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IEA heat pump center

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250 kW split air/water heat pump for a block central in Sweden, outdoor units (see page 11)

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The Future Market Potential in Sweden for Heat Pumps in Block Centrals

The Swedish Government has decided to reduce dependence on imported oil through the development and use of domestic renewable energy sources, such as heat pumps, solar collectors, and boilers for burning wood-chips and other solid fuels.

1. Introduction

Two-thirds of oil consumption in Sweden is used for heating residential areas and to satisfy the heat demand in industries. A large part of this oil is used in different types of medium-sized heating plants called "block centrals."

From the heat pump point of view, it is reasonable to ask: What is the future market potential for heat pumps in these block centrals? The question is answered in two research reports initiated by the Swedish Council for Building Research.

2. Heat Pumps for Heating Residential Areas[1]

A block central in this context is defined as a plant producing sanitary hot water and water for heating, delivered to more than one building. It is often owned by the owner of the main part of connected buildings.

In the study, the block centrals are divided into three size groups, as shown in Table 1.

To ensure optimum working conditions, some modifications of the heat distribution system are always required when a heat pump is installed in a block central.

Contents **Page**

For instance, measures should be taken to reduce the temperature level as much as possible. Accumulators might be installed and subcentrals with heat exchangers might be rebuilt. The cost for these measures can be as much as 30% of the total investment cost of the installation.

The heat pump must compete with existing alternative heating systems. For economic comparisons, many assumptions must be made. Figure 1 shows, for example, the assumed average costs for the installation of heat pumps and boilers for solid fuels in existing oil burning block centrals.

The result of the calculations in the report are summarized in Table 2. Oil and electricity prices used here are as of May 1985.

A reasonable market for heat pumps should obviously be the conversion of about 25% of small and medium-sized block centrals and 10% of the large ones into heat pump plants.

Note that about 50% of the block centrals are assumed to be connected to district heating systems. By experience, it is very difficult to compete with district heating in district heating areas, even when alternative heating systems are theoretically more economical.

		Small	Medium	Large
Heating capacity	kW	<200	200-2000	>2000
Number of installations		5800	2500	90
Total oil consumption	1000 m^3/y	570	1000	280
Water temp. supply	°C	max. 50-60	90-110	90-120

Table 1. Data on block centrals for heating residential areas

Block-central size	Plants today	Conversion to: District heating	Heat pumps	Wood chips	Continuing use of oil
Small	5,800	2,900	1,700	400	800
Medium	2,500	1,200	600	600	100
Large	100	50	10	40	0

Table 2. Market potential for heat pumps and other systems in block centrals

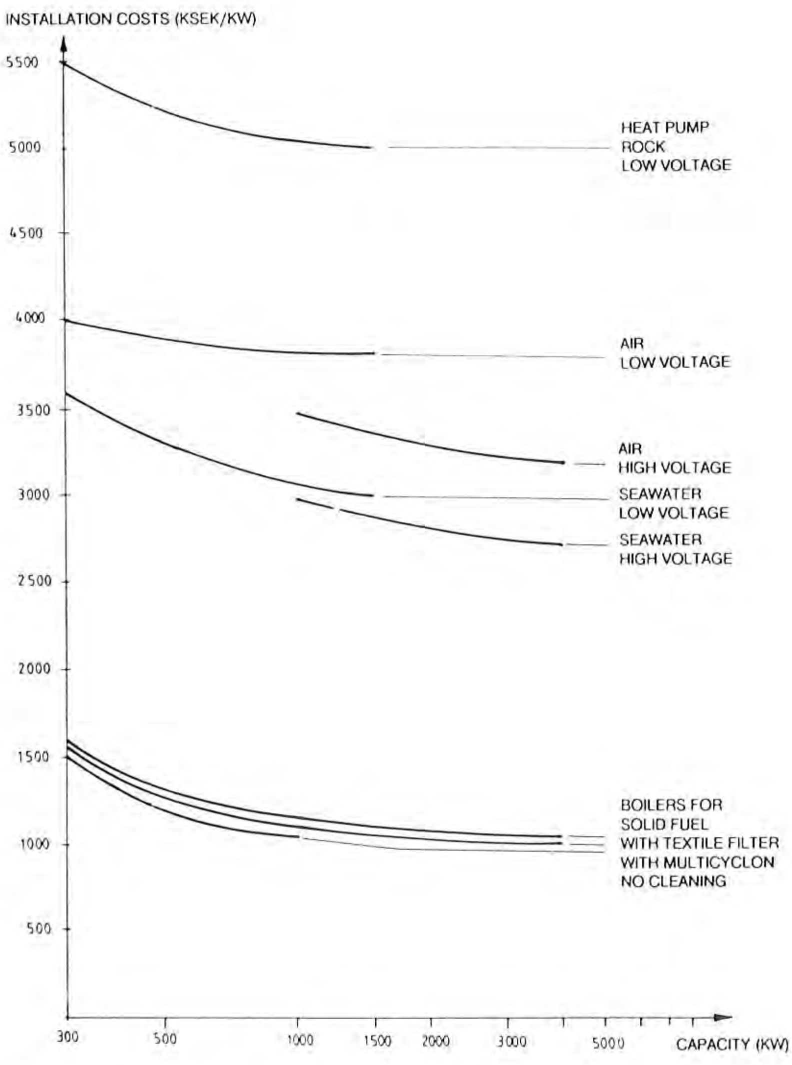

Figure 1. Installation costs

3. Heat Pumps in Industrial Block Centrals[2]

The industrial block centrals are not easy to identify from a heat pump point of view. The different industrial branches have their own special energy demands and, therefore, their own heating system design. Some branches have no presumptions for the use of heat pumps, and others have too little total energy demand to be of interest in this investigation.

Twenty-one industrial branches were finally identified, each with a total oil consumption of at least 14,000 m^3/year. All the branches also have a considerable need for low-temperature energy for their processes and/or for heating of buildings.

For each of these 21 branches, a characteristic plant was identified, among others, by means of energy demand at different temperature- and pressure-levels for different purposes, such as heating of buildings, production of hot water, different kinds of process heating, and so on. These are the industrial heating plants (block centrals) which were studied in this report.

To judge the economic potential for new energy techniques in these industrial block centrals, many assumptions were made regarding energy prices, calculated interest, decision criteria, etc. All these assumptions and limitations cannot be listed here, but it is important to bear in mind that they exist. It is also important to remember that the investigation did not aim at calculating the absolute values for energy costs but at finding the competitiveness of the different energy systems under different circumstances in the future. The results for all different cases were tabulated; an example is given in Table 3.

Industrial branches	Installed Capacity (MW) 1980/85	1986/90	1991/95
Slaughterhouses	3	7	10
Dairies	35	51	64
Fruit and vegetable preservation	8	8	13
Cooking oils and fats	1	1	1
Bakeries	-	-	-
Breweries	2	3	6
Chocolate and candy	4	4	8
Textiles	7	13	18
Lumber yards	-	-	-
Pressed wood	1	2	2
Paper and pulp	270	510	750
Chemicals	43	119	130
Graphics	-	5	10
Plastics	9	15	18
Rubber	10	17	23
Porcelain and clay	-	-	-
Tile	-	-	-
Cement	-	5	10
Iron and steel	11	22	33
Cast metal	-	-	-
Workshops	-	-	-
TOTAL	404	782	1081

Table 3. Penetration of heat pumps in industrial block centrals

Regarding heat pumps, the following conclusions can be drawn:

- By the year 1995, heat pumps with a total capacity of approximately 1000 MW might be installed in Swedish industries. Pulp and paper industries dominate (70%), but are also very uncertain as the influence from new industrial processes has not been considered.
- Considerable markets for heat pumps are to be found in industrial branches such as slaughterhouses, dairies, fruit and vegetable preservation, chemical, breweries, plastics, iron and steel, and rubber industries.
- Because of the relatively favorable price for electric energy, the market will be dominated by electrically driven closed-cycle heat pumps. However, some absorption heat pumps are assumed to penetrate into brewery industries.

4. Summary and Comments

- There seems to be a considerable market for converting from oil burning to heat pumps in block centrals in Sweden.
- Where the heat pump is an integral part of an industrial process, the calculated profitability is generally very good.
- There are heat pumps available on the market today for most applications.
- Native solid fuels and oil are the heaviest competitors of heat pumps.
- A calculated high profitability is not enough to convince the decision makers to invest in "new techniques" in block centrals. Demonstration plants are probably needed to make the heat pumps credible.

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The Heat Pump in a District Heating and Cooling Project in Norway

Baerum energiverk are in the process of constructing a district heating and cooling plant in Sandvika, on the outskirts of Oslo, Norway. The plant represents innovations in Norwegian energy supply and will be supported by the Oil and Energy Department. The plant is designed for an output of approximately 24 MW for heating and 9 MW for cooling. New buildings will be subject to compulsory connection to the district heating plant; connection to the district cooling grid will be optional. Developers of new offices/commercial buildings in the area have, however, shown great interest in being connected to the district cooling grid at prices which are very satisfactory for the energy company. When the plans are realized, the expected sale of district heating and cooling will be 60 GWh/year and 12 GWh/year, respectively. The plant will be equipped with an advanced control and monitoring system.

1. Project background

1.1 General

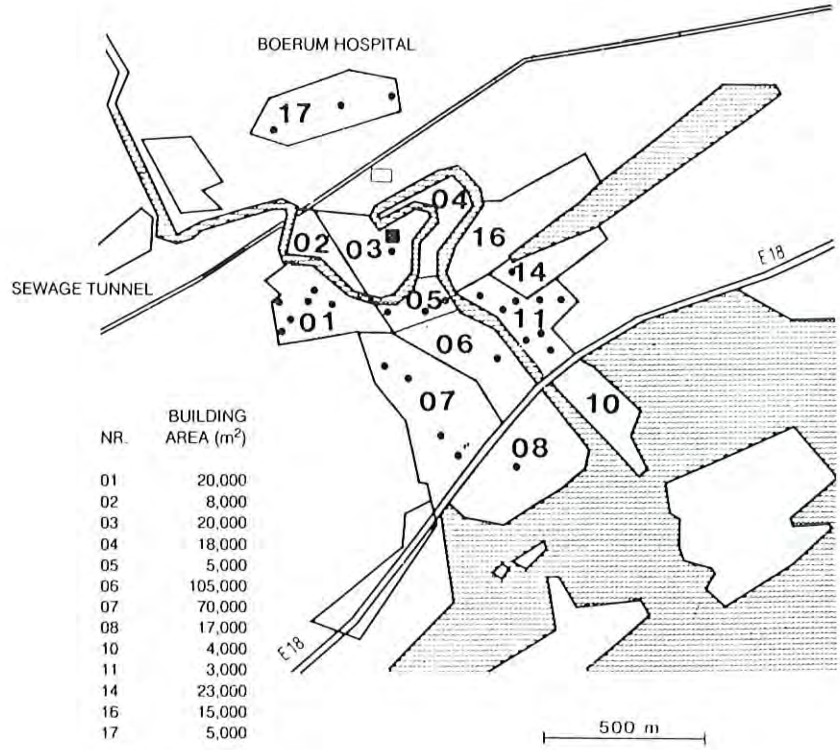
The Sandvika region, a densely populated area in Baerum, 10 km from Oslo, is on the threshold of an extensive exploitation. In total, some 300,000 m², mostly devoted to business and commerce are planned to be built.

Baerum energiverk has evaluated alternative means of supplying energy to this area.

1.2 New and existing buildings in the area

A relatively large number of existing buildings are equipped with water-circulating heating systems, comprising a total building area of 260,000 m², including a hospital with approximately 50,000 m². The area is shown in Figure 1.

Based on the assumption that all new buildings shall be constructed with water-circulating heating systems, it became apparent for the energy company that these buildings, in addition to existing buildings, would be an interesting district heating area. The location of the area in the vicinity of the Oslo fjord and an existing sewage tunnel made the alternative of a heat-pump-based district heating system particularly interesting.



NR.	BUILDING AREA (m²)
01	20,000
02	8,000
03	20,000
04	18,000
05	5,000
06	105,000
07	70,000
08	17,000
10	4,000
11	3,000
14	23,000
16	15,000
17	5,000
Total	300,000

● Existing buildings with water-carried heating
■ Peak load and emergency heating plant
□ Base load heating plant

Figure 1. Sandvika area

1.3 Energy requirements in new and existing buildings

In the existing buildings, the energy requirements which can be converted to district heating are approximately 35 GWh/year. The capacity requirement by consumers is approximately 17 MW, and the temperature demand is between 80-85°C at design temperature.

The district heating potential in the new buildings in this region has been the subject of rigorous evaluation. In domestic dwellings, schools, hospitals, etc., the district heating potential is significant. Office/commercial buildings with large internal loads and heat recovery from cooling machines are less suitable for a district heating system. The fact that there are a significant number of such buildings planned in the area was the background for the initiation of the district cooling project (see Section 2).

Based on the assumption that heat recovery from cooling machines within the buildings is neglected, the district heating potential for the new buildings within the shaded area on the map in Figure 1 is approximately 25 GWh/year and the capacity demand is approximately 13 MW. This gives a total district heating potential of 60 GWh/year with a total capacity requirement of approximately 24 MW. By connecting the area north of Sandvika, the district heating potential will increase by 20-30 GWh/year. In addition, district cooling must be considered.

2. District cooling

2.1 General

The cooling requirement will be large in the type of buildings planned in Sandvika. More than 210,000 m² will be office/business premises with a total cooling requirement of approximately 12 GWh/year and a capacity demand of approximately 9 MW.

In the evaluation of the profitability of connecting these buildings to a district heating system, a heat pump plant was considered. The costs related to the district cooling plant are therefore evaluated as marginal costs.

Cooling will be obtained from the same machine which supplies heat to the district heating grid - a combined heat pump/cooling plant.

2.2 District cooling as a necessary supplement to district heating in new office/commercial buildings

Buildings with their own systems for producing cooling will also produce heat. This heat can cover all or part of the heat requirements of the building, or may not be utilized at all. Today it is regarded as prudent to invest in equipment which can utilize this heat in the majority of new office/commercial buildings.

By connecting these buildings to a district cooling plant, heat recovery will be realized and the total heating/cooling requirement can be covered by the energy center.

Sensitivity analyses for the Sandvika project show that the district cooling plant gives a positive contribution to the profitability of the project, even with very low district cooling prices. The reason for this is that an increase in the sale of heat will be achieved. In other words, modern office buildings do not need large quantities of district heat if they are equipped with individual cooling systems with 100% heat recovery. In order to sell district heat, the energy utility must also take care of the buildings' cooling requirements.

2.3 Consumer distribution

The district cooling grid is, in principle, built up as a district heating grid. Stipulations relating to type of pipe and insulation are not, however, the same. Surface-treated steel pipe, cast iron pipe and several types of plastic pipe can be used and have competitive prices. Loss of cold in the grid is negligible in relation to the extra investment which insulation of the district cooling grid will represent. For the Sandvika plant, PEH pipes will be used.

The district cooling system is, in similarity with the district heating system, based on an indirect system. Cooling water is heated in a heat exchanger at the consumer center. The thermodynamic loss and the extra investments which this heat exchanger represents were evaluated as having less value than the advantages which the energy company would achieve with clearly defined limits of responsibility to the consumers. The temperature demand in the buildings' water system is approximately 8°C, and the temperature for the district cooling grid will be 5-13°C at design conditions.

2.4 Security of supply

The heat pump system is composed of two units which independently can supply cooling to the district cooling grid. The circulating pumps in the district cooling grid are duplicated in order to achieve high availability. Consumers with computer cooling will have a possibility to achieve an agreement with the Baerum Water Authority regarding the use of grid water as spare cooling.

2.5 Consumer prices for district cooling

In the evaluation of how district cooling ought to and can be priced, the following points are central:

- Costs of productions for district cooling (capital and operating costs)

- The price which the customers are willing to pay for district cooling instead of investing in private cooling machines

- Tariff setting for district cooling

- Value of the increased district heating sale which can be expected as a consequence of the sale of district cooling

Cost of producing district cooling

Capital and operation costs for connection and supply of 12 GWh/year cooling with a capacity of 9 MW to approximately 15 buildings distributed in areas 02, 03, 04, 05, 06, 07, and 08 (Figure 1) are estimated below. It is assumed that, since a heat-pump-based district heating system will be installed, all the costs for producing district cooling are to be regarded as marginal costs.

Investment costs (in million Kroner):

District cooling grid	7.0
Monitoring and control equipment	0.8
Additional investment in heat pump plant	2.4
Extra space requirements in rock-cavern	0.2
Circ. pumps, pressure retaining and water treatment	1.1
Total marginal investments	11.5

Annual costs:

Capital costs (10 years at 10% interest)	1.87
Operational costs (0.7 GWh/year)	0.14
Maintenance costs (1.5% of investment)	0.17
Total annual costs	2.18

By setting the above rentability to invested capital in the district cooling plant and by discounting the expected increase in the district heating sales, the specific energy costs for district cooling to the consumer is 0.182 kr/kWh.

Alternative costs for client

A client's specific cost (kr/kWh) for privately produced cooling varies largely. The following points are of great significance:

- Invested capital in cooling machine
- Capacity requirement and usage time
- Heat requirement which can be covered by recovery of cooling machine's condenser heat
- Demands for rentability of invested capital

For clients in Sandvika who evaluate these points, the specific costs for privately produced cooling can vary between approximately 0.12 and 0.70 kr/kWh. Users with the lowest specific cost are characterized by a cooling requirement of long period of use, a heat requirement which generally can be satisfied by condenser heat and low demands to rentability of capital. Similarly, for users who are characterized by short-term cooling requirements and high capital demand, the specific costs for privately produced cooling are much higher.

In order to take into account these conditions, the energy company must differentiate the price for district cooling through special tariffs.

Tariffs

Due to the varied costs in privately produced cooling, the district cooling must be priced with a capacity factor in addition to an energy factor.

The capacity factor will reflect the capital costs and the energy factor will reflect the operation costs in the two alternatives. On the basis of these criteria, the district cooling tariff for Sandvika is fixed as 210 kr/kW and 0.10 kr/kWh.

Potential value of increased district heating sales

As described in Section 2.2, the potential increase in the sale of district heating as a consequence of the sale of district cooling represents a significant value for the Sandvika projects. This recognition is an argument for setting the price of district cooling significantly lower than the selected tariff. The response from potential district cooling consumers is, nevertheless, so positive that Baerum energiverk will adopt a tariff for district cooling which gives the project a welcome economic bonus and a satisfactory total profitability.

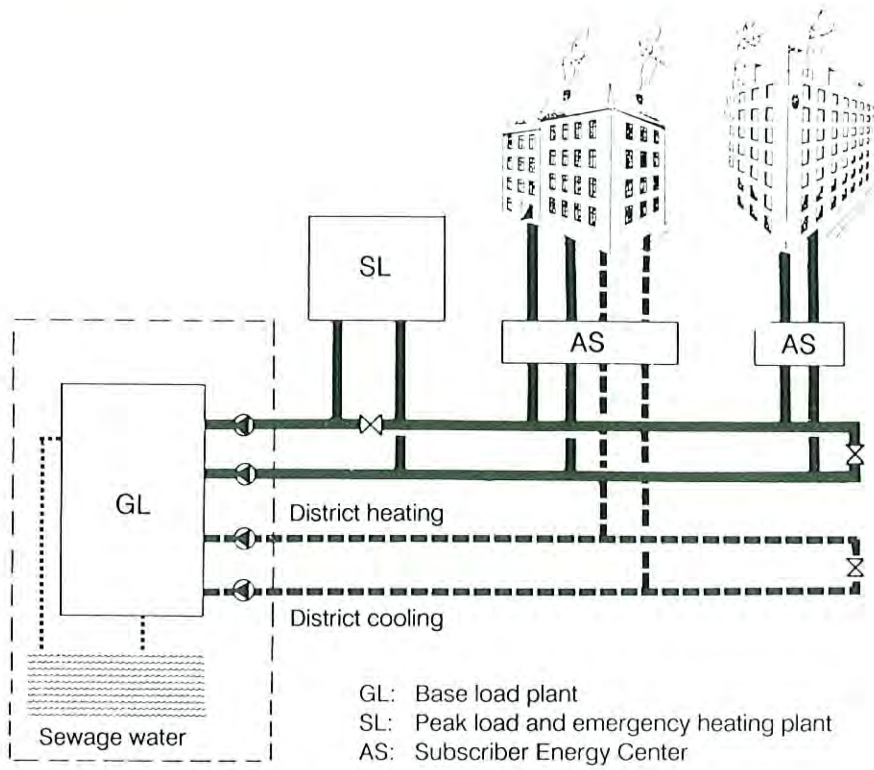


Figure 2. Heat energy distribution system

3. District heating and cooling plant - technical description

3.1 General

The production of heat in the energy center is divided into base production and peak and emergency heat production. The base load will be covered by a heat pump plant with an output of approximately 60% of the maximum capacity demand. The peak and emergency load will be covered by oil-fired boilers, with a capacity equal to maximum demand in the heating system.

A diagram of the complete system is shown in Figure 2.

3.2 Technical framework

Heat source

Mechanically treated sewage from the sewage tunnel will be the heat source for

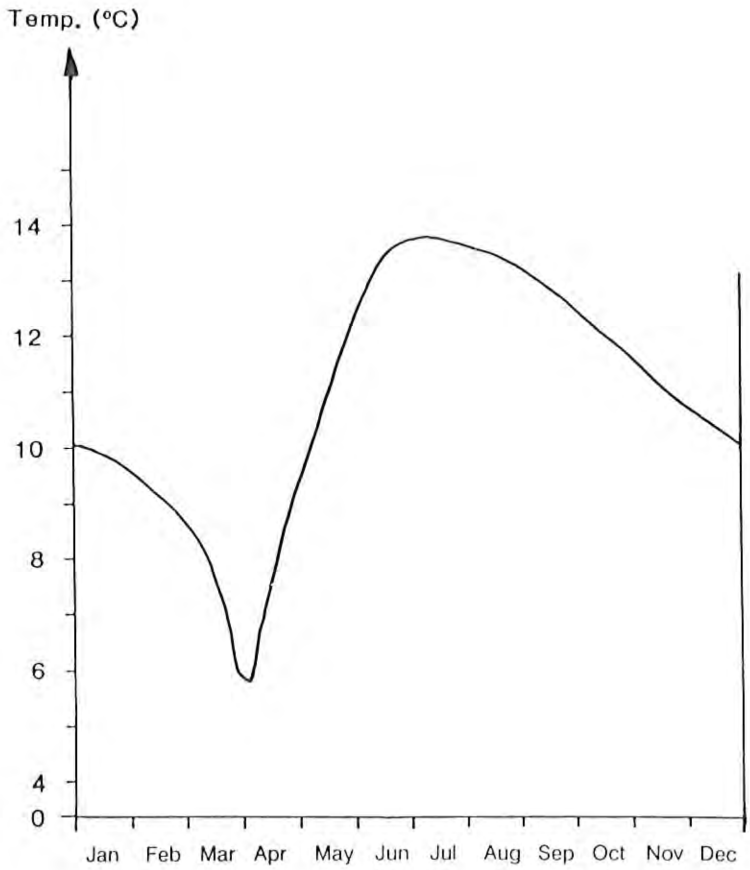


Figure 3. Waste water temperature

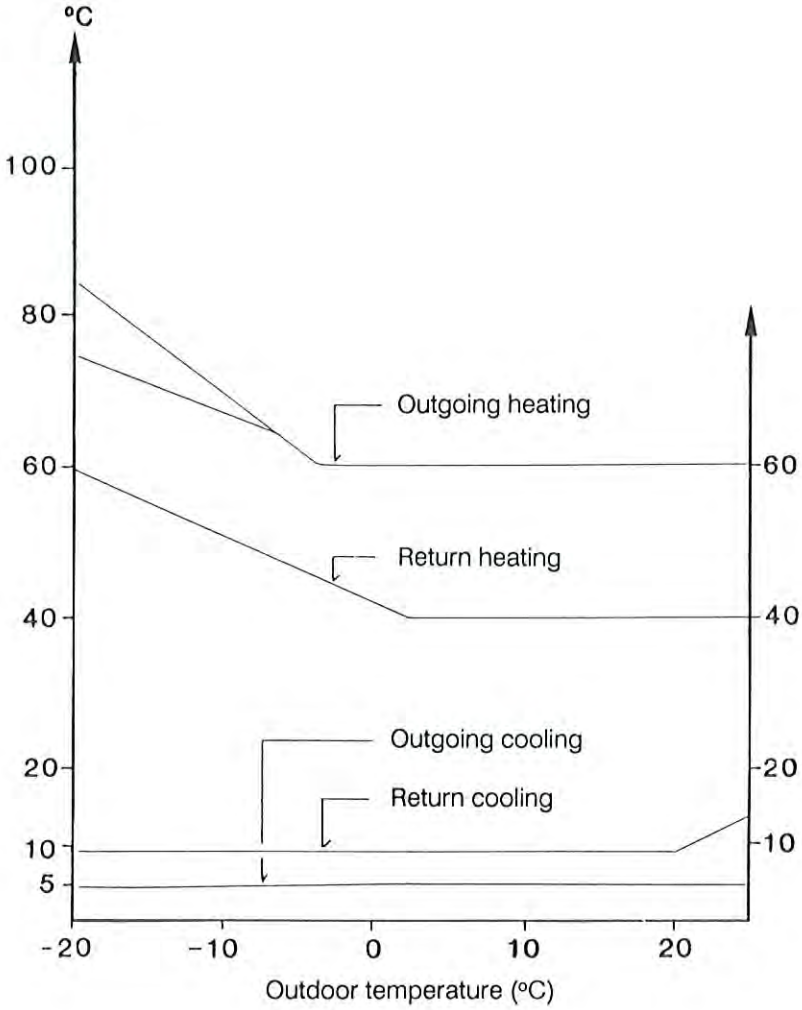


Figure 4. Heating and cooling distribution temperatures

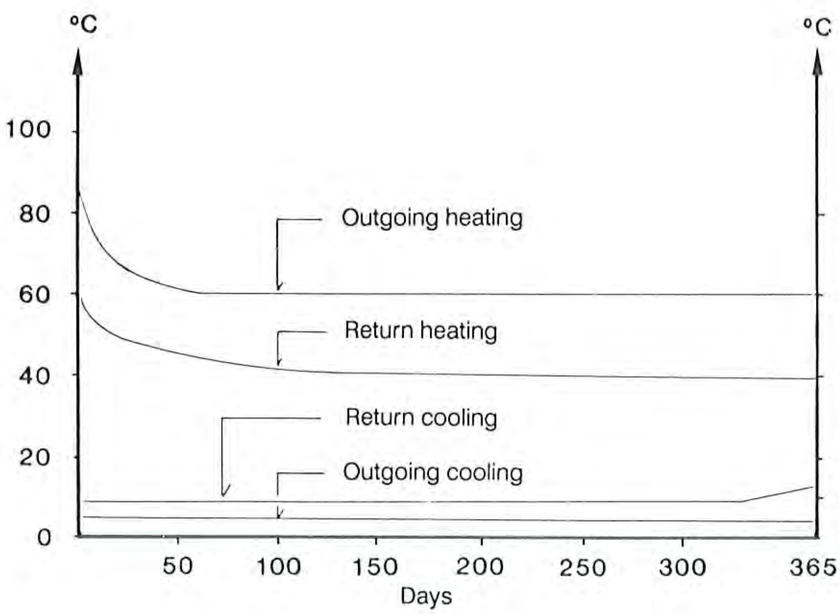


Figure 5. Temperature duration

the heat pump plant. The same heat source is at present used at Oslo Lysverkers heat-pump-based district heating plant at Skoeyen West. The temperature level of the sewage is shown in Figure 3.

Distribution grid

The expected temperature levels in the distribution grids as a function of outside air temperature and their duration time are shown in Figures 4 and 5.

Capacity and energy requirement

The district heating and cooling systems capacity/duration curve is shown in Figure 6. The curve shows that a significant variation in energy demand will occur during the year. Part load performance for both heating and cooling are thus of great importance. Day/night variations in the heat capacity demand are small throughout the year. Cooling is distributed with an almost constant load for computer cooling and a need for room climatization which only occurs in the daytime during summer months.

3.3 Energy center - base load

The energy production plant is an underground plant (see Figure 7).

The sewage screening part consists of gates (1), two parallel filters (2), and dry displayed sewage pump (3) with pipe to the heat pump units. The screens have an aperture of 1 mm. The flow of sewage is kept constant at 300 kg per second per heat pump unit.

There are two heat pump units (4). Each unit is equipped with a valve system (5) for reversing the flow of sewage in order to avoid clogging in the intake to the evaporators.

Circulating pumps (6), pressure retaining equipment and water treatment vessels (7) for the district heating and cooling grids are located in a separate room. High voltage units and ventilation equipment (8) are located at the entrance.

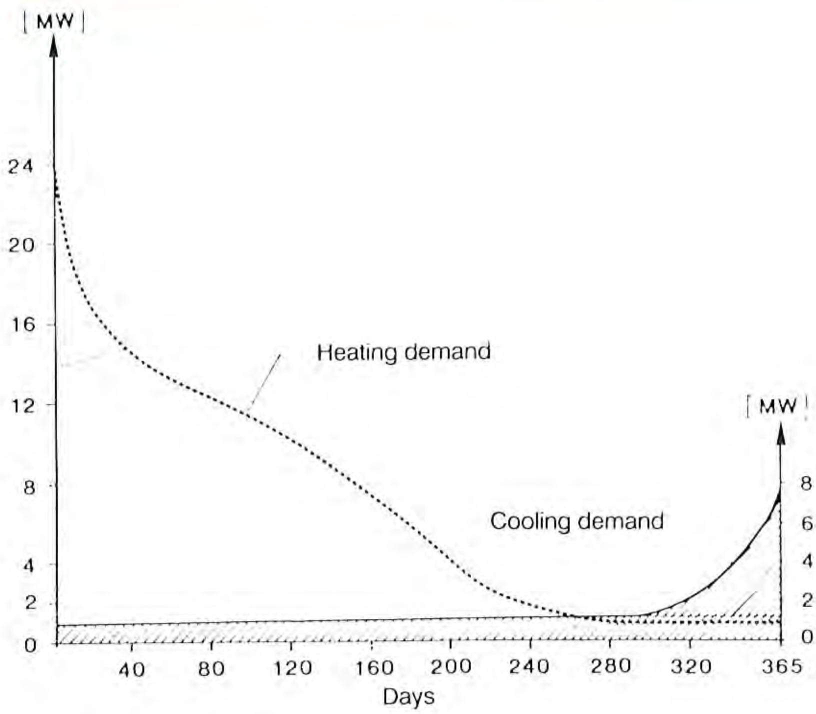


Figure 6. Capacity duration

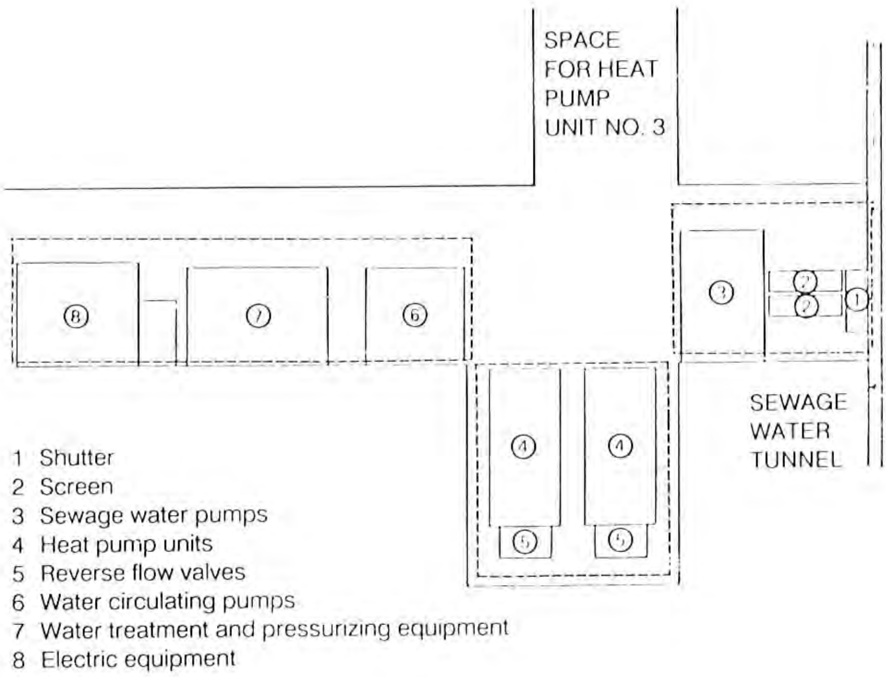


Figure 7. Base load plant - general arrangement

3.4 Energy center - peak and emergency load

The peak load center is located 400 m from the base load center in a renovated paper mill. It consists of three oil-fired boilers of 2.5 MW, 9 MW and 10 MW, respectively. The expected utilization time for the plant is very low, with an oil consumption of less than 400 m^3 liters per year. The plant is calculated to supply approximately 5% of the total heat demand.

3.5 Heat pump units

The system concept for the heat pump unit is shown in Figure 8. Each heat pump unit has the following main components: Condenser (9) with subcooler, two compressors (11, 12), which can be capacity regulated independently of each other, intermediate pressure vessel (13), combined evaporator/condenser (14) for heat exchange with sewage, an evaporator (15), and a 4-way valve (7) for redirection of sewage flow.

The outgoing temperature of the district heating grid is regulated by capacity regulating the HT-compressor. The outgoing temperature of the district cooling grid is regulated by capacity regulating the LT-compressor or by manipulating the liquid level in the evaporators.

The refrigerant level in the condenser is controlled by the level in a receiver, operating a valve (1). The level of refrigerant in the sewage heat exchanger is controlled by the level operating valve (2) or (4). The level of refrigerant in the district cooling evaporator is controlled by the level operating valve (3). The intermediate pressure is controlled by capacity regulating the LT-compressor or by operating valve (5).

There are two actual operating conditions for the plant:

-- **Heat requirement dominates, low cooling requirement.** HT-compressor is capacity regulated according to the heat requirements (outgoing temperature of district heating). LT-compressor is capacity regulated so that the intermediate pressure is optimized. The outgoing temperature of the district cooling grid is permitted to vary freely as long as it is less than 5°C.

-- **Cooling requirement dominates.** HT-compressor is regulated according to the heat requirement. LT-compressor is regulated according to the cooling requirement. The sewage heat exchanger is working as condenser. Valve (5) is controlling the intermediate pressure. Valve (4) sets the level in the sewage heat exchanger to the lowest level. (Valves (2) and (6) are closed.) Part of the refrigerant in the sewage heat exchanger is thus, via the district heating evaporator, moved to the intermediate pressure vessel.

Selection of sewage heat exchanger

The tube heat exchanger was selected, based on the following:

- Experience from several plants in Sweden with shell and tube evaporators in connection with the use of treated sewage.
- Evaluation of the quality of the sewage water in the tunnel in relation to the quality in the plants in Sweden.
- Investigations with the tube evaporators at Skoeyen West.
- Experience from Skoeyen West with plate evaporators.
- Cost consequences for the two alternatives.

The choice was made with the following specifications:

- One-pass type shell and tube heat exchanger.
- A 4-way valve is installed for recirculation in order to avoid clogging due to fiber.
- A copper-based material is used in the tubes in order to avoid growth on the heat exchanger surfaces.
- Possibility for further filtering of the sewage is included.
- Possibilities for a washing detergent cleaning system is included.

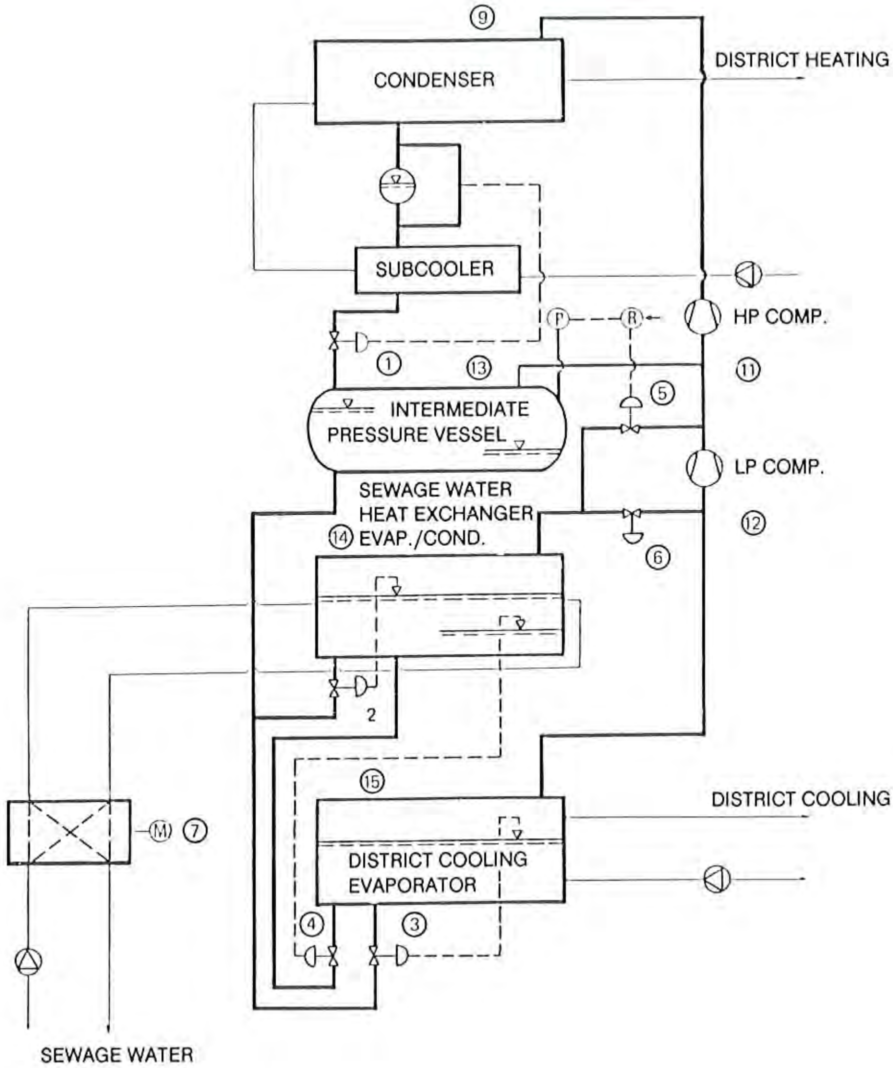


Figure 8. Heat pump unit, piping diagram

3.6 Control strategy

General

In order to minimize the personnel operating costs, the plant is based on unmanned operation outside normal working hours. There will, in addition, be consideration given to an operation monitor/alarm which will be included in the energy company's existing monitoring system.

The plant will be constructed with a view to overall operational security. This is secured by each operational section being able to function independently if the main control system fails.

The control system provides effective monitoring of the situation, systematic documentation and operation reporting.

Composition of control system

The control system consists of a computer, a communication network and numerous substations and measuring stations.

The substations are the local control systems for the following parts of the plant: heat pump, screening plant, circulation pumps, machine hall, and peak and emergency plant. The measuring stations are one per each subscriber.

Primary control functions

The master control system will generate reference values for the outgoing temperature in the district heating and cooling grids based on meteorological values and registered changes in consumer demand. The circulation pumps are controlled by the differential pressure at the consumers.

Measuring stations' task in the control system

Programmable logic control units (PLS) in the measuring stations will automatically charge the subscribers for the heating and cooling energy consumed. The immediate capacity used will be continuously checked and it will not be possible to achieve a higher capacity than that subscribed.

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Block Central Project "Achter de Muren" in Deventer

The "Achter de Muren" block central is one of two large district heating projects with heat pumps in the Netherlands. It provides hot water for heating purposes to approximately 75 consumers, such as schools, office buildings, and a theater in downtown Deventer. Originally, heat was generated by three coke-fired hot water boilers, which were later retrofitted to natural-gas firing. The heat pump was put into operation during 1984, together with a new gas-fired boiler; one of the old boilers was kept as a back-up. The block central is owned and operated by the Municipal Energy and Water Board of Deventer (G.E.W.B.).

Description of the system

Table 1 shows the main features of the heat pump plant. It consists of two gas engine-driven heat pumps and one modulating boiler. Useful heat is delivered to the district heating grid by the heat pump condenser, the engine cooler and exhaust gas coolers on both gas engines and boiler. The supply water temperature is controlled between 65°C and 80°C. Heat is extracted from groundwater, which is also used for two air coolers. The groundwater is supplied by two wells and rejected into the IJssel River.

Reduction of energy consumption and costs

From weekly data gathered by the G.E.W.B., a comparison could be made between the expected and realized energy conservation by the heat pump/boiler system in 1984 and 1985. These figures are presented in Table 2, together with a summary of energy costs (based on 1985 price level: gas fl 0.52/m^3 and electricity fl 0.24/kWh).

The large differences between the actual situation and the expectations have been caused mainly by:

- Over-estimations in the design phase of boiler and heat pump efficiencies and of the heat pump's contribution to the total heat delivered;
- The frequent part-load operation of the heat pump; and
- Incorrect operation of the gas mixing devices.

It is obvious that the expected payback time of ten years for the project will not be obtained by far. A recent study[2] indicated that a maximum annual reduction of 340,000 m^3 of natural gas may be obtained, provided that certain alterations to the installation are made (e.g., control strategy and gas mixers).

Groundwater as heat source

Groundwater is extracted from two 40 m deep wells, at a rate of approximately 130 m^3/h in total. There are, however, severe fouling problems: the heat pump evaporator is clogged several times per year, while the water wells and other parts of the installation are fouling too. This is caused by oxidation and flocculation of iron. Although the groundwater itself contains very little oxygen, it is believed that frequently water from the IJssel River, with a relatively high content of salts and oxygen, infiltrates the wells by groundwater flow. The resulting mixture then is corrosive. Several measures have been proposed to protect the installation from fouling; water treatment by dosage of sulphite will be further investigated.

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Owner	G.E.W.B. Deventer
Delivery date	February 1984
Purpose	District heating
Annual heat demand	31,700 GJ (8,800 MWh)
Peak heat demand	4,550 kW (135%)
Aver. heat load (10°C)	3,367 kW (100%)
Capacity of heat pumps	1,500 kW (45%)
Capacity of boilers	6,300 kW (187%)
Operation mode	Bivalent-parallel
Heat source	Groundwater
Heat sink	Radiator systems, air curtains
Control	Fixed temp. difference, flow controlled

Table 1. Block central "Achter de Muren", Deventer, the Netherlands

	Heat Pump eff. %	Contribution %	Electricity consumption MWh	Reduction in Gas 1,000 m^3	Reduction in Total Costs kfl
Expected	135	90	103	430	195
Actual 1984	136	59	123	282	116
Actual 1985	123	50	160	191	60

Table 2. Data on expected and actual reduction of energy consumption and costs[1]

Å. Bratt*

Experience with 250 kW Air/Water Heat Pump in a Swedish Group Central

Many heat pump plants were built in Sweden during the period 1980 to 1985, and, of course, much experience was gained by those involved. In late 1984, a medium-sized heat pump plant was begun for domestic heating where most of the experience available at that time was fed back into the project. The result was encouraging. The contracting and technical principles applied ought to be of common interest to those involved in similar projects.

Assumptions

An apartment complex in the small town of Vadstena, at the shore of Lake Vaettern in the south of Sweden, had six buildings with a total of 300 apartments. The costs for heating the buildings were unsatisfactorily high, despite measures taken for energy conservation. The owner was, therefore, interested in the installation of a heat pump.

Together with drawings of the building, the yard and the pipeline of the heating system, the owner gave the following information for the design of the recommended heat pump to be installed:

Fuel consumption	385 m^3/year		
Boiler efficiency	70%		
Water temperature needed			
Outside air	5°C	0°C	-5°C
Heating water	55°C	62°C	69°C
Energy prices			
Fuel oil	2,300 SEK/m^3		
Electricity	250 SEK/m^3		
Heat source available	outdoor air		

(SEK = Swedish kronor)

Technical and contracting principles applied

The customer was inexperienced with energy technology and heat pumps.

The main contractor was Installatoer Oestergoetland AB, a company in the Flaekt AB concern. Subcontractors were STAL Refrigeration AB, supplying the heat pump and heat pump plant knowledge and experience, and a consulting engineer who was a specialist on control systems. The cooperation in this group was broad minded and open, based on confidence and good human relations. The following principles for their work were established:

Primary principle:

- The plant should be designed for maximum profitability for the owner, considering existing assumptions.

Other main principles, based on experience:

- The plant should be built by *one* main contractor with total responsibility for the project. He should have contracting experience and knowledge, as well as access to knowledge of heat pump plants and heat pump units to be used.
- The plant should be designed with a well-specified running strategy.
- The plant should be as simple as possible.
- The heat pump itself should be of standard design, factory-built and tested, with good references.

These ideas were not revolutionary, but how many heat pump plants have been built following these principles?

Size of the heat pump

Maximum heating capacity for heating the buildings was calculated to be about 1000 kW.

The size of the heat pump to give maximum profitability in a bivalent system with the existing oil-fired boiler was calculated to be on the order of 500 kW. This heat pump should satisfy the heat demand down to an outdoor temperature of about 0°C. At these running conditions, the heat pump should be able to deliver water at 63°C, which meant that refrigerant R12 had to be used. An investigation showed that there was not enough space available for such a heat pump, neither for the heat pump unit in the machine room in the cellar of one of the buildings nor for the air coolers in the yard outside the building.

The plant was, therefore, designed with a smaller standard heat pump unit with a capacity of 266 kW, with an outgoing water temperature of 55°C, and balancing the heat demand at an outdoor temperature of about +7°C. 55°C was also the minimum temperature needed for the production of consumption water in the heat exchangers in the six different buildings. The maximum allowed outgoing water with R500 was 60°C, which meant that the heat pump could run with full load, without being limited by maximum allowable outgoing temperature, down to the conditions at an outdoor of -10°C, when the heat pump has to be stopped for other reasons. The plant was calculated to deliver 47% of the total heat demand in one year, with a mean COP of 2.3, considering the total energy consumption for running the plant.

Figure 1 shows the heating demand, outgoing water temperature demand, and the heat pump capacity versus outdoor temperature.

Plant design

Figure 2 is a highly simplified version of the piping system, showing the design of the plant under normal running conditions.

The heat pump itself is a standard water-to-water unit with two eight-cylinder semi-hermetic compressors, each with its own refrigerant system. The unit, equipped with its own safety and control systems, is factory-built and tested, and is installed in the cellar of one of the buildings.

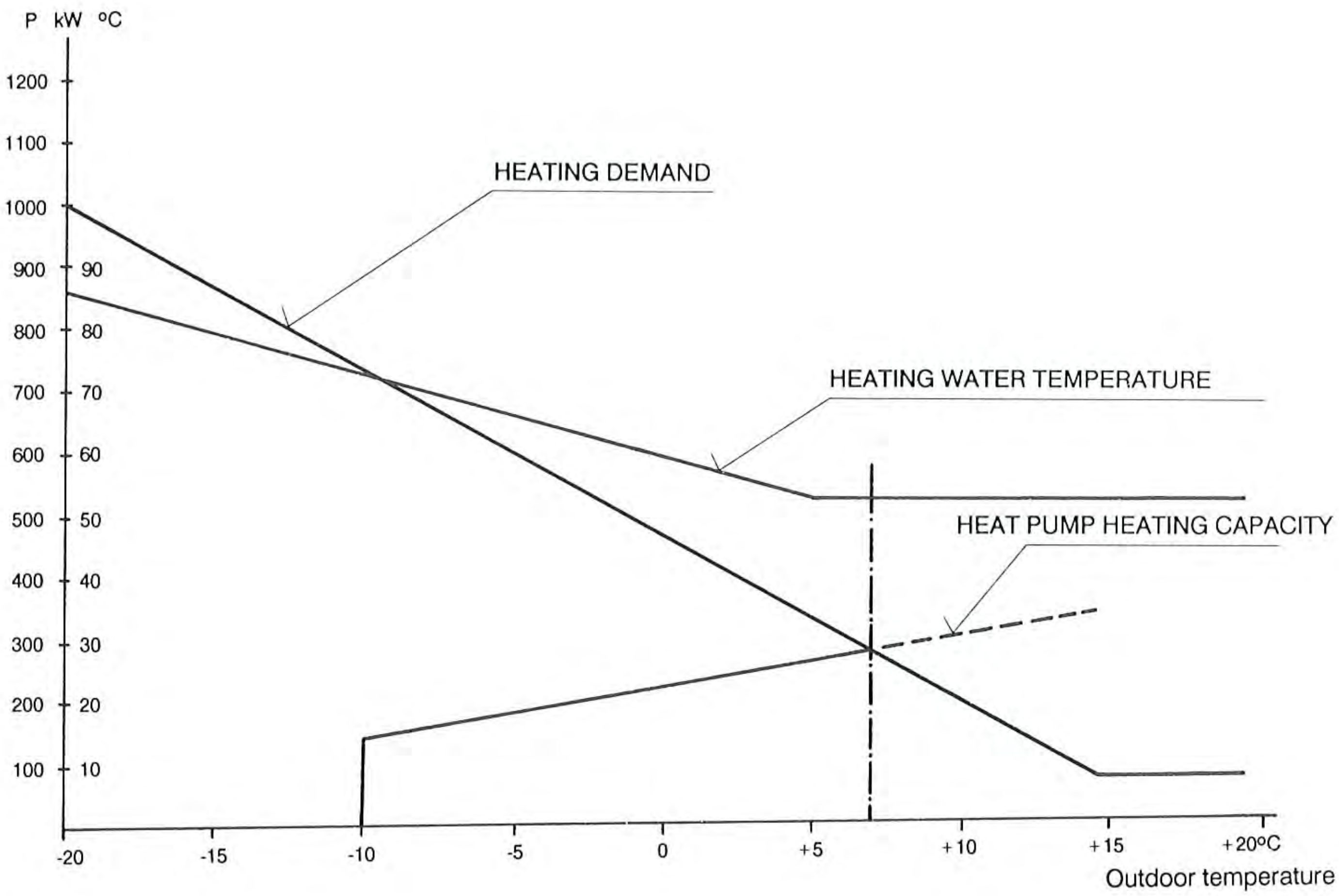

Figure 1. Heating capacity and temperature

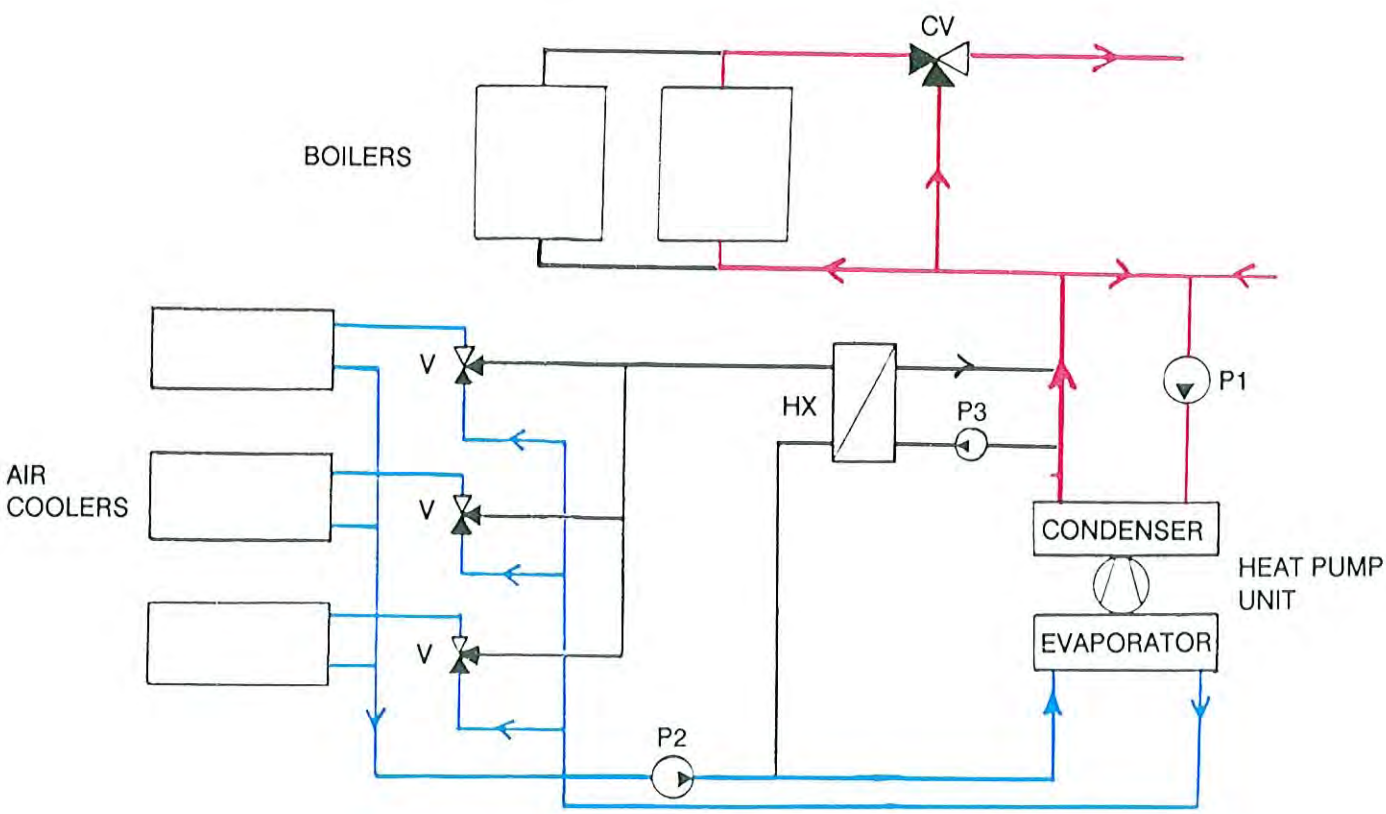

Figure 2. Piping system (normal running conditions)

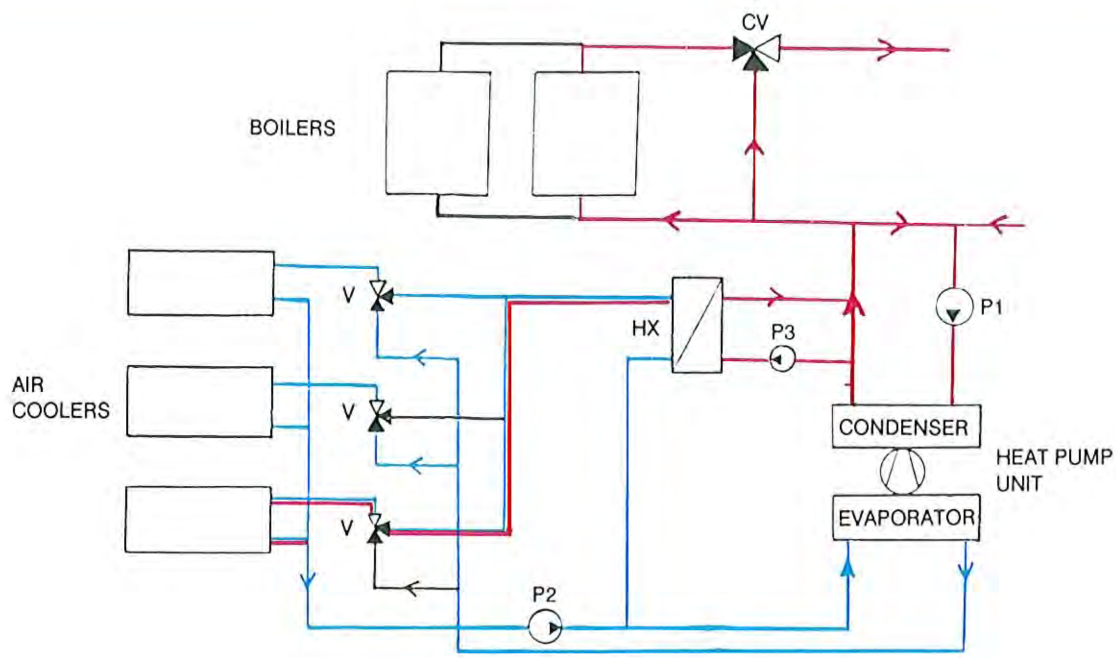

Figure 4. Piping system (defrosting conditions)

The heat pump condensers are connected with the return line in the heating system. The auxiliary pump (P1) in the connection line is designed to give a flow which is somewhat higher than the maximum flow in the heating system during the period when the heat pump alone is responsible for heat production. Heat is taken from the outdoor air through three air coolers in the yard outside the building (Figure 3 - see cover photo). The air coolers are connected to the heat pump evaporator through a brine system with a circulation pump (P2).

The air coolers are made of 1/2" copper tubes with 0.25mm aluminum fins with a spacing of 5mm to ensure good water drainage when defrosting. They are equipped with a slowly rotating fan, and a noise-insulated exhaust extension. Some design performance is listed below:

Size (length/width)	3.2/2.4 m
Number of rows of tubes	4
Fan diameter	1780 mm
Fan speed	260 rpm
Fan driving capacity	2 kW
Air inlet temperature	8°C
Brine inlet temperature	2°C
Cooling capacity	60 kW

The air coolers are defrosted by means of hot brine from the heat exchanger (HX). Defrosting is actuated by starting the circulation pump (P3), and turning the three-way valves (V) (Figure 4).

Plant running strategies

The strategy applied for running the plant was determined by the temperature conditions described in Figure 1.

At an outdoor temperature of less than -10°C, the heat needed is produced by the oil-fired boiler alone. The heat pump is not in operation, mainly because of bad heat transfer conditions due to the low temperature of the brine. This is valid for roughly 250 hours/year.

At an outdoor temperature between -10°C and +7°C, the heat pump is running continuously at full load, and the boiler is in operation. The temperature of the outgoing fluid from the condenser is increased to the necessary level due to the actual outdoor temperature by adding high-temperature hot water from the boiler. This is done by actuating the control valve (CV), shown in Figure 2.

At an outdoor temperature greater than +7°C, the heat needed is produced by the heat pump alone. The oil-fired boiler is not in operation. The capacity is controlled by running the heat pump on/off at full load. The temperature of the outgoing fluid from the condenser is permitted to fluctuate about +/- 5°C from the set mean value of 55°C. This does not cause inconvenience to the tenants because there are electrical heaters in the hot water heat exchangers, and the heat capacity of the buildings is high enough to keep the indoor temperature conveniently constant.

At an outdoor temperature higher than about +11°C, only half of the capacity is needed. The heat pump is then running with only one compressor in operation.

The control system is equipped with adjustable delay functions for starting and stopping the boiler and the heat pump according to the program above. The calibration of these delay functions is very important for the economical outcome.

Calculated economy

A customer has to calculate the profitability before he decides whether or not to make an investment. In this case, he got the following information from the contractor:

-- Total investment cost

-- Calculated energy production by the heat pump (kWh/y)

-- Calculated fuel oil consumption (m^3/y)

- Calculated COP, total average per year value
- Service and maintenance costs, based on an offered contract

It was pointed out that these values were calculated partly on factors which could not be influenced by the contractor, such as:

- Energy content in the fuel oil, kWh/ m^3
- Efficiency of the oil-fired boiler, average per year
- Heating energy needed versus outside air temperature, proportional to inside-outside temperature differences
- Outside air temperature durability per year, equal to statistical average values

and partly on the availability of the heat pump, which is dependent on the quality of the plant, but also on the quality of service and maintenance in the future.

In order to make it possible to judge the importance of these uncertainties, the customer was given sensitivity factors, showing the change in calculated performance when assumed values changed.

With this information available, and knowing the predicted capital costs, the customer found the offer attractive and ordered the plant, including a service and maintenance contract.

Running experience

After construction, the plant was put in operation and the control system was adjusted within a few days. Since then, the plant has been running with almost 100% availability.

Defrosting is started automatically, when the temperature difference between outdoor air and evaporating refrigerant exceeds a set value. According to experience, defrosting is necessary when the outdoor temperature is less than +3°C. At worst weather conditions, defrosting is done four times per day. The air coolers are defrosted in sequence for a total of about ten minutes.

The noise from air coolers tested at the wall of the house, 20m from the air coolers, does not exceed 40 dBA, which is the maximum value recommended by the Swedish authority.

The tested performance values for the first two running years are as follows:

	Calculated	1st Year	2nd Year
Total heating energy demand (MWh/y)	2900	3415	3492
Heating energy delivered by the heat pump (MWh/y)	1350	1333	1390
Elec. energy consumed by the heat pump plant (MWh/y)	582	550	548
COP	2.3	2.4	2.5

The difference between the calculated value on total heating energy demand, and those measured is due mainly to unusually cold winters, and an underestimated value of the efficiency of the boiler.

The customer had calculated a saving of 271,000 SEK in running costs. The first year he saved 287,000 SEK and his capital costs were about 70,000 SEK.

Conclusions

Much development has been made and much experience gained since this plant was built, but the main contracting and technical principles applied are still valid.

The knowledge today about environmental fouling by leaking refrigerants has strengthened the motivation for the concept of using heat pumps of standard design, factory-built and tested. This is because they usually contain less refrigerant, and because all work and all tests on the units are made at optimum factory conditions. The risk for leakage should, therefore, be less.

Other advantages of using standard factory-built heat pump units are:

- Higher reliability
- Better service
- Shorter time for construction and adjustment of the plant

If there are no standard units on the market suitable for the heat source available, an interconnecting brine system has to be supplied. This will, of course, decrease the performance of the plant due to the extra temperature difference in the heat exchanger, but careful design can reduce this disadvantage, and, as a rule, this concept is to be preferred.

As to the contracting concept, one of the difficulties is to create the *main contractor/subcontractor* constellation with all knowledge, experience, and trust needed for making a project successful. This points to the fact that the infrastructure of the system for marketing and contracting ordinary heating plants is not always suitable for heat pump plants. This is obviously one of the obstacles for the penetration of heat pumps in some of the potential market sectors. The reason for this nonadaptation is that the *heat pump plant technique*, after all, is rather young. This weakness in the marketing and contracting habits will be removed, but it will take some time. The first step towards better behavior should be to make those involved aware of these facts.

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Performance Characteristics of Small Reciprocating Internal Combustion Engines

Small reciprocating internal combustion engines are currently being considered as drivers for residential heat pumps. These small engines have characteristics which make them qualitatively similar to larger engines. This fact can be used to advantage when designing engines for use in such difficult applications as residential heat pumps. This paper is a brief review of the basic principles of engine performance especially as it relates to small engines intended for heat pump applications.

For the purposes of this paper, small engines will be defined as un-supercharged, single-cylinder, reciprocating engines with less than 800 cc displacement. These restrictions limit the power of the engines to 15-20 hp (20.1-26.8 kW) which is the largest engine that is likely to be encountered in residential heat pump applications.

Fundamental Relationships

Engine performance can be characterized by four quantities: speed, torque, power, and efficiency. These are also the quantities of greatest interest when using an engine to drive a device such as a heat pump compressor. When comparing engines of greatly different size and power, there are certain parameters which, while not dimensionless, remain relatively constant for engines which are operating at their maximum power levels. This is true even though the engines may be small 2-stroke engines operating at more than 10,000 rpm or a large marine engine that produces more than 10,000 horsepower while turning at less than 100 rpm. These parameters are the "mean piston speed" (MPS), the "brake mean effective pressure" (BMEP), the "power per unit piston area," and the "brake specific fuel consumption" (BSFC). The expressions used to calculate these quantities are shown in Table 1.

The mean piston speed is the average speed at which the piston moves during its travel up and down in the cylinder. It is equal to the distance traveled by the piston in one revolution; that is, two times the stroke, divided by the time required to complete the revolution.

The BMEP is an artificial pressure which, if applied to the piston over the entire expansion stroke, would give the same work output from the engine's driveshaft for the cycle. The BMEP is equal to the engine's output torque times the angle the engine rotates in one cycle divided by the displacement volume of the engine. The BMEP is sometimes described as a "specific torque" since the engine's torque is normalized by its displacement.

Since power is equal to torque times speed, a relationship can be developed between the engine power and the BMEP and the mean piston speed. A useful form of this relationship is the power per unit piston area which is proportional to the product of the BMEP and mean piston speed.

Finally, the BSFC is the ratio of the fuel consumed by the engine to the power produced and is a measure of the fuel required to maintain a given power output from the engine.

Mean piston speed:	MPS	$= 2\ S\ N$
Brake mean effective pressure:	BMEP	$= \frac{4\pi T}{V_D}$
Power/piston area:	P/A	$= \frac{1}{4}$ (BMEP)(MPS)
Brake specific fuel consumption:	BSFC	$= \frac{\dot{m}}{P}$

S = Stroke
N = Engine speed, rev/min
T = Torque
V_D = Displacement volume
$\dot{m}$ = Fuel flow rate

Table 1. Fundamental relationships

Design and Operating Parameters

Speed

Engines generally have a wide range of operating speeds. This can be used to advantage when driving a refrigerant compressor by using engine speed variation to modulate the compressor. Obviously, the engine must have sufficient torque capacity to supply the compressor's power requirement over the entire speed range.

There is an optimum speed for minimum fuel consumption by the engine. This speed is relatively low compared to the speed limitation imposed by inertial stresses in the engine components; therefore, engines are usually operated at much higher speeds than their optimum fuel economy point. The reason for this is to allow an engine of a given size and weight to produce the greatest amount of power. The optimum fuel economy point comes about because of the competition between heat loss and friction. The power required by the engine to overcome friction tends to increase with the square of the engine speed. However, the heat loss from the combustion gases to the combustion chamber walls tends to decrease as a fraction of the total energy supplied by the fuel as less time is available at higher speeds. This trade-off typically produces an optimum BSFC at about 60% of the engine's rated speed.

Figure 1 shows the distribution of mean piston speeds for a variety of small engines of different sizes and manufac-

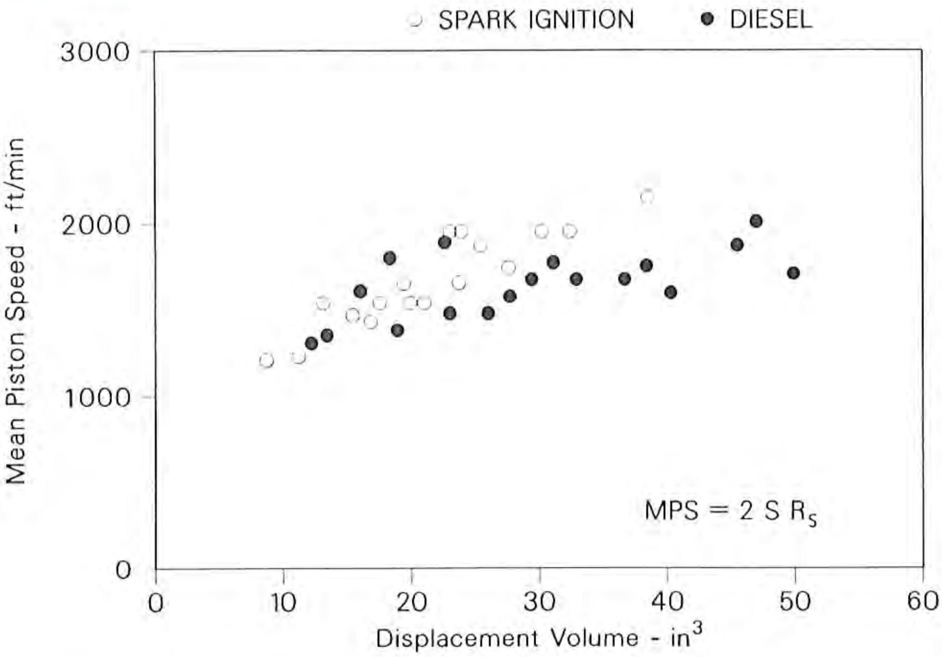

Figure 1. Mean piston speeds for single cylinder engines

turers. The mean piston speed for a wide range tends to be constant between 1600 and 2000 feet per minute (10.2 m/s). This range has been determined by experience to give the best combination of lifetime and power-to-weight ratio. Engines which operate above this level, for example, racing engines, normally have reduced lifetimes. Engines operating below this level generally are stationary engines with low power-to-weight ratio but very long expected lifetimes. It should be recognized that many of the engines in this size range are intended to be used to drive electrical generators and thus operate with a maximum governed speed of 3600 rpm (in the U.S.). Engines below 30 in displacement could operate at higher speeds but there is little incentive given their intended use.

Load

The load on the engine is normally characterized by the BMEP. When the engine is operating at a steady speed, the torque requirements of the driven device, say a compressor, dictate that the throttle be opened by the amount necessary for the engine to supply the torque. Since the BMEP is directly related to the torque, it rises also. At high BMEP levels, the peak pressures and temperatures in the cylinder are also much higher. Friction is only a weak function of load, rising very slightly at higher BMEP, so the fraction of the engine power necessary to overcome friction decreases up to maximum power where the throttle is wide open. If the fuel-air ratio is held constant, then the BSFC will minimize at the wide-open-throttle condition. Generally, the carburetor is designed to provide a richer mixture at wide-open-throttle so there is some fuel wasted due to incomplete combustion, and the BSFC minimizes before the wide-open-throttle point.

Figure 2 shows the range of BMEP for the small engines plotted earlier. The BMEP ranges from approximately 90 psi to 120 psi, which is about the maximum that can be obtained from an un-supercharged engine without special intake system tuning designed to increase air flow at specific speeds. Since the peak cylinder pressure increases at higher BMEP, the stress levels on the engine, both mechanical and thermal, are greatest at high BMEP also. Engines designed for long life usually operate at a lower than optimum BMEP with the resulting sacrifice in BSFC and power-to-weight ratio.

Figure 3 shows the power per unit piston area calculated for the mean piston speed and BMEP data from Figures 1 and 2. There is an obvious reduction in this quantity below about 30 inches to about half its value at larger displacements. This indicates that current small engines are not as heavily loaded, even on a normalized basis, as their larger counterparts. Although engine lifetime data is not readily available on these engines, it is probable that this is the major reason for the drop in this parameter. Traditional applications of small engines require them to be inexpensive and easy to manufacture. The result is lightly loaded engines with typical lifetimes as short as 250 hours for small

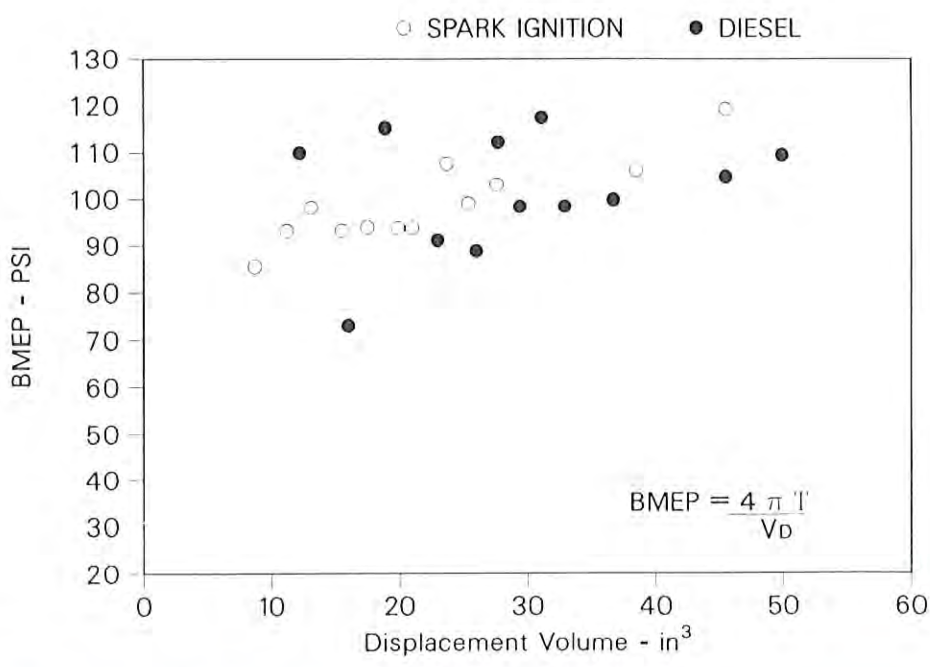

Figure 2. Brake mean effective pressure for single cylinder engines

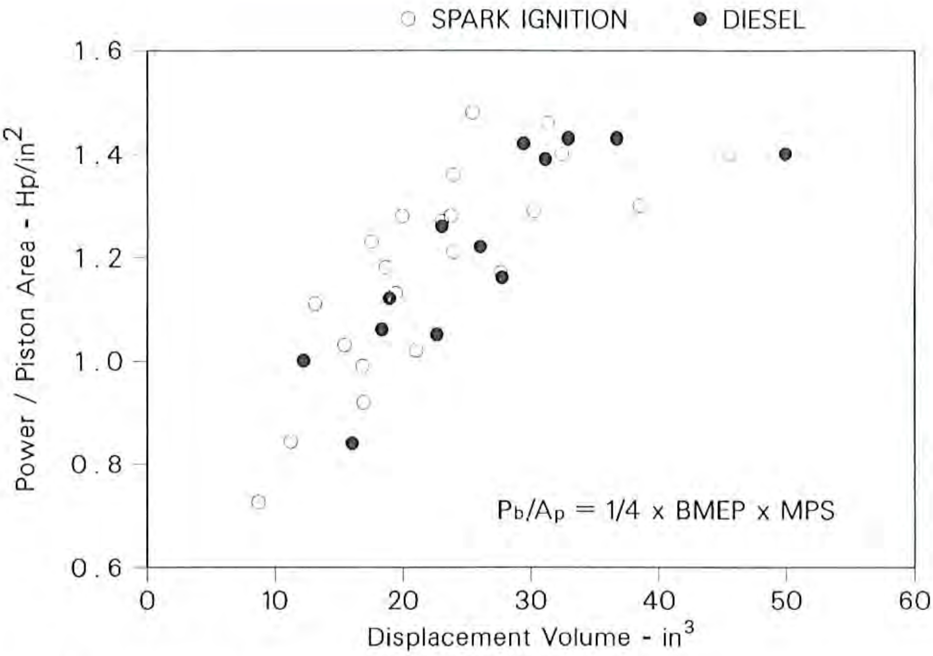

Figure 3. Rated power/piston area for single cylinder engines

also be reduced by retarding the spark timing from the point which gives the best torque or fuel economy. The reduction in nitric oxide emissions can be dramatic for only a slight penalty in fuel consumption or reduced torque.

light-duty chainsaw engines to 1000-2000 hours for stationary small generator engines.

Fuel-Air Ratio

The mass ratio of fuel and air which enters the cylinder is also a parameter which influences engine performance. The ratio must always be within the flammability limits of the fuel. Emissions, fuel economy and power constraints even more severely restrict the range of acceptable fuel-air ratios. Figure 4 shows the effect of fuel-air ratio on exhaust emissions from a typical engine. Carbon monoxide and unburned hydrocarbon emissions are very high when the engine is operating rich. Nitric oxide emissions maximize when the mixture is slightly lean and are low on the rich side or when the mixture is very lean. Hydrocarbon emissions tend to rise under very lean conditions as occasional misfiring cycles are encountered and unburned fuel is passed directly through to the exhaust.

The BSFC tends to minimize around stoichiometric, or slightly lean. Under lean conditions the flame speed of the burning mixture tends to decrease which means that the combustion occurs at a later time in the expansion stroke. Less of the fuel energy is converted to work, lowering the thermodynamic efficiency of the cycle, and raising the exhaust temperature.

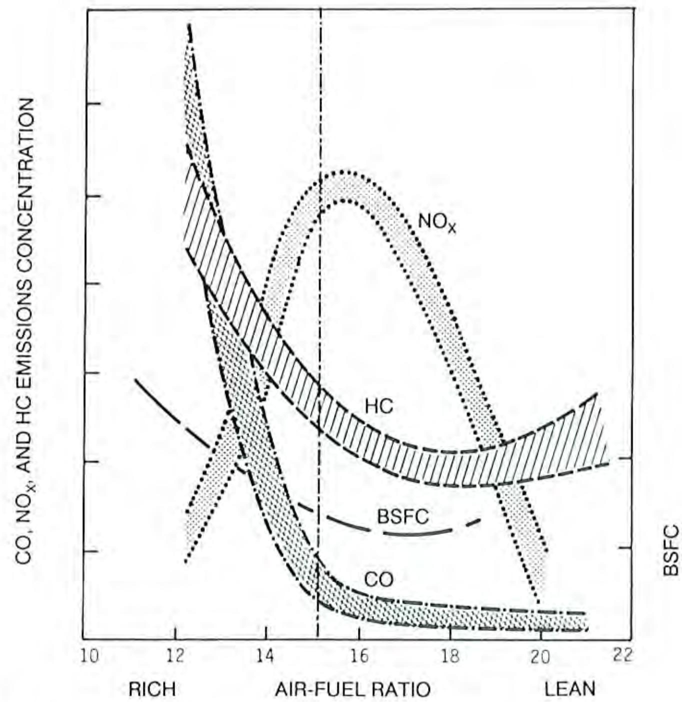

Figure 4. Effect of air-fuel ratio on emissions (Mondt, 1982)

A catalytic converter can be used to control emissions from engines but the catalyst must be operated very close to stoichiometric. Figure 5 shows the fuel-air ratio window for acceptable catalyst performance. Automotive engines deal with this requirement by operating stoichiometric or slightly rich and then adding air into the exhaust manifold as necessary to maintain stoichiometric conditions. Nitric oxide emissions can

Fuel Type

When a fuel-air mixture is compressed in the cylinder, the resulting high temperature causes chemical reactions to occur which will culminate in an extremely violent combustion event if sufficient time elapses before the spark-induced flame front consumes the mixture. This auto-ignition of all the remaining unburned fuel and the resulting vibration of the engine structure is called "knock."

The tendency of the fuel-air mixture to auto-ignite becomes stronger as the temperature and pressure of the mixture increases. Thus, as the engine's compression ratio is raised, knock becomes more of a problem. Without knock, the optimum compression ratio in most engines would be in the area of 12-16. However, knock limits the compression ratio in gasoline engines to about 8-9. Higher compression ratio improves the efficiency of engines but excessive knock destroys any gains as the severe pressure waves induced by the rapid combustion scrub the walls of the combustion chamber, disrupting the boundary layers and increasing the heat transfer.

The octane number is a measure of a fuel's resistance to auto-ignition in the cylinder. The most likely fuel to be used in a residential engine-driven heat pump application is natural gas, which is typically 90% methane. Methane is an excellent fuel for spark-ignition engines with an octane number estimated to be about 120. This compares to octane numbers of 87 for U.S. unleaded gasoline and 91-93 for unleaded premium. Use of natural gas as a fuel would allow the compression ratio of the engine to be increased above the level allowable for gasoline engines. This should result in a significant improvement in the efficiency of the engine.

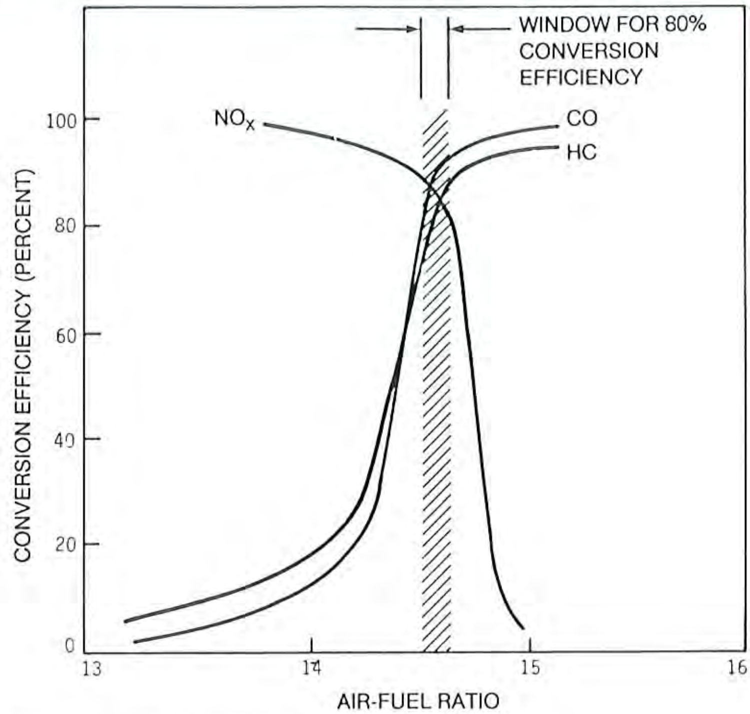

Figure 5. Conversion efficiencies for a typical three-way catalyst (Mondt 1982)

Applications of Advanced Technology

Two modern developments in engine technology which may have applications in engine-driven heat pumps are the use of charge stratification and ceramic materials. Charge stratification is an attempt to reap the benefits of both rich and lean fuel-air ratios at the same time. The charge which is in the vicinity of the spark plug is rich, either through the use of a separate chamber which has a special intake valve as shown in Figure 6, or through late addition of the fuel to the intake air, as shown in Figure 7. The major portion of the fuel-air charge is lean. Ignition of the rich charge is reliable and the rich combustion produces little NOx, although significant amounts of rich products such as carbon monoxide and partially burned fuel are produced. When these rich products are mixed with the lean charge the rich products are oxidized and the result is efficient fuel utilization with low levels of emissions.

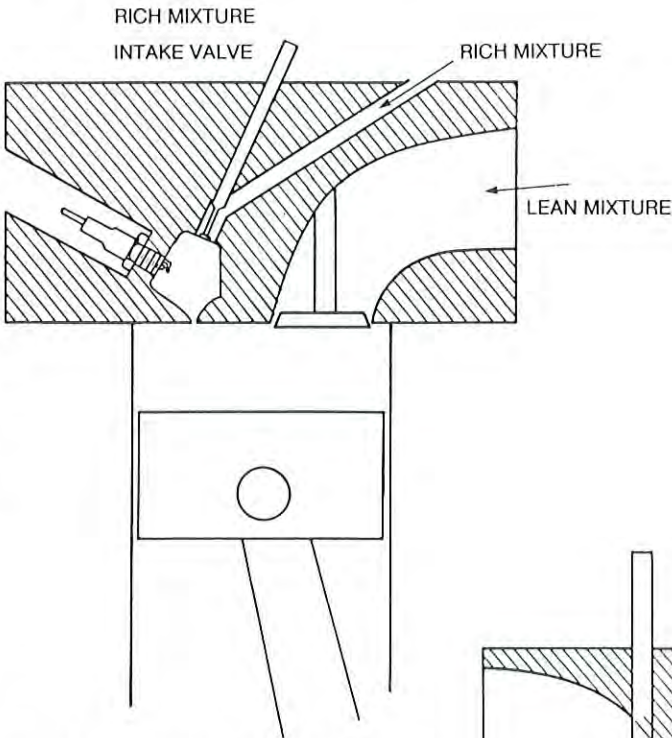

Figure 6. Divided chamber charge stratification

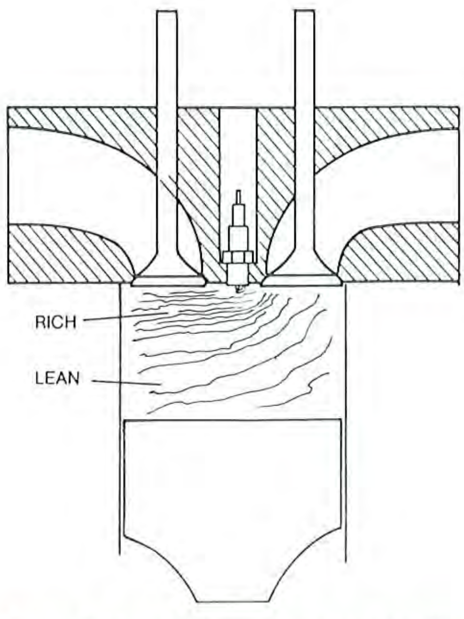

Figure 7. Charge stratification in main cylinder

Ceramic materials also offer the possibility of dramatic improvements in engine performance characteristics. Although use of ceramic materials for the combustion chamber walls has not brought about significant improvements in the efficiency of engines, it does allow the elimination of the cooling system with its inherent fuel consumption and reliability penalty. Because ceramic materials provide superior dimensional stability with temperature, they may allow the elimination of piston rings with the combustion chamber sealing problem dealt with through the use of close tolerances and gas-bearing technology. Use of extremely hard ceramic materials in the critical locations of camshaft-tappet interface, piston-wrist pin and exhaust valve seats should prolong the life and reliability of future engines. Significant problems remain with ceramics, including high temperature tribology, the cost and reliability of the ceramics and whether the ceramics should be present as coatings on conventional materials or as monolithic components. The solutions which are found to these problems will dictate the extent of ceramic use in engines, but it is certain that ceramic materials will play an important role in future engines of all sizes.

Conclusions

Small engines have many characteristics which make them desirable for use as heat pump drivers. A natural-gas-powered engine with high compression ratio operated at relatively low speed and high BMEP coupled with a heat pump should provide a system which can compete very effectively with either natural-gas-fired furnaces or electrically powered heat pumps. The application of new technology in the area of charge stratification and ceramics should enhance the competitiveness of this option. Serious problems remain in the area of engine durability and maintenance requirements, however, long-life engines are currently being produced for marine and oil-field applications and this technology should be employed in smaller applications.

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Flue Gas Condensation with Outdoor Air Heat Pump in an Oil-Fired Boiler Station

Increasing environmental awareness in today's society imposes demands on economical methods for energy conservation and environmental improvement in the production of heat. A large share of the heat production is carried out by means of combustion of fossil fuels, wood, chips, refuse, etc. Depending upon the type of fuel, the flue gases emitted to the atmosphere upon combustion of the fuel contain substances of varying composition. Examples of typical substances are SOx, NOx, chlorides, heavy metals and various kinds of particulate matter. There are different methods of cleaning the flue gases/ exhaust gases, usually through the addition of chemicals. For the most part, these methods are practically and economically viable only for large plants.

1. Introduction

Some energy saving can be achieved by lowering the temperature of the flue gases/exhaust gases via heat exchange. The heat extracted from the flue gases is then supplied to, for example, the water in a heating pipe or combustion air. The combustion of fuel then decreases to a corresponding extent, so that some reduction of emission is attained. If conventional material is used in the heat exchanger, the effect will be nevertheless only slight ($\leq$ 5%). The reason for this is that if the flue gases are cooled down appreciably, corrosive substances begin to condense. In oil firing, for example, precipitation of corrosive sulphur compounds occurs at a temperature of around 130°C. A worthwhile energy saving (10-20%) requires cooling down to below the dewpoint of the water contained in the gases. This temperature varies, but in oil firing it is usually around 40-50°C. Such a severe temperature drop gives rise to corrosion problems of considerable magnitude, but at the same time provides an opportunity for an interesting precipitation of substances that harm the environment.

Several general studies of the flue gas condensation have been carried out, among them publication 138:1985 from the Swedish Council for Building Research (in Swedish). Certain installations of different kinds have been built and commissioned, from which measurement data are also reported. One application which appears to be of special interest to study is oil-fired group centers supplemented with indirect outdoor air heat pumps.

The municipal housing company, Nackahem, in Nacka (outside Stockholm), will be installing a large outdoor air heat pump in its boiler station in Fisksaetra in 1987. In conjunction with this, the possibilities of utilizing the heat pump for flue gas condensation have been discussed. In order to obtain factual material on which to base a decision, this preliminary study/feasibility study has been carried out for an installation of this kind. The work was thus concentrated on a rapid and concrete continuation of the project.

2. Plant description

Fisksaetra boiler station serves some 2500 apartments, three schools, and a shopping center. The heat is produced in three oil boilers, two disconnectible electric boilers and a heat pump installation (Fall 1987). A 7 MW oil boiler was replaced in 1987. The total annual heat requirement is approximately 41 GWh.

The heat pump is built up around two screw compressor units, type STAL VSP73EC. The heat source is outdoor air from which heat is collected indirectly via cooling coils placed on the roof of the machine building. The heat transfer medium is $CaCl_2$, which has excellent properties at low temperatures. The temperature levels of the heat system permit the use of refrigerant R22 in one of the units. In the other, R12 is required to assure a sufficiently high supply temperature.

At the balance temperature, +8°C, output requirement is a good 4 MW, and at an outdoor temperature of 0°C the heat pump is estimated to supply approximately 3.4 MW of heat to the network. It is planned to be taken out of operation at outdoor temperatures below -10°C, implying an intentional stop of some 250 hours per year. Installation of flue gas heat exchangers will nevertheless enable the heat pump, at reduced capacity, to utilize the flue gases as the only source of heat at low outdoor temperatures. All in all, the heat pump will generate approximately 23 GWh of heat per year, corresponding to 55% of the demand.

The heat demand of the area is evident from the duration diagram, Figure 1, in which the heat deliveries from the heat pump and oil boilers and the envisaged flue gas heat recovery are also marked.

A diagram showing the system arrangement is illustrated in Figure 2, which is highly simplified in order to accentuate the central factors. It should be pointed out in particular that the cooling coils are divided into several groups which are defrosted in sequence.

All equipment for cooling of flue gases from the oil-fired boilers are placed in parallel with existing flue gas ducts. By these means, operation is assured even in the event of malfunctioning of newly installed equipment. The flue gas cooling is dimensioned to handle the flue gases from a boiler at full output, approximately 9000 Nm^3/h at an output of 7 MW. The connection principle permits handling of flue gases from any boiler.

The fan (F) located after the heat exchangers is controlled so that the flow through the heat exchangers is always somewhat greater than the flow of flue gas from the boiler in question. This gives a slight leakage of outdoor air

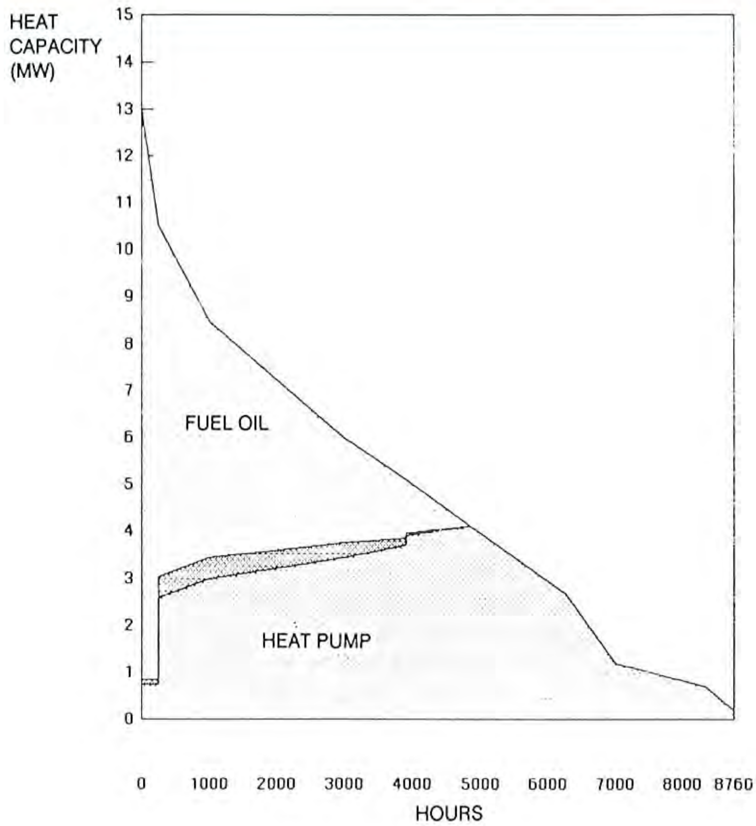

Figure 1. Capacity duration

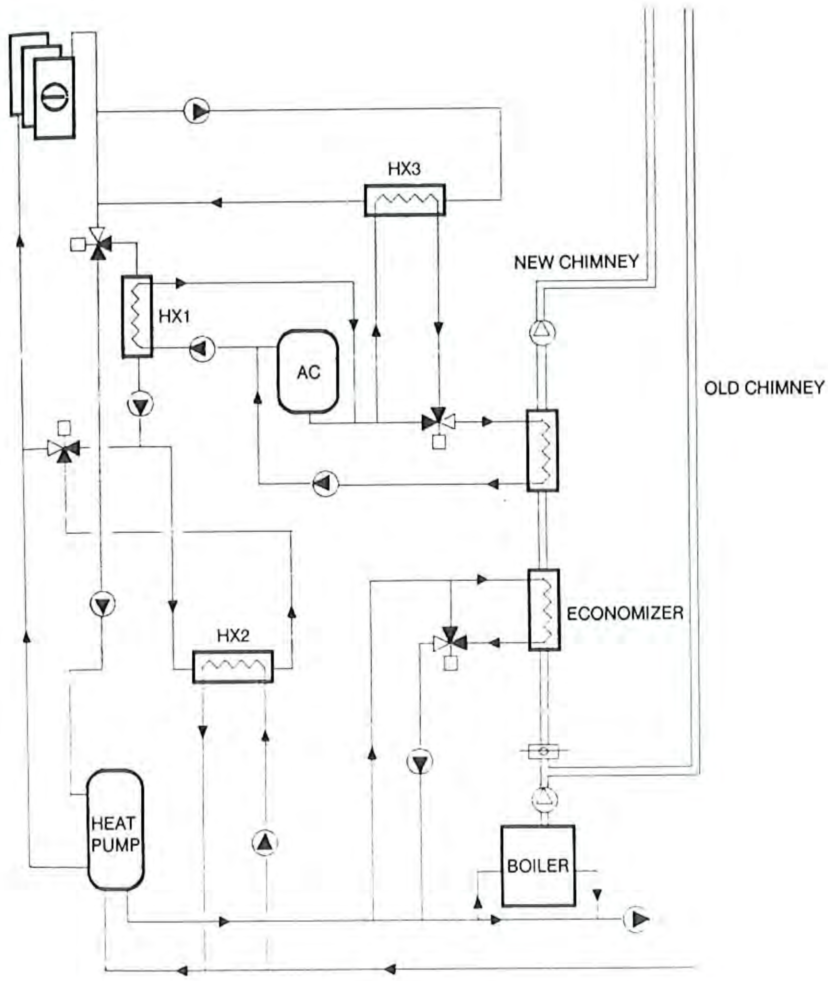

Figure 2. Piping system (simplified)

down through a chimney flue. The temperature of the flue gases, true enough, is lowered a few degrees before the first exchanger, but at the same time it is ensured that no fuel gases will make their way up through the chimney and condense inside it.

Two flue gas heat exchangers are installed. The first, in the flow direction of the flue gases, is made as a conventional economizer of carbon steel. Here, then, no condensation of corrosive substances takes place and instead the flue gas temperature is maintained at a satisfactorily high level. The heat from this exchanger, a maximum of approximately 100 kW, is supplied directly to the heat line after the heat pump.

The flue gases, which have thus been cooled to 130-140°C, are supplied to the second flue gas heat exchanger where the temperature is steeply lowered and condensation of the water takes place. The outgoing flue gas temperature varies between around 30°C and 50°C, depending on the moisture content and flow rate of the flue gases and on the coolant temperature. Approximately 300 kW of heat is obtained from this exchanger. This heat is used primarily to defrost the cooling coils of the outdoor air heat pump (heat exchanger HX1). Otherwise, the defrosting heat would have to be taken from the heat line (HX2), implying that prime heat would have to be replaced. When the temperature in the accumulator (AC) is increasing, the coolant temperature into the heat exchanger will also increase, whereupon the output decreases. This is unfavorable for two reasons. For one thing, the capacity of the installed heat exchanger is not properly utilized, and for another, moderate cooling of the flue gases results in reduction of the cleaning capacity.

This effect could be eliminated by installation of a third heat exchanger in the chimney, cooled by the heat transfer medium of the heat pump. The output from the latter will then increase as that from the second heat exchanger decreases and vice versa. The resulting increase in the heat transfer medium temperature will not pay for this additional investment. Instead, this third heat exchanger must be regarded mainly as a flue gas cleaning measure.

Another way of achieving the same result as with a third heat exchanger as described above is to recool the coolant from the accumulator with the return cool carrier via a heat exchanger (HX3). This gives numerous decided advantages; the investment cost will be lower and a definite borderline will be attained between heat transfer medium and flue gases. There is no risk of jeopardizing the heat transfer medium side of the heat pump in event of a failure in the flue gas heat exchanger.

3. Thermal engineering calculations

The annual energy yield from the flue gas heat exchangers can be found from the durability diagram, Figure 1. At a maximum output of approximately 400 kW, roughly 1.4 GWh/year, equivalent to 150 m^3 of oil, can be obtained from the flue gases. This energy can be utilized in the following ways:

a) for defrosting
b) for the cool carrier return line
c) for the heat carrier return line
d) for preheating of the combustion air

One prerequisite for exploiting a large portion of the flue gas heat ($\geq$10%), is a very low temperature of the heat absorbing medium ($\leq$10°C). This requirement is satisfied with alternatives a and b, and to some extent with alternative d. In preheating of combustion air, however, only a modest effect can be transmitted at low temperature.

From the standpoint of thermodynamics, alternative b is inferior to the others, as the heat is supplied to the cold side of the heat pump. A supply of 100 kW heat corresponds to an increase in the heat transfer medium return temperature of barely 0.3°C, which increases emission from the heat pump by 20 kW and its COP by 0.01 units, or 0.5%. In addition, a higher heat transfer medium temperature will slightly reduce the need of defrosting and prolong the operating time of the heat pump at low outdoor temperatures.

The same temperature rise can also be obtained with a larger cooling coil area. A comparison with the conditions applicable in the present case indicates that 100 kW of heat from the flue gases to the cool carrier is equivalent to an increase of around 6% in the kA-value of the cooling coils. Note that a comparison of this nature is unique for each individual plant, depending on the original dimensioning, so that no general conclusions can be drawn from this calculation.

In the first instance, then, the flue gas heat should be exploited to defrost the cooling coils of the heat pump. The combination of outdoor heat pump and flue gas condensation is obviously particularly beneficial. If the defrosting need does not suffice to exploit enough thermal capacity, extra investments may be considered.

For Fisksaetra, a system is planned in which the fuel gases are first cooled by the heat carrier, alternative c, whereupon the heat is utilized for defrosting, alternative a, and finally "dumped," in the case of surplus heat, to the return line of the cool carrier, alternative b.

4. Environmental engineering effects

Swedish legislation stipulates that the maximum sulphur content of heavy fuel oils (grades 3-5) may not exceed 1%, and the oil companies report that the sulphur content is normally 0.8-1%. According to available analysis results, the content is usually fairly close to 1%. The average density for grade 4 fuel oil is approximately 0.95, so that with a sulphur content of 1%, the emission of sulphur amounts to approximately 0.95 ton/100 m^3 of oil. If the oil consumption is 1500 m^3, sulphur emission will thus amount to about 14 tons per year.

The calculation of emissions of particulate matter is somewhat more uncertain, since this depends on the condition of the boilers, how the boilers are looked after, and the quality of the oil. From a good plant, emissions of particulate matter normally amount to 0.5-1 kg/m^3 of oil or 50-100 kg/100 m^3. If the oil consumption is 1500 m^3 per year, emissions of particulate matter will amount to

750-1500 kg/year. In addition, however, relatively large amounts of particulate matter are emitted when the boilers are cleaned.

Simple collectors, such as certain types of cyclones and inclined plates, reduce emissions of particulate matter primarily on cleaning, whereas during operation they usually have a collecting efficiency of only 10-20%.

Cleaning effects in flue gas condensation

Sulphur emissions are mainly decreased in that 80-90% of the sulphur trioxide in the flue gas is absorbed in the condensate. Sulphur dioxide, on the other hand, is absorbed to a very slight extent according to available measurement results. In total, a roughly 10% lower sulphur content in the flue gases can be anticipated after a condensation. The sulphur emission will then be about 0.86 tons/100 m^3 of oil.

If the oil consumption without flue gas condensation is 1500 m^3, it can be estimated that the consumption will have decreased to about 1350 m^3 after installation of flue gas condensation. 10% sulphur collection thus reduces sulphur emissions by 13.5 x 0.95 x 0.1 = 12.8 tons/year. Additionally, emissions from the saved 150 m^3 of oil will be entirely eliminated, giving 1.5 x 0.95 = 1.42 tons/year.

Since the uncertainty is considerable, a reduction should be regarded as lying in the range of 300-600 kg/year.

Available measurement data indicate that a 20-50% reduction of metals such as lead, cadmium and mercury may be anticipated.

In the boiler station there is a basin with a volume of approximately 5 m^3 for neutralization of the cleaning water. According to information, cleaning with water takes place once per year and the volume of water is approximately 2 m^3 per boiler. The entire basin volume can then be used for neutralization of any condensate, provided that it is emptied prior to cleaning. Since the amount of condensate will be about 4 m^3/day, it is necessary to install new regulating equipment and to arrange for continuous diversion of the neutralized water by either flowing or pumping.

Other supplementary equipment may be necessary depending upon future requirements imposed by the authorities; for example, emissions of heavy metals into water. Today, a great deal of heavy metals are contained in the soot removed in water cleaning which now go to the drains.

Hitherto, the authorities have not issued any guidelines for treatment of water from condensation of flue gases. This applies even to water cleaning, although this method is applied at a large number of boiler stations. It is, nevertheless, probable that the Swedish National Environment Protection Board, within the not too distant future, will issue guidelines for treatment of cleaning water and condensate. In this context, the same requirements will probably be imposed on the discharged water as on sewage water from engineering works with chemical surface treatment. Those responsible within the Swedish National Environment Protection Board also recommend that the treatment of condensate be elaborated in accordance with these requirements. The metal content is common for the two types of water in question.

As previously mentioned, source material relating to the amounts of metal in condensates is inadequate but nevertheless, it is perfectly clear that certain metals are present in higher contents than are permitted in, for instance, sewage water from surface treatment facilities. In this particular case, 1500 m^3 of oil is normally combusted per year, so that the average amount of condensate can be estimated at 3.5 m^3/day or about 900 m^3/year. On the basis of a normal analysis of grade 4 fuel oil, and on the assumption that the entire amount of metal ends up in the condensate (the true amount is naturally lower), the following table is obtained:

Metal	Kg/1500 m^3 oil	Max. conc. in condensate mg/l (900 m^3/year)	Probable limit value mg/l
Iron	8	9	2
Chromium	0.06	0.07	1
Nickel	10	11	1
Lead	4.6	5.1	1

The total reduction of sulphur emissions will thus amount to approximately 2.7 tons/year.

Emissions of particulate matter are probably decreased by at least 50%. If the existing flue gas cleaning equipment reduces the emissions by 20% the reduction for the same case as above, after flue gas condensation, will be as follows:

From combusted oil:

13.5 x (50-100) x 0.8 x 0.5 =	270-540 kg/year

From saved oil:

1.5 x (50-100) x 0.8 =	60-120 kg/year
Total	360-660 kg/year

No limit value for mercury and vanadium have been set for the surface industry, but is is probable that the tolerable limit for mercury will be from a few 10 g/l and for vanadium 1 mg/l. It is evident from the table that the concentrations of iron, nickel, lead, vanadium and possibly mercury in the condensate can be higher than future limit values. With conventional cleaning, pH adjustment, flocculation and coseparation, it should be possible to manage the stipulated limit values.

The existing basin should thus be divided into three or four sections:

-- pH adjustment

-- flocculation

-- sedimentation

-- possible pump pocket for diversion

The largest part, approximately 3.5 m^3, consists of sedimentation, which probably will have to be sludge-sucked 5-10 times per year.

Heat recovery with condensation of flue gases obviously gives lower emissions of environmentally harmful substances through the chimney. Nevertheless, there is every reason to study the influence on the immediate environment, on the basis of the fact that the flue gases are severely cooled.

A comparison between the influence of the temperature (displacement force) and speed (amount of movement) on the throwing height, nevertheless, reveals that the influence of the temperature is slight, being 5-10%, i.e., negligible in practice. This applies at a constant outlet speed. A lowering of the temperature of the flue gases from 450 K to 300 K, nevertheless, implies that the volumetric flow decreases by one third. Allowance for this must obviously be made when dimensioning a new chimney flue.

An improved local environment can then be anticipated since

-- the total oil consumption is reduced,

-- emissions of sulphur trioxide, which is far more readily soluble in water than sulphur dioxide and therefore precipitates first, will be appreciably lower, and

-- low temperature flue gases "attract" less water, which may then condense from surrounding air.

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Geothermal Heat Pump Application in Upper Austria

In 1918, an important hot thermal water resource was discovered in Bad Schallerbach, Upper Austria. Since 1922, the therapeutic uses of this famous sulfur thermal station expanded. Based on scientific research and preliminary projects, the use of the low-temperature geothermal energy by heat pumps started in 1985. Through the joint efforts of the management of the Bad Schallerbach Thermal Station, the Upper Austria Utility Company Oberoesterreichische Kraftwerke AG (OKA), and the local companies, the project expanded to a size of nearly 4 MW_{th}. Excellent performance results and important reductions in local pollution characterize this project as the most important of this type in Austria.

In the early decades of this century, increasing activities for oil drilling started in different countries of the Austro-Hungarian monarchy. Some of these drillings were located in Upper Austria. The results of this early phase of oil research have been modest. In 1918, an important geothermal resource with a high sulfur content was discovered at a depth of about 470m, near the village of Schallerbach (Figure 1). By 1922, the first therapeutic station was constructed and the therapeutic applications of this geothermal sulfur-water have been expanding steadily ever since (Figure 2).

A second drilling well was successfully installed in 1979, equipped with modern tubing and installations, operating at the same depth as the initial well. The water

Figure 1. Geothermal source in Schallerbach, discovered in 1918 (first bathers)

Figure 2. Thermal station, Bad Schallerbach, 1987

output is completely utilized for therapeutic applications. The total water output of both artesian wells varies between 50 l/sec in winter and 70 l/sec in summer, at a constant temperature of 38°C.

The therapeutic uses are for relief of rheumatism, sciatica, gout, paralysis, sequelae of injuries, geriatric ailments and recuperation. As an important therapeutic center with indoor and outdoor bathing facilities, Bad Schallerbach represents an internationally recommended health resort with about 378,000 medical treatments in 1986.

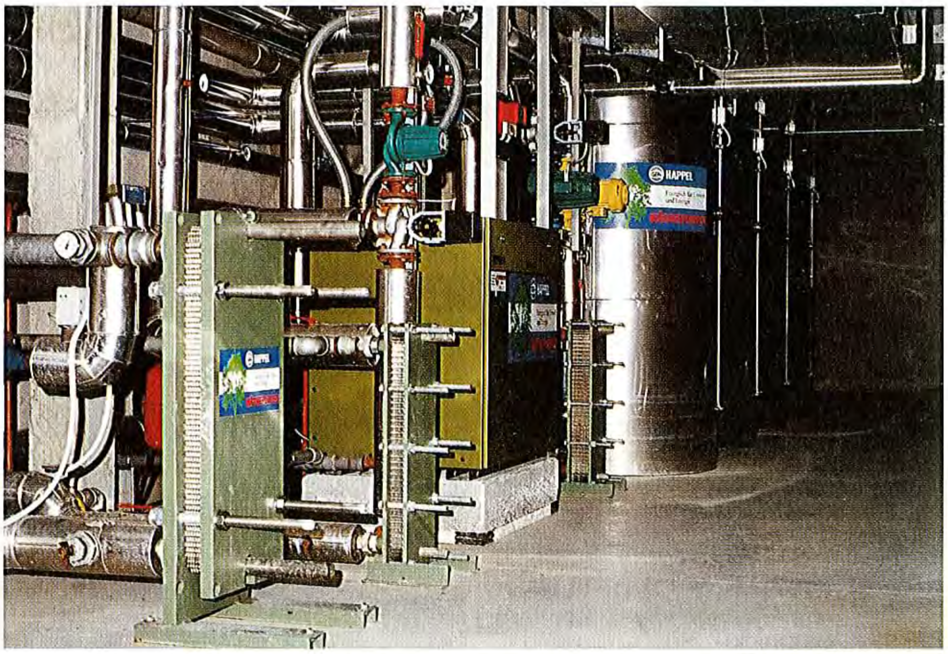

Figure 4 View of hot tap water installation, heat exchangers and heat pump (Elisabeth)

As a consequence of the first oil embargo, the idea emerged to use the thermal sulfur springs not only for therapeutic reasons for also for energy applications. Based on different preliminary studies and finally influenced by the events of the second oil embargo, the authorities of the health resort at Bad Schallerbach approved investments in a heat-pump-assisted geothermal energy project. Scientific studies at the University of Graz, Institute of Thermal Engineering, developed the basic ideas of the project, assisted by the energy consulting department of OKA. The project was conducted by local companies and manufacturers.

The project was accomplished in two phases, with a total of five heat pump installations in decentralized structures and separated for heating and hot tap water applications. The geothermal energy is used directly through heat exchangers for preheating of air and cold tap water. The heat pump installations are designed to achieve the desired temperature level of the specific system. The geothermal water after ini-

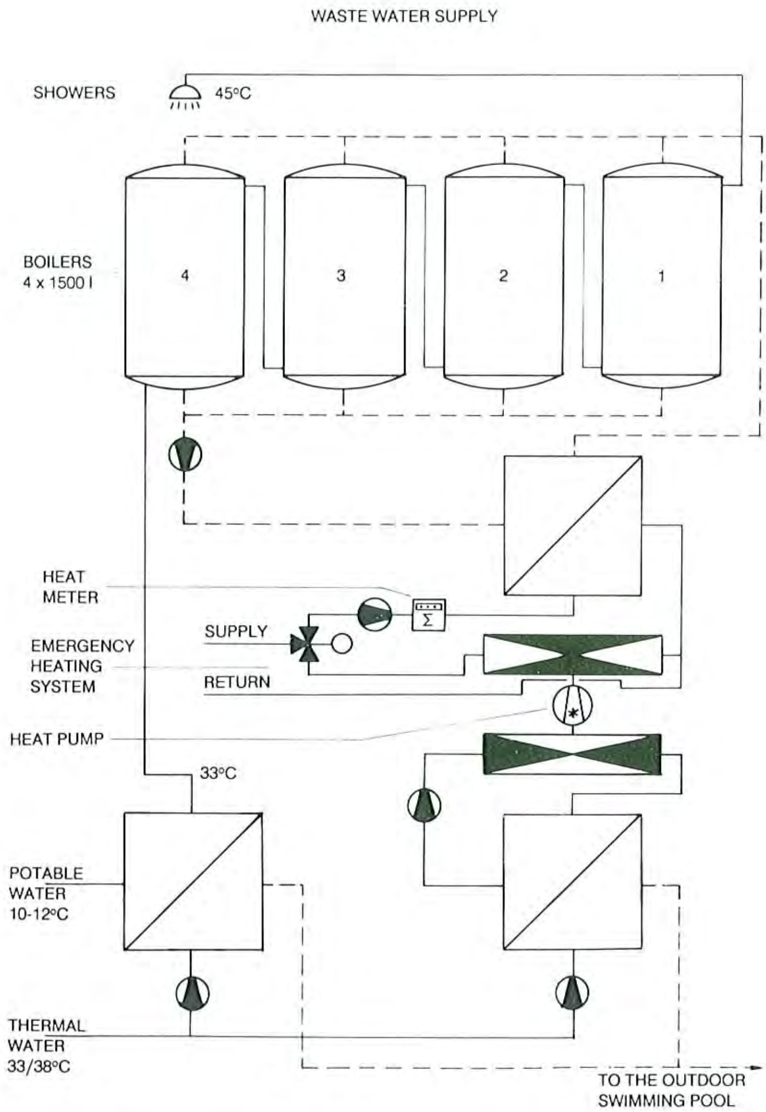

Figure 3. Hydronic scheme of hot tap water installation, indoor bath

Figure 6. Heating system indoor bath, general view of heat pump installation (Leopold)

tial therapeutic use and direct application by heat exchangers serves as heat source for the water/water heat pump installations. The existing oil-fired system has been integrated to cover the peak load of the system in a bivalent parallel operation mode.

In the summer of 1985, the first step of the project was completed for the indoor bath installations. The hydronic scheme of the system is presented in Figure 3. Figure 4 shows the total view of the installation. About 50% of the energy needs of hot tap water production is realized by the direct heat exchange and 50% by the heat pump. The total coefficient of performance is about 9.

Figure 5 shows the hydronic scheme of the heating systems of the indoor bath and Figure 6 gives a general view of the installation. An important design feature of the system has been the fresh air system. It appeared to be very difficult to change the existing heating system. Due to a necessary increase of the inlet-temperature of the heating system, a reduction of the COP from 4.5 to 3.5 resulted, including all auxiliary equipment.

Based on detailed measurements of thermal and electric parameters, and computer-assisted analysis performed by OKA, a sound basis for investment decisions on the second phase of the project existed. This important expansion of the project with 690 kW heating load and 170 kW for hot tap water production started operation in May 1987.

The specific geothermal energy potential of the Bad Schallerbach source is 250 kW_{th}/K. By the realization of phases 1 and 2, a total potential of 1300 kW_{th} is used (see Table 1). In order to simplify operation of the plant, the managing director named the five decentralized heat pump installations for the famous Babenberger's and Habsburger's of Austrian history.

The temperature loss due to therapeutic application is about 5°C and the energy use of phases 1 and 2 decreases the temperature of the water by a further

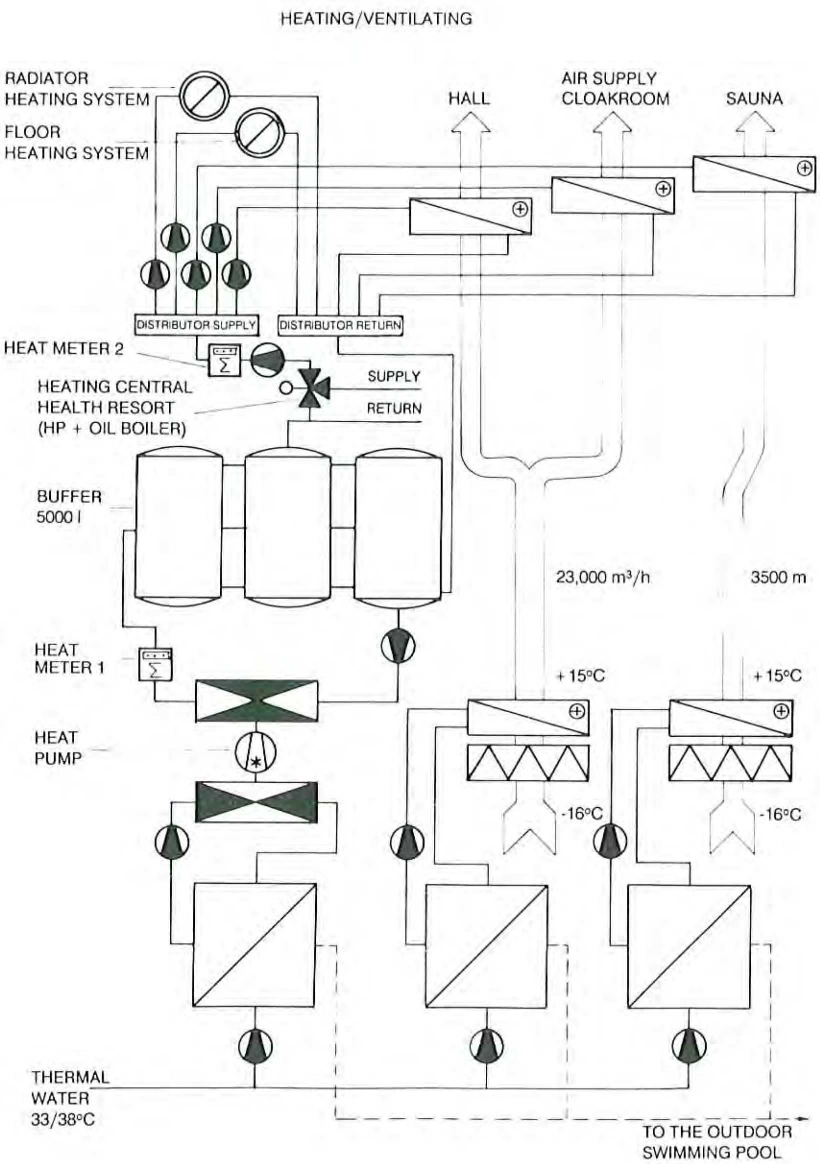

Figure 5. Heating system indoor bath, hydronic system

5°C. Therefore, the remaining temperature level of 28°C of the source water appears to be attractive for future energy use. Based on measurements, consulting and conferences with all partners involved, the decision was made to expand the geothermal energy use.

About 800m from the therapeutic center of Bad Schallerbach, a major rehabilitation center with a thermal peak load of 2.2 MW_{th} will be connected with a "cold district heat piping" of 800m length. The thermal water of 28°C serves as the heat source of a heat pump installation with six units in total. The operation of this plant is scheduled for autumn 1987.

All three projects together, the heat-pump-assisted use of low-temperature geothermal energy in Bad Schallerbach represents 3.8 $MW_{thermal}$ energy use; the reduction of emissions of the existing buildings heated by this new technique is about 95%, representing an important local environment factor. The heat pump installations of Bad Schallerbach represent the most important installations of this kind in Austria and are attracting increasing national and international interest.

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	Indoor Bath		Therapeutic Center		
	Heating System	Hot Tap Water	Heating System	Hot Tap Water for Electro-therapy	Hot Tap Water for Underwater-therapy
Name	"Leopold"	"Elisabeth"	"Rudolf"	"Margarete"	"Beatrice"
Type	GEA-Happel WPR 50	GEA-Happel HWW 12	GEA-Happel WPR 120	GEA-Happel WW 24	GEA-Happel WW 14
El. connected load incl. preheating	36 kW	12 kW	74 kW	8 kW	5 kW
Therm. load incl. preheating	395 kW	174 kW	690 kW	130 kW	40 kW
Share of delivery by heat pump	95%	100%	90%	100%	100%
Reduction of oil consumption	220 t/year		225 t/year		

Table 1. Summary of installed heat pumps and thermal outputs

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Application of Heat Pump Systems in Cold Regions

The use of air-source heat pumps in cold regions is limited by special problems. In this article, these problems are identified and solutions proposed. System simulation is being conducted at the Government Industrial Development Laboratory in an effort to develop commercially successful heat pump systems for cold regions.

1. Collection of heat from air in cold regions

In order to introduce a heat pump system using air as the heat source in cold regions, a number of technical problems must be solved. One is related to the fact that, when a fin tube type evaporator (widely used in mild regions) is used in cold regions, frost forms on the heat transfer surface of the evaporator, becoming a resistance to heat transfer and, moreover, causing an output decrease of the heat pump. To defrost the heat transfer surface when electrically driven, the heat transfer surface is heated by reversing the cycle, and engine exhaust gas is blown against the heat transfer surface when the engine is driven.[1] In any case, however, a drop of the coefficient of performance is inevitable. Consequently, to successfully utilize a heat pump system using air as the heat source in cold regions, it is important that an evaporator that prohibits frosting on its heat transfer surface be developed.

A fluidized bed heat exchanger, which is widely used for industrial purposes as solid-gas contact equipment, has a number of excellent thermal characteristics: (1) the temperature of the bed is kept at a nearly uniform value; (2) the value of the heat transfer coefficient on the heat transfer surface is more than 10 times higher than that of the forced convection; and (3) the heat transfer surface can be easily inserted into the bed. On the contrary, the fluidized bed heat exchanger has a major drawback in that the heat transfer surface in the bed is susceptible to wear, due to fluidized particles. For example, a fluidized bed combustor uses a heat transfer tube in the bed to control the bed temperature. The heat transfer tube, however, wears excessively, so that it must be replaced once a year, or a thick pipe must be used, or a protector must be provided. Conversely, if we take advantage of the drawback, i.e., if we use the fluidized bed heat exchanger as an air-source evaporator, frost that forms on the heat transfer surface can be erased by abrasion action, making it possible to yield to a high heat transfer coefficient in a frost-free state.

These characteristics, however, are realized in an environment where the temperature is higher than the normal value. In order to run a fluidized bed heat exchanger in winter temperatures and humidities (below the freezing point) in cold regions, two problems must be investigated.

One problem is freezing of the fluidized bed; the other problem is heat transfer performance and abrasiveness below the freezing point. The fluidized bed may become inoperable due to freezing of particles, e.g., if frost forms around the heat transfer tube, fluidized particles cohere around the frost. Figure 1 illustrates the results of an experiment with defrosting action of the fluidized bed[2]. The symbols o and △ in Figure 1 represent a heat transfer coefficient at the moment when the frosted heat transfer tube was inserted into the fluidized bed, and at the moment of forced convection. Judging by the fact that ten minutes later, the heat transfer coefficient shows larger values in the fluidized bed, it may be said that, since the fluidized bed has a defrosting action, no particles cohere around the heat transfer tube. The problem is that the bed as a whole may be frozen, when it is completely stopped. In this regard, its possibility must be confirmed during actual meteorological conditions. With regard to the second problem, experimental verification is required as to whether (with respect to the difference from normal temperature) only changes in physical properties of fluidizing air need to be considered.

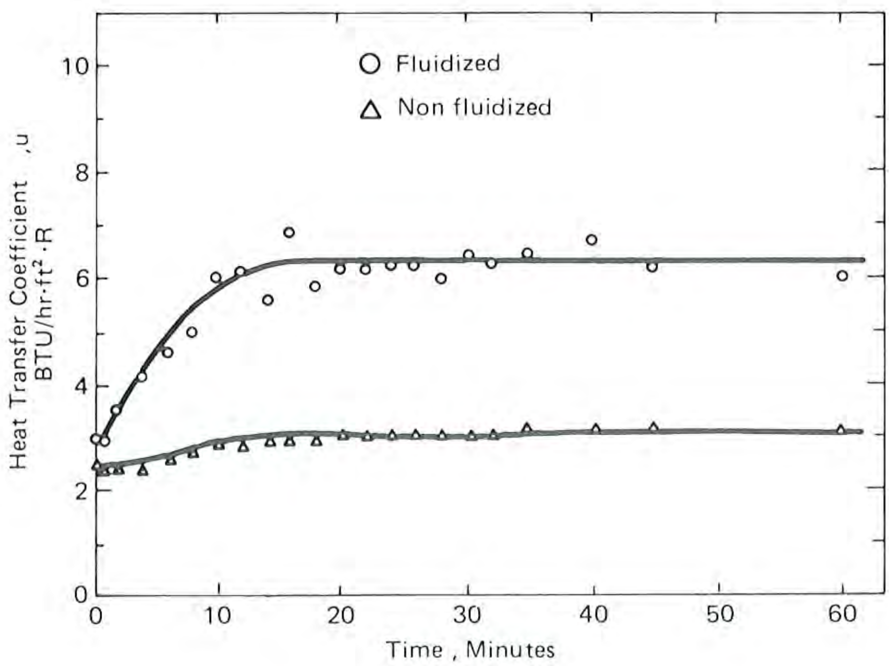

Figure 1. An example of defrosting action of the fluidized bed

Considerations were given to the possibility of a fluidized bed heat exchanger in cold regions. Though there are a number of technical problems to be solved, the fluidized bed heat exchanger is very promising.

2. Working fluid of a heat pump for cold regions

2.1 Working fluid for a low-temperature heat source

As far as using an air-source heat pump for heating purposes in a cold region is concerned, there are two problems: (1) the lower the atmospheric temperature, the larger the heating load, and the larger the compression power requirements, resulting in an efficiency decrease; and (2) if the heat pump capacity is designed for the coldest period, the necessary equipment becomes extensive, requiring a high running cost. As a step for overcoming these difficulties, much is expected from the utilization of nonazeotropic mixtures.[3,4,5]

When nonazeotropic mixtures in a compression type heat pump are used, the Lorentz cycle is applied, and the fluids inside and outside the heat exchanger run in a counterflow. During the heat exchanging processes in the evaporator and condenser, temperatures of inside and outside fluids change in the respective running directions. By choosing nonazeotropic mixtures that can approximate the outside fluid temperature, irreversible losses in the heat exchanging process can be reduced.

2.2 Possibilities of nonazeotropic mixtures[6,7]

When nonazeotropic mixtures are used, the following effects are expected to result in heat pump performance improvements:

(1) Improvement of the coefficient of performance (COP)

(2) Increased heating capacity

(3) Controllability of heating capacity

(4) Decrease in the compressor's pressure ratio and discharge temperature

Since the COP is affected not only by the nonazeotropic mixtures' thermodynamic properties, but also by the size of heat exchanger, compressor efficiency, etc., comparison criteria must be defined when compared to the cycle using a single refrigerant working fluid. The heating capacity increases when using a working fluid whose specific volume at the compressor inlet is small, and which has a large heat of vaporization. An additional effect can be gained when mixed with a working fluid that can decrease the condensation pressure or the compressor's discharge temperature, and can also move the condensation temperatures in a higher direction. In order to take advantage of nonazeotropic mixtures, the heat pump's component elements must be adequately arranged. If the condensate fluid is supercooled by the working fluid in the evaporator outlet, the evaporator's working fluid temperature range can be extended. Additionally, when the working fluid in the evaporator outlet is superheated, the evaporator's pressure level rises.

As a result, the compression ratio reduces, which enables reduction of compressor power and prevents the compressor's discharge temperature from rising. Inquiries were made into the possibility of increasing the heat pump's heating capacity by controlling the composition of nonazeotropic mixtures[8,9]. Cooper made it clear that, by using -17.8 to +8.3°C of air heat source, and by controlling the composition of R13B1/R152a within the heat pump cycle, the heating capacity can be increased when the heat source temperature is low, compared to the utilization of R22. This method is thought to be effective when using low temperature ambient air, which shows large seasonal fluctuations, as the heat source.

When using nonazeotropic mixtures, control of the expansion valve, heat transfer properties and lubricating oil must be examined.

2.3 How to combine nonazeotropic mixtures

When collecting heat effectively from a low temperature heat source, temperature difference during phase change from liquid to vapor of the working fluid must be larger than a certain value. The temperature difference is determined by working conditions of the heat pump. With regard to this point, the Sapporo district was selected as a representative cold region in order to examine a desirable combination of binary nonazeotropic mixtures to be applied when air is used as the heat source.

The heat pump's working conditions were assumed as those shown in Table 1. Energy-saving, low temperature radiant-type floor heating[10] were assumed as application mode, with a water temperature of the condenser outlet at 35°C.

The temperature range changes along

Range of ambient temperature	-15 to +14°C
Temperature difference of evaporator outlet/inlet air	10°C
Temperature difference of working fluid in evaporator outlet/inlet	5°C
Temperature of water in evaporator outlet	35°C
Temperature difference of working fluid in condenser outlet/inlet	5°C

Table 1. Working conditions of the heat pump

1	R14/R12	10	R13B1/R12B1	19	R22/R113
2	R23/R12	11	R13B1/R21	20	R22/R114
3	R23/R22	12	R13B1/R22	21	R22/R152a
4	R13/R11	13	R13B1/R114	22	R12/R11
5	R13/R12	14	R13B1/R115	23	R12/R21
6	R13/R22	15	R13B1/R152a	24	R12/R113
7	R13/R114	16	R22/R11	25	R12/R114
8	R13/R115	17	R22/R12	26	R143/R12
9	R13B1/R12	18	R22/R21	27	R143/R142

Table 2. Examples of combinations of nonazeotropic mixtures

phase changes of Freon system mixtures were determined by using Lorentz's equation[11,12]

$$\Delta t_{max} = 0.04\ (\Delta t_s)^{1.616}$$

where t_{max} denotes maximum temperature differences when the mole concentration is 50%, presupposing an ideal solution of binary mixtures, and Δt_s denotes temperature difference of boiling points in standard state. When the operational pressure of the heat pump rises, Δt_{max} decreases.

The average temperature difference, t_{av}, in the two-phase area of binary non-azeotropic mixtures were determined from the boiling point of each single refrigerant component. Relations between Δt_{max} and Δt_{av} of mixtures shown in Table 2, are illustrated in Figure 2. In Figure 2, the temperature working fluid in the evaporator inlet, t, was also shown in the case where, with the ambient temperature as a parameter, the temperature difference of working fluid, t, between evaporator inlet and outlet was set. Numbers in Figure 2 correspond to those in Table 2. It is possible to use mixtures at ambient temperature conditions similar to those shown in Figure 2. In retrieving mixtures, regarding working conditions, a simple criteria for the selection should be provided in order to facilitate the experimental investigation of optimum mixtures.

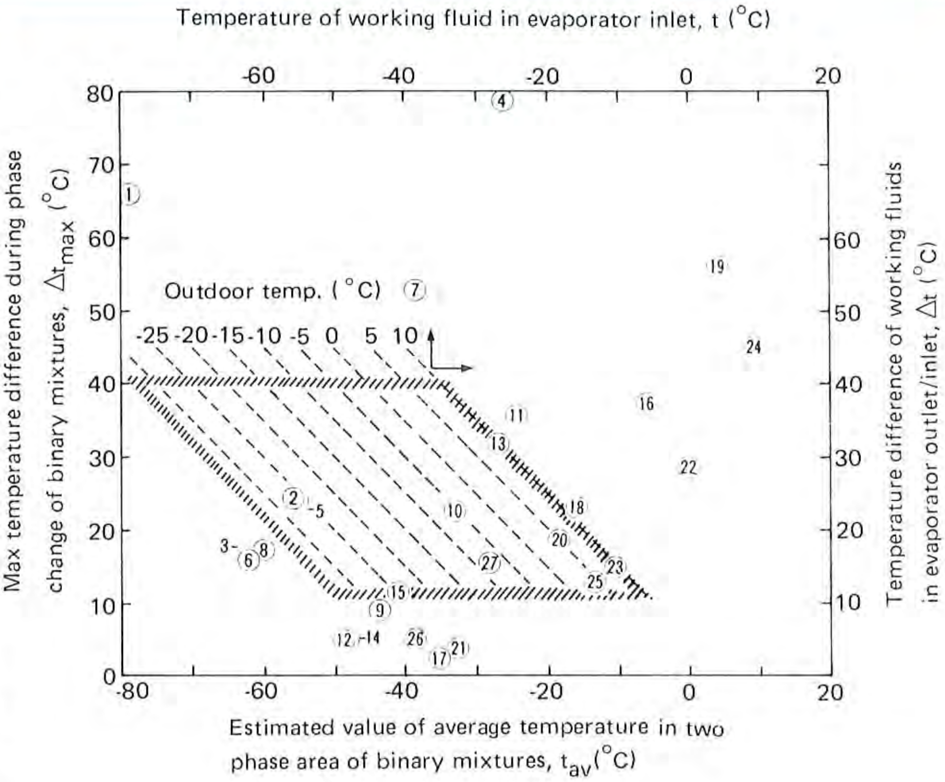

Figure 2. Relationships between Δt_{max} and t_{av}, and t and Δt

3. System simulation of the heat pump for a cold region

Problems involved in the use of air-source heat pumps in cold regions include: (1) a small COP coupled with a low temperature of air as the heat source in the coldest period in which the heating load is large; and (2) the need for a larger compression ratio of the compressor, due to an increased temperature difference from the condenser, as compared with a low evaporator temperature. The weather of Japan's cold regions, among others, is characterized by considerable differences between highest and lowest temperatures in winter and summer, for one day and one week, compared to those in European countries. The COP reduces if air-source heat pumps are installed in accordance with the maximum load (e.g., in Sapporo, the SPF is about 2.0, even when floor heating is assumed). Apart from a short period of the coldest season, they have an excessively large heating capacity, so that the rate of operation decreases.

The COP of the heat pump is determined by heat characteristics of residence, environmental conditions (atmospheric temperature, humidity, etc.), configuration of the heat pump's evaporator, condenser, compressor, expansion valve, etc., and working fluid used, as well as by operational mode, etc. However, since each of these elements allow a wide range of selection, and since they have a regional variation, it is recommended to utilize computer simulation in order to carry out its analysis, design, and development effectively. Such a computer simulation system has been developed and is in use to investigate the feasibility of diffusion of heating-only heat pumps for cold regions.

In Figure 3, an outline of the simulation model developed at this laboratory is shown. This program allows selection of an adequate working fluid of the Freon system to calculate the COP during specific conditions, and also to calculate the SPF (seasonal performance factor) through one heating period by performing dynamic simulation using three-hourly meteorological observational values in major Hokkaido cities. The results can also be charted and tabulated.

To improve the COP, it is conceivable to make a heat pump available as a multistage system, or as a composite system coupled with any other heating device. Table 3 shows an example of the simulation results: monthly heating load and efficiency of a two-stage air-source compressor-driven heat pump in Sapporo City. This presupposes 100 m^2 of floor heating with the coefficient of the overall heat transmission q = 11.72 (KJ/m^{2}h). The SPF was 2.713. Compared to a single-stage where the SPF was about 2.0, significant improvements can be achieved. In order to exceed the SPF of 3.0, side by side with multi-stage systemization and composite systemization, the following steps will need to be taken:

(1) Lower the supply temperature as much as possible, by using low-

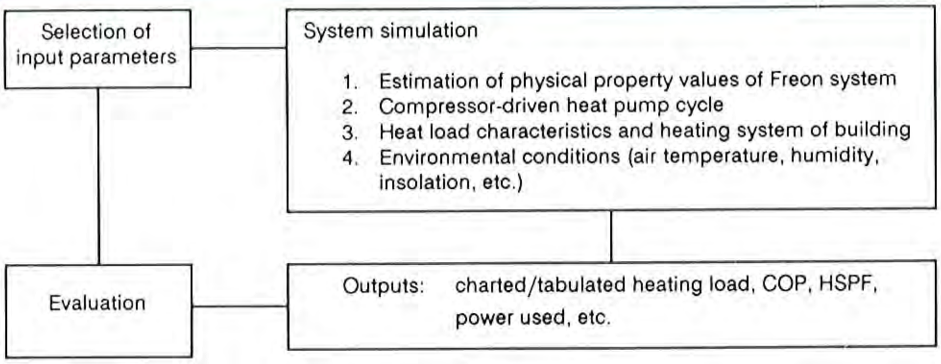

Figure 3. Outline of heat pump simulation model

Month	Heating Load	Heat Pump		Sub Heat Source	Efficiency
		Output	Input		
Oct	4663.685	4663.685	1563.502	0	2.983
Nov	9175.267	9175.267	3081.628	0	2.977
Dec	13972.684	13972.684	4882.758	0	2.862
Jan	17164.839	17164.839	7288.768	0	2.355
Feb	13294.311	13294.311	5138.828	0	2.587
Mar	12133.678	12133.678	4377.094	0	2.772
Apr	6693.355	6693.355	2247.011	0	2.979
May	4687.652	4687.652	1571.388	0	2.983
Jun	1323.826	1323.826	443.265	0	2.987
Jul	48.742	48.742	16.296	0	2.991
Aug	5.853	5.853	1.956	0	2.992
Sep	917.935	917.935	307.366	0	2.986
Season	81785.471	81785.471	30150.977	0	2.713

Heating Quantity and Efficiency (kJ/month)

1980.10 -- 1981.5 $q = 11.72$ (kJ/m^{2}h)

$Tout_1 = 253.15$ (K)
$Tout_2 = 268.15$ (K)

$Teva_1 = 246.54$ (K)
$Teva_2 = 264.34$ (K)
$Tcon = 308.15$ (K)

Table 3. Simulation results

temperature radiation heating.

(2) Separate hot water supply load from heating load.

(3) Level heat load by employing short or long-term heat accumulation.

(4) Develop a defrosting or frost-free heat exchanger. Additionally, reduce a cycling loss.

(5) Adequately utilize overheating and overcooling of the refrigerant.

(6) Effectively utilize nonazeotropic mixtures.

While improving the performance of each element of the heat pump for cold regions, efforts should be made to take the above-mentioned steps, in an effort to develop a cold regional heat pump system having performance capabilities that can be commercialized. Computer simulation is intended for this purpose.

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Å. Bratt*

3rd Scandinavian Heat Pump Conference

Within the framework of cooperation between the Scandinavian countries, the 3rd Scandinavian Heat Pump Conference was held in Reykjavik, Iceland, on 22-25 June 1987. Over 130 delegates from Denmark, Finland, Iceland, Norway, and Sweden participated.

At the opening ceremony, Mr. Pall Flygering, chief of the industrial department of Iceland, and Mr. Kenneth Larsson, representing the Scandinavian Ministry Council, called attention to the importance of cooperation between the Scandinavian countries, among others, in all questions concerning energy conservation. During the ceremony, a choir of Icelandic girls sang Icelandic and Scandinavian songs. On the afternoon of the first day, the participants were invited to a "Midsummer Party" at a traditional *tingsplats* in the beautiful, barren landscape outside Reykjavik. In ancient times, this was a place where councils met to settle civil and criminal disputes. The second day, members visited the geothermal district heating plant outside Reykjavik.

The meeting was divided into four sessions. The first session had eight contributions concerning new applications for heat pumps. Heat pumps for fish farming have become a big market in Norway; this is presumed to be an important market in other countries also. Three lectures dealt with the problem of energy conservation by condensation of waste gases. Subeffects, such as elimination of environmental fouling, can be of great importance to the future heat pump market. The second session had six contributions concerning projecting and planning. *The Choice of Contractor - the Buyer's Headache* by K.E. Madsen drew attention to questions that are of great importance for the heat pump market, questions which have not been paid enough attention today. There was also a lecture touching supercritical processes, which might be of importance for the development of heat pumps in the future. The third session had four contributions concerning environment, safety, rules and regulations. This session was followed by a panel discussion. There is a great deal of work going on today concerning both safety at the plant itself and for the people working at the plants, and concerning global environmental safety. These are subjects of great importance, but, as Prof. Lorentzen said, it is also very important that rules and regulations set by the authorities be reasonable. He also said that the technical specialists are responsible for what is going to happen. They have to make up their minds, among others, on the question of choice of refrigerant, and give the authorities all impartial information needed for their work on the solution of this problem. This is a typical subject for cooperation in Scandinavia, and the authorities concerned are waiting for this information. The fourth session had ten contributions on experiences from existing plants. A mistake can be looked at in two ways: in the pessimistic way, a mistake is a failure; in the optimistic way, a mistake is a possibility to do something better next time. The fourth session was a positive declaration from encouraging experiences as well as mistakes. This is, of course, very important for the development of the heat pump market in the future.

After the conference, there was a visit to Akureyri on the northern coast of the island, where a 1.3 MW heat pump is installed, using the return water in the geothermal district heating system as a heat source.

In total, the spirit of the conference was optimistic and realistic. There is a potential for heat pumps in new markets, and the subeffects of using heat pumps are going to be of increasing importance. Specialists today are conscious of the fact that the future for heat pumps is not only a question of technique, but to a high degree also a question of marketing the product in a broader sense. Rules and regulations, and choice of refrigerant are subject of major importance, and the specialists are responsible to see that authorities are given impartial information for making decisions with reasonable consequences. Technical development is important, but it is as important to make use of today's technology and to learn from past mistakes.

Finally, in an inspired contribution to the discussions, Prof. P.-E. Frivik from NTH (Norway's Technical High School) said that heat pumps are going to have a dominant place in the heating systems of the future, and that the technical specialists are responsible for imparting information, both technical and based on experience, to all concerned. He also delivered thanks from the participants to Mrs. Maria Gunnarsdottir and her staff, who were responsible for the conference, and invited everyone to the next conference to be held in Norway in 1989.

The arrangers, NBS-Energi Ekspertgruppen for Varmepumper, are to be congratulated for a successful conference, which brought to light major subjects of importance for the application of heat pumps.

Åke Bratt, IEA Heat Pump Center, Karlsruhe, Federal Republic of Germany

Å. Bratt*

The 3rd International Symposium on the Large Scale Applications of Heat Pumps

The following report is from an IEA Heat Pump Center consulting engineer who attended the 3rd International Symposium on the Large Scale Applications of Heat Pumps. This symposium was held in Oxford, England, 25-27 March 1987. It was organized and sponsored by BHRA, The Fluid Engineering Centre, Bedford, England.

The Symposium took place at St. Catherine's College, Oxford, with about 120 participants from 19 different countries. The delegates were invited to live at the College; a number of single student's rooms were available and meals were served in the main Dining Hall. Those who took advantage of this invitation had an opportunity to feel the atmosphere at an English college, which was quite an experience.

All the delegates and their guests were invited to attend a Civic Reception in the Town Hall in Oxford. At the reception, the Lord Mayor, in an informal speech, spoke to the multi-national delegates about their special characters and specialties. The Parliamentary Under-Secretary of State for Energy, Mr. David Hunt, MBE, MP, spoke at the Symposium dinner. The serious part of his humorous speech was devoted to the importance of energy policy with far-reaching aims, and he wished the delegates success in their important work during the Symposium.

The opening speech was given by Professor F.A. Holland, Salford University. He emphasized, among other things, the importance of energy conservation and the fact that heat pumps have great potential for energy saving in the future.

The Symposium was divided into nine sessions and a poster display. There were two sessions on district and building heating, two on process application, two on absorption systems, one on compressors and controls, and two on the future of large heat pumps.

Today's low energy prices discourage investments in heat pump plants. But throughout the industrialized world, people are aware that energy prices will change and that R & D must be done today to meet the future energy situation. Therefore, interest in heat pumps is still as high as it was some years ago, when energy prices favored the use of heat pumps.

It is not surprising, then, to find that two-thirds of the contributions to the Symposium dealt with R & D aimed at reducing the cost and improving the efficiency of heat pumps, and developing industrial heat pumps which can produce heat at higher temperatures (up to 300-500°C) than they can today. But it is also interesting to note that one-third of the Symposium contributions were concerned with experiences from existing plants in operation. The conclusions from these experiences are growing more and more important. The following is taken from *Review of Five Industrial Heat Pump Demonstration Projects* by A.W. Deakin and R. Gluckman:

> The value of Process Integration in heat pump system design emphasises that the heat pump should not be considered in isolation. It is part of a complete industrial process and must be assessed in relation to the whole process.
>
> To ensure second generation heat pumps do not suffer these faults (the same as the first generation) it is necessary to do just two things:
>
> (a) allocate adequate money for proper design; and
>
> (b) as part of the design exercise, use the experience of the last five years.
>
> The design exercise should be split into two parts. Firstly, the overall system should be considered and then detailed design of components should take place.

In the long run, this is probably as important as taking care of the results from ongoing R & D.

Proceedings of the Symposium were delivered to the participants. The complete volume may be purchased from BHRA, The Fluid Engineering Centre, Cranfield, Bedford MK43 OAJ, England, Telephone 0234 750422, Telex 825059 G, Telefax 750074.

Åke Bratt, IEA Heat Pump Center, Karlsruhe, Federal Republic of Germany

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Air-Conditioning and Refrigeration Institute*

The National Appliance Energy Conservation Act of 1987

The following is extracted from a publication of the Air-Conditioning and Refrigeration Institute (ARI) of Arlington, Virginia, USA. ARI is the trade association which represents the majority of U.S. manufacturers of heat pumps (excluding room and domestic water heating models).

U.S. President Ronald Reagan signed the National Appliance Energy Conservation Act of 1987 (NAECA) on March 17, 1987. This act covers air-cooled central air conditioners and air-source central heat pumps below 65,000 Btuh (British thermal units per hour) of single-phase power. In the U.S., a *central* air conditioner or heat pump is used to serve an entire dwelling or building from one appliance, rather than a system of individual units per room. The NAECA, U.S. Public Law 100-12, does not cover three-phase systems, packaged terminal units, water-source heat pumps or humidifiers. However, the U.S. Department of Energy (DOE) may, at its discretion, regulate other consumer products if those products meet certain energy-consuming criteria. This grant of future authority was also found in a previous law, the Energy Policy and Conservation Act (EPCA), which the NAECA amends.

Energy efficiency standards

The NAECA provides for a federal standard of 10.0 seasonal energy efficiency ratio (SEER) for split-system air conditioners and heat pumps, effective January 1, 1992; and 9.7 SEER for single-package air conditioners and heat pumps, effective January 1, 1993.

Heat pumps are also subject to federal standards of 6.8 heating seasonal performance factor (HSPF) for split-systems, effective January 1, 1992; and 6.6 HSPF for single-packages, effective January 1, 1993. For purposes of the NAECA, the manufacture and importation, not the sale, of noncomplying products is prohibited after the effective dates of the standards.

The SEER standards for split-system and single-package air conditioners and heat pumps remain in effect until at least January 1, 1999. The HSPF standards for split-system and single-package heat pumps remain in effect until at least January 1, 2002.

Future rulemakings

Under the NAECA, DOE must review the legislated standards for both split-system and single-package air conditioners and heat pumps by January 1, 1994, to determine whether they should be revised. Any revisions would not become effective until January 1, 1999, for air conditioners and the cooling side of heat pumps. They would not become effective until January 1, 2002, for the heating side of heat pumps.

The NAECA also directs DOE to review the standards a second time by January 1, 2001, to determine whether they should be revised. Any revisions proclaimed in this rulemaking would become effective on January 1, 2006.

Test procedures

Existing test procedures, such as those already developed for central air conditioners and heat pumps, will continue to be used unless and until DOE determines otherwise. DOE is given authority to revise existing test procedures and to establish new ones for additional consumer products brought under the jurisdiction of the Act pursuant to the grant of future authority for other consumer products.

Building codes

The NAECA provides that the minimum efficiency standards for central air conditioners and heat pumps found in performance-based building codes for new construction are exempted from the new rulings if established as of January 8, 1987. During the time between January 8 and the effective date of the federal standards, the Act authorizes states to adopt or revise the minimum standards in their codes to levels not exceeding the then prevailing minimum ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers) standards.

Manufacturer requirements

As in the old EPCA, DOE is authorized by the NAECA to require manufacturers to submit data regarding the energy efficiency of covered products and the economic impact of proposed standards. The NAECA supplements old law, however, stating that DOE must exercise its authority in such a manner as to minimize unnecessary burdens on manufacturers.

It also directs DOE to give due consideration to the certification programs of the manufacturer trade associations when evaluating methods of ensuring compliance with the law.

**Air-Conditioning and Refrigeration Institute, Arlington, Virginia, USA*

LITERATURE REVIEW: BLOCK CENTRALS

Stahlknecht, R., "Energy conserving HGW gas distribution system", Neue Deliw-Z. (Jun 1986) v. 37(6) p. 210-220. Conference: 3. International rational gas utilization conference, Prague, Czechoslovakia, 14 Oct 1985 (in German). Apart from natural gas supply as such the Hamburger Gaswerke GmbH (HGW) has been distributing district heat for more than 25 years. The paper reports on the development and present state of the latter. Moreover, it points out energy conservation measures applied to the gas distribution grid such as the use of plastic tubes instead of steel tubes and the extensive modernization of district heat distribution grids. Practical examples of energy conservation demonstrate the improvement of heating station and central heating system efficiencies and give access to innovative technologies on the heating sector. The final chapter is dedicated to future HGW heating station and central heating system projects and discusses future projects with respect to recovering heat from domestic waste water.

Chant, V.G.; Arthurs, D.M.; Le Veuvre, T.; Proceedings of ASES 86 annual meeting. New York, NY: American Solar Energy Society, 1986, pp. 513-516. Conference: 6. International conference on high power particle beams, Kobe, Japan, 9 Jun 1986 (in English). With the IEA/SHACP/Task VII experience in CSHPSS as background, this paper describes how one of the more promising configurations would apply in Canada, using local weather, performance and economic parameters at three locations: Toronto, Winnipeg and Fredericton. The system chosen for application under Canadian conditions has aquifer seasonal storage, low temperature distribution and a heat pump. Although cost effectiveness of the overall CSHPSS concept appears attainable when compared to an all-electric alternative, central heating plants based on fossil fuel or alternative energy sources could be more cost effective in Canada.

Lawaetz, H.; Lange, M. "Solar collectors in a collective renewable energy system", North Sun '86, Solar energy at high latitudes. 1986. pp 65-70 (in 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. Furthermore, 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 heating system are periodically very low, and consequently quite special house installations have to be designed. The solar collector area is 300 m^2 and placed on stands at the ground in front of the district heating station, facing south. The annual output from the solar system has been calculated to be 290 kWh per year in the Danish test reference year (TRY) and with an inlet temperature at 60°C.

"Installing heat pumps in rented houses is partly subsidized in The Netherlands", Energiebeh. Afvalbeherr (Aug 1986) v. 7(8). p. 19-21. (in Dutch) Since 29 July 1986, installation of heat pumps and cogeneration units in rented houses is partly subsidized (33%) by the government. In this article, calculations are set up for a collective central heating system with gas-motor-driven heat pumps for blocks of 500 house-equivalents. The heating system consists of two heat pump units, each having a power capacity of 750 KWth. The system is compared with having two new boilers equipped with flue gas heat recovery apparatus. Extra investments for the heat pump option have a payback period of 5.7 years; with subsidy, 3.9 years. Estimated potential in the nineties for these extra investments: Dfl. 175 x 106.

Mirandola, A., "Methane expansion and geothermal water exploitation combined plant for energy recovery and distict heating", Proceedings of condensed papers of the 7th Miami international conference on alternative energy sources, Miami Beach, FL, USA, 9 Dec 1985 (in English). A combined plant has been designed to recover energy from two alternative energy sources: the expansion of methane from a transport pipeline pressure to the distribution pressure in a two-stage radial turbine, to generate electric power of 1189 kW and a peak power of 1587 kW; the exploitation of a geothermal water bed, 1500 meters deep, supplying 27 Kg/s of water having a temperature of 70°C. The electrical energy produced will feed some users of the Municipal Managing Company and a set of heat pumps, which will improve the temperature of the geothermal water and supply a district heating network; a fraction of the geothermal energy will be required to heat the methane before each expansion stage, to prevent condensate presence in the pipes. The overall size of the buildings to be heated is 537,000 m^3. In this way an integrated system has been conceived which enables a saving of energy.

NEWS BRIEFS

Proposition in Sweden for Penalty Tax on Dangerous Refrigerants

The journal *Scandinavian Refrigeration*, No. 3/87, reported that the Swedish government has announced its intention to propose to the 1988 parliament a penalty tax on chlorofluorocarbon (CFC) refrigerants. CFC refrigerants (such as R12), which are thought to be harmful to the atmospheric ozone layer, would be assessed an extra tax, amounting to 50 SEK/kg. It is hoped that this measure would decrease the use of these components by about 25%. The revenue generated would be used to support research on finding substitutes for the CFC medias. If the motion is approved, the tax would be paid from July 1, 1990.

Call for Papers

Variable Speed Heat Pumps

ASHRAE Technical Committee 7.6, *Unitary Air Conditioners and Heat Pumps*, is seeking papers on residential heat pumps and air conditioners employing variable speed drives for a symposium at the 1988 ASHRAE Annual Meeting to be held in June 1988 in Ottawa. Field and laboratory evaluations, as well as analyses, are of interest. Abstracts should be sent by November 15, 1987, to James M. Calm, Post Office Box 671, Berkeley, California 94701, USA.

Sizing of Unitary Air Conditioners and Heat Pumps

ASHRAE Technical Committee 7.6, *Unitary Air Conditioners and Heat Pumps*, is seeking papers on sizing methods and impacts for residential air conditioners and heat pumps for a symposium at the 1989 ASHRAE Winter Meeting to be held in Chicago. Of particular interest are consideration of latent cooling, multiple and adjustable capacity equipment, and (for heat pumps) effects on supplemental heat use. Abstracts should be sent to David J. Young, Ontario Hydro, 800 Kipling Avenue, Toronto, Ontario M8Z 5S4, Canada.

ASHRAE Standards

New Standard

ASHRAE has announced the publication of ASHRAE Standard 114-1986, *Energy Management Control Systems Instrumentation*.

According to Donald H. Spethmann, chairman of the committee that wrote this new standard, ASHRAE Standard 114-1986 provides guidelines for specifying measurement requirements and for recommending methods of verifying accuracy of instrumentation of energy management control system (EMCS). This standard of recommended practice was written to give guidance to satisfy the different needs and viewpoints of specifiers, manufacturers, installers, and users who select and verify end-to-end accuracy in EMCS. These EMCS relate to the control, energy management, and management information functions of the heating, ventilating, and air-conditioning processes in a building.

The price of ASHRAE Standard 114-1986 is U.Š. $30.00 (ASHRAE member price is U.S. $20.00.

Revised Standards

ASHRAE has announced the publication of ASHRAE Standard 58-1986, *Method of Testing for Rating Room Air Conditioner and Packaged Terminal Air Conditioner Heating Capacity*. This standard is a revision of one originally published in 1974 and includes advances in instrumentation and packaged terminal air conditioners.

Standard 58-1986 prescribes test methods for determining the heating capacities and air flow quantities for room air conditioners and packaged terminal air conditioners designed for room heating.

For purposes of this standard, room air conditioners and packaged terminal air conditioners are designed primarily to provide free delivery of conditioned air to an enclosed space. Air conditioners provide a primary source of refrigeration and dehumidification, means for air circulation, air cleaning, heating, and may include means for ventilation, or even humidification.

The price of ASHRAE Standard 58-1986 is U.S. $16.00 (ASHRAE member prices if U.S. $11.00).

New Committee to Revise Existing Standard

ASHRAE has announced the formation of Standards Project Committee 20-70R, *Methods of Testing for Rating Remote Mechanical-Draft Air-Cooled Refrigerant Condensers*. The purpose of this committee is to revise an existing standard originally published in 1970. Standard 20-70 prescribes methods for rating mechanical-draft and remote air-cooled refrigerant condenser, which are self contained, waterless, refrigeration system components. This standard specifies procedures, apparatus, and instrumentation by which the condenser capacity can be determined with accuracy and satisfactory for commercial rating.

Qualified individuals interested in committee participation, which includes attending meetings at their own expense and paying meticulous attention to committee work assignments, are invited to submit a biographical data form to ASHRAE, Manager of Standards, 1791 Tullie Circle, NE, Atlanta, Georgia 30329, USA.

Draft Standards

ASHRAE has announced a sixty-day public review (from 15 August to 13 October 1987) for proposed ASHRAE Draft Standard 28-78R, *Method of Testing Flow Capacity of Refrigerant Capillary Tubes*. This standard establishes a uniform procedure for testing capillary tubes by determining the dry nitrogen flow capacity under specific test conditions and is designed for use by test laboratories. ASHRAE Standard 28-78R provides a basis for agreement between capillary tube manufacturers and users, and it establishes a uniform procedure to check all other methods of testing.

Although the results obtained by use of this procedure are not intended to define the flow characteristics of the tube in an actual refrigerating cycle, the nitrogen flow capacity does correspond to the refrigerant flow capacity. Even though the mathematical relationship between the two conditions cannot be precisely expressed, a high nitrogen flow capacity resulting from this procedure characterizes a high refrigerant flow capacity in an actual refrigerant cycle. The same is true for a low nitrogen flow capacity.

A copy of Draft Standard 28-78R can be obtained for U.S. $8.00.

ASHRAE also announced a sixty-day public review (from 15 August to 13 October 1987) for reaffirmation with minor editorial revisions of proposed ASHRAE Draft Standard 96-1980, *Methods of Testing to Determine the Thermal Performance of Unglazed Flat-Plate Liquid-Type Solar Collectors*. This standard establishes test methods for determining the thermal performance of unglazed, flat-plate, liquid-type solar energy collector modules that heat a liquid for low temperature applications. It also provides test methods and calculation procedures for determining steady-state and quasi-steady-state thermal performance and angular response characteristics of the solar collectors.

Standard 96-1980 applies to solar collectors to be used in low temperature applications, and to those in which a liquid enters the collector through a single inlet and leaves the collector through a single outlet.

Collectors containing more than one inlet and more than one outlet may be tested according to this standard provided that the external piping can be connected to effectively provide a single inlet and a single outlet.

A copy of the reaffirmation Standard 96-1980 can be obtained for U.S. $15.00.

All ASHRAE standards are developed in accordance with consensus procedures recognized by the American National Standards Institute (ANSI). ASHRAE makes a concerted effort to resolve all comments resulting from the public review of its draft standards, and replies to all comments received.

To order Standards 114-1986 and 58-1986, send payment in U.S. funds to ASHRAE Publication Sales, 1791 Tullie Circle, NE, Atlanta, Georgia 30329, USA. Credit card orders (VISA, MasterCard, and American Express) are accepted by telephone (404-636-8400) or by mail (include full credit card number and expiration date). To order Draft Standards 28-78R and 96-1980, send payment in U.S. funds to ASHRAE, Manager of Standards, 1791 Tullie Circle, NE, Atlanta, Georgia 30329, USA.

ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers), founded in 1894, is an international organization of 50,000 persons. Its sole objective is to advance through research, standards writing, and continuing education the arts and sciences of heating, ventilation, air conditioning, and refrigeration for the public's benefit.

Selected Book and Report Reviews

Review of the most recent publications on heat pumps

1987 HVAC Systems and Applications. ASHRAE, Atlanta, Georgia, 1987 (English).

This is the latest volume in the handbook series published by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), focusing on HVAC systems and applications. It provides engineers, architects, contractors, and service personnel with sixty chapters of comprehensive information on all aspects of heating, ventilating, and air-conditioning systems and applications. This volume of the ASHRAE Handbook series combines revised and updated chapters from the *1982 Applications* volume and the *1984 Systems* volume. A wide variety of subjects such as residential, industrial, transportation, and public facilities heating and air-conditioning applications are covered in this single volume. Some of the more general topics include automatic control, sound and vibration control, testing, adjusting and balancing, owning and operating costings, and fire and smoke control. New chapters not included in the previous volumes of the handbook series are thermal storage, nuclear plants, and underground mine air-conditioning and ventilation. This edition contains inch-pound (I-P) units and the International System of units (SI).

The cost of this volume is U.S. $100.00. It may be ordered by sending payment (in U.S. funds) to ASHRAE Publication Sales, 1791 Tullie Circle NE, Atlanta, Georgia 30329, USA.

Other current volumes in the Handbook series are *1986 Refrigeration Systems and Applications*, *1985 Fundamentals*, and *1983 Equipment*.

Stirling Engine Technologies in Japan. Heat Pump Technology Center of Japan, 1987 (English).

This is the proceedings of the International Symposium on the Stirling Engine and its Application to Heat Pump Systems, etc., held in Tokyo on 1-2 December 1986. The symposium was sponsored by the Agency of Industrial Science and Technology (MITI), the New Energy Development Organization (NEDO), and the Heat Pump Technology Center of Japan. Participants included governments, national laboratories, universities and industries from the USA, Sweden, and Japan, all of which are participating countries of IEA Annex XI. The symposium was held in conjunction with the Annex XI Workshop, which was held also in Tokyo after the symposium.

This book contains twelve proceedings from Japanese presentations. In Japan, the research and development of Stirling engines has been performed chiefly as one of many national projects, and has been approached from the aspect of applying Stirling engine technology to heat pump systems. Thus, most of these presentations discuss heat pump systems, either in specific detail or in general terms. Most see the future prospects of the Stirling engine as bright and hopeful in the area of heat pumps.

ASHRAE Publications Catalog 1987-88. ASHRAE, Atlanta, Georgia, 1987 (English).

This catalog provides a complete listing of information available from ASHRAE including videotapes, computer software, handbooks, technical data bulletins, compilations of all technical and symposium papers presented at ASHRAE Annual and Winter Meetings, standards, books, conference proceedings, psychrometry brochures and charts, publications of the International Institute of Refrigeration (IIR), and professional development seminar texts.

This catalog is free of charge and may be obtained from ASHRAE Publication Sales, 1791 Tullie Circle, NE, Atlanta, Georgia 30329, USA.

Investment in Saving Energy Costs. An investigation into the deliveries by Dutch trade and industry. Van Wees, F.G.; Boswinkel, H.H.; Diujves, K.A.; Gerbers, D.; Kruijswijk, B.J.; Van der Veen, J.C., 1986, Energie Studie Centrum, Netherlands (Dutch).

This book presents the results of a study to determine the expected share of Dutch domestic firms in the production and delivery of energy saving equipment installed in the Netherlands for the period 1990-2000. Thirteen important energy-saving techniques are considered. Based on savings realized in terms of energy investment, the profitability of these techniques is calculated. Based on theoretically available market potential and the additional constraints, the total market penetration level is estimated for the period 1990 to 2000. For each technique the share of imports and domestic production is established. In addition to the production of equipment, the contribution of Dutch engineering and installation firms is also taken into account. By means of this procedure it is stipulated that Dutch firms may contribute about 80 percent of the total value of energy saving equipment in the period considered.

Schedule of Conferences and Trade Fairs

August 24-29, 1987

Vienna (Austria); **International Congress for Refrigeration** (Internationaler Kongress fuer Kaeltetechnik). Sponsored by the Internationales Institut fuer Kaeltetechnik. Contact: XVII, Internationaler Kongress fuer Kaeltetechnik, Mondial Congress, Boesendorferstrasse 4, A-1010 Wien, Austria, telephone 0222-659629, telex 132597 mreis a, telegram wienkaelte.

September 7-9, 1987

Brighton (U.K.); **Distillation and Absorption 1987**. An event of the European Federation of Chemical Engineering, jointly organized by the IChemE Fluid Separation Processes Group and the EFCE Distillation Absorption and Extraction Working Party. Contact: The Conference Section, The Institution of Chemical Engineers, Attention: Mrs. Julie Tayler or Miss Ann Hughes, 165-171, Railway Terrace, Rugby, Warwickshire CV21 3HQ, U.K., telephone (0788) 78214, telex 311780.

September 8-12, 1987

Beijing (People's Republic of China); **China Refrigeration '87** (International Refrigeration, Heating, Ventilation and Air-Conditioning Exhibition and Conference). Sponsored by the International Institute of Refrigeration (I.I.R.), the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), and the Japanese Association of Refrigeration (JAR). Contact: Gabriele Abbott, Director, Public Relations, Asia Team Communications Ltd., Suite 2902, Windsor House, Causeway Bay, Hong Kong, telephone 5-768730, telex 60262 atcl, telefax 5-768917.

September 13-18, 1987

Hamburg (F.R. Germany); **ISES Solar World Congress and Exhibition 1987**. Sponsored by the International Solar Energy Society. Contact: ISES Solar Weltkongress 1987 e.V., c/o Hanseatic Congress Management GmbH, Am Weiher 23, D-2000 Hamburg, F.R. Germany.

September 23-25, 1987

New Orleans, Louisiana (USA); **Meeting Customer Needs with Heat Pumps**. Sponsored by the American Public Power Association, Edison Electric Institute, Electric Power Research Institute, and the National Rural Electric Cooperative Association. Contact: David P. Ross, Policy Research Associates, Inc., 12121 Basset Lane, Reston Virginia 22091, USA, telephone (703) 620-1008.

September 28-October 2, 1987

Lausanne (Switzerland); **3rd International Congress on Building Energy Management** (ICBEM-3 87). Sponsored by Ecole Polytechnique Federale (Lausanne). Contact: ICBEM '87 Secretariat, Prof. Andre P. Faist, EPFL-LESO Building, CH-1015, Lausanne, Switzerland.

October 1-2, 1987

Valdonne (France); **The First International Seminar on Thermal Energy Storage**. Sponsored by the International Center for Energy Storage (ICES) (Centre International Pour Le Stockage D'Energie [CISE]). Contact: ICES/CISE, Ms. Birgitta Kordel, Seminar Organisation Manager, Place Sophie Laffitte, Sophia Antipolis, F-06560 Valbonne, France, telephone 33-93-653565.

January 26, 1988

Toronto, Ontario (Canada); **Industrial Applications for Heat Pumps**. Sponsored by the Ontario Ministry of Energy and the Ontario Research Foundation. Contact: Dr. H.G. McAdie, Ontario Research Foundation, Sheridan Park Research Community, Mississauga, Ontario, L5K 1B3, Canada, telephone (416) 822-4111, telex 06-982311.

January 30-February 3, 1988

Dallas, Texas (USA); **1988 ASHRAE Winter Meeting**. Sponsored by the American Society of Heating, Refrigerating and Air-Conditioning Engineers. 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 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.

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- **Nonazeotropic Refrigerant Mixtures as Working Fluids in Compression Heat Pumps, A Bibliography**, Report No. HPC-B1, second edition, December 1985, DM 220 (U.S.$ 110). This report is a comprehensive bibliography of over 200 reports with complete citations and abstracts.

- **Literature Reviews:** literature surveys with complete citations and abstracts in the following areas:

  -- **Environmental Aspects of Heat Pump Applications**, Report HPC-LR1, May 1986, DM 25 (U.S.$ 13).
  -- **Industrial Heat Pump Applications**, Report HPC-LR2, July 1986, DM 40 (U.S.$ 20).
  -- **Sorption Heat Pump Systems**, Report HPC-LR3, October 1986, DM 40 (U.S.$ 20).

A new literature review is compiled in conjunction with the focus topic of each issue of the HPC Newsletter.

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Industrial Heat Pumps

The September 1987 issue of the Heat Pump Center Newsletter will focus on industrial heat pumps.

In addition, the following contributions are planned:

- Studies on chemical heat pump systems using ammonia in Japan
- An example of heat pump utilization at a photo film manufacturing plant
- The development of a natural gas-fired Stirling engine-driven heat pump

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 the subject of industrial heat pumps, please send it to the Heat Pump Center.

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