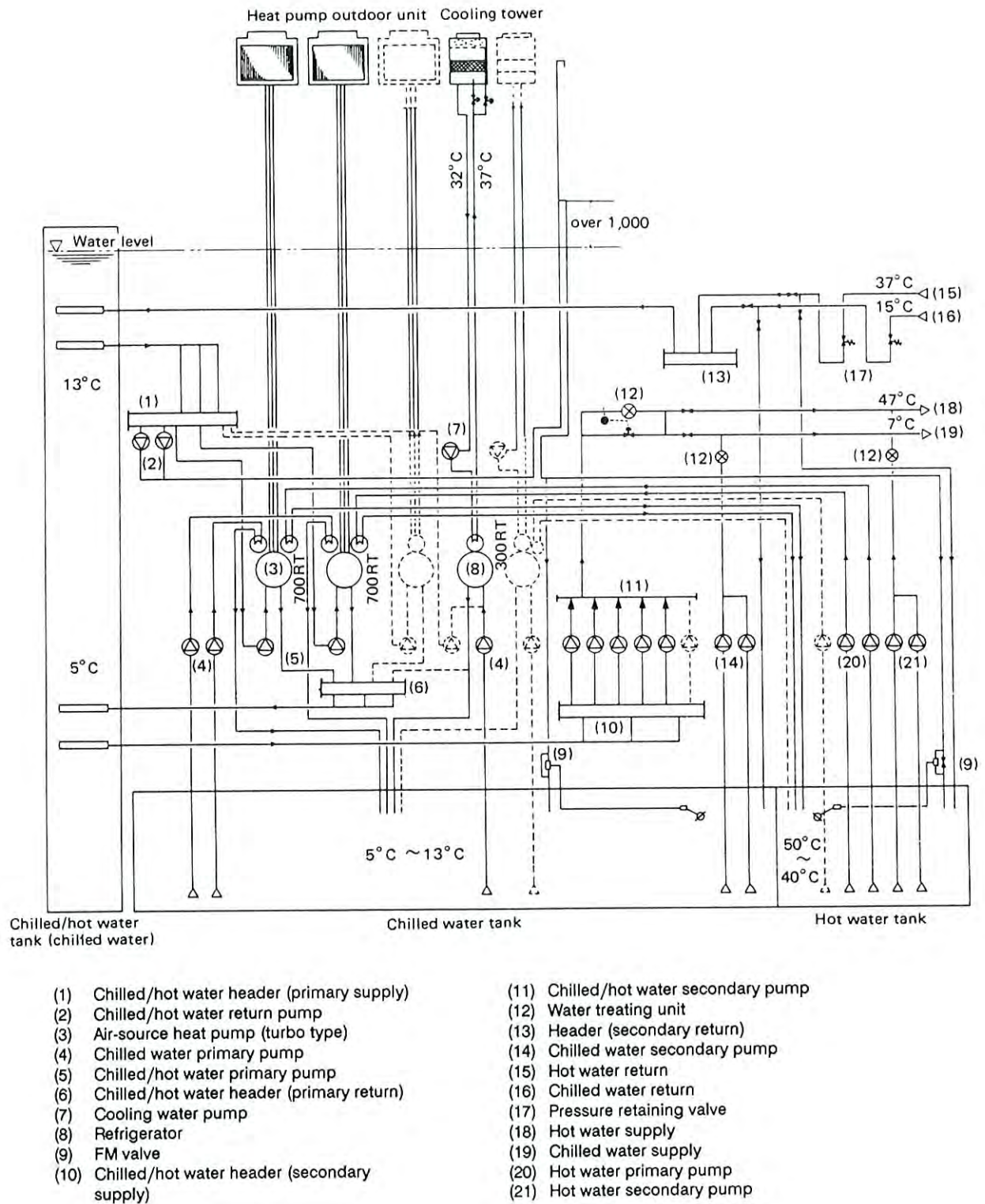


Figure 3. Plant system diagram (summer operation)

Machine	Main Specifications	Qty
Heat pump	Air-source Turbo compressor, for chilled water 750kW, for hot water 750kW (2-stage compression) Refrigerating capacity: 700 RT (2.46 MW) Heating capacity: 1.8 Gcal/hr (2.9 MW) Heat recovery: Hot water in summer - 230 Mcal/hr (267 kW) Chilled water in winter - 200 Mcal/hr (232 kW)	2
Refrigerator	Turbo compressor: 215 kW Water-cooled: 300 RT (1.06 MW)	1
Cooling tower	Cross-flow square type Energy-saving ultra-low noise type: 300 RT (1.06 MW) Flow rate of cooling water: 3,900 ℓ /min.	1
Heat storage tank	Chilled/hot water tank (stratified type) Chilled water tank (communicating vessel) Hot water tank (immersed-pipe type)	4350m ³ 950m ³ 350m ³
Chilled/hot water supply pump	Change-over of chilled and hot water: One side suction whirl-type 6000 ℓ /min 3000 ℓ /min 1500 ℓ /min	4 1 1
Chilled water supply pump	One-side suction, multi-stage whirl-type 950 ℓ /min. (One of them is such that its RPM is controlled by an inverter.)	2
Hot water supply pump	One-side suction, multi-stage whirl-type 500 ℓ /min. (One of them is such that its RPM is controlled by an inverter.)	2
Chilled/hot water return pump	For interconnection between heat storage tanks. One-side suction, multi-stage whirl pump 950 ℓ /min.	2

Table 3. Primary machine components

Other equipment

The chilled and hot water supply piping consists of 4 routes with 3 of these originating at the 2nd plant. The chilled and hot water pipes have an outside diameter of 450mm and 400mm, respectively, and are insulated and installed in protective outer pipes. Whenever possible, the piping runs on private property to minimize disturbance to public roads.

The system is monitored and controlled at 630 local points by a centralized microcomputer system. When completed, the first plant will function as a general monitoring center. In the plant's design, special attention has also been paid to the color scheme of the machines, piping, and interior to avoid the conventional dark and oily machine-room look.

In the future, these plants will use treated water from the nearby sewage treatment plant as a heat source. This has already been accounted for in the design of the second plant.

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J. Tawara*

District Heating and Cooling by Electricity for Hibiya Area

The Hibiya area, which is located adjacent to Tokyo's business and amusement zone, is undergoing changes as part of the city redevelopment program. The district covers an area of 5.3 ha. Its nine buildings, constructed between 1926 and 1974, as well as the newly constructed Toho Hibiya Shanty Building, inaugurated in October 1987, create an area with a high heat demand density. For this reason, it has been designated as a district heating and cooling (DHC) area to be supplied by an electric heat pump system with thermal storage.

This area consists of a new building and older buildings which will eventually be renovated. Additional buildings for the area are also planned. Because of this,

the DHC system will be implemented in two phases. In Phase 1, the heat supply service will be provided to the newly constructed buildings. Phase 2 will

cover existing buildings which have been renovated to allow connection to the DHC system. The connection of these older buildings is expected to begin in October 1990. Table 1 summarizes the system's capacity.

The plant is located in the fourth basement of the Toho Hibiya Shanty building. The main components of the system include an air-source heat pump, a heat-recovery heat pump, cooling and heating towers, supply piping, and a central monitoring room. Table 2 lists the plant's main components and Figure 1 is a schematic of the system's summer operation mode.

A new type of air-source heat pump utilizing a heating/cooling tower is used in this plant. It is more efficient than a conventional heat pump and uses brine

Plant area	2,346 m ²
Area served:	
Area of district served	5.3 ha
Initial area served	86,171 m ²
(total floor area, supplied to 5 buildings)	
Planned total floor area	278,185 m ²
Supply capacity (MW):	
Initial:	
Chilled water:	9.20
Hot water:	6.16
Planned:	
Chilled water:	26.2
Hot water:	18.3
Heat media:	
Temperature	
Chilled water supply	7°C ±1°C
Hot water supply	48°C ±2°C
Pressure	1.5-4.0 kg/cm ²

Table 1. System capacity

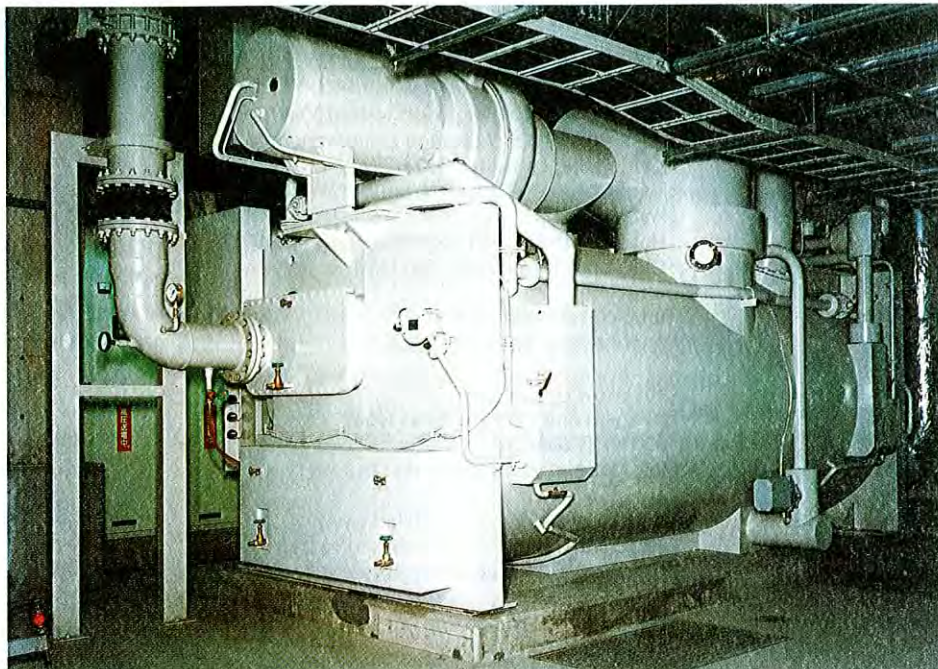


Figure 2. Air-source heat pump

as the intermediate heat transfer liquid. The brine comes into direct contact with the outdoor air during the heating and cooling mode. Figure 2 shows the air-source heat pump unit.

A heat-recovery heat pump is used for cooling in summer and waste heat recovery in winter. The heat pump is designed to effectively recover waste from an electric room, district substation, computer room, TV studio, theater, shop, etc., and produce both chilled and hot water simultaneously. The cooling and heating towers consist of one unit for the heat-recovery heat pump and another unit for the air-source heat pump. The former is used as a cooling tower for rejection of excess heat during the summer, and the



Figure 3. Heating/cooling towers

		Capacity	Initial	Extension	Planned Total
Heat source facilities	Turbo heat recovery heat pump	Cooling (MW)	2.11 x 1 unit	3.23 x 1 unit	5.34/2 units
		Heating (Gcal/h)	1.21	4.25	5.46
	Turbo air-source heat pump	Cooling (MW)	2.43 x 1 unit	2.43 x 1 unit	4.86/2 units
		Heating (Gcal/h)	1.86	1.86	3.72
	Turbo refrigerating machine	Cooling (MW)		3.52 x 1 unit	3.52/1 unit
		Cooling (MW)	4.54	8.51	13.05/5 units
	Total	Heating (Gcal/h)	3.07	6.11	9.18
Heat storage tank	Initial	All year round:			
			Chilled water tank: 3,520 m ³	Hot water tank: 2,520 m ³	
	When extended	Summer:	Chilled water tank: 6,000 m ³	Hot water tank: 1,100 m ³	
		Winter:	Chilled water tank: 1,200 m ³	Hot water tank: 5,900 m ³	

Table 2. Main plant components

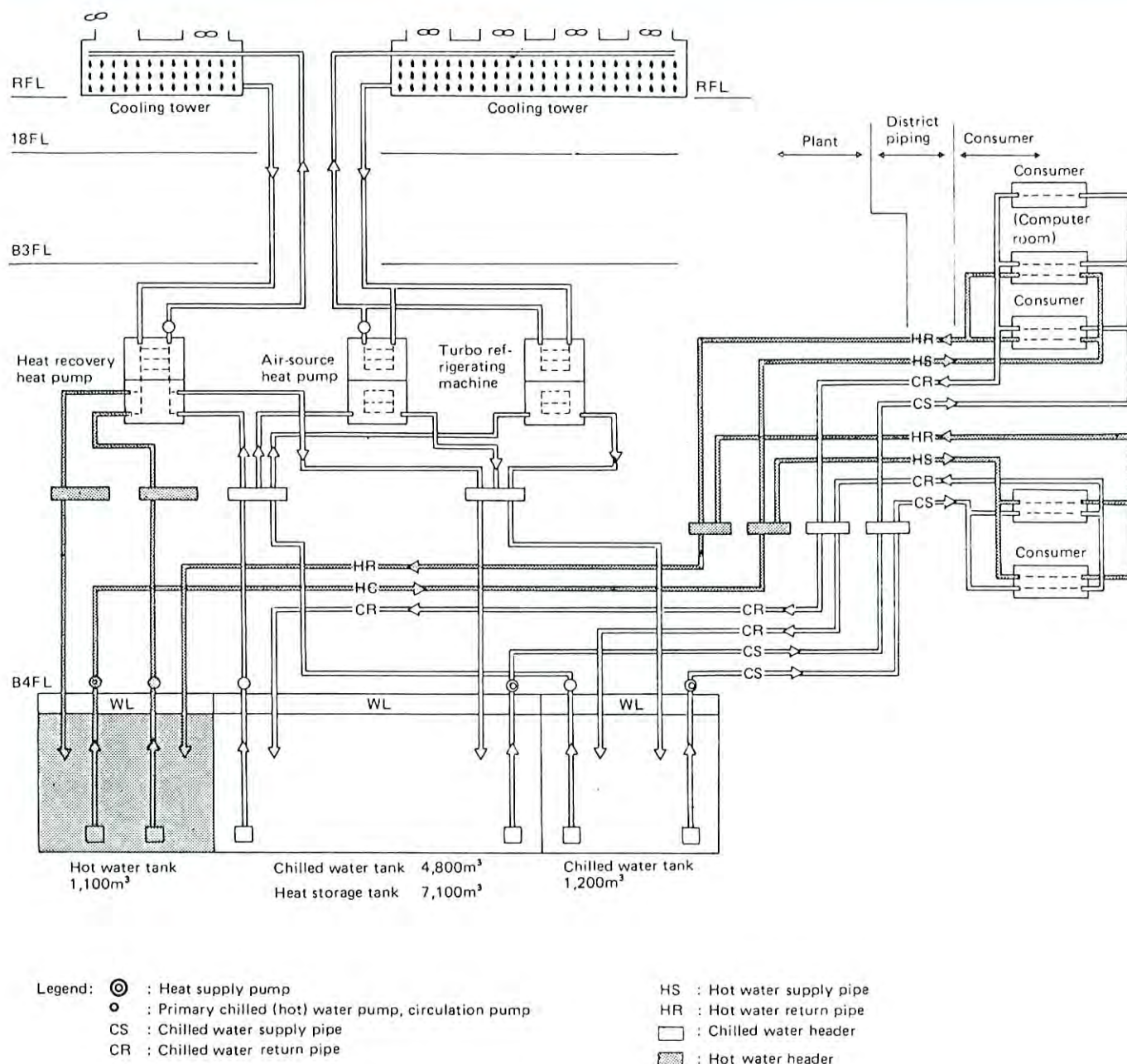


Figure 1. DHC plant system diagram (summer operation mode)

latter is used for heat rejection during the cooling season and heat recovery during the heating season. Both units are located on the roof of the building. Figure 3 shows the two units; the unit located to the left is the one used by the heat-recovery heat pump.

Each building is fed chilled and hot water from a central supply station. Flow rates are controlled according to the load to ensure efficient pump operation at all times. A central monitoring room is used for operation of the facility

as a whole by monitoring and controlling information necessary for optimum plant operation. Information which is monitored includes the heat demand, heat recovered, water supply temperature, and heat storage tank temperature.

This is Tokyo's first fully electric heat storage type heat pump system combining a new high-efficiency air-source heat pump and a heat-recovery heat pump with a large thermal storage tank (7,100 m³). By using night-time power

and storing the heat for daytime use, the system helps level the utilities electric power load. The recovery of city waste heat from various sources, such as substation facilities, saves energy and improves the system's economy, and contributes to a comfortable, clean, and attractive city environment.

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Research by the U.S. Department of Energy on Application of Heat Pumps to District Heating and Cooling Systems

There have been great advances in the state of technology of heat pumps. We have already seen this in regards to conventional electric heat pump technology, and these advances are just now beginning to find their way into the market. In the case of thermally activated heat pumps, there have been some exciting improvements and developments in the past few years. In industrial heat pumps, there is steady progress and new applications. District heating and cooling (DHC) systems utilizing hot water or steam can benefit from advancements made in heat pump technology. As such, the U.S. Department of Energy (DOE) is conducting two research programs aimed at developing advanced heat pump technologies for applications in DHC systems.

Quasi-open-cycle heat pump

District heating systems provide thermal energy to their customers in the form of hot water or steam. These systems can use one or more types of heat sources to meet the thermal load, including boilers, cogeneration systems, or low-grade heat sources in conjunction with a heat pump.

Most large-scale heat pumps operate using the closed-cycle concept, and usually use a chlorinated fluorocarbon as the working fluid. An alternative to this approach is the quasi-open-cycle heat pump. The quasi-open-cycle heat pump concept deals with the use of low-grade local energy resources in district heating systems that use hot water as the transport media. The particular low-grade energy resources of interest in this concept are water resources, such as flooded mines and sewage ponds, and waste heat resources, such as low-pressure waste steam, hot water, and exhaust gases from industrial processes.

Figure 1 illustrates the quasi-open-cycle heat pump concept. A fraction of the water returning from the distribution network is bled off, throttled in an expansion valve, and injected into the evaporative heat exchanger. The heat exchanger is connected to the low-grade energy resource which transfers

heat to the low-pressure water, and produces saturated steam. The steam produced in the heat exchanger is then compressed in the compressor. The superheated steam that exits the compressor then enters a direct-contact condenser where the remaining fraction of the district heating system return water is added. The water that exits the direct-contact condenser is then pressurized by the district heating system pump where it enters the prime mover's waste heat recovery equipment and, if applicable, is distributed throughout the district heating system where it transfers its thermal energy to various users.

The quasi-open-cycle heat pump is "open" in the sense that the compressor working fluid is the same as the district heating hot water transport media. The system, however, resembles the closed cycle in that two heat exchangers, a direct-contact condenser and a surface-area type evaporator, are used.

The quasi-open-cycle heat pump has several advantages over the closed cycle. First, the working fluid is water, which is nontoxic and has excellent thermal properties. Second, since the high temperature heat exchanger is a direct-contact condenser as opposed to a primary surface heat exchanger, the capital cost of this heat exchanger is much lower.

DOE is sponsoring a study performed by Thermoelectron Corporation on open-cycle heat pump development for local resource use to evaluate the use of a water-based, quasi-open-cycle heat pump system as a means of recovering and upgrading heat from low-grade waste resources for use in hot water district heating systems. As part of this work, systems analyses were performed to determine the independent parameters that have the greatest effect on a heat pump that uses the steam/water system as its working fluid, as well as the most appropriate type of heat resources. In addition, a survey of equipment manufacturers of the various system components was done to predict the impact of their individual performance on the entire heat pump system operating under various conditions.

As a result of this work, it was found that the quasi-open-cycle heat pump is best suited for the higher temperature heat resources such as those found in the waste streams of industrial processes. This is due mainly to the thermodynamic properties of steam. At low temperatures, the vapor specific volume is quite large and, because of the pressure-temperature relationship, a reasonable temperature rise is obtained when the compressor operates with a large pressure ratio.

As a result of a survey that was conducted on appropriate heat sources for the quasi-open-cycle heat pump, the chemical industry (SIC 28) was found to reject large enough quantities of heat at temperatures that are high enough to make it an attractive industrial candidate for waste-heat recovery by this heat pump concept.

Based on this data, a preliminary case study was performed using the Monsanto Chemical Company in Massachusetts as an appropriate industrial heat source and a nearby United States Postal Service facility as the user. This case study was used as a basis for an economic evaluation of a quasi-open-cycle heat pump system capable of supplying 822 kW of hot water into the heating system. The economic analysis indicated a three- to four-year payback, which is considered good for a system of this size.

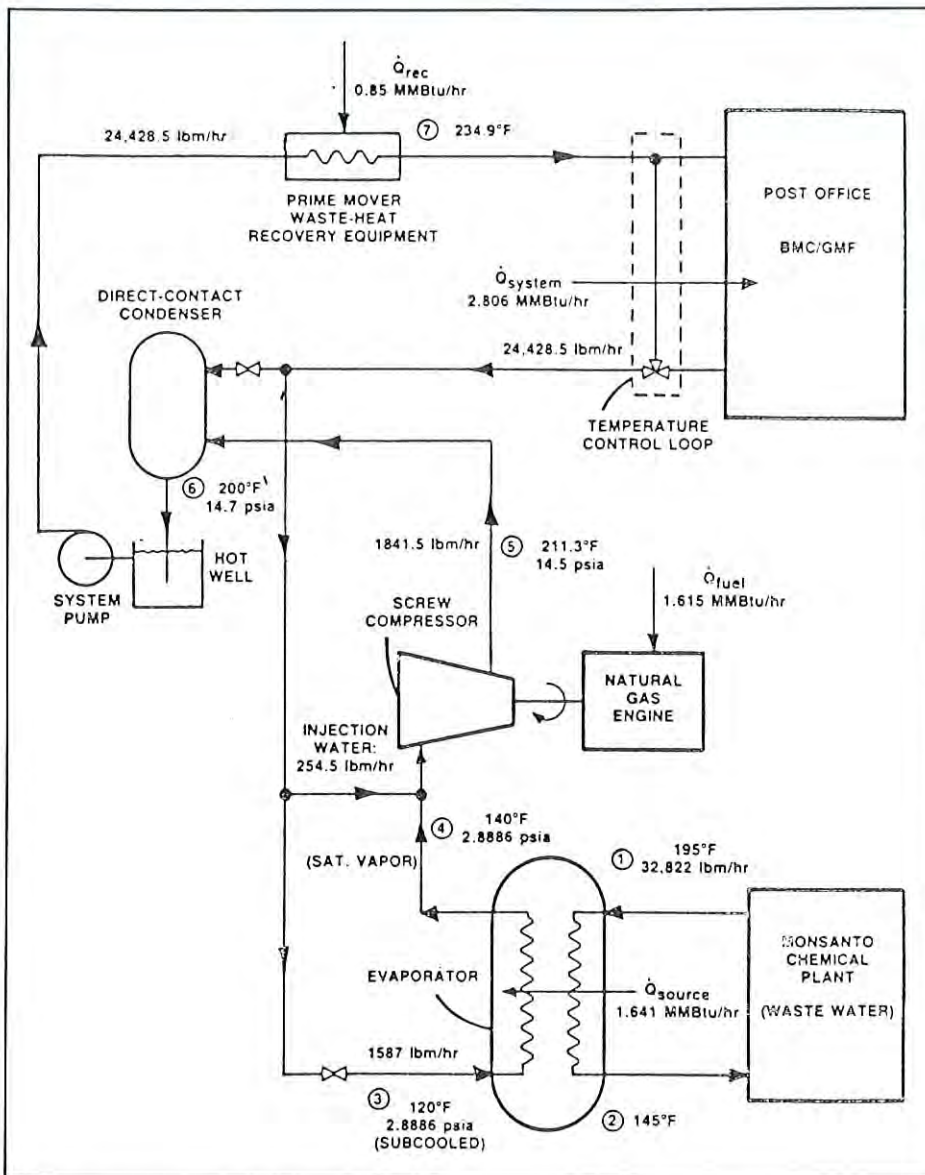


Figure 1. Quasi-open-cycle heat pump

The performance of the quasi-open-cycle heat pump using a representative industrial heat source (the Monsanto Plant) was determined by analyzing estimated performance data. The result of the analysis is as follows:

Amount of heat recovered from source	481 kW
Amount of heat supplied to district heating system	822 kW
Maximum system supply temp.	113°C
Compressor power input	123 kW
Prime mover input (based on higher heating value)	473 kW
COP fuel based (f)	1.734
COP based on compressor power input (c)	6.68
Compressor inlet volumetric flow rate	$1.54 \text{ m}^3/\text{sec}$

Advanced heat pump cycle

DHC systems have a great potential for performing energy conversion consid-

erably more efficiently than many individual units due to the large scale of plants. DOE is sponsoring research efforts by the University of Maryland focusing on further reducing the cost of energy conversion by introducing a very versatile, high-efficiency heat pump cycle, which, of course, can provide cooling simultaneously or alternatively. This will be accomplished by applying an advanced Rankine cycle because of its great potential for better performance and its widely extended application range. The cycle will be driven by an internal combustion engine.

The base of most of the new features is the use of working fluid mixtures rather than pure substances. Mixtures provide more degrees of freedom. The main feature of the proposed cycle is that it will provide a much larger tem-

perature lift than conventional systems at considerably higher COP's. This enables heat pump operation at very unfavorable outdoor temperatures while still achieving higher temperature levels.

The proposed cycle uses an internal combustion engine as the main power source. The waste heat of the engine can then be utilized for heating purposes as well. While conventional combinations of internal combustion engines with heat pumps achieve a COP of 1.5 to 2, the proposed system promises a COP which is up to 33% higher, especially under extreme temperature requirements.

Frequently, it is implicitly assumed that the phase change of the working fluid mixture is complete, i.e., no liquid phase is left when the fluid leaves the evaporator. This does not have to be the case. As a matter of fact, the cycle becomes much more versatile and flexible, and the number of candidate working fluid mixtures much larger when the mixture must not evaporate completely. The remaining liquid phase is then conveyed into the condenser by a pump through a separate liquid pipe, bypassing the compressor. The vapor generated from the evaporator is compressed by the compressor. The remaining liquid is recirculated to the condenser by means of a pump. Within the condenser, the liquid phase absorbs the compressed vapor delivered by the compressor. In order to increase the efficiency, a liquid/liquid heat exchanger is included.

Using the single-stage cycle as a building block, more advanced versions can be created which show very surprising features such as reduction of the pressure ratio by 45% compared to the conventional systems. In heat pumping, it is very desirable to have the pressure ratio across the compressor as small as possible, in order to achieve reasonable compression efficiencies. On the other hand, in technical applications it is very desirable to accomplish heat pumping across large temperature differences between heat sink and heat source. This is especially true in DHC systems.

When a Rankine cycle with solution circuit is employed, one can enjoy the

great advantage of staying with one compressor and staging the solution circuit. The condenser of the first stage rejects its heat to the evaporator of the second stage. This heat transfer is completely internal, i.e., there is no heat exchange with any outside heat reservoir. Thus, a large temperature difference can be overcome at a very moderate pressure ratio.

The following advantages can be expected from the proposed advanced versions of the Rankine cycle compared to a conventional system with regard to heating and/or cooling applications:

- COP improved up to 50%. In special cases, up to 100% as shown by a pilot plant in a DHC system in Germany.
- Extended heat pump range due to reduced pressure ratios or temperature lift
- Simultaneous heating or cooling (on two temperature levels if so desired), or simple switchover between the heating and cooling mode

-- Wide range of capacity control for constant displacement rate compressors offering the opportunity for simple load management

-- Adjustable gliding temperature intervals

District heating systems in general offer several advantages over individual boilers in terms of adverse environmental impact. Since a large central heating plant can be operated more efficiently than smaller individual units, less fuel is needed to meet the required heating duty of a particular region. A reduction in the amount of fuel burned naturally results in a reduction of emissions discharged into the atmosphere. This reduction in fuel consumption can be very dramatic for a heat-pump-based DHC system. For example, if the DHC system is competing with individual boilers that convert 75% of the fuel's energy to useful heat, then by definition, the boilers have a fuel-based COP of 0.75. Analysis of the heat pump has shown that COP's of over 1.8 can be achieved. Thus, DHC systems using a heat pump can deliver the required amount of heat and burn 2.4 times less fuel than many

individual boilers. When compared to the most efficient gas-fired pulse combustion furnace with an efficiency of 95%, the heat pump offers potential energy savings of 50% or more of the energy consumed for heating in the U.S. residential and commercial sectors by the year 2010.

Summary

These two research efforts are the first at DOE specifically directed toward application of heat pumps in district heating and cooling systems since 1981, when a previous effort entitled "Heat Pump Centered-Integrated Community Energy Systems" was concluded. The current effort reflects the focus of DOE on longer range, higher risk technologies and the judgment that such technologies can be effective in increasing the potential of district heating and cooling in energy conservation.

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Systematic Research on Condensation of Binary Mixture of Vapors for Rankine or Heat Pump Cycle

The condensation heat transfer performance of a binary mixture of vapors was systematically investigated. The results were used to calculate the thermal efficiency of a binary mixture heat pump cycle. Various flow and condensing surface conditions were used and an optimal condition for condensation was clarified. A reduction of the heat transfer coefficient when using a binary mixture occurs in the small ΔT region of free convective condensation. However, in the high ΔT region of free convective condensation and in the forced convective condensation region, the reduction is negligible. For condensation in a pipe, a temperature difference is needed to perfectly condense the binary vapor. In spite of these shortcomings, the cycle efficiency still increases when using a binary vapor as the working fluid.

Introduction

To increase heat pump cycle efficiency,

the Lorentz cycle has been proposed. In this cycle, the condensation and evaporation of the working fluid occur

over a temperature range. This condition is usually achieved by using a binary mixture of volatile vapors as a working fluid. It has also been reported that a proper mixture of two working fluids results in higher cycle efficiencies for the Rankine cycle.¹ These analyses assumed that the condensation and evaporation heat transfer coefficients were the same as that for a pure fluid. However, in the condensation of a mixture of vapors, a volatile vapor concentrates near the vapor-liquid interface and creates a diffusion layer. This reduces the heat transfer coefficient, and hence, the cycle efficiency.

There have been many theoretical works concerning the condensation of a binary mixture of vapors. Sparrow² obtained a similar solution for the free convection condensation on a vertical plate. Forced convection condensation of a binary mixture was numerically analyzed by Denny.³ It has also been determined that when a finned surface is used for heat transfer enhancement, the heat transfer performance changes in accordance with the ratio of the thick-

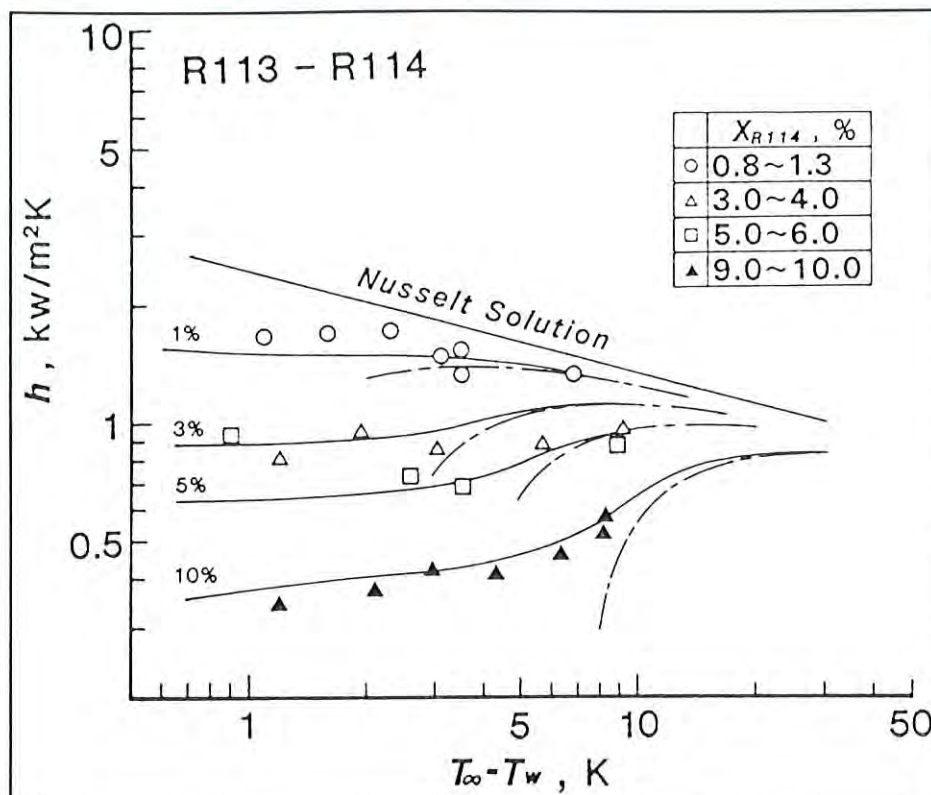


Figure 1. Free convective condensation on a flat plate

ness of the vapor diffusion layer to the fin height. The actual effect in this case on heat transfer performance has not yet been clarified.

In the case of forced convection condensation, as the flow direction of the condensate does not always coincide with that of the vapor flow, the condensation performance of a binary mixture of vapors was systematically investigated by changing the flow condition, the phase equilibrium condition, the condensing surface shape and the bulk condition of vapor concentration. By using these results, the efficiency of a binary vapor Rankine cycle is discussed, which is also applicable to the heat pump cycle.

Free convective condensation

Experimental apparatus

Experiments were carried out in a closed acrylic resin chamber, whose volume was $2 \times 10^4 \text{ cm}^3$. Two volatile liquids were put into the chamber and heated by sheath heaters located at the bottom of the chamber. The generated vapor was condensed on a cooled vertical plate located on the side wall of the

chamber. Condensing plates were made of copper and had a 50mm square projected surface area. The heat flux was calculated from direct measurement of the condensate flow rate. The mole fraction of the volatile vapor was analyzed by a gas chromatograph. In order to keep the concentration in the main flow constant, vapor was mixed by a microfan but its forced convection effect was kept to a mini-

mum by adjusting the power input to the fan.

Mixtures of R113-R114 and R113-R11 were tested. For the azeotropic mixture, a mixture of water and pentafluoro-propanol (PFP) was used. A flat plate and two finned surfaces having two-dimensional parallel fins in the vertical direction were used. Surface 1 had a tall fin height (3mm in height and 2mm in pitch) to reduce the effect of the vapor diffusion layer. On the other hand, Surface 2, having a small tip radius, was designed as a typical enhanced surface for pure vapor condensation (0.8mm in height and 0.8mm in pitch).

Condensation on a flat plate

Condensation heat transfer coefficients h of binary mixtures of R113-R114 on the flat plate are shown in Figure 1 for various concentration of the less volatile vapor in the main flow $X_{2\infty}$ (X_{R114}) and temperature difference between the cooled wall and the main flow ΔT . The solid line is the Nusselt solution for pure R113 vapor which exactly agreed with the experimental results for the pure vapor.

The experimental result shows that the heat transfer coefficient of the binary mixture decreases with an increase of $X_{2\infty}$ in the small ΔT region. However, at higher ΔT 's, the reduction from the Nusselt solution of pure R113 vapor is

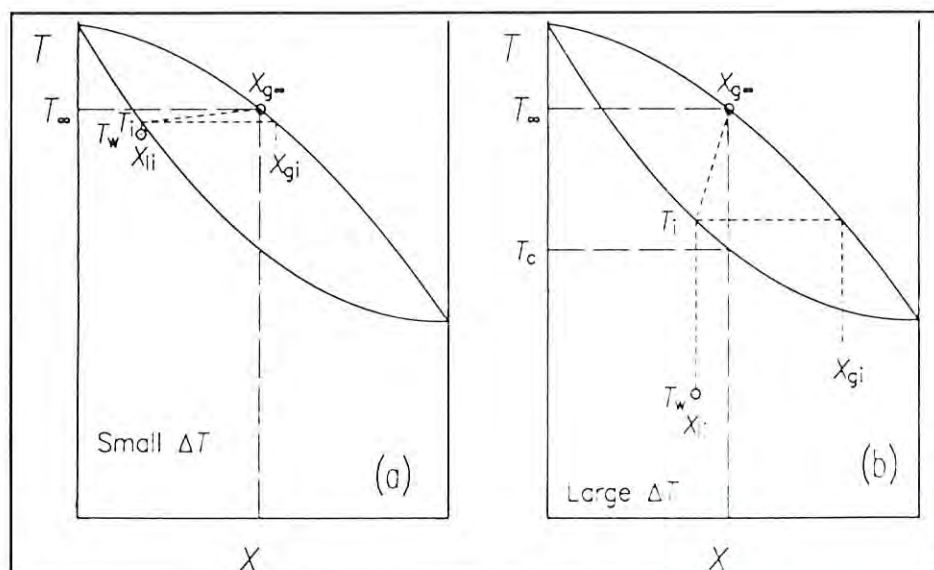


Figure 2. Phase diagram and interface temperature

small and all the data tend to the Nusselt solution, as ΔT increases.

The thermal resistance of the condensation of a binary mixture of vapors consists of two resistances existing in series; these are the vapor diffusion layer and the condensed liquid film. The relative magnitude of these resistances changes in accordance with ΔT . In the small ΔT region, the concentration of R114 vapor at the vapor-liquid interface is nearly equal to that in the main flow, however, the concentration of R114 at the liquid surface is less than that in the vapor phase, because the concentration in the vapor phase X_{2g} is on the condensation curve and that of the liquid phase X_{2l} is on the boiling curve at the same temperature as the vapor, as shown in Figure 2(a). In order to satisfy the mass conservation at the vapor-liquid interface, the oncoming velocity of R114 vapor must be lower than that of R113. This velocity difference between R113 and R114 vapors brings about the binary diffusion of vapors. Therefore, the thermal resistance of the vapor diffusion layer predominates over liquid film thermal resistance, and the heat transfer coefficient decreases with an increase of the concentration of volatile vapor.

On the other hand, when decreasing the wall temperature under constant T_∞ , the concentration of R114 at the liquid surfaces increases and approaches that of the R114 vapor in the main flow, which means that the oncoming velocity difference between R113 and R114 vapor at the interface must decrease. This requires the reduction of the vapor diffusion resistance. Then, the liquid film becomes a main thermal resistance and the heat transfer coefficient approaches the Nusselt solution of mixed vapors. When T_w is sufficiently low, the temperature at the vapor=liquid interface T_i finally reaches the temperature T_c which is shown in Figure 2(b). Therefore, the lower limit of the interface temperature T_i is T_c and the heat transfer coefficient becomes Equation 1, as follows:

$$h = h_o \{ (T_c - T_w) / (T_\infty - T_w) \}^{3/4}$$

where h_o is the heat transfer coefficient of Nusselt solution for film condensation. Equation 1 is plotted in Figure 1 by chain-dotted lines. They clearly demon-

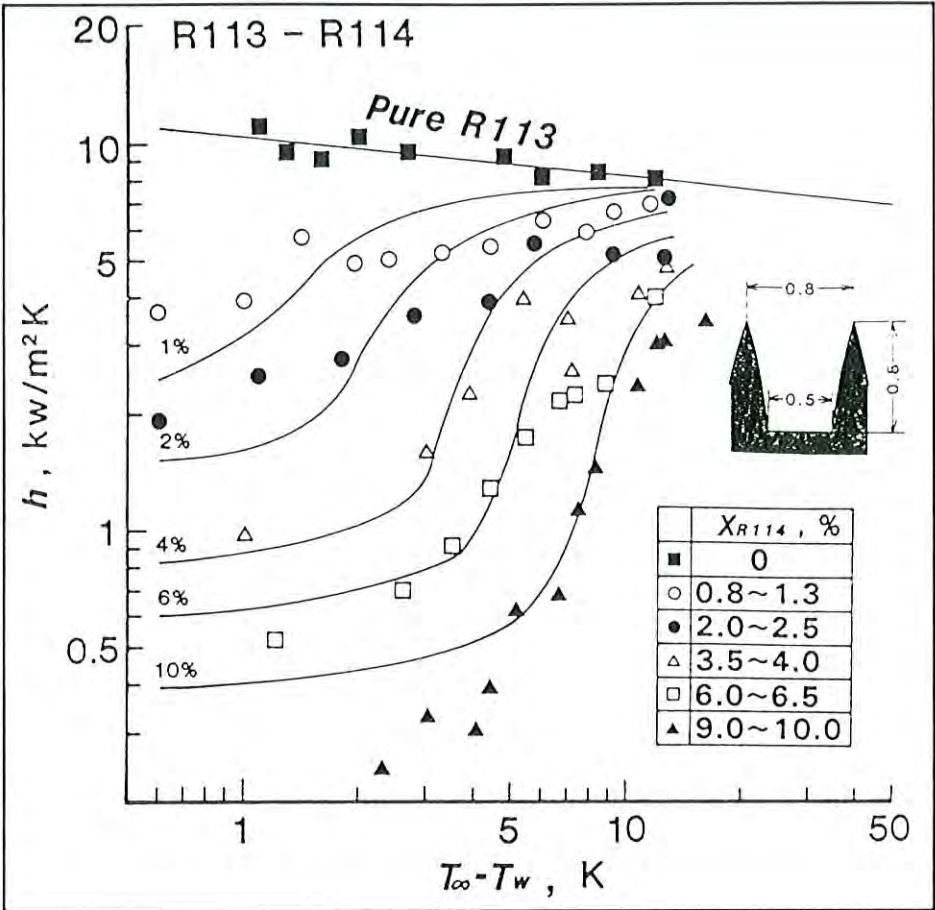


Figure 3. Free convective condensation on Surface 2

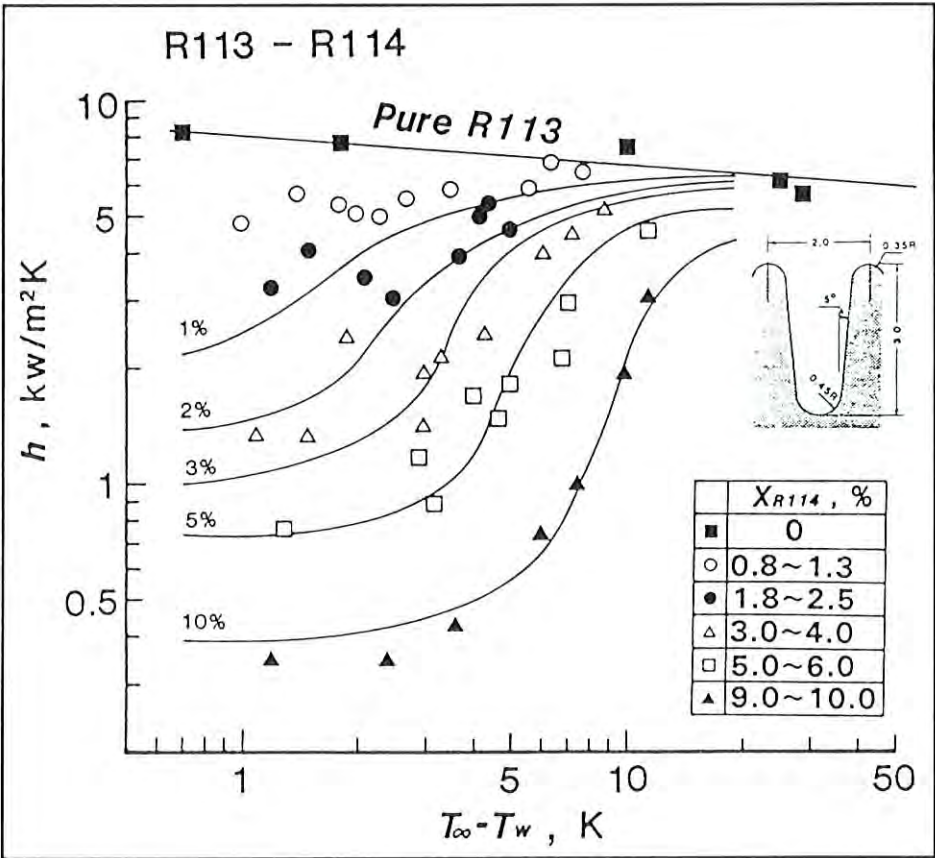


Figure 4. Free convective condensation on Surface 1

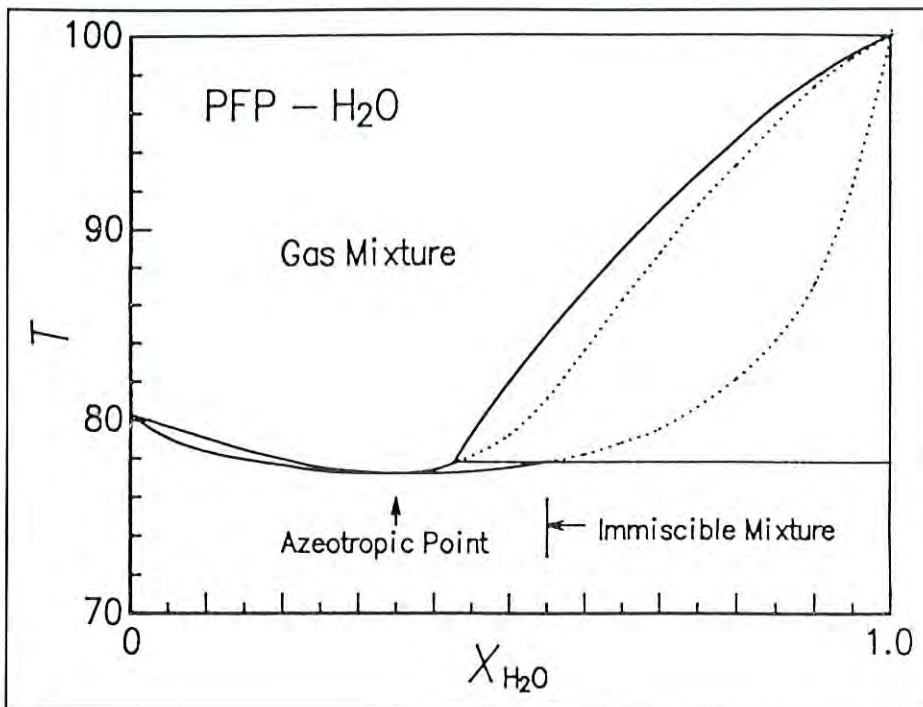


Figure 5. Phase diagram of mixture of PFP and water

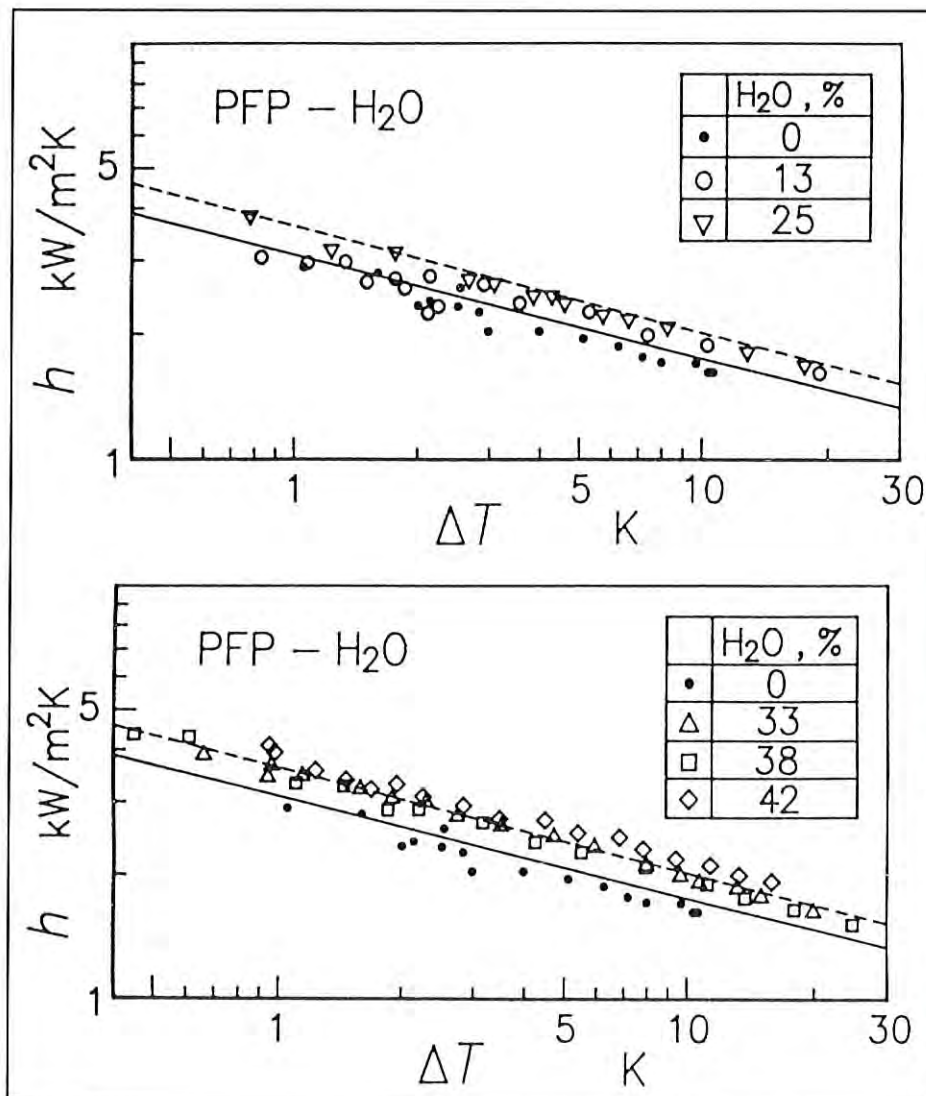


Figure 6. Condensation of azeotropic mixture

strate the tendency of the experimental data in the high temperature difference region.

In order to estimate the mean heat transfer coefficient, an imaginary constant temperature is assumed at the interface. Thermal resistances of the liquid film and the vapor diffusion layer are calculated independently from each other and the mean heat transfer coefficient is obtained by assuming they exist in series; this is called a pseudo-similar solution. The pseudo-similar solution is plotted in Figure 1 by solid curves and it is shown that the theory can well explain the experimental results in a wide range of $X_{2\infty}$ and ΔT .

In the case of R113-R11 mixture, it was found that the reduction of the heat transfer coefficient with an increase of the concentration of R11 is smaller than that of the R113-R114 mixture. The difference of the molecular weight between two vapors is larger for R113-R11 than R113-R114, which means that the buoyancy force is larger and the thermal resistance of the diffusion layer is smaller in the R113-R11 mixture for a given main flow concentration of volatile vapor.

Condensation on a finned surface

In order to clarify the effect of the shape of the condensing surface, two kinds of finned surfaces were tested whose ratios of the actual surface area to the projected area were 3.45 and 2.66 for Surfaces 1 and 2, respectively. The results for the R113-R114 mixture are shown in Figures 3 and 4. For pure R113 vapor, the heat transfer coefficient of Surface 2 is higher than that of Surface 1; even though the actual surface area is smaller. This is because the shape of Surface 2 was designed to have high condensation performance for a pure vapor. For a finned surface, the thermal resistance of the liquid film is quite small compared with that of a flat surface. Thus, the reduction of the heat transfer coefficient by the vapor diffusion layer is greater for small ΔT .

We can consider two limiting conditions. In Case A, the thickness of the diffusion layer is small compared with the fin height, and the effective surface area of the diffusion layer is considered

to be the same as the actual surface area. In the other limiting condition B, the diffusion layer is thicker than the fin height, and the effective surface area for vapor diffusion is equal to the projected area. Since the thickness of the vapor diffusion boundary layer is proportional to $(X_{2\infty})^{1/2}$ and $\Delta T^{-1/4}$, the behavior of the diffusion layer approaches Case B as the temperature difference decreases and the concentration of volatile vapor increases. Therefore, the heat transfer coefficient of a finned surface at ΔT and large $X_{2\infty}$ tends to that of the flat plate, which is clearly seen from the comparison between Figure 1 and Figure 3 or 4.

As the diffusion layer thickness of R113-R114 is sufficiently large compared with the fin height of Surface 2 in the whole range of experimental conditions, the results must be compared with the theory of Case B which is shown by solid curves in the figure. In the case of Surface 2, theoretical results clearly explain the experimental ones. However, in Surface 1, experimental results are a little higher than the theory at small T 's and low concentrations of R114. This discrepancy can be explained by the relatively thin diffusion layer compared to the 3mm fin height.

Condensation of azeotropic mixture

Since the condensation performance at azeotropic concentration is quite the same as that of pure vapor, it may be appropriate to the working fluid of the Rankine cycle. For a mixture of PFP and water, the azeotropic concentration of water is 34.9 mol% and its boiling temperature is 77.3°C. When the water concentration X_{w1} is larger than 55%, excess water becomes immiscible and exists as a dispersed phase. This phase diagram and the condensation performance on the vertical flat plate are shown in Figures 5 and 6, respectively.

The solid and broken lines are the Nusselt solutions for pure PFP and for the azeotropic mixture. The experimental results well agree with those solutions except for $X_{wg} = 13\%$. In this case, the heat transfer coefficient becomes lower as ΔT decreases slightly, because the thermal resistance of the diffusion layer becomes important. This does not occur for the pure and azeotropic mixtures. When the concentration of

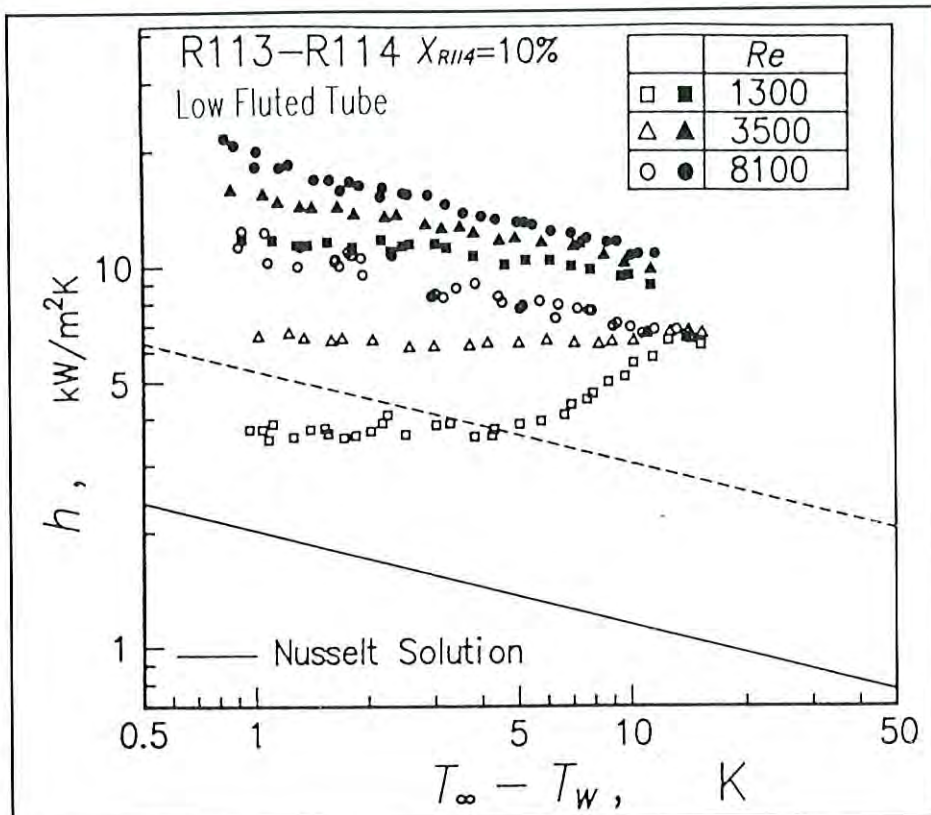


Figure 7. Forced convective condensation on a vertical cylinder

water in liquid phase exceeds the miscible limit (55%), the concentration of steam is kept constant at 42%. Therefore, the mixture of water and PFP is considered a very good working fluid under a wide range of concentrations.

For the finned surface, it was found that the condensation characteristics were the same as those of the flat plate. In this case, the surface tension effect was less effective in improving the convection performance.

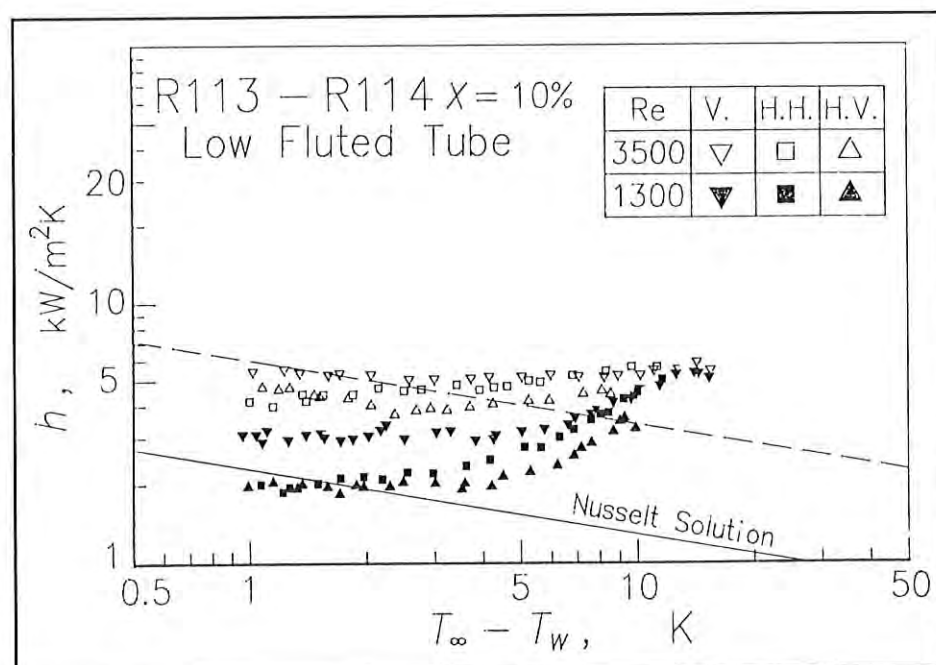


Figure 8. Condensation on finned tube for various configurations

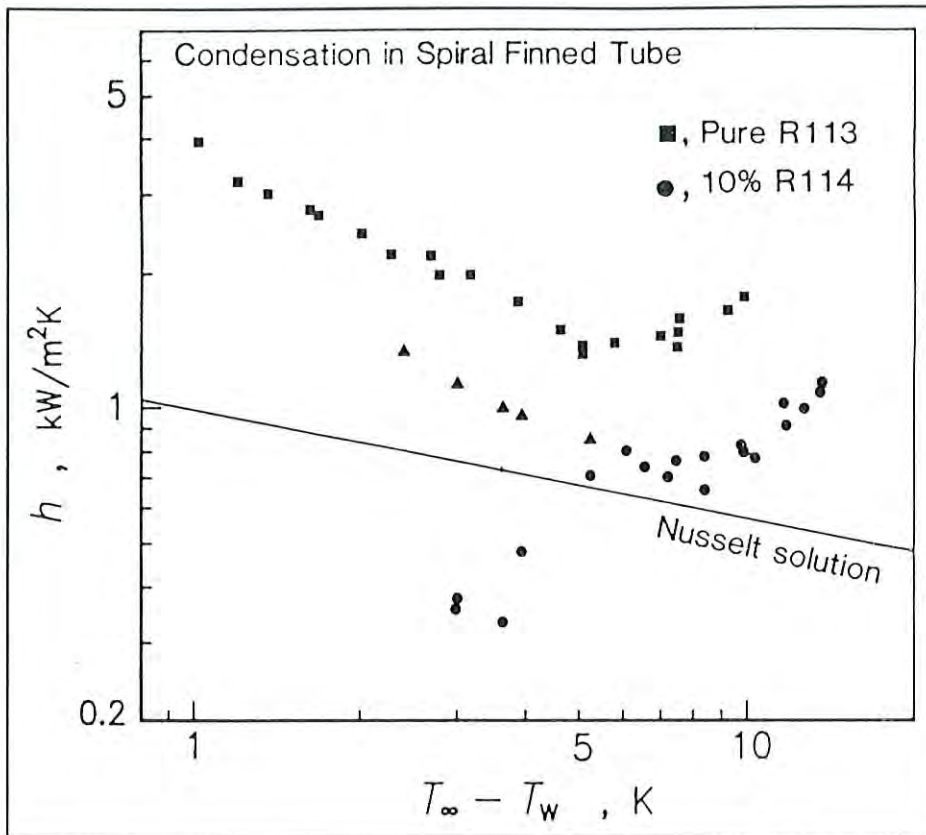


Figure 9. Condensation in a spiral finned tube

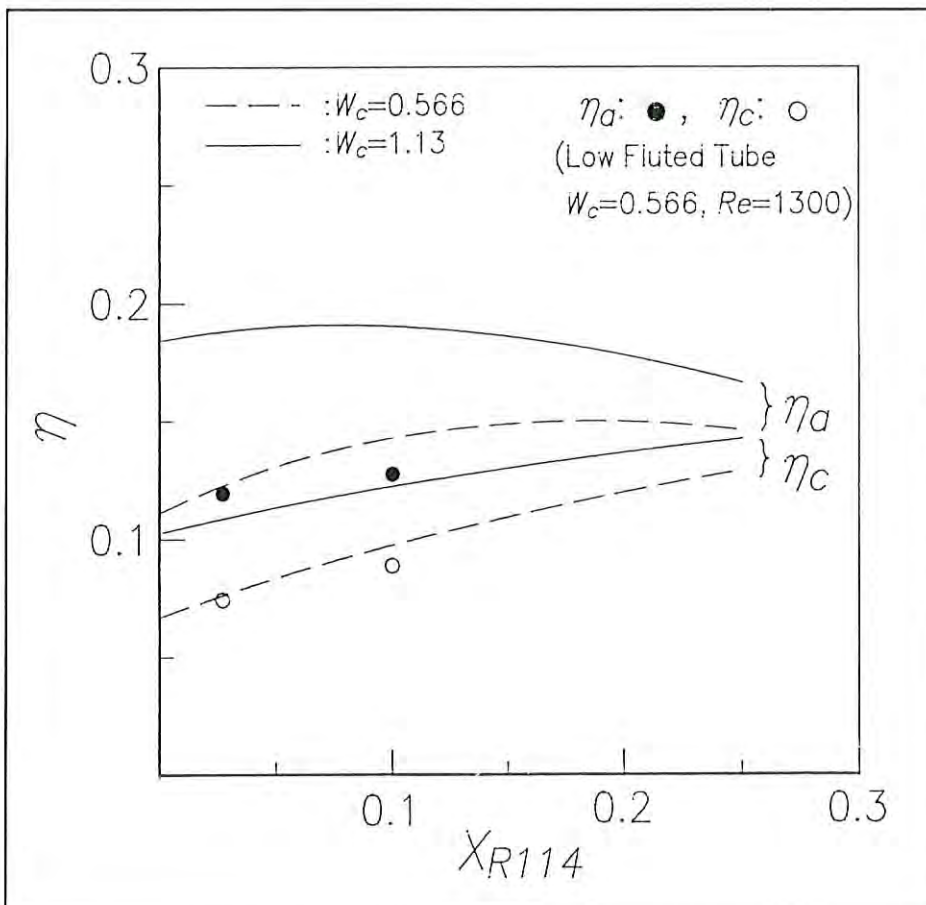


Figure 10. Efficiency of a binary vapor Rankine cycle

Forced convective condensation on cylinder

Experimental apparatus

A binary mixture of R113 (90%) and R114 (10%) was generated at the boiler and the vapor flowed into the test section having a 140mm x 60mm cross section. The condenser tube was a copper cylinder of 21mm in outer diameter and 150mm in length, cooled by water inside. The system pressure and mole fraction of R114 were kept constant. The vapor velocity was varied up to 57.5 cm/s. The condensing surfaces were smooth surface, "high wavy fin" and "low fluted tube." "Low fluted tube" had a small fin pitch of 0.7mm (surface area increase was 2.64) and a sharp fin tip, while the "high wavy fin" was 3.0mm in height with a fin pitch of 2.0mm (surface area increase was 3.21). Axial and circumferential fins were used for the vertical and horizontal tubes, respectively.

Condensation on a tube

Figure 7 shows the results for the condensation heat transfer coefficient of pure R113 (closed symbols) and a binary mixture of R113 (90%) and R114 (10%) (open symbols) on the low fluted vertical tubes in the horizontal flow. The solid and broken lines in Figure 7 represent the Nusselt solutions for smooth and finned tubes, respectively. For pure vapor, the heat transfer coefficient increases with increasing Reynolds number. This is because of the surface wave due to the shear force of the vapor flow. The figure also shows that the dependence on ΔT for the binary mixture and pure fluid is nearly the same. The reduction of the heat transfer coefficient due to the binary mixture becomes significant with decreasing ΔT and Re . Although it is not so remarkable for $Re=8100$.

From theoretical considerations, an increment of the mass transfer coefficient due to the suction by the vapor condensation is inversely proportional to the square root of the concentration difference between the bulk and the interface. By using this relation, the experimental result for the smooth vertical tube is compared with the theory. When $Re=1300$, the estimated Nusselt

number is 1.8 times larger, because of the interfacial wave.

In order to clarify the effect of flow direction and tube orientation, three configurations were tested. These were vertical tube and horizontal flow (shown by V.), horizontal tube and horizontal flow (H.H.), and horizontal tube and vertical flow (H.V.), respectively. The effect of the flow configuration and tube orientation is shown in Figure 8 for "low fluted tube" and two different Re numbers (open and closed symbols). The solid and broken lines represent the Nusselt solutions for the smooth and low fluted finned tubes, respectively. In case V., the fin is positioned perpendicular to the flow. This causes more turbulence in the vapor which enhances the heat and mass transfer. For this reason, case V. has the highest heat transfer coefficients.

Other important characteristics which were determined for the heat transfer performance are as follows:

1. For pure vapor condensation on a smooth tube, the heat transfer coefficient in the vertical downward flow is 20% higher than that in the horizontal one, but its dependence on the vapor Reynolds number is the same as that of the vertical tube.
2. For pure vapor condensation on a finned tube, the heat transfer coefficient does not depend on the flow direction.
3. For condensation of the binary mixture on a smooth tube, the heat transfer coefficients do not depend on the flow direction, which means that the mass transfer in vapor phase rate controls the total heat transfer process.

Forced convective condensation in a tube

The experiment was carried out with a vertical 14.4mm I.D. tube with spiral fins on the inner surface. The fin pitch and height were 0.5 and 0.2mm, respectively. The pure R113 and a binary mixture of R113 (90%) - R114 (10%) were used as an inlet condition, and at the exit of the tube, the vapor was perfectly

condensed. The results are shown in Figure 9.

In the case of pure vapor, laminar and turbulent flow regions are separated at a ΔT of about 6 or 7°C, where the liquid film Reynolds number is approximately 400. Due to the vapor flow, the heat transfer coefficient is 2 to 3 times higher than the Nusselt solution. However, in the case of the binary mixture, the heat transfer coefficient suddenly decreases in the small ΔT region. This is because the volatile component accumulates in the bulk along the flow direction. In the small ΔT region, it is very hard to condense perfectly.

The triangular symbols in Figure 9 show the calculated heat transfer coefficient for the binary mixture based on a corrected surface area. The corrected area excludes the surface area of the tube occupied by the condensed liquid. The dependence on ΔT is then similar to that of the pure vapor.

Efficiency of binary vapor Rankine cycle

In order to clarify the effect of the condensation performance of a binary mixture of vapors, the efficiency of the Rankine cycle for a geothermal plant was calculated. The working fluid was a mixture of R113 and R114. These results also apply to the heat pump cycle. Fluid temperature of the heat source and heat sink were assumed to be 120°C and 25°C, respectively. The pinch point temperature difference at evaporation side was assumed to be 10°C.

In Figure 10, the cycle efficiency η_c and available energy efficiency η_a are plotted against the mole fraction of R114 by broken and solid lines, under two nondimensional cooling water flow rates of $W_c = 0.566$ and 1.13. For these lines, the reduction of the condensation heat transfer coefficient due to the binary mixture is not considered. When W_c is small, the efficiencies increase with increasing mole fraction of R114. When $W_c = 0.566$, η_c and η_a using the condensation performance of the low fluted vertical tube are shown by open and closed circles, respectively. Even though a reduction of the heat transfer coefficient occurs due to the binary

mixture, the efficiencies increase with increasing mole fraction of R114. Therefore, the binary mixture has a slight advantage as a working fluid for the Rankine cycle.

Conclusions

In order to clarify the effect of a binary vapor working fluid on the Rankine or heat pump cycle, the condensation performance was investigated for various flow conditions and fluid materials. The following conclusions were obtained:

1. Experimental results for the free convective condensation of a binary mixture of vapors can be clearly explained by the "pseudo-similar solution."
2. In the small ΔT region, the heat transfer coefficient is rate controlled by the diffusion layer. In the high ΔT region, the thermal resistance of the diffusion layer decreases and the heat transfer coefficient can be estimated by the pure vapor theory.
3. In forced convective condensation, a vertical finned tube with an axial groove is preferable to enhance the mass transfer in the vapor phase.
4. Under some working conditions, the efficiency of binary vapor cycle is higher than that of a normal one, even though the condensation performance is lower.

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District Heating With Dual Cogeneration and Heat Pump Installations

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District heating systems incorporating a combination of cogeneration and heat pump installations are shown to have better economic prospects than systems with either of these installation types alone. One characteristic of such a dual system is that cogeneration operation takes precedence during periods with a high electricity price (medium- and peak-load demand) and heat pump operation during off-peak periods with a low electricity price. The improved economic prospects are the result of a reduced sensitivity to unfavorable developments in electricity and fuel prices.

Introduction

Cogeneration and heat pumps are different means of generating heat that can be used for district heating purposes. Which of these options is to be preferred will depend on many factors, of which one may be regarded as being of special importance. For a given fuel price level, this will be the ratio between the price of electricity and the price of fuel.

Uncertainty regarding future changes in this price ratio can make it difficult to choose between these two processes. Although present-day values may clearly favor one of the alternatives, there is no guarantee that the situation might not change drastically at some future date.

The objective of this article is to demonstrate that the probable answer to this problem lies in choosing a combination of cogeneration and heat pump installations rather than one option alone.

Cogeneration

Installations for the combined generation of electricity and useful heat are known in many different forms. The main differences concern the magnitude of the output capacity, the type of prime mover for converting fuel to electricity and heat, the type and quality of

the fuel, and finally whether the primary function of the installation is to produce heat or electricity.

The type to be considered in this context will be characterized by its primary function of producing heat and the type of fuel used being of the same high quality as would otherwise be required for the individual heating of each dwelling. The prime mover would typically be a gas motor or gas turbine running on natural gas or light oil, having a thermal capacity that can range up to several tens of megawatts.

A typical energy conversion diagram for such an installation incorporating, for

example, a gas motor, is shown in Figure 1. The shaft efficiency is set at a relatively high level of 35% and the heat recovery efficiency at 50% (of the lower calorific value of fuel). Assuming an electric generator efficiency of 93%, this corresponds to a total thermal efficiency of approximately 83%.

Heat pumps

Heat pumps are available in many varieties. The differences mainly concern the thermodynamic process (compression or absorption cycle), the drive energy (fuel or electricity), and the heat source (ambient air, ground or surface water, waste heat, etc.). In this context, attention is limited to compression cycle heat pumps being driven by either an electromotor or a gas motor. The heat pumps are assumed to be of relatively large capacity with an as yet unspecified heat source. Typical energy conversion diagrams are shown in Figure 1.

For the electrical heat pumps consisting of an electromotor driving a compression cycle heat pump, the coefficient of performance (COP) of the latter section is defined as the ratio between the amount of heat produced and the amount of drive-shaft energy required. Assuming this COP value to be 3.5 and taking an electromotor efficiency of 93%, this results in an overall COP of 3.25. This is somewhat higher than normal for electrical heat pumps in district heating applications (generally from 2.8 to 3).

System	Per Dwelling with 10kW Connected and 49 GJ/yr Annual Heat Demand					
	Base Load Output (kW)			System Investments (fl)		
	COG	EHP	FHP	Base Load Installation*	Other Addl. Investments**	Total Addl. Investments
Electrical heat pump	-	3	-	1800	4000	5800
Fuel heat pump	-	-	3	2100	4000	6100
Cogeneration	3	-	-	3600	4000	7600
Dual	3	2	-	4800	4000	8800

*Specific installation costs:
 -- Electric heat pump (EHP): 600 fl/kW
 -- Fuel heat pump (FHP): 700 fl/kW
 -- Cogeneration installation (COG): 1200 fl/kW
 **Mainly for heat distribution system

Table 1. Output and investment assumptions

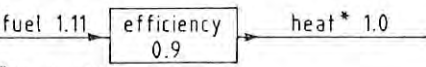
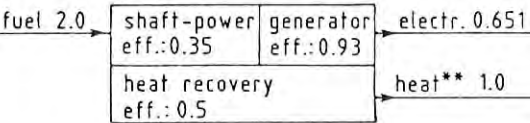
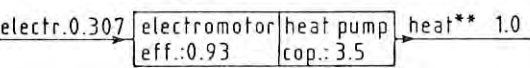
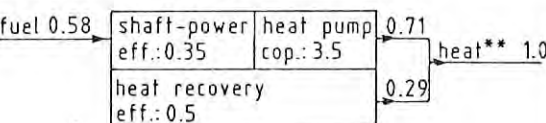
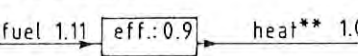
Energy conversion diagrams	Specific energy costs (per kWh heat output)
<u>Reference boiler</u>  *to dwelling	$C_{ref} = 1.11 p_f$
<u>District heating installations</u> <u>Cogeneration</u>  <u>Electrical heat pump</u>  <u>Fuel-driven heat pump</u>  <u>Peakload boiler</u>  **to district heating system	$C_{cog} = 2 p_f - 0.651 p_{el}$ $C_{ehp} = 0.307 p_{el}$ $C_{fhp} = 0.58 p_f$ $C_{pb} = 1.11 p_f$ p_f = fuel price per kWh p_{el} = electr. price per kWh
Relative energy cost savings = $\frac{\text{total energy cost savings}}{\text{energy costs reference system}}$ $\cdot \text{heat distribution eff. } n_{distr.} = 0.9$ $= 1 - \frac{1}{n_{distr.}} \cdot \frac{1}{C_{ref}} \cdot \sum \left(\frac{Q_n \cdot C_n}{Q_{tot}} \right)$ $\frac{Q_n}{Q_{tot}}$ = relative contribution to total heat demand $(n = cog, ehp, fhp, pb)$	

Figure 1. Basis for energy cost calculations

For the fuel-driven heat pump consisting of a gas motor driving a compression cycle heat pump, the indicated energy conversion diagram has been derived from the preceding cogeneration and electrical heat pump diagrams by omitting the electrical equipment. Because the overall effect is to convert fuel energy to useful heat, this type of installation is referred to as a fuel-driven heat pump.

Cost comparison

The economics of district heating systems are generally judged by making a

cost comparison with a reference system consisting of each dwelling having its own boiler or furnace installation. Energy cost savings that are the result of a more efficient method of heat production must then be weighed against the additional investment and maintenance costs resulting from the more complex heat production equipment and the addition of a heat distribution system.

Energy cost savings

The equations presented in Figure 1 indicate the different origins of the en-

ergy cost savings. In the case of cogeneration, these savings result from benefits for the produced electricity from which must be subtracted the costs of a higher fuel consumption (relative to that of the reference system). In the case of the electrical heat pump, energy costs are reduced by the cost of electricity for heat pump drive being lower than the cost of fuel for the reference system. The energy cost savings of the fuel-driven heat pumps are the direct result of the heat pump requiring less fuel than the individual boiler installations.

Using these equations, the energy cost savings (Figure 2) have been calculated for district heating systems following the typical heat load duration curve, shown in Figure 3.

The cogeneration and heat pump installations with a capacity equal to 30% of the total connected load operate in a base load capacity to deliver approximately 88% of the total amount of required heat. The remaining 12% is produced by an additional boiler operating in a peak load capacity.

The results are expressed in terms of a proportion of the fuel costs of the reference system. The values of these relative energy cost savings are presented as a function of the electricity-to-fuel price ratio in Figure 2a. This provides a clear indication of the importance of this price ratio for making a choice between these methods of heat generation.

Return on investment

In order to relate the energy cost savings to the additional investment and maintenance costs, it is obviously necessary to make some assumptions with regard to the values of energy prices and installation costs.

With regard to additional investments compared to the reference system, Table 1 lists the assumptions which were used for the purpose of this analysis. The given values indicate the additional investment per reference dwelling with a maximum heat demand of 10 kW (the connected load of the dwelling).

The total additional investment is determined as the sum of the investments for

base load heat production capacity and an investment figure which represents the sum of all other additional investments. The cost of the heat distribution system is by far the most important item determining the magnitude of this latter item. The investment for base load heating capacity is calculated as the product of capacity and specific investment costs.

With regard to energy prices, the price of fuel is set at 0.065 fl/kWh, conforming to approximately 18 fl/GJ, which represents the highest level of the small-consumer price for natural gas in the Netherlands (1985). Although the large-consumer price will generally be lower, the same value is assumed for the fuel price of the district heating system.

On the basis of these cost assumptions, it is possible to calculate the yearly energy cost savings as a percentage of the additional investment and as a function of the electricity price. The resulting values are presented in Figure 2b as a function of the electricity-to-fuel price ratio (left-hand scale).

The return on investment can then easily be determined by subtracting the additional maintenance and other costs (overheads, taxes, etc.) as a percentage of the same additional investment. This is indicated in Figure 2b for an assumed 4% value of these costs (right-hand scale).

From a comparison of the three systems, it is evident that cogeneration yields the best results if the electricity-to-fuel price ratio is high enough. However, there is also the danger of the return on investment dropping to below zero if this ratio decreases to 2 or less.

The electrically driven heat pump can also provide relatively good results if the electricity price approaches the fuel price. At present energy price levels, this will, however, only occur in rather exceptional cases.

The economics of the fuel-driven heat pump system are shown to be inde-

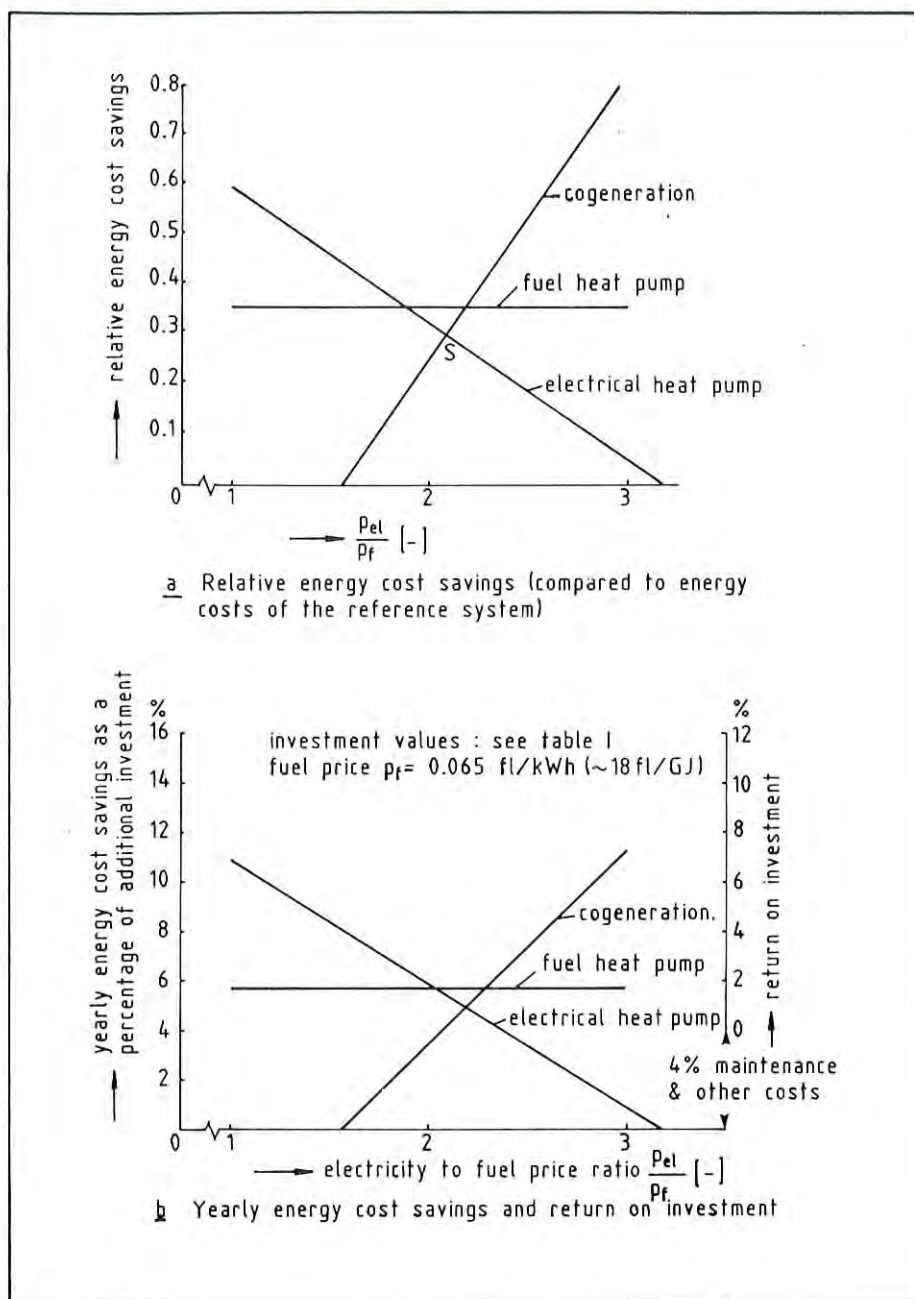


Figure 2. General cost comparison between cogeneration and heat pump systems

pendent of the price of electricity. However, for the assumed values of fuel price and investment and maintenance costs, the return on investment seems unacceptably low, even in the case of the district heating system being owned by a non-profit utility company.

Dual cogeneration and heat pump system

In the preceding cost comparison, the price of electricity was regarded as having a year-round constant value. In reality, this price will vary with the level of the electricity demand. The price

during off-peak periods will largely be determined by the cost of base load electricity generation and the price during peak periods will reflect the higher costs of medium- and peak-load electricity generation.

Recognizing the existence of this price variation is one of the reasons for looking more closely at the combined application of cogeneration and heat pump installations. A characteristic operational feature of such a dual system would be that cogeneration operation takes precedence during periods with high-price electricity and heat pump

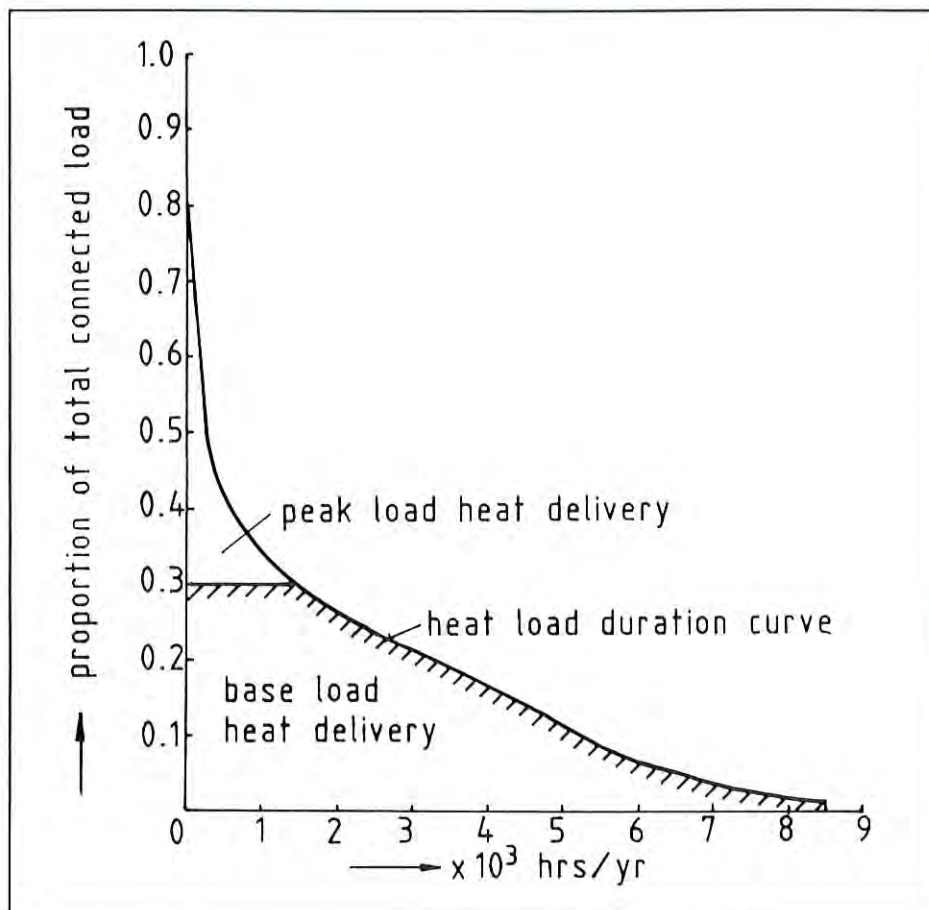


Figure 3. Heat load duration curve and relative heat contributions

operation during periods with low-price electricity.

Again, a simplified method of making a cost comparison is used for illustrating the advantages of such an arrangement. Apart from the various assumptions made earlier, this includes a simple method for distinguishing between various electricity price levels. This takes the form of defining a high- and a low-level electricity-to-fuel price ratio and allocating periods of time spent at each of these levels.

According to this approach, the high- and low-level values of the price ratio are considered to represent a time-weighted mean of all ratio values that are higher and lower, respectively, than the value corresponding to point S in Figure 2a (in other words, ratio values that favor cogeneration or heat pump operation, respectively).

With regard to allocating periods of time to each of these levels, a further distinction is made between a high and a low (electricity price) scenario. The high

scenario presupposes conventional electricity to be predominantly based on the use of a relatively high-cost fuel such as oil or natural gas. In this situation, the high-level electricity price is assumed to occur during 16 day-time hours and the low-level price during eight night-time hours of all days of the year. This results in a yearly total of 5840 high-level and 2920 low-level hours.

The low scenario presupposes conventional electricity generation to be predominantly based on the use of a low-cost fuel such as coal or uranium. In this situation, the high-level price is assumed to occur only during 16 day-time hours of the four coldest months of the year. This results in a yearly total of approximately 1950 high-level and 6810 low-level hours.

For each of these two scenarios, the relative energy cost savings are presented in Figures 4a and 4b for values of the high-level electricity-to-fuel price ratio ranging from 3 to 4, and for values of the low-level price ratio ranging from

1.5 to 2. It is evident that on the basis of energy cost savings alone, the dual system clearly offers the best results. Cogeneration comes in a close second in the high scenario case (Figure 4a). The relative advantage of the dual system increases, however, as the electricity-to-fuel price ratio is reduced. This is particularly evident in the low scenario case presented in Figure 4b.

The relative advantages of the dual system become somewhat less when the higher investment and maintenance costs of this system are taken into account. These costs being higher is a consequence of the total base load capacity being larger than for any of the other systems (Table 1).

Compared to the electrical and the fuel heat pump systems, the dual system is shown to be better over the whole range of considered electricity-to-fuel price ratios. This leads to the conclusion that, although the dual system gives better results over a wide range of energy price ratios, the magnitude of the cost advantage very much depends on the actual values (and duration) of these price ratios. However, more important than the magnitude at any given energy price level is that an advantage can be maintained in the face of considerable changes in these price levels. This can be illustrated by first determining the return on investment for certain reference values of the fuel and electricity prices and then considering the effect of interdependent changes of these energy prices. Results of such a procedure are presented in Figures 5 and 6.

The reference price of fuel is set at 0.065 fl/kWh and the corresponding reference values of the high and the low level electricity price at 0.22 at 0.11 fl/kWh, respectively (Table 2).

The interdependency of fuel and electricity price changes is assumed to be different for high and low level electricity prices. A 1% change in fuel price is assumed to correspond with a 2/3% change in high level and a 1/3% change in low level electricity price. Although these values are somewhat arbitrarily chosen, the distinction is made in order

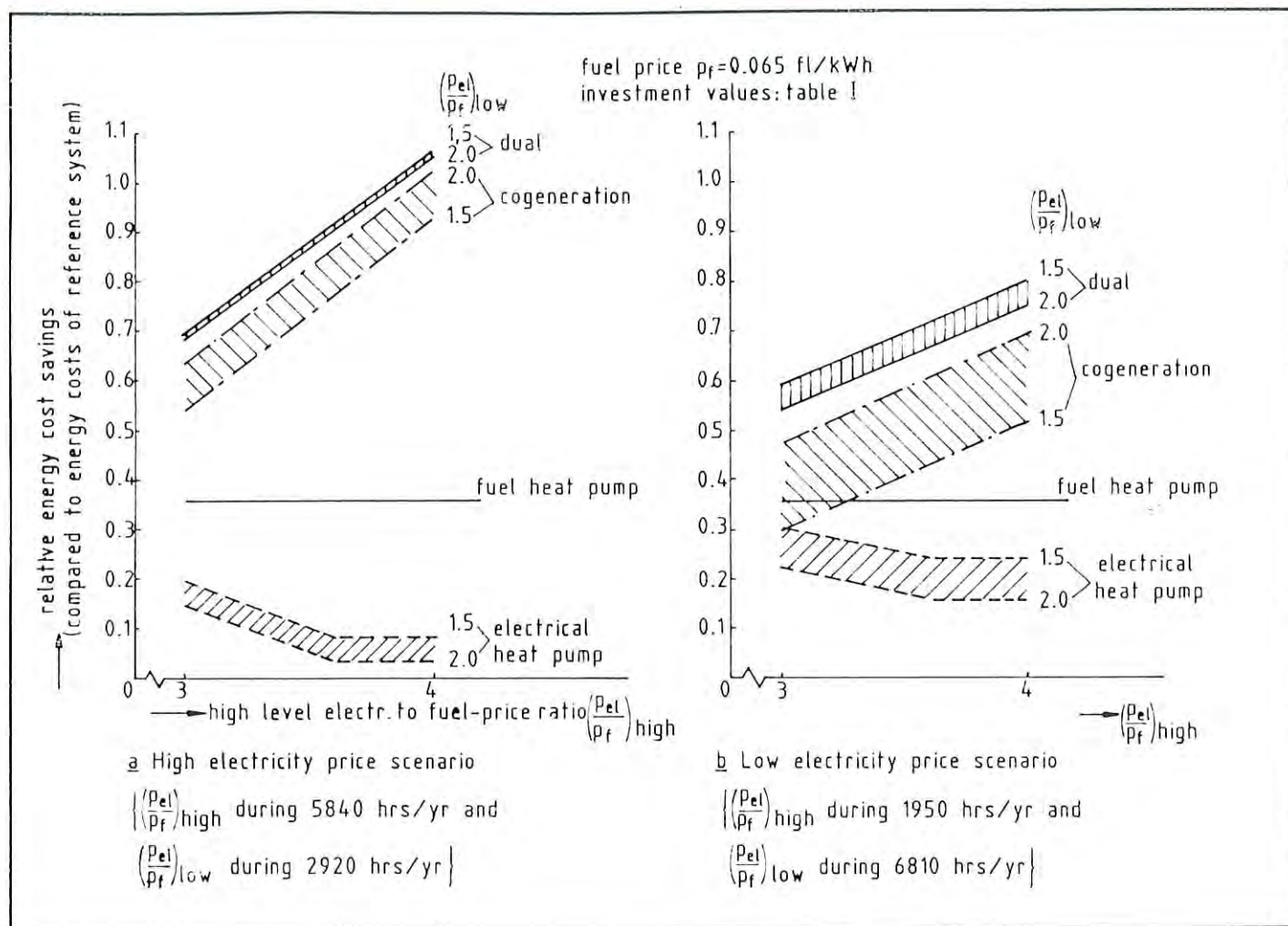


Figure 4. Relative energy cost savings as a function of electricity to fuel price ratio

to take into account the usually different methods of peak load and base load electricity generation. With the use of coal-fired or nuclear installations as base load units, the corresponding costs of electricity generation will be much less sensitive to changes in gas and oil prices than in the case of electricity being generated by (often oil-fired) middle or peak-load units.

The effect of these price changes as illustrated in Figures 5a and 5b suggests that the dual system combines the best of cogeneration and heat pump systems.

With cogeneration, it shares the advantage of a relatively higher return on investment at reference and lower energy price levels. Under such conditions, the electrical and fuel heat pump systems prove to suffer a more severe decrease in return on investment.

In comparison with cogeneration, the very worst case for the dual system is when the heat pump section becomes entirely superfluous due to even the low-level electricity price being too high for economical heat pump operation. However, even then, the energy cost savings as a percentage of additional investment will only be 14% lower (same energy cost savings, additional investment of fl. 8,800 instead of fl. 7,600 per dwelling).

At the other end of the fuel price scale, the dual system shares with both heat pump systems the advantage of being able to benefit fully from a general increase in energy price levels. Under such conditions, the cogeneration system is at a clear disadvantage. Although a certain improvement is still possible in the high scenario case (Figure 5a), there is an evident decline under low scenario conditions (Figure 5b). The latter scenario is considered to be

more meaningful because a strong increase in fuel price may be expected to cause a shift from high- to low-cost fuels for conventional electricity generation.

The resulting "safeguarding" against the effect of unfavorable energy price changes may also improve matters in an indirect way. It may, for instance, persuade utility companies to lower the minimum requirements with regard to return on investment. This, in turn, could form an incentive for equipment manufacturers to undertake the developments that would lead to further cost reductions and a consequent increase in market penetration.

Figure 6a illustrates the effect of such cost reductions amounting to 40% of the total additional investment and maintenance costs. Taking 6 to 8% return on investment as the minimum acceptable to a utility company, the figure shows that such requirements

reference fuel-price $p_f = 0.065$ fl/kWh (~ 18 fl/GJ)
 electricity price: see table II

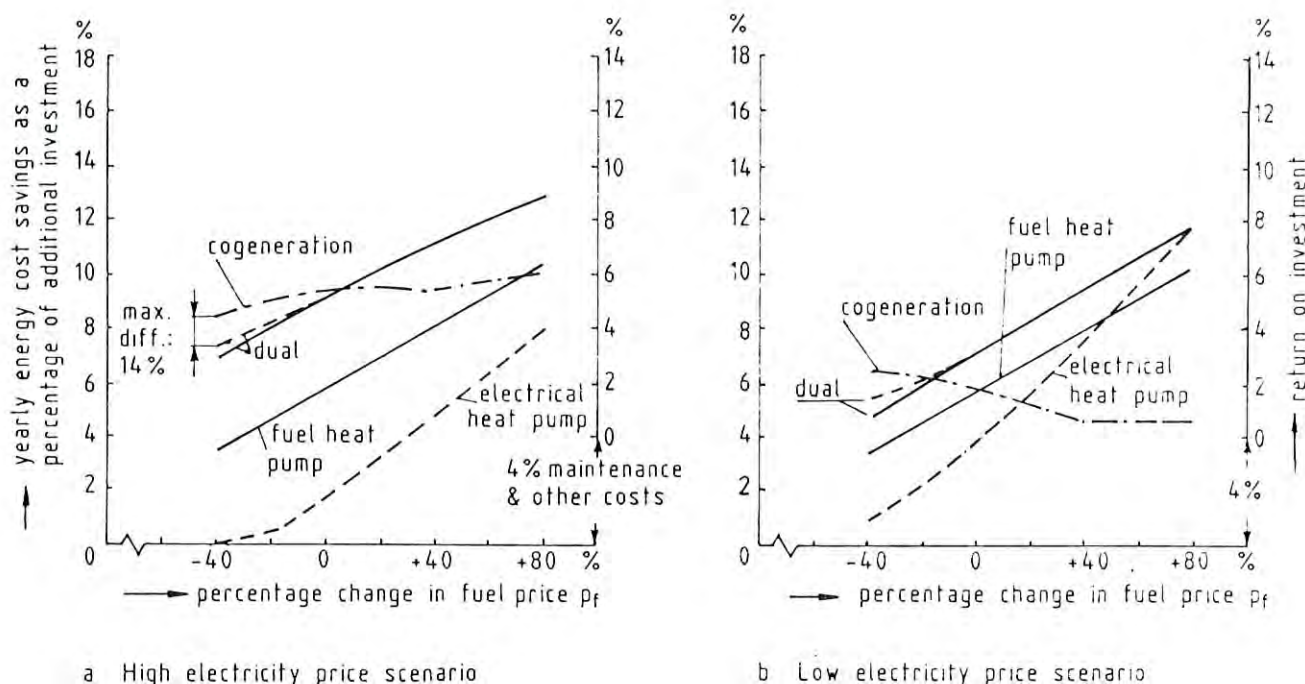


Figure 5. Return on investment as a function of changing electricity and fuel price levels

could be met by the dual system at energy prices around the reference levels.

Another aspect that will be of importance to determining market penetration is indicated in Figure 6b. This figure illustrates the effect of a 40% reduction in investment (and maintenance) costs of only the cogeneration and heat pump units while leaving the other additional investments per dwelling the same. It is clear that this has the effect of increasing the advantage of the dual system relative to the other systems. For example, at a fuel price slightly above the reference value, the return on investment for the dual system would be 6% as opposed to only 3.3% for the best of the other systems (under low scenario conditions).

Such comparative results are representative for any application in which there is an increase in the costs of heat distribution relative to the costs of the heat

production equipment. This is the case when district heating is applied in areas with existing housing (relatively higher heat demand but generally an even greater increase in heat distribution costs) or in areas with highly insulated new houses (relatively lower heat distribution costs but an even greater decrease in magnitude and duration of the heat demand).

Because future district heating applications will generally be faced with one or the other of these relatively unfavorable conditions, it can be appreciated that this characteristic of the dual system may be of prime importance in deter-

mining the future market penetration of this type of district heating system.

Finally, it is important to consider what the effect would be of changes in the many assumptions which have been made in order to make a quantitative cost comparison. These assumptions mainly concern the characteristics of the heat demand, the installation performance, the interdependency of fuel and electricity prices, and the investment and maintenance costs of the various system components. It will be obvious that any changes in these assumptions will result in a different outcome of the cost calculations. It is considered

High-level electricity price:	
- Reference value corresponding to fuel price $p_f = 0.065$ fl/kWh	0.22 fl/kWh
- Electricity price change per 1% change in fuel price	2/3 %
Low-level electricity price:	
- Reference value corresponding to fuel price $p_f = 0.065$ fl/kWh	0.11 fl/kWh
- Electricity price change per 1% change in fuel price	1/3 %

Table 2. Electricity price assumptions (Figures 5 and 6)

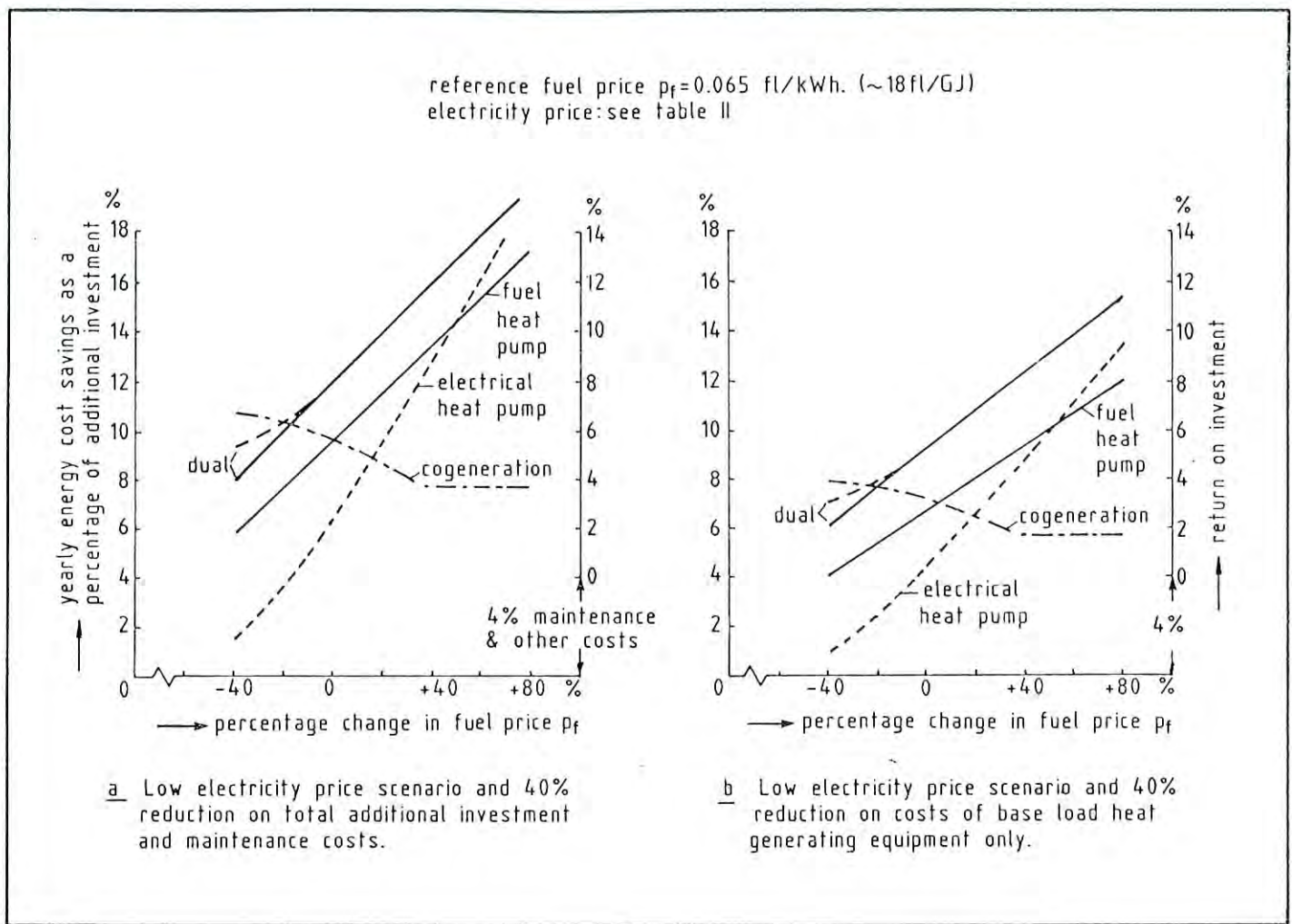


Figure 6. Effect on investment and maintenance cost reductions on return on investment

unlikely, however, that such differences will seriously affect the conclusions reached on the basis of the present results.

Combined heating and cooling

In order to avoid complicating matters, the system concepts which have been chosen for analysis purposes were in certain respects oversimplified. For instance, no consideration has yet been given to the effect of adding heat storage means or to improving matters for the dual system by a certain integration of cogeneration and heat pump installations.

Another aspect that deserves attention is the use of the collective energy system for both heating and cooling purposes. It will be evident that in this case the systems which incorporate heat pumps will have an added advantage. An interesting arrangement for such an application includes a heat pump which

is made to also operate as an ice-making machine. By providing means for diurnal storage of ice, the heat pump can operate on cheap night-time electricity to produce ice for cooling purposes during the day. In this context, the cogeneration section of the dual system could be utilized as a pure peaking unit, without heat recovery, during day-time hours in the summer.

A full analysis of such possibilities, however, falls outside the scope of this article.

Conclusions

District heating systems incorporating a combination of cogeneration and heat pump installations have better economic prospects than systems with either of these installations alone. At low energy price levels, such a dual system has the better economic characteristics of the cogeneration system and at high

energy price levels, the better cost savings potential of the heat pump system.

These basic advantages can be enhanced by a greater scope for cost reductions and performance improvements. One of the reasons is that equipment manufacturers are provided with a greater incentive for achieving such goals. Another is associated with the application of the dual system for both heating and cooling purposes.

The relative advantages of the dual system increase in the case of district heating being applied in areas with existing housing or highly insulated, newly built housing. This may prove of vital importance in determining the market penetration of this type of district heating system in the future.

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Experience from Three District Heating Installations in Sweden

Many heat pump plants were built in Sweden during 1980-1985 as part of an extensive R,D&D program commissioned by the Swedish Parliament. Part of the project is to follow up on the plants, measure results, draw conclusions, and learn for the future. It is of great common interest that this experience be made available for everyone involved. Therefore, the Swedish Council for Building Research (BFR) is writing reports on the results from the separate projects. The following is a summary of three BFR reports on three different heat pumps in district heating systems.

Plant Kullaskogen, Vaxholm (BFR Report R12:1987)

At this installation, 150 dwellings are connected to a small district heating system in Kullaskogen. There are both one and two-family houses as well as apartment houses, with a total area of 13,000 m². The dwellings are heated by a low-temperature central district heating system. Design conditions are 700 kW with 60°C delivery and 40°C return temperature. The mean heat energy demand is 1.9 GWh/year. Heat is supplied by three heat pumps and four electric boilers. A schematic of the heating system is shown in Figure 1.

Heat is delivered to the separate houses by means of a piping system. Sanitary water heating and heating water temperature control is controlled separately in each house. The heat source for the heat pumps is sewage water delivered through a 1500m long pipe, a portion of which runs along the bottom of a lake. Low-temperature heat energy is transferred from the heat source to the heat pumps by means of a plate heat exchanger and a glycol-water system.

As shown in Figure 1, the heat pump evaporators are connected in parallel and the condensers in series. The three heat pumps are factory-built standard units, each equipped with an eight-cylinder piston compressor. Heat pump no. 1 uses refrigerant R500 and nos. 2 and 3 use R22. The total capacity of the heat pumps is about 330 kW.

Test results for the period November 1984-October 1985 are as follows:

Heat delivery:	
Total	1889 MWh
From the heat pumps	1223 MWh
Energy consumption:	
Total	1169 MWh
For the heat pumps	415 MWh
For heat pump auxiliaries	124 MWh

The mean value per year of COP for the heat pump units was 2.9, for the heat pump units and auxiliaries 2.2, and for the heating plant 1.6. Use of the heat pump decreased electric energy consumption by 720 MWh/year (38%) although the projected decrease was 1200 MWh (63%).

There are three main reasons for the differences between the calculated and actual values of heat energy delivery from the heat pumps:

- The heating system's hot water return temperature was too high.
- The part-load control on the heat pump units did not work properly.
- The heat pump unit sequence control system did not work properly.

Due to these factors, the heat pump units were often stopped by the high pressure limit controllers, allowing the heat pumps to produce only about 65% of the heat energy rather than the calculated 92%.

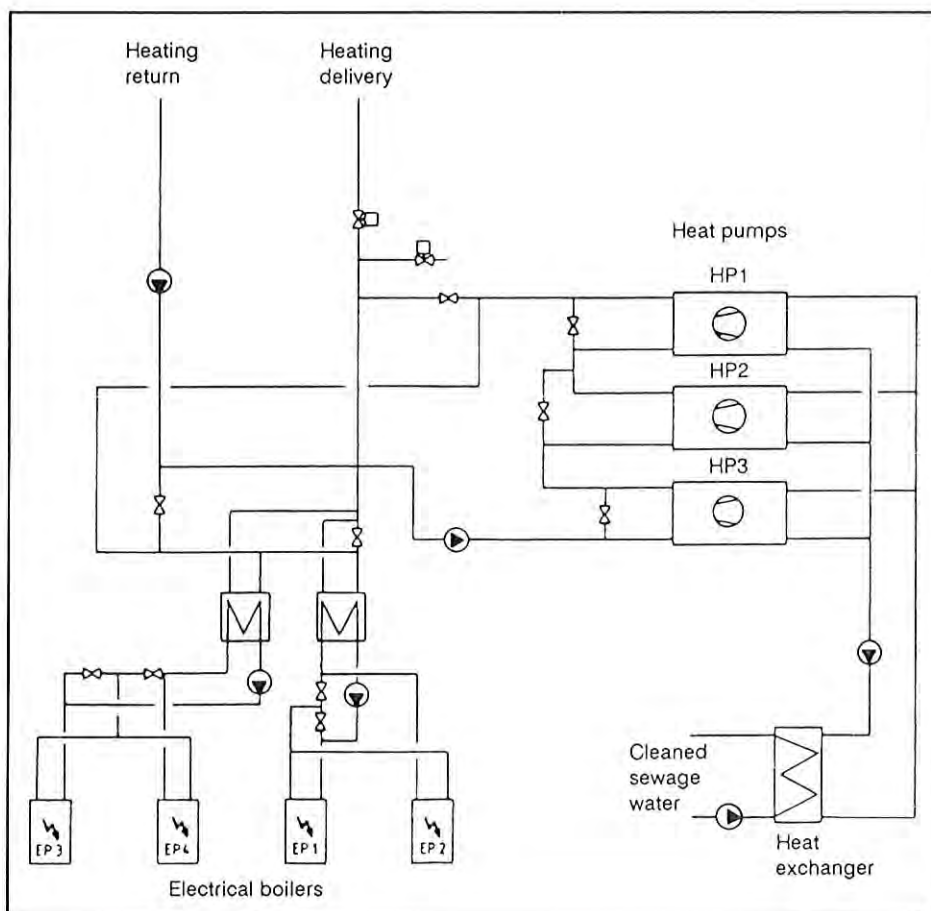


Figure 1. Heating plant schematic

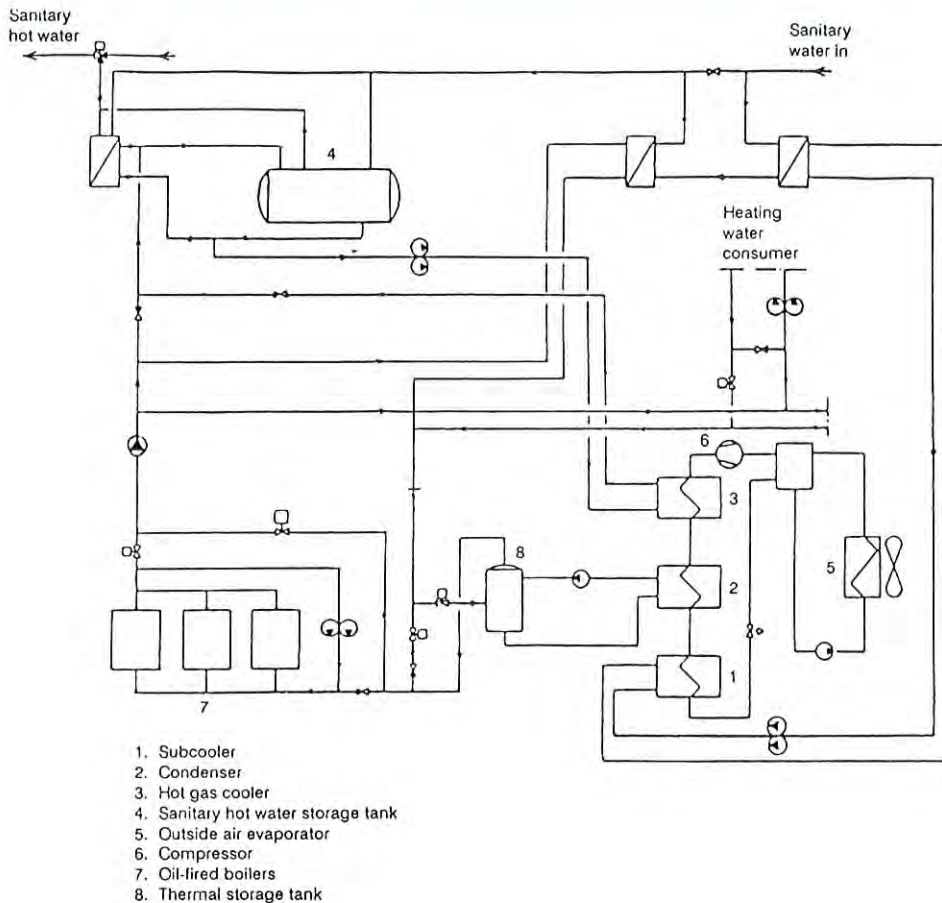


Figure 2. Heating plant schematic

Plant Brunnsberg, Varberg (BFR Report R24:1987)

The small district heating system in Brunnsberg serves 374 private dwellings and a few shops and garages. The area heated is 24,000 m². The design conditions for the heating system are 2.1 MW with 90°C delivery and 70°C return temperatures and a heat energy demand of 4.7 GWh/year. The heat is supplied by three oil-fired boilers and a heat pump installation. The heating system schematic is shown in Figure 2.

Sanitary hot water production is centralized. Incoming cool sanitary water is preheated by subcooling the refrigerant after the condenser and is superheated by means of a refrigerant hot gas heat exchanger. The heating water temperature is controlled separately in each house. There is also a thermal storage tank connected to the heat pump condenser. The heat source for the heat pump is outside air. The heat pump installation is site built. A schematic for this heat pump system is shown in Figure 3.

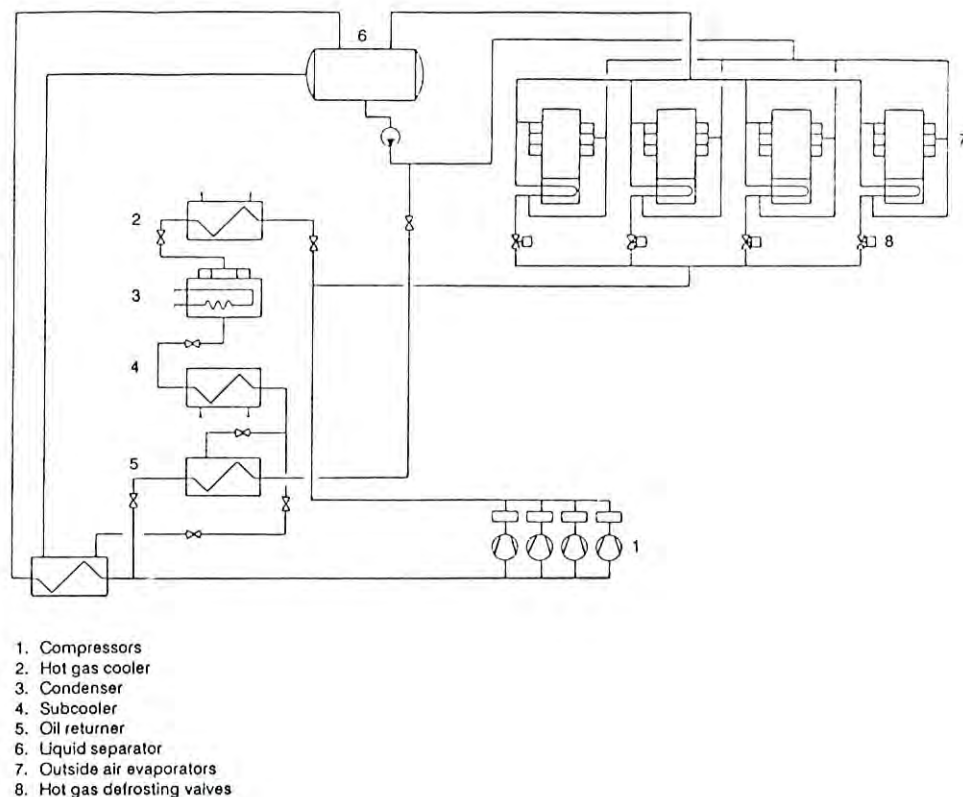


Figure 3. Heat pump schematic

There are four piston compressors working with refrigerant R12 with a combined capacity at design conditions of about 1.1 MW at +2°C outside air temperature. Capacity control of the heat pumps allows the temperature of the water leaving the condenser to be maintained at 60°-70°C, depending on the outside air temperature. There are eight flooded single-fan air-cooled evaporators with a combined cooling capacity of 710 kW at an evaporation temperature of 7.5°C connected to a refrigerant liquid separator. The system includes about 3,000 kg of refrigerant.

The evaporators are air defrosted at outside air temperatures higher than 8°C, and hot gas defrosted below this temperature. Defrosting is controlled automatically in sequence and is initiated by the pressure difference across the evaporators. In summer, when the heat load is low, the evaporator fans must not be in operation. At outside air temperatures less than -8°C (about 6% of the year), the heat pump is turned off.

Test results for the period April 1984-March 1985 are as follows:

Heat delivery from:

Heat pump		
Hot gas heat exchanger	0.34	
Condenser	2.6	
Subcooler	0.32	3.3
Boilers		1.4
Total		4.7 GWh

Energy consumption of:

Heat pump		
Compressor motors	1.2	
Fans	0.22	
Auxiliaries	0.08	1.5
Boiler (oil)		1.9
Total		2.4 GWh

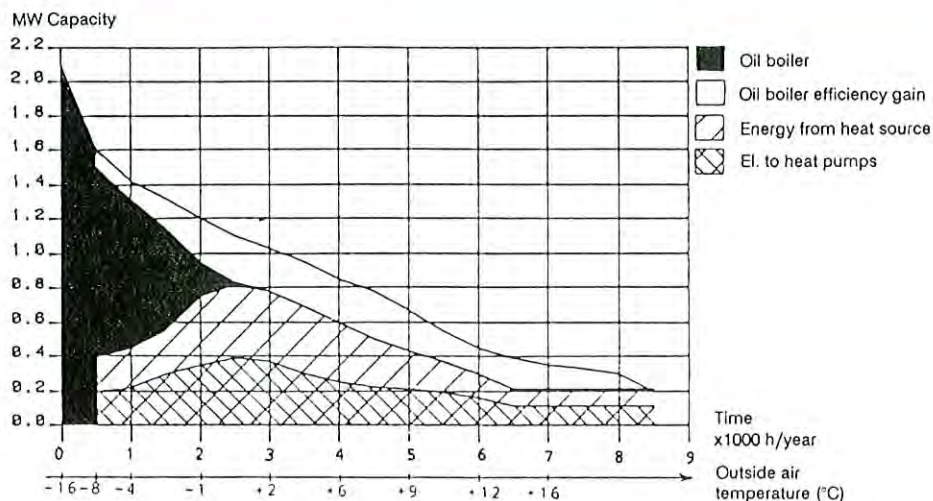


Figure 4. Energy balance and saving diagram

The COP mean value per year was 2.2, with monthly mean values between 4.2 and 1.8. The energy balance and savings are shown in Figure 4. The economics of this heat pump installation are shown below. The capital costs are calculated from an interest rate of 15%, inflation of 9% and thus a calculated interest rate of 6%. Depreciation time is assumed to be 22 years, thus the annuity is 8.3%. The investment was 5.9 MSEK, including consulting and construction costs.

Oil-fired boiler only (SEK):

Oil	(600m ³)	1,500,000
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With heat pumps (SEK):

Oil	(186m ³)	465,000
Electricity		
Fixed		66,000
Variable	(578 MWh)	111,000
	(412 MWh)	89,000
	(442 MWh)	145,000
Capital costs		489,000
Service		118,000
Total		1,483,000

Since this is an R,D&D project, the owner received a loan of 5.5 MSEK from BFR, free of interest and amortization, during the evaluation period. Thus, there are no real capital costs and the payback period is 11.5 years.

Part of the logbook from the plant is included as an attachment to the report. The air-cooled evaporators have been rebuilt to get a more even distribution of the refrigerant and better defrosting performance. There have also been other problems mostly concerning oil leakage in shaft seals, refrigerant leakages, and shut-downs due to low oil pressure and high oil temperature. The fans have also experienced many mechanical problems.

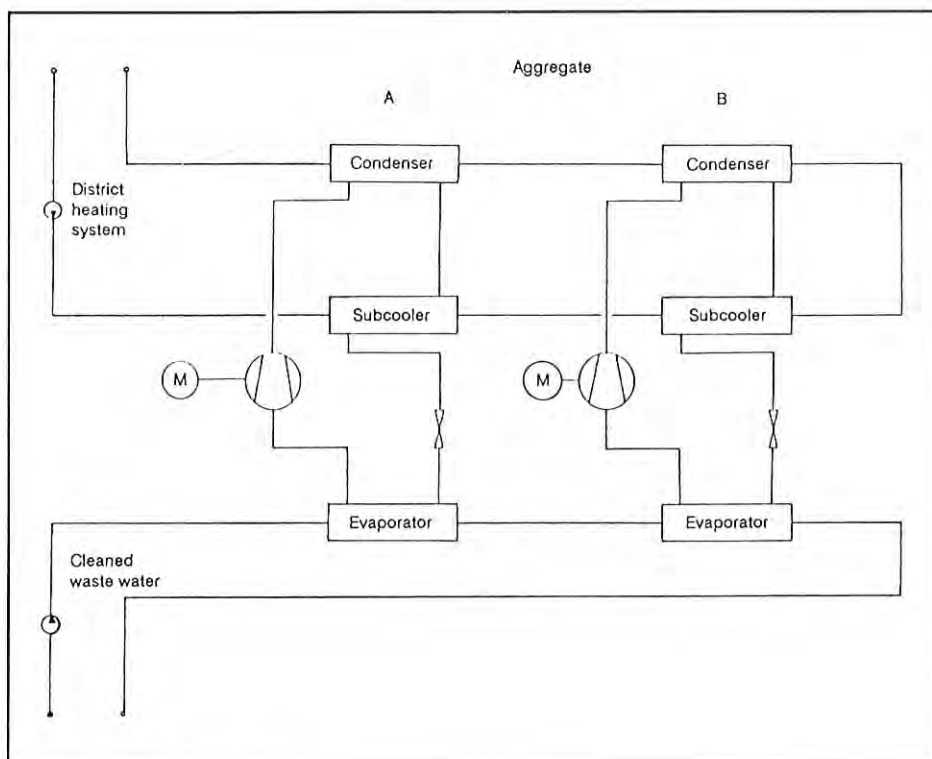


Figure 5. Heating plant schematic

Plant Loudden, Stockholm (BFR Report R16:1987)

The heat pump plant in Loudden was installed in 1981 and is connected to the large district heating system in Stockholm with a return water temperature of 40-60°C. The heat source is cleaned sewage water (10-20°C), which is delivered through a 1000m pipe. The plant consists of two turbo compressor heat pump units with subcoolers. The combined capacity is 2.2-2.6 MW, depend-

ing on temperature and heat source flow. A schematic of the heating system is shown in Figure 5.

Tests were conducted during the period December 1982-November 1983.

The mean running time for the two units was about 70%. Only 5% of the off-time was due to limitations caused by the heat source and sink, and by normal service and maintenance. The remaining 25% was due to faults in the heat

pump units, primarily gear boxes and bearings.

The use of waste water causes deposits on the evaporator tubes. These deposits decrease the water flow rate and increase the heat transfer resistance, causing the capacity of the heat pump to decrease by about 1.5 kW per day. Cleaning of the evaporator tubes is necessary at regular intervals. The optimal interval is about 50 days, based on actual energy and cleaning costs during the required 12-hour shut down.

A simple economic calculation, valid for the two test years, is as follows:

Energy production	55.8 GWh
Alternate energy price	0.2 SEK/kWh
Alternate energy costs	11.16 MSEK
Heat pump elec. energy	17.5 GWh
Elec. energy price	0.15 SEK/kWh
Elec. energy costs	2.63 MSEK
Energy cost savings	8.53 MSEK
COP mean value	3.19

The investment was approximately 12 MSEK. Calculated payback time is about 2.8 years, despite the relatively low availability of 75%, or 6500 h/year.

Conclusions

Most of the problems experienced are not with the heat pump units themselves. If there are problems with the heat pumps, they are either due to the fact that the heat pump was not a standardized but site-built unit or that it was a factory-built unit which was not originally designed for this purpose.

The conclusions drawn are as follows:

- Use high quality equipment in the auxiliary systems. Paying 1% more for the plant to get 10% higher availability is a good investment.
- Use standardized factory-built units which are designed for their intended purpose.
- Use experienced heat pump specialists during project planning.

These are the general lessons learned from these R,D&D projects. The details of the individual projects are left to the specialists to analyze.

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District Heating and Cooling of Urban Areas Case Study: Ginza 2- and 3-chome Area

Operation of the district heating and cooling (DHC) system in the Ginza 2- and 3-chome area began at the end of April 1984. This system differs from conventional DHC systems in that it is installed in an existing building in a heavily populated urban area, is fully electric, and utilizes ice for part of its thermal storage. This plant is presented as a possible configuration for new DHC plants in urban areas.

Area supplied and supply conditions

The area supplied is part of Ginza 2- and 3-chome and covers about 2.6 hectares. This is the second smallest designated DHC area in Tokyo. However, many large buildings are concentrated in this area so that the density of heat demand is high. Currently, there are three users connected to the system, comprising a total floor area of 43,000 m². Table 1 shows the chilled and hot water supply conditions and demand period. Chilled water is required year round by the department stores whereas the heating demand is seasonal. The DHC plant is designed to supply the required chilled and hot water for cooling and heating. High temperature water or steam is not supplied. By using a hot water temperature lower than 50°C, the heat loss through the piping is small when compared to a conventional high temperature system.

Features of the plant

The purpose of using DHC plants in urban areas is to reduce air pollution from individual buildings during the winter heating months. Since the present plant is all electric, air pollution

problems are completely eliminated. In addition, the hazards associated with fuel use, storage, and transport are avoided.

The heat sources used by the plant during the winter are air and waste heat from the department stores and from the transformers at a district substation. The waste heat is recovered by a heat recovery type refrigerating machine (Figure 1) and is stored in a thermal storage tank. In the future, waste heat from underground electric cables will also be utilized. Figure 2 is a schematic diagram of the system.

The use of thermal storage is one of the special features of the plant allowing a reduction in the installed refrigerating capacity and at the same time decreasing the peak electrical demand. Cost savings result from both of these factors. The size of the storage tank in this system was limited to 650m³ due to space restrictions. It can store about 20% of a summer day's peak load, thereby allowing the refrigerating machine's capacity to be 25% lower than if no thermal storage had been installed. Ice is also used for a portion of the thermal storage. Its high latent heat

	Supply	Return	Demand Period
Chilled water temperature (°C)	6 +/- 1	13 +/- 1	All year (8:00 - 21:00)
Hot water temperature (°C)	47 +/- 2	40 +/- 2	Nov. 1-Apr. 20 (8:00 - 21:00)
Pressure (kg/cm ²)	1.5 - 2.5		

Table 1. Supply conditions and demand period

of fusion allows the heat storage capacity of the plant to be about 4.5 times larger than if only chilled water storage had been used. About 27,000 m of ice-forming polyethylene coil 1 cm in diameter is installed in the storage tank. Brine at 5 to 10°C is circulated through the coil to form ice on its outer surface. With an ice thickness of 30 mm on the coil, about 60 m³ of ice can be stored. The brine refrigerating machines are equipped with two evaporators, one to refrigerate brine at night for forming ice, the other for producing chilled water during the day. The ice-making operations are performed at night since the outdoor temperature is lower and off-peak electric rates are available. This helps improve the system's efficiency and reduce operating costs. Nevertheless, the operating and capital costs for an ice storage system are higher than for a chilled water system. For this reason, ice storage may be said to be suitable for applications where thermal storage space is limited.

Features of the equipment

Due to climatic conditions and building type, the peak load for cooling is about three times larger than for heating. A portion of the hot water requirements is met by the heat recovery refrigerating machine which at the same time also supplies chilled water. Additional air source heat pumps insure an adequate supply of hot water during the winter months. Figure 3 shows schematically how heat is recovered by the system. The refrigerating machines (turbo type and brine refrigerating machines) were selected based on the remaining cooling load which the heat pumps could not meet. The selection process also took into account the thermal storage capacity.

In this plant the chilled and hot water supply pumps are controlled in sets of three. In each set, one of the three pumps is equipped with an inverter to allow speed control resulting in a reduction in power requirements during part-load operation.

The thermal storage system consists of three tanks. One is used exclusively for chilled water while the other two can be used for either chilled or hot water. The ice-forming coils are installed in the chilled water and one of the chilled/hot

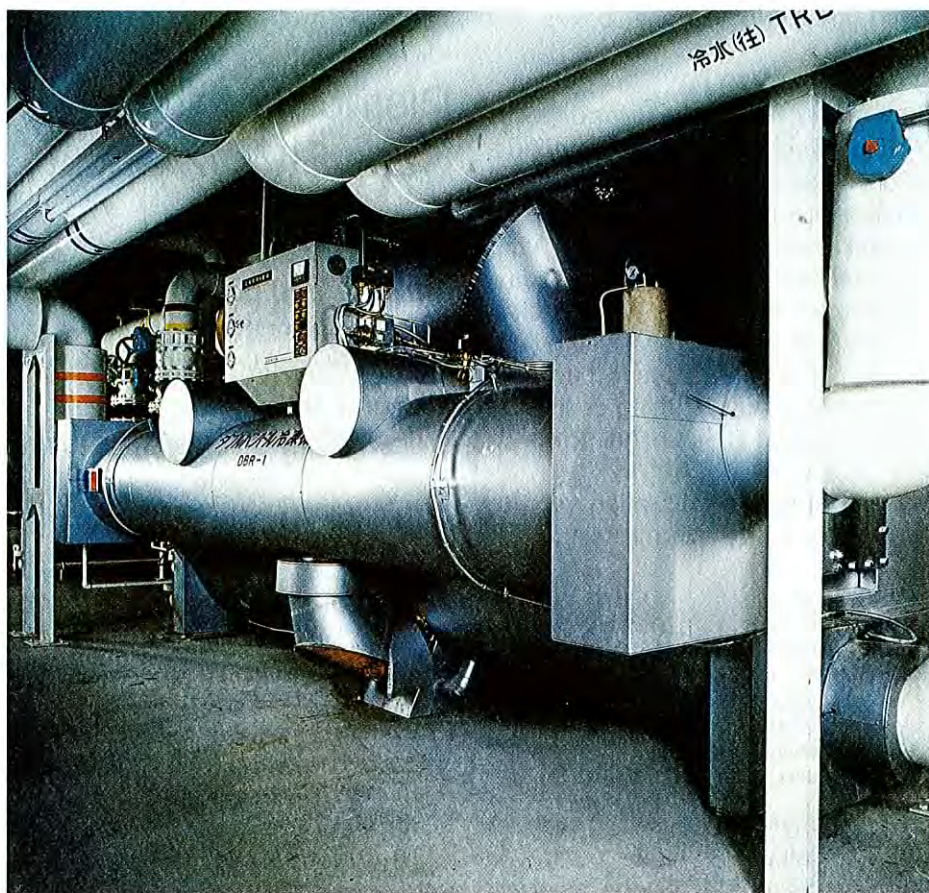


Figure 1. Heat recovery type refrigerating machine

Name	Qty	Main Specifications
Heat pump	1	Air-source heat pump (Screw type, 340 kW) Heating capacity: 640,000 kcal/h (0°C) (740 kW) Cooling capacity: 250 RT (32°C) (880 kW)
	1	Air-source heat pump (Screw type, 190 kW) Heating capacity: 440,000 kcal/hr (510 kW) Cooling capacity: 160 RT (563 kW)
Refrigerating machine	1	Heat recovery type refig. machine (Turbo type, 290 kW) Heating capacity: 1,002,000 kcal/h (44/49°C) (1.16 MW) Cooling capacity: 250 RT (10/5°C) (880 kW)
	1	Turbo type (320 kW) Cooling capacity: 400 RT (1.4 MW)
	2	Brine refrigerating machine (Reciprocating type, 84 kW x 2) Cooling capacity: -10°C 90 RT (ice) (317 kW) +5°C 175 RT (chilled water) (615 kW)
Cooling tower	1	Square type, ultra low noise 700 RT (2.5 MW) Cooling water: 9,100 t/min.
	1	Square type, ultra low noise 400 RT (1.4 MW) Cooling water: 5,200 t/min.
Heat storage tank	1	Chilled/hot water tank: 300 m ³ , with ice-making coil
	1	Chilled/hot water tank: 180 m ³
	1	Chilled water tank: 100 m ³ , with ice-making coil
Chilled water supply pump	3	Centrifugal pump 4,200 t/min. (One of the three comes with an inverter)
Hot water supply pump	3	Centrifugal pump 2,250 t/min. (One of the three comes with an inverter)

(RT = tons of refrigeration)

Table 2. Main units

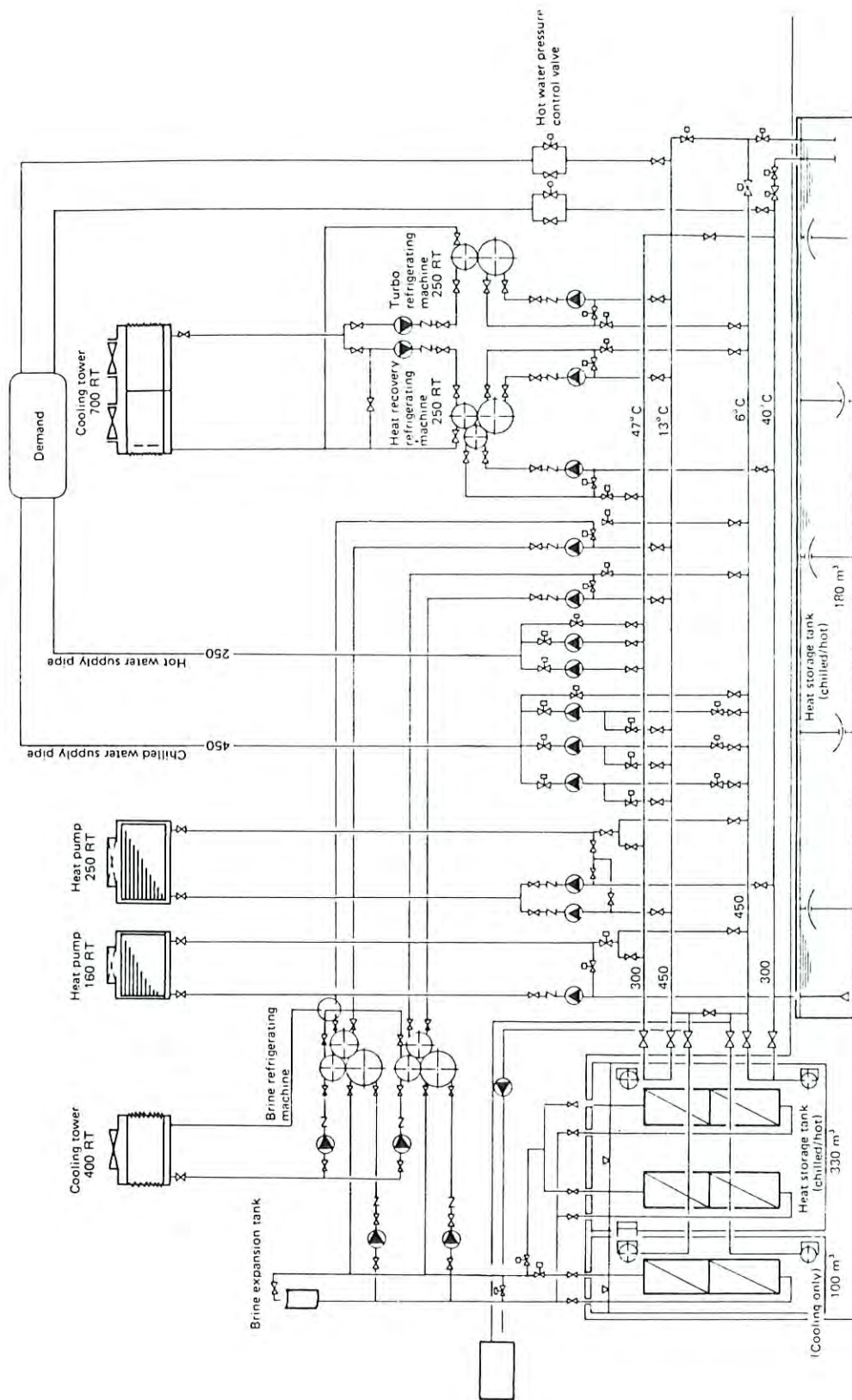


Figure 2. System schematic

water tanks. Table 2 summarizes the specifications of the plant's equipment.

The power supply for the plant consists of two 1000kVA transformers installed to convert the 22kV from the electric supply network to 415V and 240V. The plant's water supply and return lines consist of 450mm diameter pipe for supplying chilled water and 250mm diameter pipe for supplying hot water. Both pipes are about 160m in length and run either underground or in special trenches.

Central monitoring for the plant is done by a microcomputer. Its functions include generating daily reports, general monitoring, manual and automatic control of system components, electric demand monitoring, and power factor improvement.

Installation

Since the plant was installed within an existing building, both the layout of the machines and the installation work were restricted in various ways. In this area of Tokyo, piping installation was especially difficult. Below the roads lie electric, gas, and water supply lines as well as drainage pipes and telephone lines. These extend over the road's width as well as vertically. Since all of these pipes are quite important, disruptions had to be avoided. Installation was therefore carried out at night. In addition, the securing of routes for the piping, any necessary changes, the replacement or protection of existing lines, etc., was carried out to comply with each utility's standards. For the above reasons, piping installation work

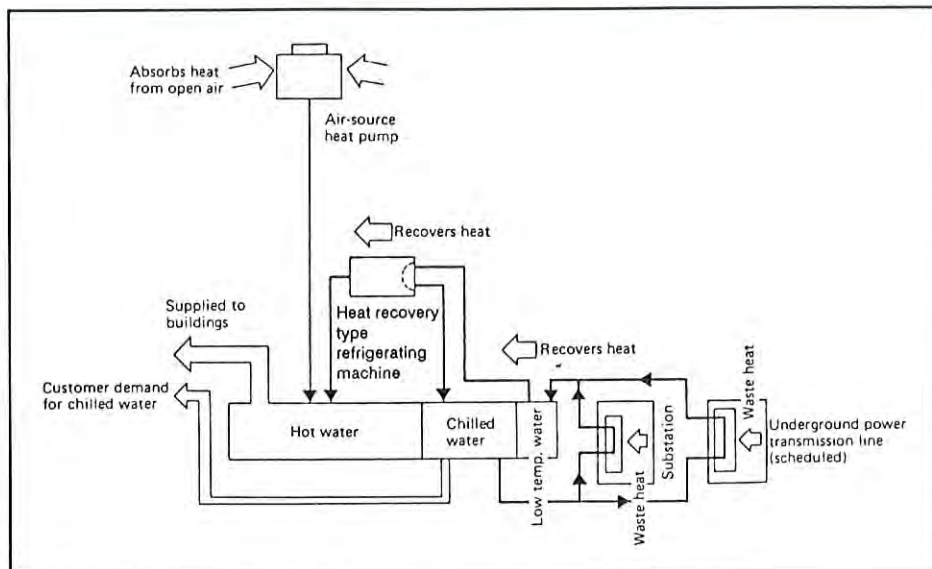


Figure 3. Conceptual drawing of waste heat recovery

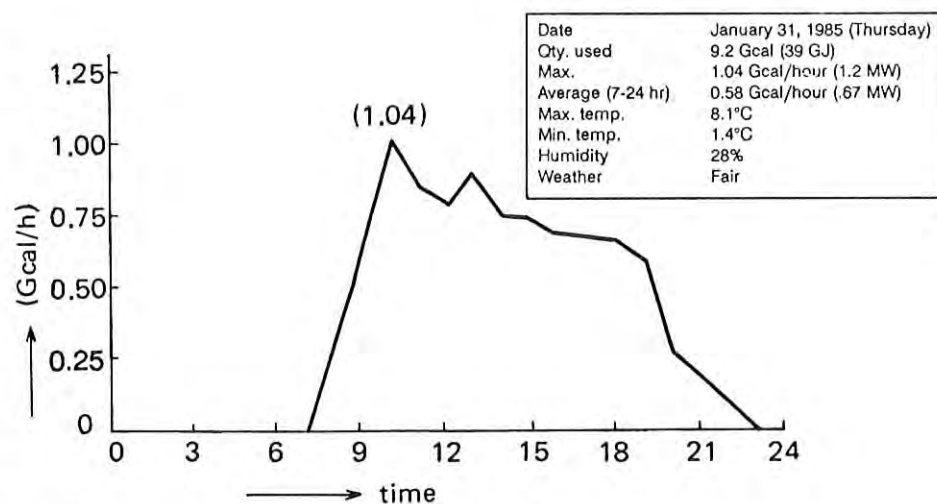


Figure 5. Daily load curve for high temperature heat

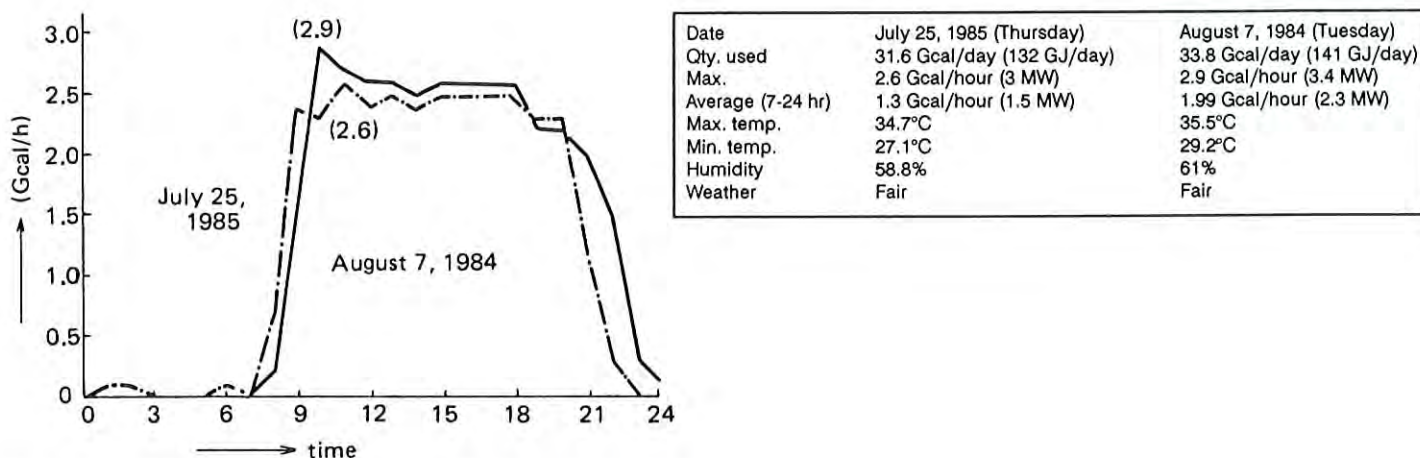


Figure 4. Daily load curve for low temperature heat

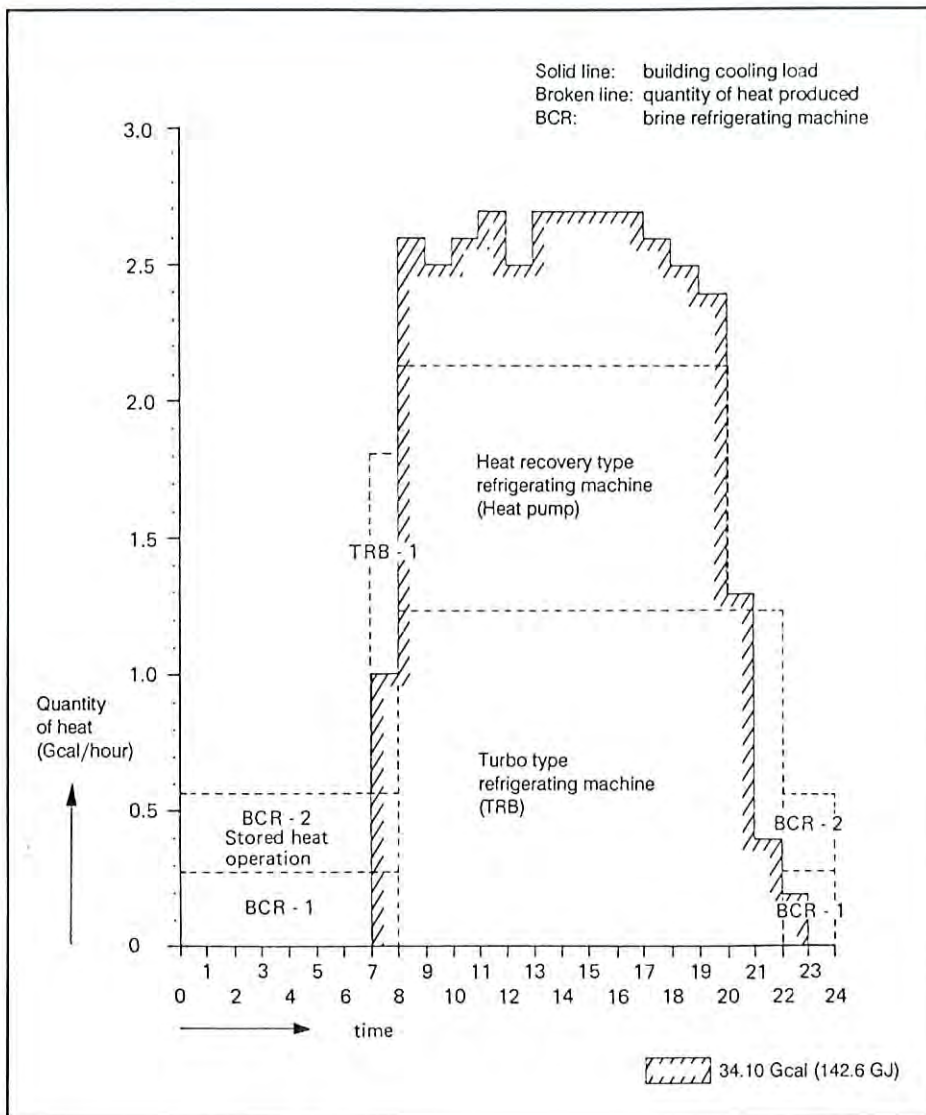


Figure 6. Machine operation pattern in summer

for DHC systems in urban areas is an important factor which influences costs and construction scheduling.

Operating results

Figures 4 and 5 show daily load curves for cooling and heating for two summer days and a winter day. A peak usually occurs during the morning hours and, as seen in these figures, it is relatively small. Annual demand has not yet reached the planned values so that ice storage is currently being used only a few days of the year. As an example, Figure 6 shows the cooling load and machine operating pattern for a representative summer day. In this case, it was most efficient to store chilled water at 5 to 10°C by use of the brine refrigerating machines. During winter operation of the system, the heat recovered from cooling the department store and the transformers at the district substation has provided about 20% of the total heating demand.

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Advanced Heat Pump Systems Foundation for Planning 1987/88-1989/90 at the Swedish Council for Building Research (BFR-Report G21:1986)

The Swedish Council for Building Research (BFR) is a body that undertakes research in the field of buildings; it is subordinate to the Ministry of Housing. The Council is responsible for overall planning, coordination, appraisal, and financing of research and development work in the building sector. The BFR itself does not actually carry out the research work. The following is a summary of BFR-Report G21:1986.

Introduction

This report is produced by a group responsible for technical and economic evaluation of ongoing projects and is to be a foundation for decisions on continuing research.

The price for electric energy is low in Sweden today and heat pumps driven by electric motors are of primary interest. However, electric energy prices are expected to increase when the nuclear power stations, which are responsible for about 50% of Sweden's electric power generation, are finally withdrawn in the year 2010. Other types of heat pumps will then be of interest, too. In a

study of national energy systems for the future, different kinds of heat pumps must, therefore, be considered, and they must compete with solid fuels, cogeneration, etc.

It is assumed that the price for electric energy will increase 50-60% within the next 15-20 years. If the oil price is assumed to be unchanged, then the electricity-to-oil price ratio will be about 1.5. From the user's point of view, this is optimal for the use of heat pumps. At a lower price ratio of about 1.0, heating directly with electricity is more economical. At a price ratio of 2.0 and above, combustion-engine-driven heat pumps, cogeneration systems, and similar systems are of highest interest.

It is obviously a difficult matter to judge the future potential for heat pumps in Sweden. However, in future scenarios considered in a BFR study, heat pumps will be of considerable importance. They will be used in group centrals for the heating of buildings, in heating centrals in industries, and in smaller district heating systems.

Background

Heat Pump Installations

The goals for past work have continuously changed. They have evolved from the development of new heat pump systems and the support of installations, which could immediately decrease oil consumption, to the evaluation of the results from plants installed and support of development of conventional heat pumps systems to, finally, R&D on advanced heat pump systems.

All this occurred very quickly. At the end of 1985, there were about 130,000 heat pumps in operation, producing some 5-6 net TWh of heat energy per year. An additional 1-2 TWh will be produced by heat pumps installed in 1986-1987. Installations in single-family houses are most numerous. Based on installed capacity, heat pumps in district heating systems are of the same magnitude. Costs and performance for common, well-designed, and reliable installations are shown in Table 1.

	Annual Average COP	Investment SEK/kW	All Energy Prices SEK/kWh ¹
Single-family houses	2.0-2.5	4,000-7,000	0.15-0.25
Group centrals			
100-300 kW	2.0-3.0	4,500-8,000	0.15-0.20
300-3000 kW	2.0-3.0	3,500-6,500	0.15-0.20
Multi-family houses with mechanical ventilation	2.5-3.5	4,000-8,000	0.15-0.25
System with sewage waster > 2 MW	2.3-3.5	1,500-4,000	0.10-0.20
System with waste water > 2 MW	2.3-3.5	1,500-4,000	0.10-0.20
(6% interest rate and 15 years depreciation) ¹			

Table 1. Cost/performance data of Swedish installations

Natural Heat Sources and Environmental Pollution

Parallel to the development of heat pumps, research has been done on natural heat resources, their locations, and the available energy potential in the ground and in water. Research on the ecological effects of energy emission and leakage of heat-carrying fluids have received high priority.

As a result, rules for design of heat source systems are almost completely available. As to environmental impact, the problems have generally been small, except for the question of the influence of CFC refrigerant leakage, the answer for which is not yet known.

Present Situation

General guidelines from the Swedish agency for environmental protection (SNV) are available concerning emission of heat from the ground, from water, and from air (SNV PM 1833). Codes concerning allowable indoor and outdoor noise levels are also in effect (SBN 1985, SNV 1978:5) and several communities have their own additional rules.

The economic support from government for heat pump installations was discontinued in 1984. Thus, the capital costs have almost doubled. The price for electric energy has increased by 10% and the oil prices have decreased by 30% since the beginning of 1986. The profitability of new heat pump in-

stallations for heating of buildings is, therefore, bad and relatively few new installations are being made.

There are 1.6 million single-family houses in Sweden, 1.0 million of which have their own heating systems, 0.1 million are connected to some kind of district heating system, and 0.5 million are heated by electricity.

At the end of 1985, there were 103,500 small heat pumps in single-family houses, 3500-4000 heat pumps with a 25-3000 kW capacity in apartment houses and group centrals, and about 100 heat pumps with a 3-75 MW capacity connected to district heating systems.

Due to their rapid development, the availability, performance, and thus, profitability, are less than calculated in many plants. The potential for further savings of running and service costs in these installations is considerable.

The obvious need for training and information about heat pump technology is being met. BFR and the Swedish State Power Board have initiated a training program with special courses for various people, such as homeowners, consultants, installers, operating and maintenance personnel, inspectors, planners, teachers, etc. More than 3,000 people have participated in these courses.

Many projects, mostly long-range, supported by BFR are going on at various universities.

Development Possibilities in Sweden

Any "technical jumps" are not foreseen in heat pump technology. Higher COP's for combustion-engine-driven heat pumps for heating of buildings are not to be expected within the next decade. Absorption heat pumps driven by cheap industrial waste heat can possibly be applied favorably. Any new heat sources which would radically change the situation are not foreseen.

The most profitable R&D is, therefore, probably on absorption heat pumps and on design and installation of electric heat pumps for higher overall COP.

The heat pump market will probably decrease. However, the use of heat pumps is very effective for decreasing oil consumption and is particularly effective concerning utilization of electric energy. Improvement of financing possibilities and/or a large increase in oil prices are necessary to highlight these advantages.

Knowledge on service and maintenance of heat pump installations must be increased. An increase of profitability of at least 10% is often possible through experience. There is a likely possibility for specialized companies to take care of this, especially for larger installations.

International Development Possibilities

The international business and the exchange of technical knowledge in the area of heat pumps are not only a matter of the heat pump itself. The availability of different kinds of energy, energy prices, and the traditional methods of designing systems for heating and ventilation of buildings and heating of sanitary water are of equal importance and vary considerably between different countries. National and local energy supply systems are also growing and are becoming more integrated and complex. On top of that, the consciousness of environmental problems connected with the use of energy is rapidly growing. The international outlook is, therefore, somewhat complicated.

As to smaller heat pumps, the large international manufacturers are not concentrating on heat pumps designed for cold climates such as Sweden's. Therefore, there is the possibility that Swedish assemblers will buy cheap standard components on the international market and build applied heat pumps for cold climates such as Sweden's. The development of systems and installation techniques must be made where the heat pump is to be used.

Resources Available for R&D

The heat pump itself has been paid comparatively little attention to, calculated in terms of governmental support. This work has been done mainly by the manufacturers themselves. One of the reasons for this is that the expertise is concentrated in a rather small group of technicians.

There are fairly well-equipped laboratories available at universities, but there is a need for a larger number of competent research guiding personnel. Some large laboratories have been built during the last five years, as well as the National Testing Institute in Borås, at the State Power Board in Älvkarleby, and at the bigger industries in this branch. The limitations are mostly on competent personnel for R&D. Sweden also takes part in international R&D by being a member of the International Institute of Refrigeration (IIR) and the International Energy Agency (IEA).

Continuation of Governmental Support

Although many heat pumps have been installed, heat production by means of heat pumps is still comparatively low, and the market is decreasing. Considering the possibilities of decreasing oil consumption and of utilizing electric energy effectively, as well as environmental aspects in the future, governmental support to the heat pump branch is highly motivated.

The support should be both direct, such as financing demonstration plants, investing in production plants, training

installation and service personnel, and indirect, such as supporting traditional R&D, promoting education on all technical levels, providing information to communities, building owners, etc., and supporting international cooperation efforts.

A consistent policy is very important.

Experience gained from the first generation of heat pumps must be taken into account, organized, and considered for the second generation.

Alternatives for houses heated directly by electricity will need to be addressed when electricity prices increase. Cooperation of all people involved in a heat pump installation and the work on new rules and practices applicable to the new technical presumption, such as contract formulations, guarantees and testing procedures, must be supported.

R&D Plan

The potential for development in heat pump technology is considerable. R&D is recommended to be concentrated on projects in the following areas:

- Increasing of performance
- New heat pump processes such as alternative drives, absorption heat pumps and advanced systems with electrically driven processes
- Improvement of conventional installation techniques for control systems, optimization of operation strategies and system dimensioning, and development of components, especially evaporators
- New applications, especially for industrial processes
- Continuation of ongoing work for application of heat pumps in multi-family houses and smaller group centrals
- Noise and CFC leakage

**Åke Bratt, IEA Heat Pump Center, Karlsruhe, Fed. Republic of Germany*

LITERATURE REVIEW: DISTRICT HEATING

Anonymous. **Use of Decentralized Heat Pumps With a Common Heat Reservoir.** (Anvendelse af decentrale varmepumper med fælles anlæg som varmereservoir.) Teknologisk Instituts Forlag, EFP-84, Energiministeriets Varmepumpeforskningsprogram, Rapport 39, Denmark, 1986. (Danish)

Based on an extensive future expansion of collective district heating systems for smaller villages in Denmark, Cowiconsult Consulting Engineers and Planners AS has prepared a theoretical analysis of the economic/technical consequences of using decentralized heat pumps with a common heat reservoir in future district heating schemes. The system consists of a common heat reservoir, e.g., a groundwater drilling, an extreme low temperature distribution pipe system made of plastics and decentralized heat pump units at the individual consumers. The analysis shows examples of alternative configurations of systems and lists technical and economic characteristics for the different production, distribution, and consumption parts of the system. As production units, solar collectors, soil-tube collectors, groundwater drillings, and heat storages for waste heat are investigated. The analysis concludes with calculations of the net present value over a 20-year period (NPV20) for the alternative systems in focus. Individual gas oil boilers and a traditional district heating system based on heavy fuel are used as reference.

Lawaetz, H., and M. Lange. **Solar Collectors in a Collective Renewable Energy System.** NORTH SUN '86, International Conference, Copenhagen, Denmark, p. 65-70, 10 June 1986. (English)

A district heating system based on renewable energy sources is established in a small town consisting of 100 houses. The heat is produced by collectors, a heat pump with energy absorbers and embedded coils, a straw burner, and an oil burner. Additionally, two small windmills have been erected to produce electricity for the operation of the heat pump through the public grid. The operation is very untraditional as the temperatures of the district heat-

ing system are periodically very low, consequently special house installations must be designed. The solar collector area is 300m² and is placed on the ground in front of the district heating station, facing south. The annual output from the solar system is calculated to be 290 kWh per year in the Danish test reference year (TRY), with an inlet temperature of 60°C.

Saari, M. **Heat Pump in a Block of Flats With Warm Air Heating.** (Laempöppumppu ilmalaemmitisessae asuinkerrostalossa.) Valtion Teknillinen Tutkimuskeskus, Espoo, Finland, December 1986. (Finnish)

Energy and water consumption as well as the operation of the HVAC-system has been studied in a building complex located in Hyvinkäeae. The building complex consists of five separate buildings which have a common district heating plant. In addition, each building is supplied with an exhaust air heat pump. The heat distribution from the heating plant to the buildings is done by a one-water system. In each flat there is a warm air heating unit. The volumetric (building volume 6975 m³) heating energy consumption was 45.1 kWh/m³/a. 39% of the total heating energy was produced by district heating. The mean annual COP of the pumps was 2.1. According to measurements, the low value of the mean COP is due to insufficiently operating evaporator, high pressure drop in refrigerant cycle and high condensation temperature. In winter, there have been some complaints about low room air temperature. The rate of complaints was significantly reduced by balancing the warm air heating units. To some extent, there were also complaints about the noise caused by the warm-air heating units. As a whole, the HVAC-system was working properly. During the heating period, the heat pumps were operating continuously and the additional heating power was taken from the district heating network to feed the total heat demand of the buildings. During the summer period, the heat pumps were sufficient to produce all the heating power needed.

Iihola, H., T. Ala-Peijari, T. Tinell, J. Kaiharju, and M. Heikkiläe. **Groundwa-**

ter as Energy Source of Heat Pump in District Heating in Vieremae, Forssa. (Pohjavetäe kaeyttäeväe laempöppumppu Forssan Vieremaen alue-laempöeverkossa.) Imatran Voima Oy, Helsinki, Finland, December 1986. (Finnish)

The heat pump plant was connected in series together with a boiler using heavy fuel oil. The plant and the boiler are situated at a short distance from the district heating network. The heat pump heats the district heating water up to a maximum of 75°C and the oil boiler raises the water temperature to a level required by the outdoor temperature, the maximum temperature being about 100°C. The heat pump needs groundwater of some 20 kg/s. The water is returned after cooling from the natural temperature of 6°C to 4-2°C back to the same aquifer. The intake well for water and injection wells are situated at a distance of about 1 km from the heat pump plant. Final test operation of the plant was started in May 1985 and commercial operation in November 1985. Two water intake wells had to be built during the construction, as the iron c. 2 mg/l in the water pumped from the first well caused clogging in injection wells. So far, the first well has been used as an injection well for cooled water, as it lies at a better distance than the actual infiltration wells. The heat pump plant reduces the use of oil by some 350 tons/year. Sulphur oxide emissions will be decreased by approximately 17 tons and nitric oxide emissions by c. 4 tons/year in the Vieremae area. The heat pump operation has not caused any changes in the groundwater. The experiences that have been gained show that groundwater can be utilized by a heat pump connected into district heating networks. However, in order to keep the costs moderate, the aquifer and heat consumers must be at a short distance from each other. During the autumn of 1986, price relations between heavy fuel oil and electricity indicated that a groundwater heat pump was not economically viable, even if the experiences of this project were utilized and extra costs were avoided.

Lund, P., J. Routti, S. Peltola, R. Keinonen, B. Oestman, and L. Flyer. **Energy Consumption in Kerava Solar Village 1983-1985. The Final Report.** (Keravan aurinkokylaen energian

kaeytoen tutkimus 1983-1985. Loppuraportti.) Ministry of Trade and Industry, Helsinki, Finland, 1986. (Finnish)

Kerava solar village (KSV) is the first Finnish community solar heating system with seasonal heat storage. It is also the largest experimental building area in Finland. The overall investment costs of the heating system have been 6.6 million FIM (U.S. \$1.00 = 6 FIM, 1985) and total project costs including building construction were 20 million FIM. Energy flow measurements were performed during 1983-85. The Kerava solar village comprises 44 detached houses. For seasonal storage, a rock cavern of 1,500m³ filled with water and 11,000m³ of vertical pipe storage in rock are employed. The measured yearly heat recovery efficiency of the seasonal heat storage is 85%. The storage is charged by heat from flat-plate solar collectors with a total surface area of 1100m². The yearly solar yield has been 250-280 kWh/m². A heat pump is used for efficient storage. Some problems in the heat pump's operation have been encountered, thus decreasing the utilization of the stored solar heat. The new energy technologies employed have satisfied about 40% of the yearly heat consumption in the village. The fraction of stored active solar heat in the energy balance has been slightly over 30%. Because of its experimental nature, the cost of delivered heat in KSV is about 1 FIM/kWh. Due to large governmental subsidies, the homeowners pay about the same as in the case of conventional heating sources. By system optimization and larger heat load, the cost of heat could be reduced significantly.

Huhtinen, M., J. Linnavuori, L. Saukkoripi, L. Maekelae, B. Landtman, and S. Sirvioe. **Heat Pumps in District Heating. The Potential of Heat Pumps Utilizing Waste Heat from Municipal Sewage Water Treatment Plants and Industrial Plants in Finland.** (Kaukolaempoepumput. Yhdyskuntien ja teollisuuden jaetelaempoeae kaukolaempoeoen hyoedyntaevien laempoepumpujen potentiaali Suomessa.) Ministry of Trade and Industry, Helsinki, Finland, 1986. (Finnish)

The purpose of this study was to determine the prospects for large heat pumps in district heating in Finland.

Waste heat from industrial plants and from municipal sewage water treatment plants are considered as possible heat sources. The study was carried out on a case-by-case basis for all localities where district heat load and heat source of at least 0.5 MW heat pump application existed. Feasibility of heat pump installations for district heating is affected by local conditions such as district heating load, utilization time, temperature level, type and size of heat source, distance between heat source and DH-distribution network, investments required in heat source, competing heat production systems and existing heating capacity. Information concerning these local conditions was collected by an inquiry sent to all communities and to 200 large industrial plants. Detailed economic analysis was conducted for 146 sites that can utilize waste heat sources of industrial plants (ca 60) and/or sewage water treatment plants (ca 130). This analysis was performed with a computer program that determines optimal heat pump systems for DH-network and heat source pairs. The feasibility study pointed out that the potential for heat pump installations for district heating is very sensitive to the future price level of electricity. In the economic analysis, it was assumed that heat pumps can be shut off 500 hours annually during peak loads of electricity network. In this case, the best estimate for electricity price is 180 mk/MWh. Using that electricity price and fuel prices of October 1984, the potential in the 1990 situation is estimated to be 97 heat pump plants totalling 370 MW.

Mucic, V. and B. Kalkum. **Generating Steam From a District Heat Return Pipe Using a Dual-Agent Compression Heat Pump.** *Fernwaerme Int.*, v.15(1), February 1986, p.34-6. (German)

The authors investigate the marginal conditions under which it can be economical to generate steam year round using a return district heating pipeline as heat source and with the aid of a dual-agent compression heat pump. The costs of heat generation in relation to the energy costs are given for steam up to a pressure of 3 bar and at return flow temperatures of 55 and 70°C. An electrical and a combustion engine are used as alternative sources of power for the heat pump.

Siefert, E., and R. Zimpfer. **Including An Electrical Heat Pump for Heating an Open-Air Swimming Bath in a District Heating Concept. Fundamentals Shown by a Specific Case.** *Solarer Schwimmbadkongress. Statusbericht*, pp. 65-73, 1986. (German)

In the context of an energy supply concept for a district heating center situated near the open air swimming bath at Schlitz in Hessen, investigations are at present being carried out as to whether the open air swimming bath should be used as a buffer store and the heat demand of the open air swimming bath should be covered by cogeneration plants (BKHW). First estimates of economy show that the heat supply of the open air swimming bath can be represented economically by a second BKHW module. A precondition for this is a supply of natural gas for the BKHW at 4.2 Pfennig per kWh for the natural gas. The inclusion of the heat pump in the district heating concept can be done via the return pipe of the heat distribution network.

Anonymous. **Combined Heat and Power Production in Local Communities. Production, Potential, Prognoses.** (Kraftvaerme i kommunerna. Produktion, potential, prognos.) Statens Energiverk, Stockholm, Sweden, February 1986. (Swedish)

The Swedish government has commissioned the National Energy Administration to study the development of industrial backpressure and combined heat and power (CHP) production in district heating (DH). At present there are CHP plants ranging from 7 MW to 250 MW of electricity in 19 municipalities. Most of the plants are municipally owned, while others are owned partly or entirely by power companies. District heating deliveries in 1984 in these municipalities totalled 18.7 TWh. It is estimated that in 1990, district heat production in these municipalities will amount to at least 22 TWh, 2.8 TWh of which is derived from combustion of waste, 5.0 TWh from heat pumps and 1.8 TWh from industrial waste heat. Summing up, it may be said that the possible CHP production around the year 1990 in the existing CHP plants in the 19 municipalities is 3.6-6.2 TWh of electricity and the CHP potential in nine large municipalities without CHP plants is 1.5-2 TWh of elec-

tricity. The economic possibilities for further expansion of combined heat and power are difficult to survey. The cost pattern with regard to investments may be defined with a reasonable degree of accuracy, but it is difficult to specify the future revenues from the production of electricity. The National Energy Administration will be investing SEK 15 million in research and development work concerning small combined heat and power plants during the period 1984-1987. The Administration has recently awarded a grant for a prototype and demonstration facility for 4 MW of electricity and 12 MW of heat in one municipality.

Anonymous. Proposed Haymarket District Heating and Cooling System, Lincoln, Nebraska. Phase 1, Feasibility Report. City of Lincoln, Nebraska, and Lincoln Electric System, USA, October 1986. (English)

This study attempts to examine the prospects of serving a 16-block down-

town area with a new district heating and cooling system. The system would deliver hot water and chilled water to the various buildings in the district who choose to be connected. The core of the system would be large water-source heat pumps which would utilize wastewater effluent from the Theresa Street Plant. By incorporating the heat pump into this system, the wastewater effluent becomes the primary source for heat extraction and rejection for the district system. The advantages to the building owner include: (1) hot water and chilled water that can be purchased at a rate comparable or below that of operating their own boiler and chiller; (2) the need for a boiler and chiller are eliminated which substantially reduces capital cost of equipment and ongoing maintenance costs; (3) the elimination of related costs such as water treatment, insurance and maintenance labor; (4) the gain of space in the building which would be occupied by the equipment that could be utilized for alternative purposes.

NEWS BRIEFS

Study Tour to Japan

The editorial staff of *Clima Commerce International* magazine is organizing a study tour to Japan and Singapore on the occasion of the 1988 Japan Heating, Ventilating, Air Conditioning and Refrigeration Exposition and the 1988 JAR International Symposium on Recent Developments in Heat Pump Technology. The trip will take place March 8-18, 1988, and will include tours of Japanese HVAC production plants, pilot projects, solar and heat pump systems, and building HVAC installations. For more information, contact Frau Gilb, Promoter Verlags- und Foerderungsges. mbH, Postfach 21 10 53, D-7500 Karlsruhe 21, Federal Republic of Germany.



Selected Book and Report Reviews

Review of the most recent publications on heat pumps

Impact of Technical and Economic Boundary Conditions on Technologies for Rational Energy Use in Industrial Nations - A Comparison. Final Report. Jochem, E., H. Grupp, O. Hohmeyer, F. Jaeger, W. Mannsbart, H. Groener; Bundesministerium fuer Forschung und Technologie, Bonn, Federal Republic of Germany, September 1986.

In some industrialized countries, new technologies for energy conservation and the use of renewable energy sources have been introduced on a wider scale than in the FRG in the last ten years. For example, district heating is used three to five times more in Denmark and Finland, Denmark has three times as many wind energy converters and France has twice as many electrical heat pumps as the FRG. This study considers the causes of the differences of market diffusion for several technologies producing electricity and heat. A considerable part of the differences in application of these technologies can be explained by the influences of factors that result

from climate, industrial structure and historical development that cannot or should not be changed for energy economic reasons. However, a number of opportunities exist that determine the diffusion rates of new technologies and have apparently been used more intensively in the other countries than in the FRG. The report proposes some measures that could be taken by organizations representing the groups involved, the federal legislative or state governments (e.g., simple contract conditions for private electricity generators, improved information and consulting for small- and medium-sized businesses, support for cooperative generating plants, simplification of electricity sales from private generators to neighboring businesses, personnel increases and interdisciplinary enhancement of the public energy administration). 371 pages, 273 refs, 88 tables, 26 figures.

Symposium Proceedings - Large Scale Applications of Heat Pumps. BHRA, Cranfield, Bedford, England,

ISBN 0 947711 23 6, December 1987.

BHRA, The Fluid Engineering Centre, held the 3rd International Symposium on the Large Scale Applications of Heat Pumps in Oxford, England, March 25-27, 1987. The symposium proceedings are now available from BHRA, Publication Sales Department, Cranfield, Bedford MK43 0AJ, England. Market proposals for industrial heat pumps have been adversely affected by the recent drop in fossil fuel prices, and reports of poor performance from some users. Yet when problems are investigated, the main reasons for this poor performance are over-sizing and under-utilization. Heat pumps which are carefully designed and matched to the process and the end-user's requirements are economically viable. The symposium discussed methods of improving the performance and cost-effectiveness of existing designs. Practical contributions from a wide range of international authors provide an outstanding review of current developments.

Schedule of Conferences and Trade Fairs

January 30-February 3, 1988

Dallas, Texas (USA); **1988 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 (404) 636-8400, telex 705343 ASHRAE.

March 21-24, 1988

Lisbon (Portugal); **1st European Conference on Industrial Furnaces and Boilers**. Sponsored by INFUB. Contact: Secretariat of INFUB, c/o Prof. Albino Reis, Praça Dr. Pedro Teotónio Pereira, 125-Esc. 33, 4300 Porto, Portugal, telephone 564849, telex 26261 ENTEC P.

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 Engr., Purdue University, IIR, and other cooperating groups. Contact: Ms. Vicki Delaney, Conf. Secretary, Ray W. Herrick Laboratories, Purdue University, West Lafayette, IN 47907, USA, telephone (317) 494-2132, telex 4930593 PHERLUI.

February 1-3, 1988

Dallas, Texas (USA); **Southwestern Air-Conditioning, Heating, Refrigerating Exposition**. Sponsored by ASHRAE and the Air-Conditioning and Refrigeration Institute (ARI). Contact: Judy Marshall or Judith Breese, ASHRAE International Headquarters, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA, telephone (404) 636-8400, telex 705343 ASHRAE.

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 (615) 574-2012.

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.

February 4-6, 1988

Dallas, Texas (USA); **Absorption Experts 88 Meeting**. Sponsored by the U.S. Department of Energy and the Gas Research Institute. Contact: R.C. DeVault, Oak Ridge National Laboratory, P.O. Box Y, Bldg. 9102-2, Oak Ridge, Tennessee 37831, USA, telephone (615) 574-0738, telex 854481 UD.

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.

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.

March 9-10, 1988

Tokyo (Japan); **1988 JAR International Symposium on Recent Developments in Heat Pump Technology**. Sponsored by the Japanese Association of Refrigeration (JAR). Contact: JAR, San-ei Bldg., 8 San-ei-cho, Shinjuku-ku, Tokyo 160, Japan, telephone 03-359-5231, cable REITOJAR, TOKYO, telefax 03-359-5233.

June 27-29, 1988

Ottawa (Canada); **1988 ASHRAE Annual Meeting**. Sponsored by ASHRAE. Contact: Judy Marshall or Judith Breese, ASHRAE International Headquarters, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA, telephone (404) 636-8400, telex 705343 ASHRAE.

August 28-September 1, 1989

Sarajevo (Yugoslavia); **CLIMA 2000** (2nd World Congress on Heating, Ventilating, Refrigerating and Air Conditioning). Sponsored by the Federation of Representatives of European Heating and Ventilating Associations (REHVA), ASHRAE, and IIR. Contact: Organizing Committee of CLIMA 2000, Masinski fakultet, Prof. Dr. E. Kulic, 71000 Sarajevo, Omladinsko setaliste bb, Yugoslavia, telephone 071/642071, telex 41 529 IPES YU.

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Update 1985, Report No. HPC-R2-1, May 1986, DM 75 (U.S.\$ 45). This report contains half-page project status summaries for over 700 heat pump RD&D projects completed and ongoing world-wide.

■ **Nonazeotropic Refrigerant Mixtures as Working Fluids in Compression Heat Pumps, A Bibliography**

Report No. HPC-B1, second edition, December 1985, DM 220 (U.S.\$ 130). This report is a comprehensive bibliography of over 200 reports with complete citations and abstracts.

■ **Proceedings of the Workshop on Ground-Source Heat Pumps**

Report No. HPC-WR-2, 1987, DM 50 (U.S.\$ 30). This publication contains papers presented at the Workshop on Ground-Source Heat Pumps, Albany, New York, USA, 27 October - 1 November 1986.

■ **National Reports on the Status of Heat Pumps**

Report No. HPC-WR3, 1987, DM 40 (U.S.\$25). This publication contains reports presented at the 2nd Experts' Meeting on Heat Pumps, 13th Congress of the World Energy Conference, Cannes, France, October 1986.

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Volume 6 (1988)

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- Volume 6, Number 1 (March 1988): Air-to-Air Heat Pumps
- Volume 6, Number 2 (June 1988): Working Fluids
- Volume 6, Number 3 (September 1988): 10th Anniversary of IEA Implementing Agreement
- Volume 6, Number 4 (December 1988): Absorption Heat Pumps

Our regular features (book reviews, schedule of conferences and trade fairs, news updates) will be included.

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

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