

IEA Heat Pump Centre

NEWSLETTER

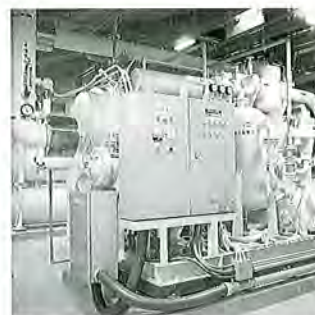


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Front Cover: Water-source Centrifugal Heat Pump/Chiller, supplying hot and cold water simultaneously (see article on pages 8-11). Photo courtesy of Sakuya-Konohana-Kan.

International Energy Agency

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

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

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Editorial

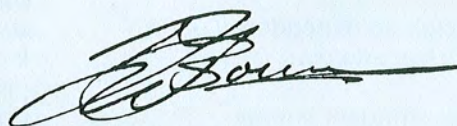
Solving operational problems, often environmental in nature, could become the primary approach taken to achieve a wider acceptance of heat pumps, both in space conditioning and industry, including in the agricultural industry. Nowadays, firms increasingly have to meet requirements regarding waste heat utilisation and emission reduction. Heat exchangers and heat pumps can often provide economically sound solutions for these problems through heat recovery.

A number of advanced technologies are available today for the recovery of heat. Various applications of heat recovery heat pumps already exist, which is something many technicians and decision-makers are not aware of. These installations operate satisfactorily and do not attract a great deal of attention. They include established heat recovery heat pumps used in the timber and fish drying industries. Such systems have been realised worldwide.

By definition, heat recovery heat pumps have good prospects because the system provides both for heating and cooling, together with favourable temperature and timing conditions. Apart from the economical benefits that can be gained, using heat recovery heat pumps can also result in higher productivity. An example of this is shown in an article dealing with salmon smolt hatcheries.

Often heat recovery systems are not realised due to particulates present in waste heat streams which may result in fouling and the internal parts of the system being clogged. As is shown in an article about a laundry, adequate and often simple methods exist to handle such a problem. In some cases however, heat exchange equipment require special designs. After all, an optimum system design is very important.

A major challenge for the heat pump industry and utilities is the commercialisation of heat recovery heat pumps. This newsletter describes some good examples of how heat can be recovered in an economically justifiable manner and it is hoped that awareness of the potential that heat pumps can offer in the building sector, district heating and industrial sectors, where many opportunities for this type of system still exist, will increase. Perhaps the suggestion offered by the authors of the US article could be helpful, i.e. industrial heat pumps, being primarily process waste heat recoverers in a non-reversing closed design, should be given a different name in order to prevent confusion with space conditioning designs. Any other suggestions are always welcomed.



Jos W.J. Bouma
General Manager HPC

A New National Policy on the Utilisation of Urban Waste Energy for Heat Pumps in Japan

*T. Maekawa

INTRODUCTION

This article introduces an approach to, and steps to be taken for, "utilisation of unused energy", both of which are strongly supported by the Japanese Government. The article also refers to some pioneering cases which have been primarily promoted by electric power companies.

During the past year, the words "unused energy" have become widely recognised among those involved in the field in Japan. It denotes heat sources for heat pumps which have conventionally been called "waste heat", and heat sinks such as river and sea water. That is, it indicates a medium of which the temperature, compared to that of air, is stable all year round and which is capable of allowing a heat pump to achieve a high coefficient of performance (COP). In countries like Japan, where cooling is often more important than heating, it becomes necessary, not merely to pursue "utilisation of waste heat", but to also utilise the types of heat sinks that may help to realise efficient cooling.

In Japan, air-source heat pumps occupy an overwhelming share of the total, irrespective of scale, with fewer water-source heat pumps used. This is said to be mainly attributable to reasons such as restrictions on pumping up underground water, and limitations on the use of river, sea and other available water sources by government regulations. Recently, utilisation of these mediums has drawn increasing attention. In the eyes of countries where the air temperature is low and water generally serves as a heat source for heat pumps, Japan's interest in water may seem to lack innovativeness. However, it is quite unique in the history of Japan's

energy policies that such a theme on the demand-side has been taken up as an energy policy, this is comparable to nuclear power generation.

BACKGROUND OF UNUSED ENERGY UTILISATION

The utilisation of unused energy has been adopted as an energy policy because of the rapid changes in energy circumstances which are taking place. These are detailed below:

- crude oil imports may be delayed or cancelled due to the Gulf situation in the Middle East;
- reflecting the growing movement against the use of atomic energy, a possible shortage of energy supply equipment may develop;
- in response to the growing concern for the greenhouse effect, energy consumption may be subjected to restrictions;
- demand for energy is increasing at a rate higher than expected.

In the summer of 1989, after a 14 year interval, the National Committee on Energy Policy meeting was held by the Japanese Government. The main points of the long-term energy demand/supply forecast discussed were as follows:

- review of growth of energy demand;
- high expectations for new energy;
- increased dependency upon nuclear energy.

These stringent energy circumstances have been the prerequisites for putting the

utilisation of unused energy into the limelight of Japan's energy policy. In order to secure its position as an element of policy, however, technical support and real achievements are necessary. At about the same time as the National Committee on Energy Policy meeting was held, construction of a district heating and cooling plant using river water had been completed in the Hakozaki area. Prior to that, some district heating and cooling plants utilising waste heat had already been in operation. Thus, much progress had already been made.

Since district heating and cooling businesses are enterprises which are run privately, their operation is naturally profit-oriented. There is not much incentive to excessively invest to improve energy efficiency alone. Herein lies the reason why the scale of existing district heating and cooling plants utilising waste heat is not very large. Thus governmental involvement is significant in developing large-scale unused energy utilisation systems.

UTILISATION OF UNUSED ENERGY AS A NATIONAL POLICY

The fiscal year 1991 budget for unused energy utilisation has been set and will be implemented as follows:

The Ministry of International Trade and Industry (MITI)

- *Construction costs subsidised*
Where a heat supplier is going to utilise unused energy, 15% of the total construction cost incurred is subsidised. The scope of the subsidy covers all equipment, including heat source water pipes, plus plant equipment and district conduit pipes.

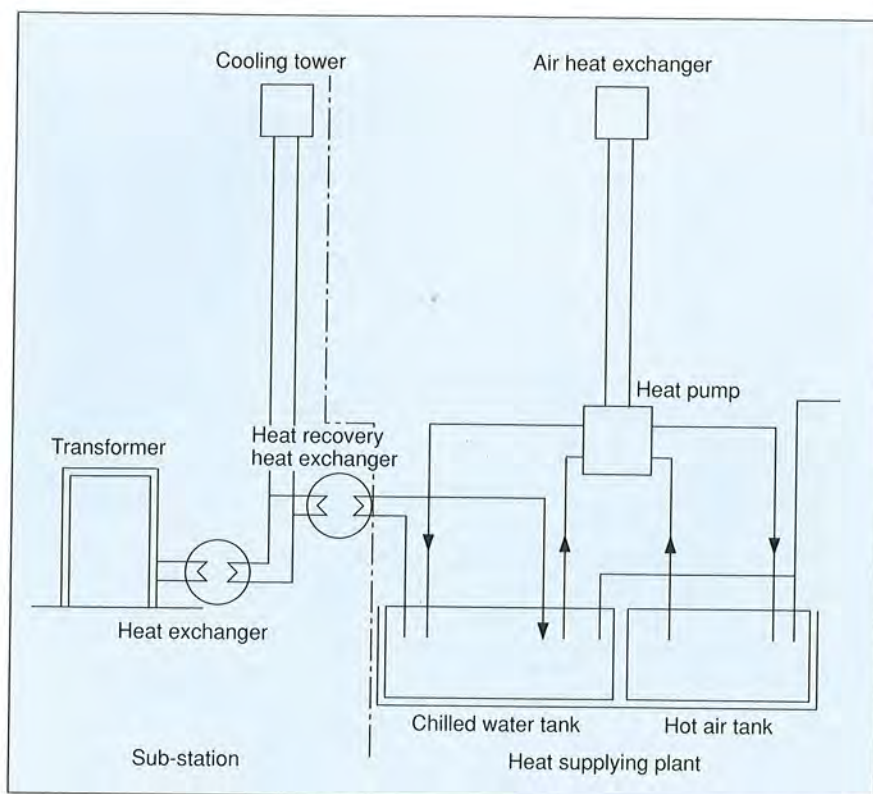


Figure 1: A System for Recovering Waste Heat from Transformer Cooling.

- *Feasibility study costs subsidised*
50% of the cost of a feasibility study conducted prior to the implementation of a district heating/cooling installation utilising unused energy will be covered.
- *Technological development subsidised*
Concerning new technological development projects aimed at improving competitiveness of district heating/cooling utilising unused energy, 50% of the R&D costs will be paid for from the national budget.
- *Diffusion and public awareness subsidised*
A subsidy is granted for both diffusion and public awareness programmes.

The Ministry of Construction

The Ministry of Construction controls a huge amount of unused energy resources, and has created the "Committee for Promoting the Utilisation of Unused Energy" within the Ministry. One of the main pillars is the "Plan for Utilising Unused Energy in Each District". The Committee is supposed to

work out a plan for the utilisation of unused energy for each district, based on a balance between heat demand and unused energy sources in the district. Each bureau within the Ministry will organise a committee for working out a policy for unused energy utilisation, according to each department.

The Ministry of Transport

The Ministry of Transport has set up a committee to examine any possible district heating supply systems which can utilise unused energy derived from sea water found in ports and harbours.

The Ministry of Health and Welfare

For the utilisation of waste heat generated from waste incineration plants, the Ministry will furnish the plant builder (local government) with a subsidy.

ACTUAL EXAMPLES OF UNUSED ENERGY UTILISATION

I. HEAT SOURCES

Many kinds of unused energy which have been utilised as a heat source for heat pumps include:

Power Distribution Sub-stations

Sub-stations have so far been utilised the most. As shown in Figure 1, in the case of a large-scale sub-station, a system is employed in which cooling water, circulating through the transformer oil cooler, is used as a heat source. In this case, the upper limit of the cooling water temperature is about 35°C. The cooling water has a stable temperature and flow rate, so that it can serve as an excellent heat source for a heat pump.

In the case of a small-scale sub-station (about 60,000 kVA), the oil cooler is cooled by air ventilation. In this case, the air temperature increases by about 3 - 5°C relative to that of the air intake. Even when a heat exchanger is installed in an air passage, the quantity of heat

collected by this system is not large.

Waste Incineration Plants

Since the waste heat from waste incineration plants has a high temperature level, it can be utilised for power generation, heating, cooling and hot water supply. There are a number of examples in which waste heat from waste incineration plants is used as a heat source for district heating/cooling. Many of them, however, utilise the high temperature, in its original form, to supply heat.

When taking into account the quality of energy, however, it is not appropriate to use the waste heat (steam) from a waste incineration plant having a high temperature level, in its original form, to meet the demand for low temperature heat. After using it, firstly, for power generation, the resulting warm waste heat should be utilised. For example, a system based on this idea is operating at Hikarigaoka Park Town.

Underground Railways

An underground railway tunnel is not a very good heat source. The main source of waste heat is heat emitted from railway carriages. Since heat is recovered from the air, as in the case of a small sub-station for power distribution, it is difficult to expect a high heat recovery rate. According to field measurements made by the author et al. [1], the quantity of heat recovered in winter was 116 kW/km. However, since underground railway lines are extensive, they may be considered a desirable heat

source for heating and hot water supply, if heat is recovered from tunnels which are long enough for heat recovery requirements.

Attempts can be made to use underground railway heat so that an absorption chiller can be driven indirectly. This could be accomplished by using a system which consists of an electrically-driven heat pump which uses air from the underground railway as a heat source and supplies 44°C water. This water is then used as a heat source for a steam turbine - driven heat pump to produce 90°C water. The hot water drives the absorption chiller. Although this is a unique application, it requires a heat sink as well, and it is inferior to conventional systems (see Table 1). Thus, apart from areas where there is a high demand for heating, a cooling system using underground railway waste heat is not to be recommended for areas in which there is a relatively larger demand for cooling.

II. HEAT SINKS

The recent demand for heating/cooling is mainly characterised by a decrease in demand for heating and an increase in demand for cooling. In large buildings, all year-round cooling has already become necessary, and in many 'intelligent' buildings, demand for cooling in winter exceeds that for heating. This tendency is projected to accelerate in the future, and a system assuring high efficiency during cooling will become increasingly important. In this context, unused energy sources, which serve as heat dumps (i.e. heat sinks), are gaining in importance.

River Water

The major advantage of river water being utilised as a heat source and heat sink for district heating/cooling at Hakozaiki (see Figure 2) lies in the fact that its temperature is more stable than the outdoor temperatures. Another advantage is that no replenishment of cooling tower water is necessary. Although city water charges are only about 5% of the heat supply cost, any decrease will surely lead to overall cost reductions. Since cities suffer from chronic water shortages during the summer, the elimination of cooling tower water replenishment is an effective step for reducing water consumption.

Sewage

In Sweden there are many cases of district heating using **treated** sewage as a heat source, while in Norway there are many cases of district heating/cooling using **untreated** sewage as a heat source [2]. Since sewage temperatures are usually more stable than river temperatures, sewage is a more preferable heat source for district heating/cooling. In Japan, sewage operators have already developed their own technology for utilising treated or untreated sewage as a heat source for heat pumps to heat/cool buildings and there is also a growing tendency towards multi-purpose utilisation of sewage. Therefore, sewage utilisation will continue to increase. Although the utilisation of treated sewage is limited to areas around sewage treatment plants, it is not subjected to any legal control. As for untreated sewage, there still remain certain institutional restrictions. These have been stipulated from a sanitary point of view, but should hopefully be updated to respond to current needs.

Sea Water

In Sweden, there are many heat pumps using sea water as a heat source. In Japan, many of the thermal power plants use sea water as cooling water. This demonstrates that there are no technical

Table 1: Comparison of System Efficiency. Note that the figures in brackets indicate efficiencies relative to those of conventional systems.

	Waste heat utilisation system		Conventional system	
Cooling	Absorption chiller	COP = 1.20 (68%)	Turbo chiller	COP = 1.76 (100%)
Heating	Steam turbine heat pump	COP = 1.10 (118%)	Gas boiler	COP = 0.93 (100%)

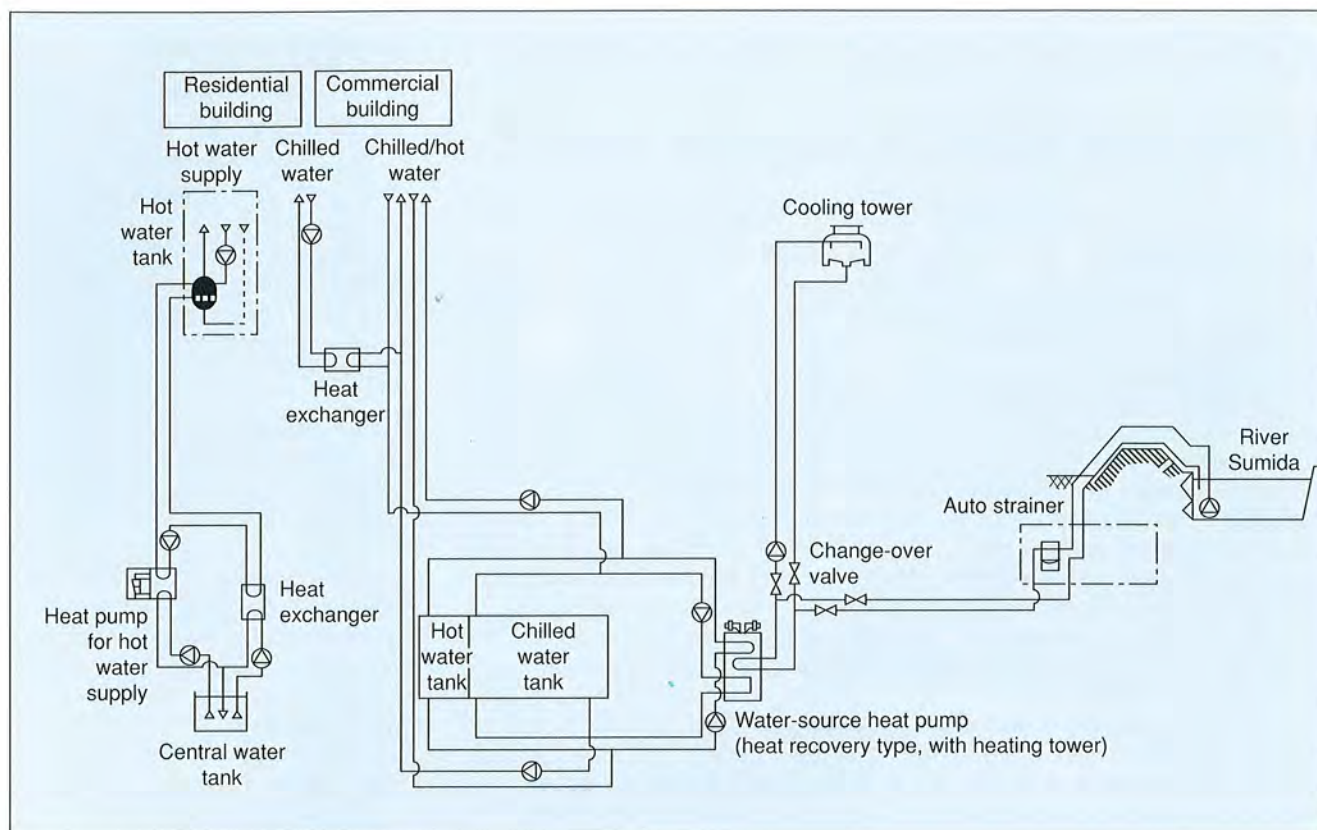


Figure 2: Heat Supply System Utilising River Water.

problems arising from sea water utilisation. Because sea water is used safely at many thermal power plants, there are no institutional problems. Additionally, since the environmental capacity appears to be larger than river water in many cases, use of sea water is expected to increase along with the development of waterfront areas.

CONCLUSION

It may be said that, in its short history in Japan, district heating/cooling systems are enjoying their heyday. However, if Japan remains idle during the boom hardships will be experienced once again. As long as district heating/cooling systems rely solely upon simple intensification of the use of boilers and chillers, they can never acquire true competitiveness. District heating pipes are shouldered as part of fixed expenses, and distribution pump power as part of variable expenses. So, unless district heating/cooling systems compensate for these with an "extra benefit that can be derived only from district heating/cooling",

they cannot be better than individual heating/cooling systems. One of the keys is the "utilisation of unused energy". A system designed to utilise unused energy incurs increased fixed expenses, but requires noticeably reduced variable expenses. Moreover, fixed expenses are also expected to decrease due to national financial support. As a consequence, external economic effects such as energy saving and environmental preservation can be derived.

At present, requests are being made for improvements to infrastructure to be effected by the 21st century. So, together with improvements in conventional infrastructure, such as the preservation of national land and industrial infrastructure, it is essential to improve "district heating/cooling equipment utilising unused energy". This improvement, in itself a social asset is intended to solve any further problems that may arise in the environmental area.

Literature:

- [1] Kuromoto et al., "Study on Utilisation of Waste Heat from Subway Tunnels", (Paper presented at conference on Air-Conditioning and Refrigeration, November 1990).
- [2] Report on Unused Energy Utilisation in Europe; Survey Mission Dispatched by the Federation of Electric Power Utilities, June 1990.

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Air-Conditioning of a Large Greenhouse Utilising Urban Waste Heat

*S. Miyamoto

INTRODUCTION

This article reports on Japan's first example of a heat pump using treated sewage as a heat source for air-conditioning the "Sakuya-Konohana-Kan", a permanent pavilion installed for the "International Garden & Greenery Exposition" held in Osaka. Amidst the recent tendency of cities to form "heat islands" due to increasing heat density, and increasing NO_x and CO₂ emissions, applying a heat pump system using treated sewage as a heat source for air-conditioning a large greenhouse is significant. It serves as an indicator for a desirable way to coexist with nature. One year has passed since its installation, and the projected efficiency has been achieved. From now on, it will help to facilitate heat pump applications for other heat-source recovery systems.

Being one of the biggest in Japan, the "Sakuya-Konohana-Kan" greenhouse reproduces natural plant environments in eight zones, from the tropical to alpine and polar zones, centring around the flower hall. In addition to a multi-purpose ventilating system and underground heating and cooling equipment, the greenhouse advantageously utilises sewage treated at the Moriguchi Sewage Treatment Plant, about 900 m away, as a heat source for its heat supply equipment. It also receives electricity generated at the Tsurumi Plant by utilising heat from waste incineration, thus achieving effective utilisation of city waste heat.

The pavilion consists of:

- an exhibition building - a large greenhouse, with a floor area of 6,890 m²;
- a culture building of 1,243 m², and;
- an equipment building of 1,062 m² floor area.

SYSTEM OUTLINE

Sewage treated at the Moriguchi Sewage Treatment Plant is fed through an underground water pipe, 300 mm in diameter, to the equipment building, and stored in the collected water tank within the equipment building. During the summer, treated sewage is utilised as cooling water for the heat pump as the heat sink, and during the winter as the heat source water, before being discharged into a sewage discharge waterway.

As illustrated later on in this article, the temperature of treated sewage during the summer is normally below, and during the winter above that of the outside air. Therefore, it is an ideal heat source and, other than is the case of well water, there is no fear of depletion. The system is designed to utilise treated sewage, and is composed of five water source heat pump units on the low temperature side and two on the high temperature side. In the greenhouse various types of climates suitable for a variety of plants must be created. Since alpine plants, among others, require all year-round cooling, preference is given to heat recovery operation by employing a heat pump type which simultaneously delivers chilled and hot water.

Heat Supply Equipment (See Figure 1)

- Water source heat pump chiller (reciprocating heat recovery type) x 5 units. Cooling capacity: 407 kW (4°C); heating capacity: 475 kW (50°C).
- Water source heat pump chiller (centrifugal type) for high temperatures of up to 80°C. Heating capacity 1,395 kW x 1 unit, 930 kW x 1 unit.

- Water-water heat exchanger (titanium plate type) for exchanging heat between treated sewage and heat source water. Capacity 850 kW x 2 units.
- Cooling tower to be put into operation when sewage supply stops. Cooling water flow rate 240 m³/h (37 → 32°C) x 1 unit.
- Gas-fired vacuum hot water heater reserved to serve when sewage supply stops. Heating capacity 2,325 kW x 1 unit.
- Collected water tank for storing treated sewage. Capacity: 450 m³.
- Chilled water tank (4 ~ 7°C). Capacity: 200 m³.
- Hot water tank (50°C). Capacity: 200 m³.
- Auto strainer; treated water rate 492 m³/h x 2 units.
- Ozone injector; injecting 150 g each time (1.0 ~ 1.3 ppm), injection cycle: twice/day.
- Pumps (x 20 units); 1 set.

The system is monitored and controlled from a central control room.

Basic Operational Patterns

- Heating cycle (during the winter and intermediate seasons)
See Figure 1. Treated sewage at 12 ~ 15°C is fed through a sewage supply pipe and is stored in the collected water tank. After exchanging heat through the water-water heat exchanger, it is discharged into a waterway. Heat source water at 10 ~ 13°C, which has absorbed high temperature heat through the heat exchanger, is then used by the low temperature heat pumps,

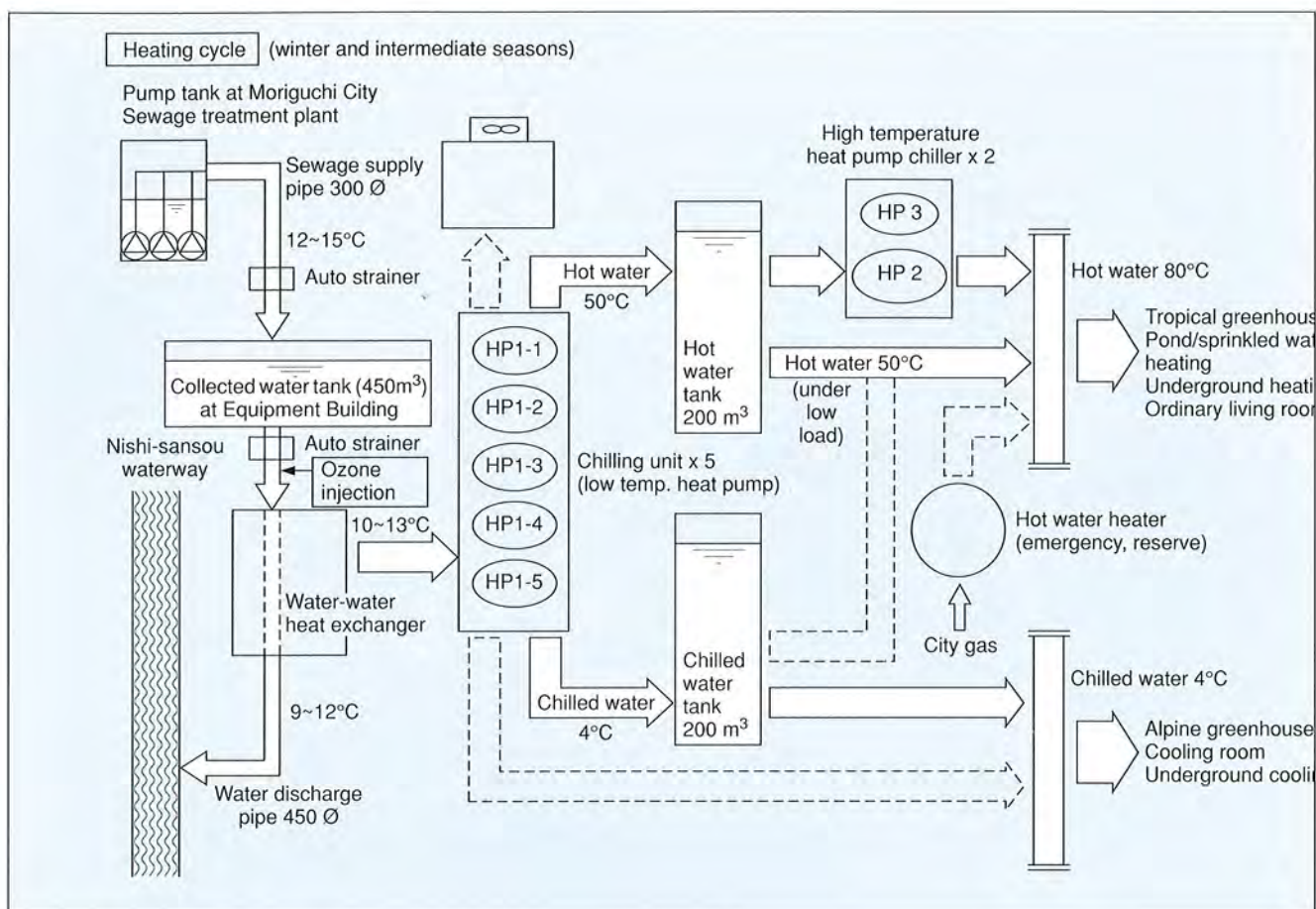


Figure 1: Heating Operation.

and hot water at 50°C is supplied to the hot water tank. Chilled water at 4°C is produced by selectively operating the five heat pumps according to the demand for chilled water and fed to the chilled water tank. 50°C hot water acts as the heat source for a high temperature water heat pump, effecting the second stage temperature increase so that hot water at 80°C is fed into the greenhouse. Chilled water is directly fed into the greenhouse from the chilled water tank. During the intermediate seasons, when the system load is low, it is possible to directly feed hot water at 50°C from the hot water tank into the greenhouse. In those cases where the sewage temperature drops excessively (i.e., below 9°C), or the sewage supply stops, a gas-fired hot water heater supplements the operation.

• Cooling cycle.

See Figure 2. Treated sewage at 25 ~ 28°C fed by the sewage supply pipe is stored in the collected water tank. Chilled water at 4°C and 7°C is produced from heat sink water at 30 ~ 33°C, the heat being discharged to the sewage water through the water-water heat exchanger. Chilled water at 7°C is stored in the chilled water tank, and then supplied to the greenhouse, while chilled water at 4°C is supplied directly to the greenhouse. If the sewage supply is interrupted, the cooling tower functions to discharge heat.

EVALUATION OF HEAT SUPPLY EQUIPMENT

Heat Exchanger Slime Prevention

One of the technical problems involved in the use of treated

sewage has been a drop in the heat exchanging efficiency caused by the adherence of organic matter to the heat exchanger plate. As a countermeasure, the system uses both an auto strainer and intermittent ozone injections based on the results of preliminary tests at the Moriguchi Sewage Treatment Plant. Ozone is injected for five minutes at a time, twice a day. In May 1989 the system was disassembled and inspected, and in the summer of 1990, when actual load levels were high and the flow rate increased, performance of the heat exchanger did not decrease. The heat exchanger approach temperature remained at its design value ($\approx 2^\circ\text{C}$). It may be concluded that intermittent ozone injections are excellent slime preventive measures.

Sewage Utilisation System

Sewage and outdoor temperature fluctuations are shown in Figure 3. Although outdoor temperatures

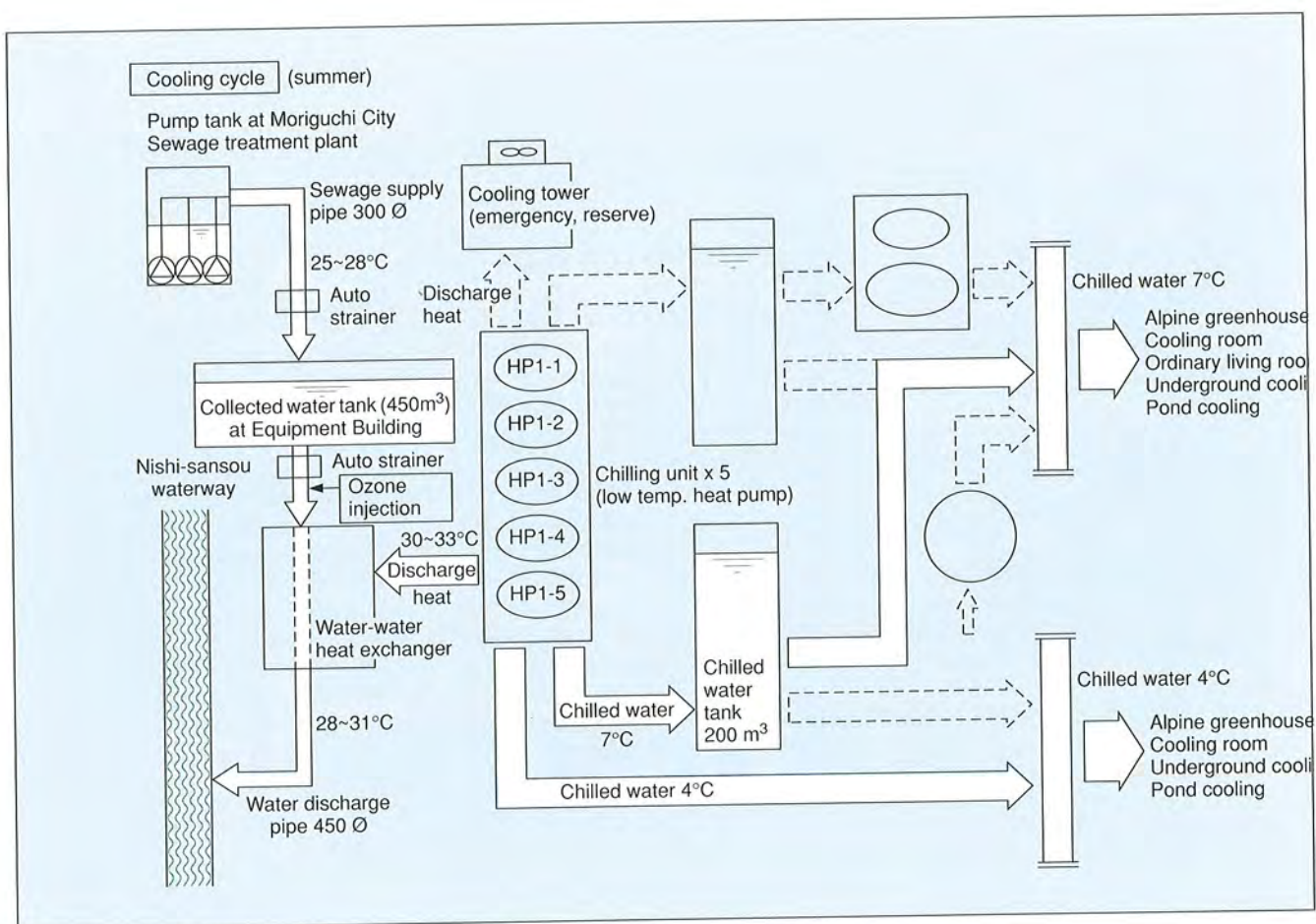


Figure 2: Cooling Operation.

were comparable to those in normal years, sewage temperatures were a little higher than normal. According to more detailed data, in July, outdoor air temperatures are a little higher than sewage temperatures. During the winter when the sewage temperature was at its lowest, it remained at 15.2°C, or about 9°C higher than the lowest average value of outdoor air temperatures. Thus, in a recovery system like this, which primarily depends on heat recovery, sewage serves as a more effective heat source than air.

Heat pumps

One of the low temperature heat pumps was evaluated and proved to perform according to design values for all modes of operation. Since the high temperature heat pumps were operated at a low 30 to 60% load, they showed low COP values, i.e. 50 ~ 70% compared to factory test values or design values. However, since the

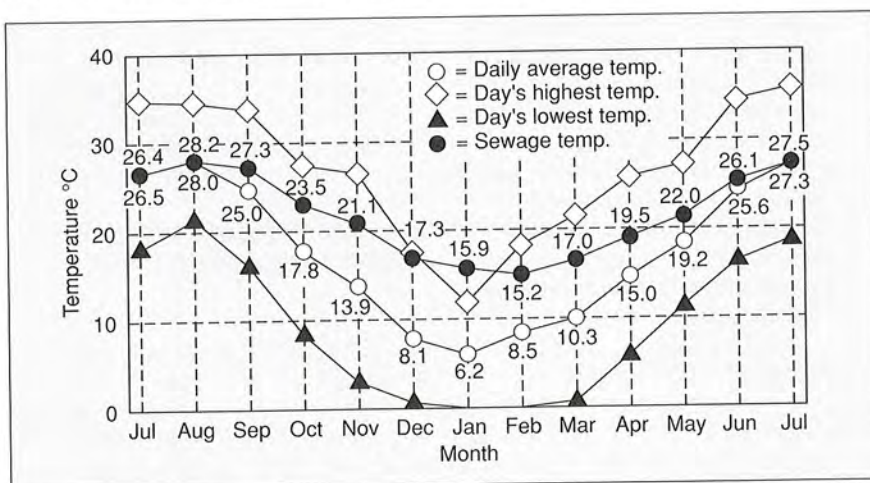
compressor features continuous capacity control, it was confirmed that the COP was nearly the same as the design value when compared with the characteristics when operating under capacity control. In March, this year, when the system incurred a low load, the system control method was

improved from the conventional parallel operation of two units to the operation of only one unit.

EVALUATION OF SYSTEM CHARACTERISTICS

The system COPs are shown in Table 1. In the cooling mode

Figure 3: Treated Sewage Temperature in °C from July 1989-July 1990.



(column A), the load from day to night varied between 60 ~ 70%. The COP of the heat supply equipment was 3.42 - an initially planned value. Taking into account the 12% additional power input by the heat source pump, the system COP was 3.03, while the COP derived by adding about 15% power input from the water intake pump became 2.68. Thus, it was concluded that the system water intake pump power must be reduced.

The heat recovery operational mode, (column B) consists mainly of heat recovery in the daytime and 50°C of heating at night. The COP of the heat supply equipment was 5.24, while the load was only 8% during heat recovery and 20% during heating. Thus, the heat

source pump's input ratio increased to 22%, resulting in a system COP of 4.08. Incidentally, despite the extremely low load, the COPs of the heat source units were nearly equal to the design values. Thus, it was confirmed that a system comprising plural heat source units was functioning efficiently.

Column C in Table 1 shows the COP of the high temperature water system. The high temperature delivery system features an increase of temperature in two stages up to 80°C, involving a heat pump for high temperatures under continuous capacity control. The COP of the heat source units was 1.94 under 45% load condition. Thus, under full load conditions a COP of about 2.3 to 2.6 for the

heat source unit can be expected, and a system COP of more than 2.0 also becomes possible.

CONCLUSION

As a result of the evaluation, the following findings were made:

- the temperature of treated sewage is more stable than the outdoor air temperature and is specially effective during the winter;
- the slime adherence prevention effect of the intermittent ozone injections upon the heat exchanger is noticeable;
- the heat supply to the load side is stable, and the control system for the equipment building, etc., operates as designed;
- the heat pump performance satisfies the design specifications;
- the heat source units have operated normally for the one year test period, and the one year of normal operation since, thereby demonstrating a high reliability rate.

At present, global environmental problems are extensively being discussed as serious social problems. In a city suffering from NO_x and CO_2 emissions, an overall heat utilisation system using urban waste heat has been commercialised. This is very significant, and will serve as a step for the diffusion of heat pump applications in other heat recovery systems.

Operating Mode*			A	B	C
Date			Aug.7, '90	Nov.5, '90	Mar.5, '90
Time-span (hours)			13:00-17:00	19:00-23:00	19:00-23:00
Cooling capacity (kWth)			1,428	534	1,030
Input (kWe)	Each input	1 HP (heat source unit)	418	102	532
		2 Heat source unit's pump	53	29	63
		3 Water intake pumps	62	4	10
	i) 1+2		471	131	595
	ii) 1+2+3		533	135	605
COP	Heat source unit		3.42	5.24	1.94
	System 1 (heat source unit + heat source unit's pump)		3.03	4.08	1.73
	System 2 (system+water intake pump)		2.68	3.95	1.70
* Operating Modes: A = Cooling B = Heat recovery/50°C heating delivery C = Heating/heat recovery - high temp. delivery					

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Laundry Dryer Exhaust Heat Recovery for Preheating Wash Water

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ABSTRACT

A heat recovery system, to reclaim energy from the exhaust of commercial-sized laundry dryers and preheat the water used for the washing machines, has been designed, built, and installed on a test site. The system layout incorporates both a passive and active heat collection loop. The passive loop circulates water directly through two sets of finless coils placed within the heat stream while the active loop employs a 5 kW heat pump and collects energy via an evaporator placed down-stream of the passive coils. The system has been operating for approx. three months and preliminary results indicate an overall energy saving in excess of 50% of the original setup.

INTRODUCTION

Among other energy conservation alternatives, commercial waste heat recovery is an option that has been currently considered in many applications. The attractiveness of this technology stems from the high efficiency of the systems developed and consequently their potential for significant energy savings and power demand reduction[1].

Heat recovery processes can displace other energy sources for space heating, cooling, and water heating. A conventional system is based on a very simple mechanism - energy, in the form of heat, is retrieved from the rejection products of a process, transferred to an intermediary storage, and then used in another process with an energy need. Success strongly depends on a suitable match between the heat source and heat sink which should be located geographically close, provide and require the same grade of energy, and operate on roughly coinciding schedules of supply and demand.

A typical laundry is an excellent candidate for heat recovery as it satisfies the above criteria. Energy from the hot moist air discharged during the drying process and from the warm used wash water ("grey" water) can be reclaimed and used immediately on site to preheat domestic fresh water for the washing machines. An integrated system can save needlessly wasted energy, reduce power demand, and substantially lower the operating costs.

OBJECTIVES

With consideration to the potential for commercial heat recovery, a field project is being conducted by Ontario Hydro Research Division to demonstrate the capabilities of advanced heat pump technology in commercial laundries for waste heat recovery, water heating, and energy management. Heat recovery from the dryer exhaust of an all-electric laundry was of primary consideration in the design, installation and operation of this system; however, waste water heat recovery from washing and rinsing may also be considered in the future.

The objective of this project is to investigate the performance and reliability of a heat recovery system for a medium sized laundry facility. Reclaiming heat from clothes dryer exhaust has generally been avoided due to the perceived difficulties with lint build-up on the heat exchangers. Consequently, special attention was given to the design of system components exposed to lint accumulation. It is therefore expected that the experimental results will also demonstrate that this problem can be considerably reduced by using large fin spacings on the heat exchanger coils and relying in part on the self cleaning effect of condensate run-off.

DESCRIPTION

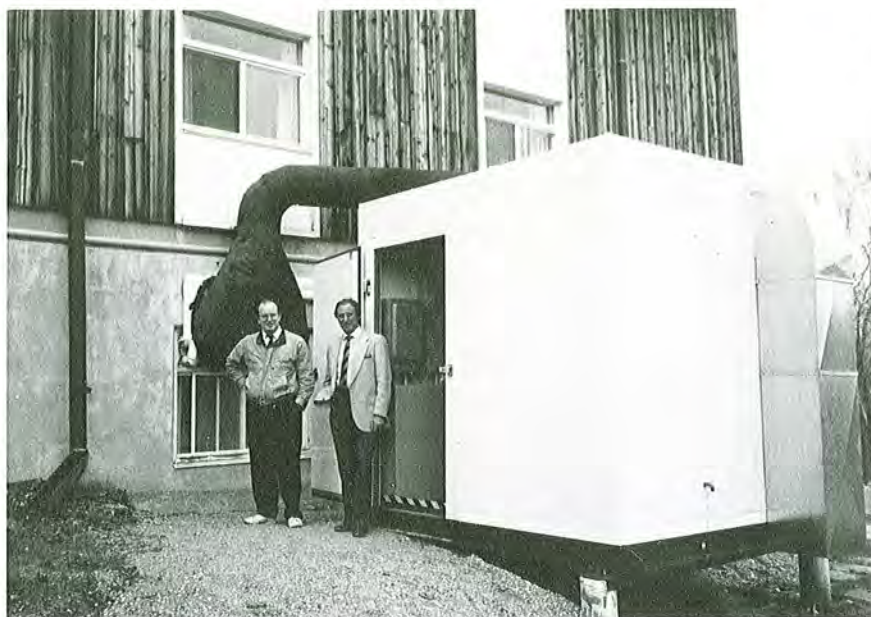
Test Site

A suitable location for the heat recovery system was chosen at the laundry facility of the Ontario Hydro Conference and Development (C&D) Centre in Orangeville, Ontario. This centre provides management and professional training to Ontario Hydro employees and it operates like a medium size hotel with 223 individual rooms, offices, recreation and dining facilities, training facilities, and conference rooms.

Kitchen and room linen is washed every day by the centre's on-site laundry facility. The laundry operates over a typical eight hour working day, during which approximately 450 kg of linen are processed. Over the course of a typical day, the laundry requires an average of 4,500 litres of hot water; however, daily demands as high as 6,200 litres have occasionally been recorded. A typical average daily hot water demand profile is illustrated in the performance section of this article.

Heat Recovery System

To facilitate the heat recovery equipment, an insulated module was assembled and placed near the dryer exhaust ports immediately outside the laundry (see Photo 1). This circumvented the unusually low space availability within the laundry and also allowed research staff unobtrusive access to the equipment. Cold water is drawn from the main line and directed into the module where it is preheated by the heat recovery system and the backup heaters. The "wash-ready" hot water is then piped back into the laundry room to be used by the washing machines. A manual override of the heat recovery system was installed



Photograph 1: Laundry Heat Recovery Facility.

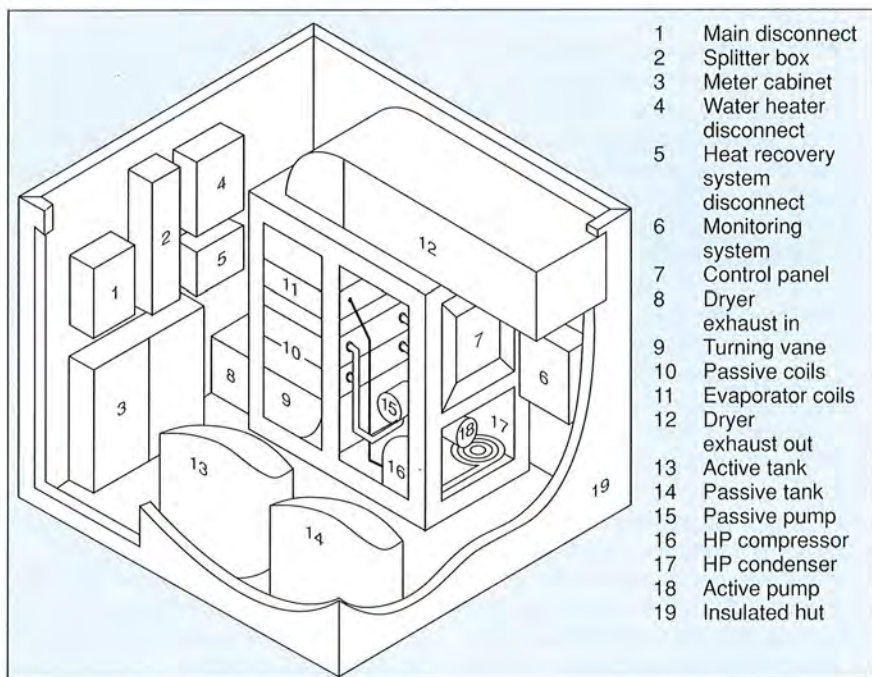


Figure 1: Layout of a Heat Recovery Module.

to avoid disruptions in the laundry operation schedule in the event of a failure. The override also allows for different modes of operation which may be investigated in the future.

As shown in Figure 1, manifolded laundry exhaust products entering the module are diverted upwards by a turning vane, pass through the heat recovery section, and are finally discharged into the outside environment. The recovery system can be subdivided into two separate subsystems - a "passive"

heat exchange system, and an "active" system. The passive system circulates water from the bottom of the passive storage tank, through finless heat exchanger coils, and then back into the tank. The active system incorporates a 5 kW heat pump which collects energy from a large fin spacing evaporator placed within the dryer exhaust immediately downstream of the passive coils. Water is taken from the bottom of the "active" storage tank, circulated through the heat pump condenser, and then back into the tank. A simple water flow

diagram for both the passive and active loops is presented in Figure 2.

The laundry occasionally demands a very high volume of hot water in a short amount of time; some draws have been recorded above 350 litres over a 15 minute interval. Consequently, the active tank is equipped with two 18 kW backup electric resistance heaters which are set to operate when the tank temperature drops below the minimum required by the washing machines.

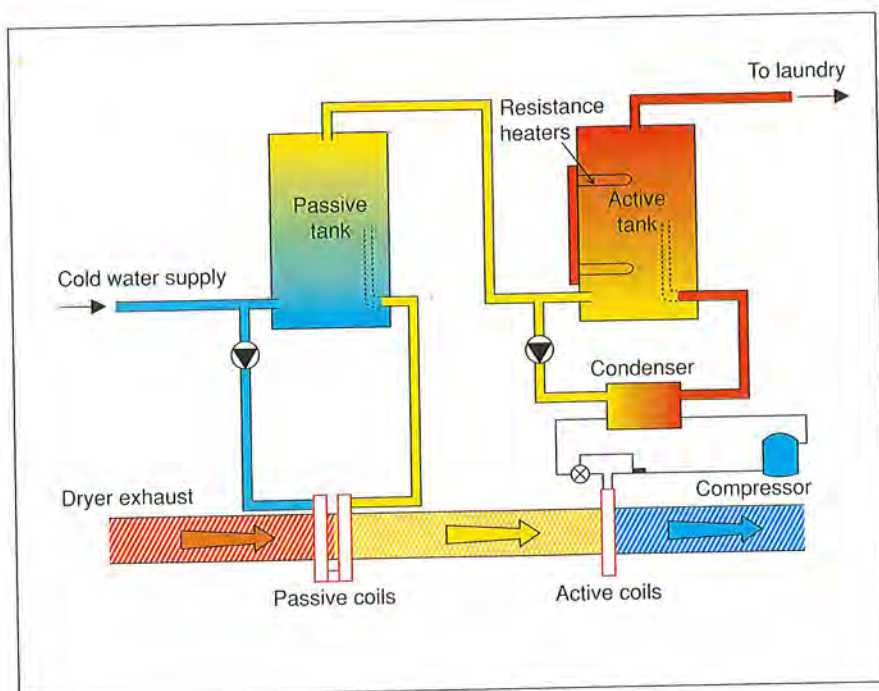


Figure 2: System Water Flow Diagram.

Lint Control

Although the dryers are equipped with internal lint filters, the exhaust products still contain a fair amount of lint which *may* have the ability to plug the heat exchanger and the evaporator coils. As shown in Figure 1, the passive and active coils were positioned horizontally to allow the lint to fall off easily. It is also expected that the condensate, which forms on the upper active coils, will assist the self cleaning process by continually washing off the lint which collects on both sets of coils.

As a backup, a spray wash bar connected to a timer was installed above the active coils to be used in the event that the condensate wash-off procedure proved to be ineffective. The spray wash bar sprays a pre-specified volume of cold water down onto the active coils at an adjustable time interval. The force of the water running down the active and passive coils should provide adequate cleaning.

Monitoring

A monitoring effort is now underway which provides information on site parameters, system components, and system performance. Relevant quantities are scanned every five seconds and the averages

and totals are stored in the data-logger memory every fifteen minutes. This information is collected by a computer at the research office for further processing and analysis.

PERFORMANCE

At the time this article was written, the heat recovery system was commissioned and had been operating for a period of approximately three months. Preliminary results are very encouraging and show a 50% reduction in the total water heating energy consumption by the laundry and an average power demand of 17.5 kW which is approximately half of the demand required for direct resistance heating. Average daily energy savings of 142 kWh have been achieved so far. To illustrate some of the "typical" operating characteristics of the system, Figure 3 presents average hourly data over the week of April 29 to May 3, 1991, on hot water demand, system thermal output versus electrical input, and the estimated power demand reduction.

A direct measurement of the demand reduction realized by the system proved inappropriate. This is due to the original setup of the laundry, for which hot water was

supplied by a 1800 m³ (4,000 Imperial Gal) electric boiler that also supplies all the hot water requirements of the C&D Centre including showers and kitchen use. The laundry therefore had a relatively moderate impact on the boiler performance, and due to the continually variable operation of the training centre, a typical daily demand profile could not be established.

The passive loop has performed exceptionally well and is usually providing 25 to 35% of the overall water heating energy. Performance for this loop is enhanced by the cold temperature of approximately 5 to 15°C of the incoming fresh water. The heat pump has shown an average COP of approximately 3.5 and provides about 25% of the overall water heating energy. The remainder of the energy required for water heating is provided by the backup resistance heaters which are extremely important during periods of high draw.

The mechanical reliability of the system, over the experimental period since its commissioning, has been very good. Except for several minor adjustments no major interventions or special maintenance procedures were required.

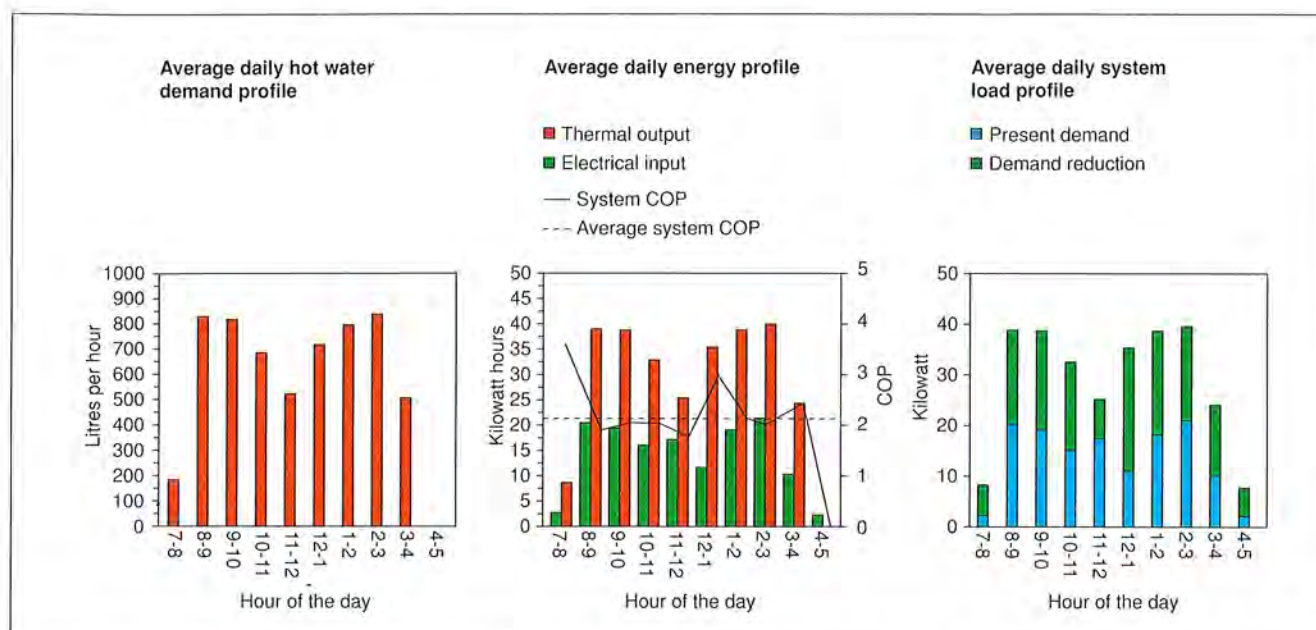


Figure 3: Average Hourly Performance Data for Week of 29 April to 3 May, 1991.

The system was initially operated for two months with a set of washable pre-filters installed upstream of the passive heat exchanger coils. However, continuous inspection indicated that lint build-up on system components was minimal and that the filters themselves were collecting only a moderate amount of lint. This is probably due to the effectiveness of the internal lint filters within the dryers. Consequently, the pre-filters were removed and the visual inspection of the coils was resumed.

Remarkably, both the passive and active coils have remained devoid of any lint accumulation. It appears that the condensate run-off is effective enough to wash the coils and collect the lint at the bottom of the drain pan. The back-up spray bar has not yet been used, and it appears that it will not be required. This is encouraging from the system operation standpoint and represents an important feature for the acceptance of this technology by the personnel involved in the operation and maintenance of the laundry facility.

FUTURE WORK

This is an on-going project and monitoring is expected to continue

for at least another year. During this time typical hot water and energy use patterns will be developed and the performance and mechanical reliability will be assessed. Appropriate adjustments will be made to optimise the system operation.

To gain more practical experience with laundry heat recovery systems, observations on lint accumulation will be conducted on a regular basis and the effectiveness of condensate run-off will be assessed. In addition, all major electrical loads will be monitored, energy savings and power demand reduction will be quantified, and recommendations on potential energy management strategies will be made.

It is expected that the results of the monitoring and lint observation will provide valuable insight for the heat recovery role that advanced heat pump technologies will play not only in commercial laundries, but other particulate filled exhaust environments.

The preliminary results have indicated that these technologies can indeed save energy, reduce power demand, and increase customer satisfaction. In addition, they represent an environmentally

friendly alternative in the management and use of low grade energy.

Reference:

- [1] Perlman, M.; Bell, J.M.; Chiarelli, M.; Elphick, I; *Heat Recovery Applications in Commercial Establishments*, Presented at : CLIMA 2000 - Second World Congress on Heating, Ventilating, Refrigerating and Air-Conditioning, Sarajevo, Yugoslavia, August 1989.

*Authors: Glenn Moore and Maier Perlman, Ontario Hydro, Canada.



Heat Pumps in Salmon Smolt Hatcheries

* P. Johansen

ABSTRACT

The first heat pump system in a salmon smolt (young salmon) hatchery was built in 1979 in Norway. The purpose was to warm the water up to 12°C from February onwards in order to begin "start-feeding" early, and to keep the fish in warmed water until the natural water temperature exceeded 10°C in the spring. This was quite successful. "Start-feeding" loss was reduced and the amount of smolts ready for further rearing after one year of being in the hatchery (one year smolts) was increased.

During the first few years, this new process met with scepticism from farmers and scientists, claiming that this unnatural process would result in reduced resistance when smolts were put into salt water, and problems with prematurity would arise.

However, after a number of years, experience had shown that there were no reasons for this scepticism, and the one year old smolts

provided good results. In 1984 there were approximately 50 heat pumps installed in salmon hatcheries worldwide, the majority in Norway. From that time onwards things developed quite rapidly, and presently, the total number in Norway and other countries is estimated to be approximately 230 units.

SALMON HATCHERY HEATING UNIT

The heat pump is connected to a heat exchanger at the water intake of the hatchery. Typical temperatures of the process are shown in Figure 1.

The 2°C inlet water passes the heat exchanger and is warmed up to 13°C. Passing through the fish tank, results in a heat loss of approx. 0.3°C, and the effluent water temp. drops to 12.7°C. A pump takes the effluent water through the condenser, where it is warmed up to 15.5°C. The water is now warm enough to exchange heat with the inlet water. Passing

the heat exchanger the effluent water is cooled down by 11°C to 4.5°C. Having equal flows on both sides of the heat exchanger, allows the inlet water to gain 11°C, the same as the temperature decrease on the effluent side. From the heat exchanger, the effluent water of 4.5°C passes the evaporator and is rejected at a temperature of 2.3°C.

The heat exchange and power consumption figures of the process are:

Evaporator:	$Q_0 = 110 \text{ kW}$
Compressor:	$W = 25 \text{ kW}$
Condenser:	$Q_c = 135 \text{ kW}$
Heat exchanger:	$Q_{HE} = 535 \text{ kW}$
Pump:	$Q_p = 5 \text{ kW}$

The coefficient of performance of the total system is:

$$\text{COPs} = Q_{HE} / (W + Q_p) \\ = 535 / (25 + 5) = 17.8$$

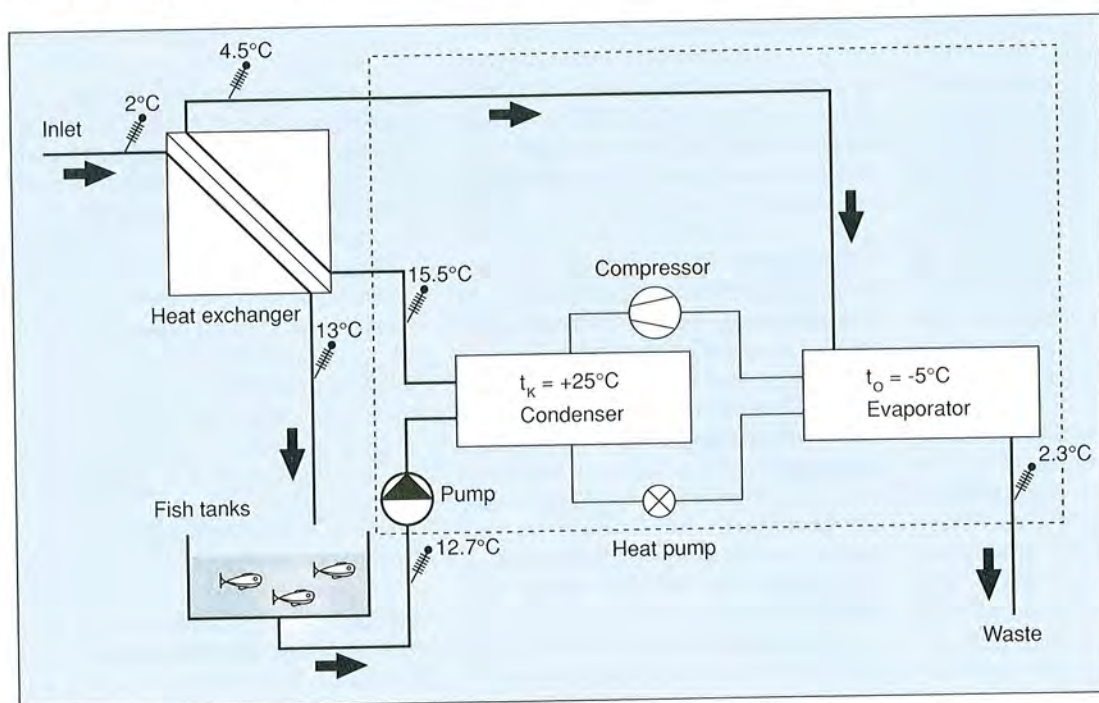


Figure 1:
Diagram of a
Hatchery Process
Using a Heat
Pump.

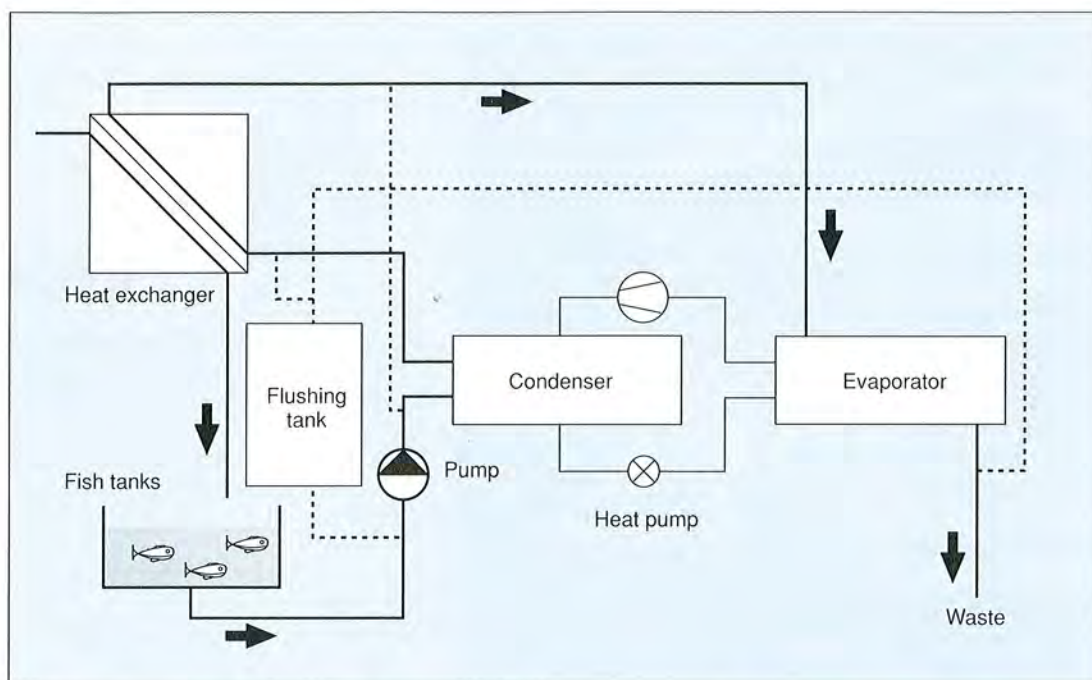


Figure 2:
Backwashing
System.

OPERATIONAL PROBLEMS

The waste water contains feed and excrements, which coat the heating surfaces. A "standard type" heat pump would clog up within a short period. It is therefore very important to use condensers and evaporators that are specifically designed for operation with contaminated waste water. The design of the plate-heat exchanger must minimise fouling. However, even if a special design is used, periodic cleaning is still necessary.

An effective backwashing system is therefore integrated in the plant (see Figure 2). Heated water with fat solvents is circulated from a flushing tank in a closed cycle. The heat exchanger, condenser and evaporator are backwashed separately one after the other. The backwashing operation takes 10 minutes, and must be performed every 4-5 days.

The recent introduction of filters between the fish tank and the pump has prolonged the running

period between backwashings to 15-20 days. The Unic Rotary Filter in Figure 3 and the Triangle Filter in Figure 4, both operate satisfactorily. It is now always recommended to install filters to screen the waste water coming from the fish tanks.

As the water is heated when passing the heat exchanger, super-saturation conditions will occur. The water must therefore pass a de-gassing unit before it goes to the fish tanks. Many home-made de-gassing units have been used with unsatisfactory results. Too high levels of nitrogen saturation do not allow the fish to prosper and result in a high mortality rate. It is advised to keep a record of the saturation values, and a saturation meter should be available at the site at all times.

Sites having low temperature water supply in the winter, can expect freezing in the heat pump evaporator if the water temperature drops below 1.5°C. This temperature includes a safety margin of 1°C, because unstable conditions occur when tanks are drained for cleaning, or after backwashing. The best way to handle low temperatures is to heat the supply water with sea water. If sea water is not available, frost protection of the heat pump by electric heaters is

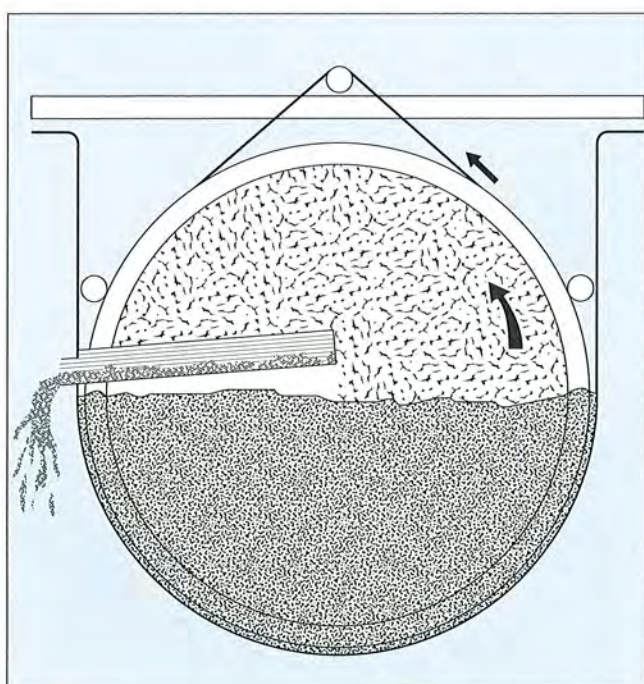


Figure 3: Unic
Rotary Filter.

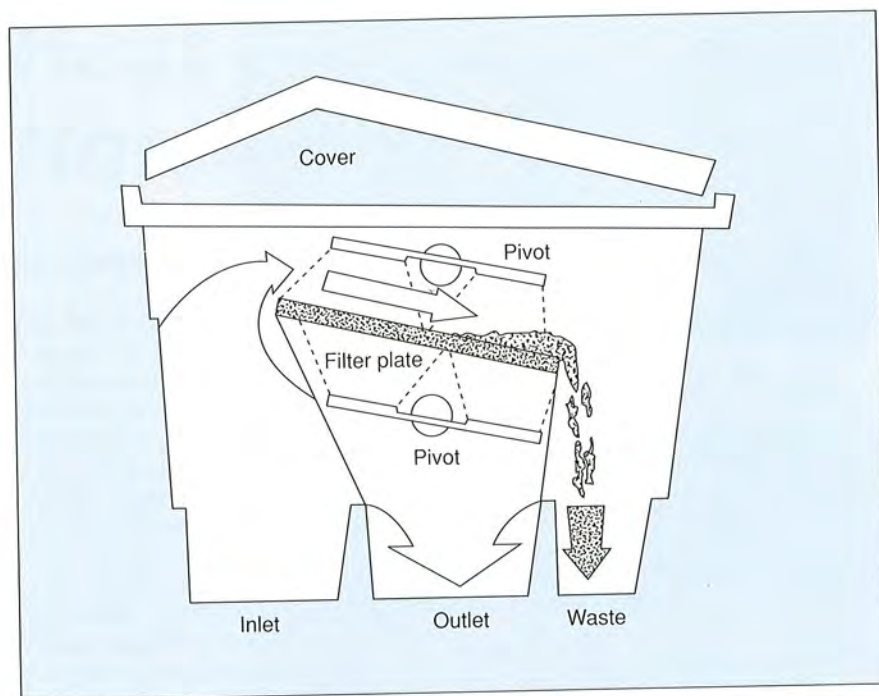


Figure 4: Triangle Filter.

necessary. For a heat pump of the size explained, a 30 kW heater would be sufficient to secure operation down to 0°C. This means that the power consumption is doubled and the COP of the system is reduced below 10.

SEA WATER HEAT EXCHANGERS

Hatcheries having access to deep sea water can warm up large flows of fresh water at low cost by using sea water heat exchangers. Shores which are influenced by the Gulf Stream have 7 - 8°C salt water available at a depth of 20-30 m, during the coldest winter period. A sea water heat exchanger is of

the plate-heat exchanger type. The plates used are of a Titanium material which resists sea water corrosion. The heat exchanger connected to the heat pump uses stainless steel plates. A plate-heat exchanger is highly efficient and flexible and can be opened for cleaning. Plate-heat exchangers can be delivered in many sizes, from small harmonicas to 30 tonne units.

The normal running period (Norwegian locations) of these exchangers is from November to May. Sea water heat exchangers are also used to cool fresh water in case of high summer temperatures and/or high fish stocking densities.

Furthermore, sea water heat exchangers are used to compensate day/night fluctuations during the spring and autumn. The following example is given to illustrate investment and running costs of a sea water heat exchanger:

Fresh water temp.:	0.5°C
Sea water temp.:	7°C
Warmed fresh water:	5.3°C
Fresh water flow:	180 m ³ /h
Sea water flow:	180 m ³ /h
Transferred heat:	1000 kW
Pumping power:	20 kW

Total investment: NOK 350,000 (approx. US\$ 50,000).

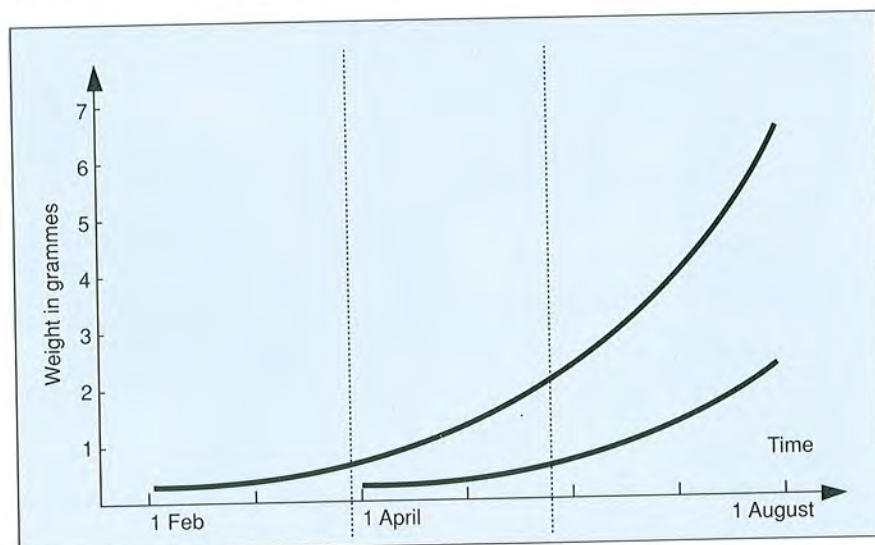


Figure 5: "Start-feeding".

The investment cost is highly dependent upon the length of the sea water intake pipe and tide differences at the location. But this is the cheapest energy there is - only 20 kW pumping power is needed to obtain 1000 kW of heat.

THE ONE YEAR SMOLT PROCESS

It is important to have a large share of one year smolts to achieve profits in smolt rearing, because:

- production capacity can be doubled at the same tank volume;
- feeding costs are lower;
- labour is less;
- the payback period is shorter.

To make "start-feeding" possible with minimum losses, the water

temperature should be about 12°C. The fish must thereafter be kept at 12°C water until the natural water temperature rises in the spring.

The curves in Figure 5 show how important early "start-feeding" is. When "start-feeding" begins from February 1, an average fish weighs approximately 7 grammes by August 1, compared to only 2.5 grammes when "start-feeding" begins on April 1.

A second heating period begins in the autumn when the natural water temperature falls below 8-9°C. Since the larger biomass (fish) requires a much higher water flow, heating is only applied to the smallest fishes. This helps them to grow and become smolts the following spring. Heating during the autumn also yields a larger weight smolt, which, in turn, results in a better profit. This

process also utilises the heating equipment. A heat pump is a big investment, but the running cost is low. A longer running period gives a better investment payback, as long as heating improves growth and production.

TYPICAL ANNUAL COST

This is a difficult subject to tackle, especially in fish farming where different running conditions have a strong influence on the productivity. The figures in Table 1 are taken from a middle-sized Norwegian smolt unit.

CONCLUSION

Experience with a great number of installations gained in Norway over more than a decade, has shown that the application of heat pumps in salmon smolt hatcheries does not pose technical problems, yields a higher fish production rate, and is economically attractive.

Table 1: Economic comparison of a heat pump versus an oil boiler for an annual production of 600,000 smolts (average size 70 grammes).

Heat Pump	
Total investment whole smolt unit:	NOK 6,100,100
Heat pump investment:	NOK 818,000
Annual running hours:	4,300 h
Power consumption:	65 kW
Heating power:	1,080 kW
Utilisation factor HP:	0.7
Electricity price:	NOK 0.35/kWh
Interest rate:	12%
Amortisation factor 10 years:	0.17
Running costs:	NOK 68,470
Capital cost HP	NOK 139,060
Total annual costs:	NOK 207,530
Cost per smolt:	NOK 0.35
Oil boiler	
Total investment:	NOK 5,720,000
Boiler plant:	NOK 470,000
Running hours:	4,300 h
Oil consumption:	36 l/h
Electric power consumption:	10 kW
Oil Price:	NOK 2.3/l
Running costs for oil:	NOK 250,000
Running costs for electricity:	NOK 10,500
Capital costs for boiler:	NOK 79,900
Total annual costs:	NOK 340,400
Cost per smolt:	NOK 0.57
(Note: 1 NOK (Norwegian Krone) = approximately 0.142 US\$)	

* Author: Per Johansen, Aquaterm, Norway.



Effective Use of Waste Heat by Closed Cycle Industrial Heat Pumps

*J.S. Gilbert and R.C. Niess

ABSTRACT

Acceptance of the industrial heat pump in the United States has been adversely affected because too much attention has been placed on "project economics" and not enough on the real reasons an industrial plant might select this heat recovery option. This article discusses this aspect and outlines an approach that addresses both the buying motives and an effective economic analysis method, using a textile plant as an example. The concepts advanced are based on both authors' many years of experience in applying industrial heat pumps in plant facilities throughout the United States.

INTRODUCTION

Acceptance of the Industrial Heat Pump (IHP) in the United States (US) has been slow in the industrial and commercial non-HVAC markets. Setting aside your probable interest in the IHP itself for the moment, let's start with the reasons why most industrial and commercial firms are **not** interested in the device:

- **Energy Efficiency and Conservation:** These firms are not interested in these altruistic motives. They are interested in saving money. And, ahead of even that, they look for opportunities to save money while solving operating problems. Therefore, the primary driver for industrial heat pump adoption is that it solves a problem.
- **Technological Correctness:** Most firms do not really care whether the device is the right choice or even if it is located at the optimal point in the process. They simply want to rid themselves of the operating bottleneck.

Another observation that can be made, based on many field contacts with potential users, is that we should probably stop calling the device a heat pump - that is a sure way to set the conversation on the wrong track. Most US customers confuse the IHP with space conditioning designs. We are talking about a closed cycle design that recovers process waste heat and therefore does not reverse. In addition, the IHPs discussed here are fairly large - 350 kW would be a small one. Some IHPs installed in the pulp and paper industry are measured in megawatts (MW), not kilowatts (kW)!

Finally, most IHPs installed in the US are in combination with heat exchangers (HX). This application, if properly implemented, requires an optimisation of each of the two segments. This optimisation is different from that of "placement" as epitomised by the pinch technology. One project caveat is "do not combine a winner with a loser to obtain a marginal winner". This concept will be developed later in this article.

KEYS TO GAINING ACCEPTANCE

What industrial firms are interested in is *solving an operating problem*. Today these problems often are environmental in nature, and waste heat and emissions top the list. They must do something about their waste heat, and reduce emissions.

Minimise or Recovery of Waste Heat
Simply put, the firm can either no longer reject waste heat or has run out of the capability of rejecting waste heat to the environment. This commonly occurs in food processing plants when they run out of evaporative condensing or cooling tower capacity, in textile finishing mills when the local

township puts tighter restrictions on waste water discharge temperatures.

Reduce Emissions

The major source of emissions in many plants results from the combustion of fossil fuels in the boilers. Reduction of boiler load reduces emissions.

Heat recovery techniques can help solve both of these problems. These can do it with a return on the investment, avoiding a negative cash flow simply to satisfy a government mandate or perceived social need.

If either an IHP or a HX is to be considered, two conditions must exist: 1) a source of waste heat, and 2) a need for the recovered waste heat. Note that if the two do not occur simultaneously, thermal storage must also be considered. This latter technique will not be discussed in this article.

WHERE DOES THE IHP FIT IN THE FAMILY?

It is essential that the IHP be considered as a member of the heat recovery family. Outside of dehumidification kilns, it is seldom installed without a heat exchanger. Therefore, it is usually a good idea to begin the evaluation with the search for cost-effective heat exchange. Then, when heat exchange has run its economic course, the next step is to consider the heat pump.

For example, suppose the following textile mill situation exists in a plant:

- water stream of 31.5 l/s of 43.3°C (110°F); environmental and other restrictions mandate the discharge temperature can be no higher than 26.7°C (80°F).

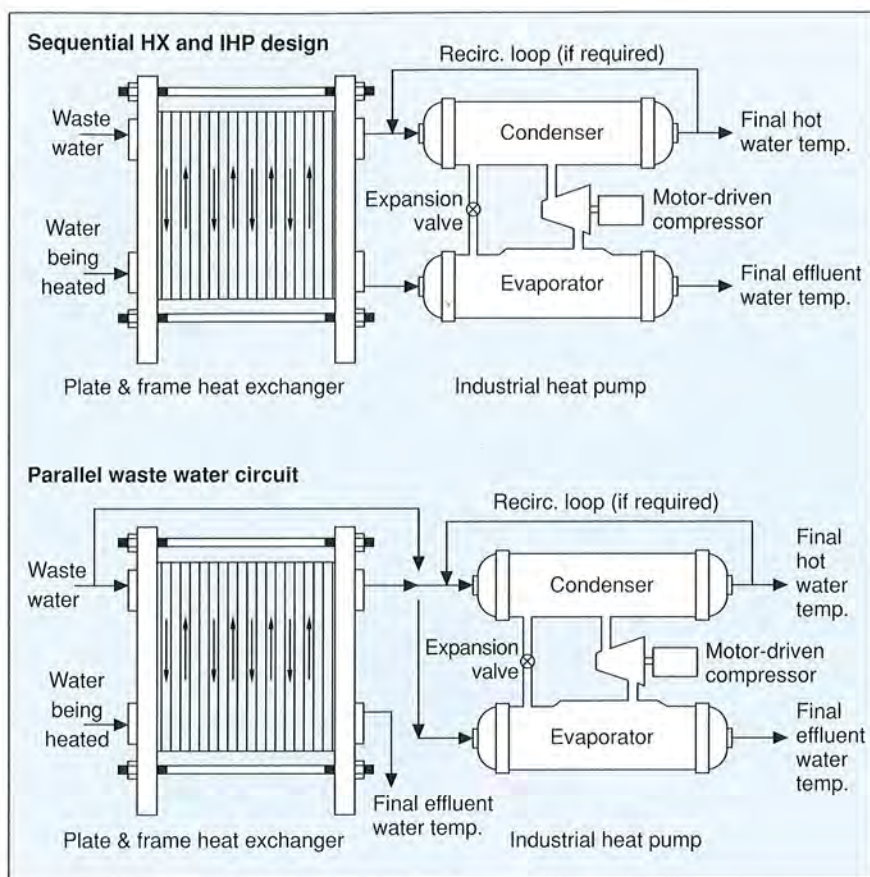


Figure 1: Basic Combination Heat Pump - Heat Exchanger Diagram.

- Hot water is needed for the dye and bleach operations: 15.75 l/s at 54.4°C (130°F) (heating clean 15.6°C (60°F) ground-water).

Since the incoming stream is much smaller than the waste water stream, there are two basic choices as shown in Figure 1:

- take all of it sequentially through a heat exchange and heat pump pair, or
- split the waste water stream to take part of it through a heat exchanger and part through the heat pump, a parallel waste water circuit.

Note the advantage of splitting the waste heat stream is that the IHP coefficient of performance (COP) will be higher, but the disadvantage is that both heat exchangers have to tolerate freshly discharged, contaminated mill waste water.

This usually requires a specialised heat exchange, designed to

minimise fouling and plugging. In the US, this design typically is one where the contaminated mill water flows through continuous tubes having no velocity change other than that at the inlet and exit. In addition, a four way valve is used to alternate the mill water flow direction over a selected time interval. For example, the flow is reversed for five minutes every half hour. This design minimises operating problems in reclaiming waste heat from contaminated water. The metallurgy of the tubes is another site-specific design decision, based on plant experience or technical review of the mill water contaminants. The clean water being heated flows through the HX shell and contacts the outside of the continuous tubes.

In most cases, and for this discussion, the assumption used is the typically common choice: take all the waste water through the heat exchanger and heat the incoming cold water for the dye and bleach operations.

OPTIMISE THE HEAT EXCHANGER SELECTION

In considering an energy project that solves an operating problem, one often-encountered **error trap** is considering only the overall project payback period. It can be readily shown that HXs of various sizes can meet the investment criteria (for example, two year payback period (PB)) and yet be a very unwise investment. The overlooked element is the *incremental* payback of the last few square meters of heat exchanger surface. Table 1 illustrates this point, using a specialised shell-and-tube heat exchanger designed to minimise fouling. The stream flows and temperatures and other input assumptions are shown. Starting with a 297.5 m² HX, incremental additions of 10% are made, and the results recalculated.

The HX cost assumption used for this example was a fixed amount plus an additional US\$ per m² of HX area. The HX effectiveness measures the calculated heat recovery divided by the maximum (Q_{max}). The Incremental Payback is the added cost of the last 10% area divided by the increase in annual savings due to its addition.

Note that while the project PB is excellent (0.6 yrs) for a 848.5 m² HX, the more proper selection would be about 541.1 m², where the incremental PB is about 3 years. Varying any of the assumptions will vary the analysis outcome. The annual hours of use and value of the recovered heat have a marked effect.

OPTIMISE THE HEAT PUMP - HEAT EXCHANGER SELECTION

As **additional** HX does not meet the temperature reduction criteria of the waste stream, the IHP should now be considered. Adding a heat pump to the recovery process does not change the concept of incremental PB; it remains, and is an even more important factor due to the larger unit cost of the IHP in comparison to the HX. The analysis becomes somewhat more

Table 1 - Heat Exchanger Optimisation - Water-to-Water.

INPUTS								
	Fluid #1			Fluid #2				
Inlet temperature	43.3°C			15.6°C				
Flow rate	31.5 l/s			15.75 l/s				
Annual hours of HX use:				6,000 hr/yr				
Value of recovered heat:				US\$ 3.79/GJ				
				Estimated "U" = 681.6 W/m².°C				
				Minimum Hx area = 297.5m²				
				HX Cost = US\$ 86.4/m² + US\$ 20,000				
ANALYSIS								
	Flow Rate:			31.50 kg/s	15.75 kg/s			
	Q Max = 6.594 Gigajoules (GJ)/hr =1.832 kW							
HX Area sq m	Fl. 1 out °C	kW recovery	Total HX cost US\$	Annual savings US\$	PB yrs	Incr PB	Fl. 2 out °C	HX eff
297.5	31.1	1,611	45,613	131,925	0.3	----	40.0	0.88
327.2	30.8	1,645	48,174	134,762	0.4	0.9	40.5	0.90
359.9	30.6	1,677	50,991	137,332	0.4	1.1	41.0	0.92
395.8	30.4	1,705	54,090	139,630	0.4	1.3	41.1	0.93
435.4	30.2	1,729	57,499	141,655	0.4	1.7	41.8	0.94
479.0	30.1	1,751	61,249	143,410	0.4	2.1	42.1	0.96
526.9	29.9	1,769	65,374	144,904	0.5	2.8	42.4	0.97
579.6	29.8	1,784	69,912	146,150	0.5	3.6	42.6	0.97
637.6	29.7	1,797	74,903	147,165	0.5	4.9	42.8	0.98
701.3	29.6	1,806	80,393	147,971	0.5	6.8	42.9	0.99
771.4	29.5	1,814	86,432	148,594	0.6	9.7	43.1	0.99
848.5	29.5	1,820	93,076	149,059	0.6	14.3	43.2	0.99
Approximate Heat Exchange Solution at Desired Increment								
526.9	29.9	1,769	US\$ 65,374	US\$ 144,904	0.5	2.8	42.4	0.97
541.1*	29.9	1,774	US\$ 66,604	US\$ 145,278	0.5	3.0	42.5	0.97
579.6	29.8	1,784	US\$ 69,912	US\$ 146,150	0.5	3.6	42.6	0.97
Determination of Streams Entering IHP								
Stream being cooled: 2				Temp: 29.88 °C 15.75 l/s				
Stream being heated: 1				Temp: 42.46 °C 31.50 l/s				
* Interpolation of the two selections bracketing the input payback (PB) point.								

complicated with the addition of the IHP to the flow stream.

Heat Exchanger Assumptions:

These are the same as presented in Table 1.

IHP Assumptions:

Performance and cost assumptions for the IHP are also required. With Closed Cycle IHP designs available in the US, a "Heat Pump Efficiency" of 60% of Carnot is achievable, based on an 4.44°C (8°F) design small temperature difference in the heat pump evaporator and condenser. The installed cost estimate is made on the basis of a fixed amount plus an additional US\$ per kW heating capacity.

General Assumptions:

These include annual operating hours, value of heat recovered, the incremental cost of power to operate the IHP, and the selected PB criteria (typically three years or less).

An important economic factor affecting the PB is the relative cost of electric power for the IHP and the value of the heat recovered. The latter is typically measured by the boiler unit fuel cost divided by the incremental conversion efficiency.

Counter-intuitively, this incremental efficiency is typically about 80 to 85% in conventional

boilers, as the boiler efficiency declines as they are unloaded from the normal operating point.

The "Incremental Cost of Power" for the IHP is usually taken as the unit energy cost only, in US\$/kWh. This cost typically does not include a demand (US\$/kW-month) charge, because the IHP is typically controlled to not set a new demand for the plant. The IHP is shut-off for the few high-demand hours a day and either the boiler or thermal storage is used to supply the process during that time.

Table 2 illustrates the **Inputs** and the **Output** of a combined analysis of the HX and the IHP for this

example. The Break Even COP is calculated to be that COP where the cost of electric power equals the Value of Heat Recovered on a US\$ per kJ basis, and is for information purposes only.

Note the discharge stream temperature is now below the 26.7°C limit and the total project pays back in less than 3 years.

The analysis procedures to determine the output values are similar to those used for the HX

only. If the final output hot water temperature is below that needed by the process, boiler steam is used to do the final heating using a steam-to-water HX located following the IHP condenser. This final HX is usually sized to do the entire heating load, and is used for this should the HX and/or IHP be out for service.

Such a system must be carefully controlled to assure that the boiler does not "steal" the load from the IHP.

CONCLUSION

The wider acceptance of the IHP, the IHP and HX, and the IHP, HX and thermal storage will be gained by increased exposure of plant owners, operators and engineers to the option of solving environmental problems in this manner. Then projects can be defined to meet the financial criteria of the user, using techniques similar to the incremental payback concept illustrated in this discussion.

Table 2: Heat Pump - Heat Exchanger Optimisation.

Table 2: Heat Pump - Heat Exchanger Optimisation.			
INPUTS			
Stream flow and temperature data			
Inlet flow (stream 1):	31.5 l/s	Inlet temp:	43.3°C
Inlet flow (stream 2):	15.75 l/s	Inlet temp:	15.6°C
Heat exchanger			
Estimated "U" value:	681.6 W/m ² °C		
HX Cost:	US \$86.40 /m ²		
Fixed cost:	US\$ 20,000		
Heat pump			
Heat pump efficiency:	60.0%		
Heat pump variable cost:	US\$ 136 /kW		
Heat pump fixed cost:	US\$ 50,000		
General			
Value of heat recovered:	US\$ 3.79 /GJ		
Annual hours operation:	6,000 hrs/yr		
Incremental power cost:	US\$ 0.0250 /kWh		
Design max. incremental payback:	3.0 years		
ANALYSIS OUTPUT SUMMARY			
Size of HX:	541.1 m ²		
Heat recovered by HX:	1,774 kW		
Stream 1 outlet temp:	29.9°C		
Stream 2 outlet temp:	42.5°C		
Installed cost:	US\$ 66,604		
Annual savings:	US\$145,278		
Total payback:	0.46 yrs		
Heat recovery heat pump results:			
Break even COP:	1.83	Actual COP:	5.60
Power used:	117.7 kW	IHP size:	659 kW
Heat produced by IHP:	659 kW	Installed cost:	US\$ 140,000
Stream 1 outlet temp:	25.8°C	Annual savings:	US\$ 36,352
Stream 2 outlet temp:	52.5°C	IHP payback:	3.85 yrs
Combined heat exchanger & heat pump performance summary:			
Total installed cost:	US\$ 206,603	Total payback:	1.14 yrs
Total annual savings:	US\$ 181,630	Heat saved:	2,433 kW

Reference:

Gilbert, J. S. and Niess, R. C. 1989. "Marketing the Industrial Heat Pump," Edison Electric Institute Pub. No. 07-89-21, Washington, D.C.

NOTE: This paper was written using English units and then converted to SI units. Some calculations may not be exactly equal due to rounding errors introduced in the conversion. Conversion units used:

gpm	* 0.0631	= l/sec
sq ft	* 0.0929	= m ²
US\$/ton	* 0.284	= US\$/kW
million Btu	* 1.055	= GJ
Tons	* 3.515	= kW
million Btu/h	* 293	= kW

*Authors: Joel S. Gilbert and Richard C. Niess, Gilbert & Associates, Atlanta, GA and Gloucester Point, VA, USA.

Energy Saving with Heat Pumps in a Dry Powder Metal Factory

*O. Oberhumer, H. Halozan

INTRODUCTION

Heat pump applications in industry usually imply high temperature heat pumps integrated into processes. But other applications exist, e.g., that of cooling industrial processes combined with heat utilisation. The majority of heat pumps installed in industry are of the second type. These types utilise both the cold side as well as the hot side of the heat pump.

COOLING OF INDUSTRIAL PROCESSES

In the case where a below-ambient temperature level is required, the common method used in industry is to use ground water and cooling reservoirs. But this method has some serious disadvantages:

- The availability of ground water is not unlimited, i.e., in the case

of a plant expansion, ground water can be a limiting factor. Licensing of the ground water extraction rate by the authority is required.

- The utilisation of ground water in open cooling water circuits leads either to calcination and corrosion of the cooling devices or to the necessity of water treatment. If calcination and corrosion does occur, production interruptions may result, which means a reduction in profits. On the other hand, water treatment involves investment and operating costs.
- Finally, heat which could otherwise be used within the factory is wasted into the environment.

The only possibility of avoiding all these disadvantages is to change

from an open cooling water circuit to a closed system, using a heat pump.

A representative schematic of the principal flow is shown in Figure 1. A heat pump unit is connected to two storage tanks, the evaporator to a cold store and the condenser to a hot store. The cold store supplies cold water for process cooling and condenser heat in the hot store can be utilised for heating purposes and service water production.

The operation of the heat pump is controlled by the cooling demand required. Therefore, the production of heat in the condenser will not always match the demand for heating; the excess heat has to be dissipated by means of cooling towers, and a lack of heat has to be covered by an additional boiler.

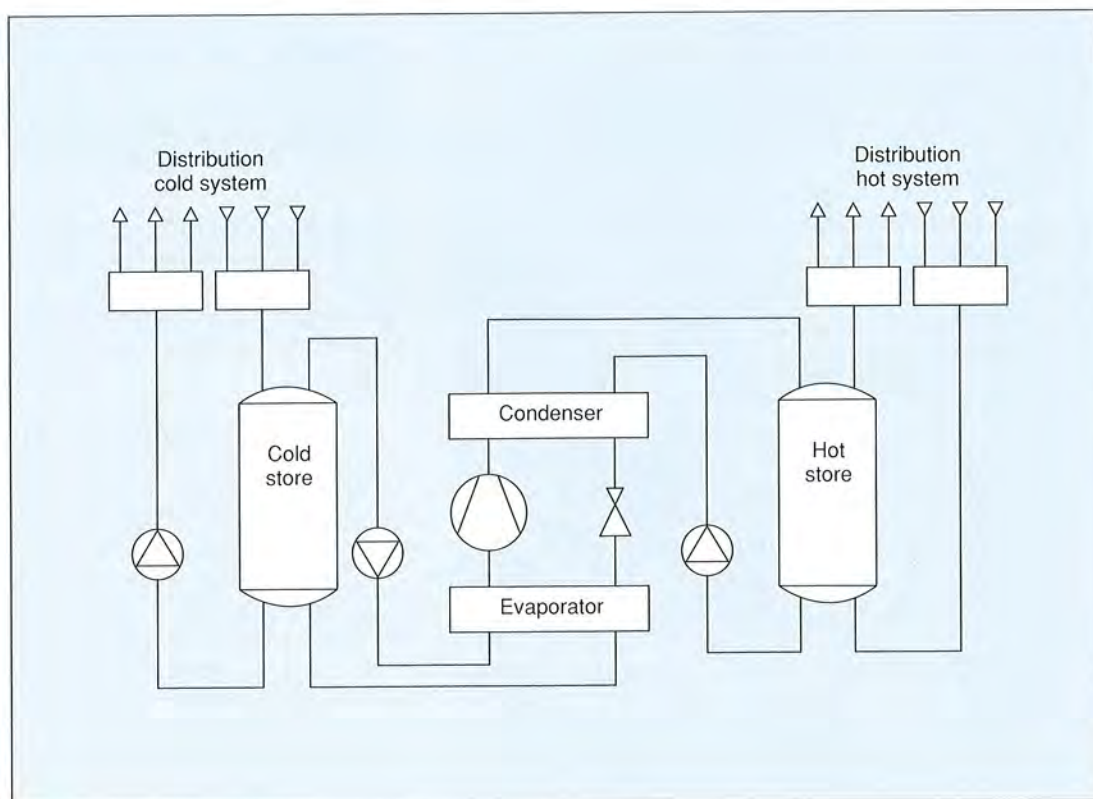


Figure 1: Flow Diagram of a Heat Pump Cooling System.



Photograph 1: Heat Pump Unit with Three Refrigerant Circuits.

A small disadvantage of this simple arrangement is the temperature level in the hot store: it must always cover the requirements of the consuming device with the highest temperature level, even when an excess amount of heat is available. For this reason, more sophisticated flow sheets are used in larger systems.

HEAT PUMP INSTALLATION IN A DRY POWDER METAL FACTORY

A sophisticated type of installation was implemented at a highly specialised supplier of the car industry. Its specialty is, apart from other products, in the field of powder metallurgy. In view of an envisaged expansion of the production capacity, by increasing the number of sintering furnaces, the decision was made to change from an open cooling circuit to a closed cooling cycle with a heat pump. The existing cooling system which worked by means of ground water and cooling reservoirs no longer sufficed, and could not attain the specified increase in cooling capacity to 530 kW.

Apart from the economic aspects, utilisation of the condenser heat and improved production capability with the closed cooling circuit, the decision was also taken with respect to environmental considerations, covering both the saving of ground water and the

reduction of environmental pollution due to flue gas emissions.

DESCRIPTION OF THE PLANT

The main components of the new cooling system are the two heat pump units (see Photograph 1). Each unit has three separated refrigerant circuits consisting of a reciprocating compressor, two condensers arranged (on the refrigerant side) in series and one evaporator (see Figure 2).

The design for integrating these units into the factory's cooling and heating system aimed at achieving a high reliability rate, as well as low operating costs. The three evaporators of one unit are hydraulically arranged in parallel and placed in one shell; they are connected to the cold store which provides cooling water for the sintering furnaces.

The first condensers (CO1/1, 2/1 and 3/1) in each refrigerant circuit are hydraulically arranged in series and connected to the hot store. This store provides hot water for heating the office, production shops, laboratories as well as for the preparation of service water. The arrangement in series was chosen in order to decrease the overall condensing temperatures as heating the hot water occurs in three steps. Only the last compressor, i.e., the compressor connected to evaporator 1, has to

be operated with a condensing temperature corresponding to the hot water outlet temperature required; the condensing temperature of the other compressors is lower. This results in an increase of the COP.

The second condensers (CO1/2, 2/2 and 3/2) in each refrigerant circuit are hydraulically arranged in parallel and connected to a dry-type air cooler system. The condenser outlet temperature varies between 40°C and 25°C depending on the load and the outside air temperature. The minimum condenser outlet temperature of 25°C which is required for condenser operation is maintained by means of a three-way mixing valve. The heat transfer medium used in the cooling tower system is an anti-freeze solution.

OPERATION OF THE PLANT

The heat pump operation is principally controlled by the cooling requirement of the sintering furnaces. Cooling water with a temperature of 30°C is taken from the cold store and cooled down to a temperature of 15°C. The utilisation of the condensing heat depends on the heat load of the consumers connected to the hot store. In principle, there are two operating modes:

- during the heating season the condensing heat of the first

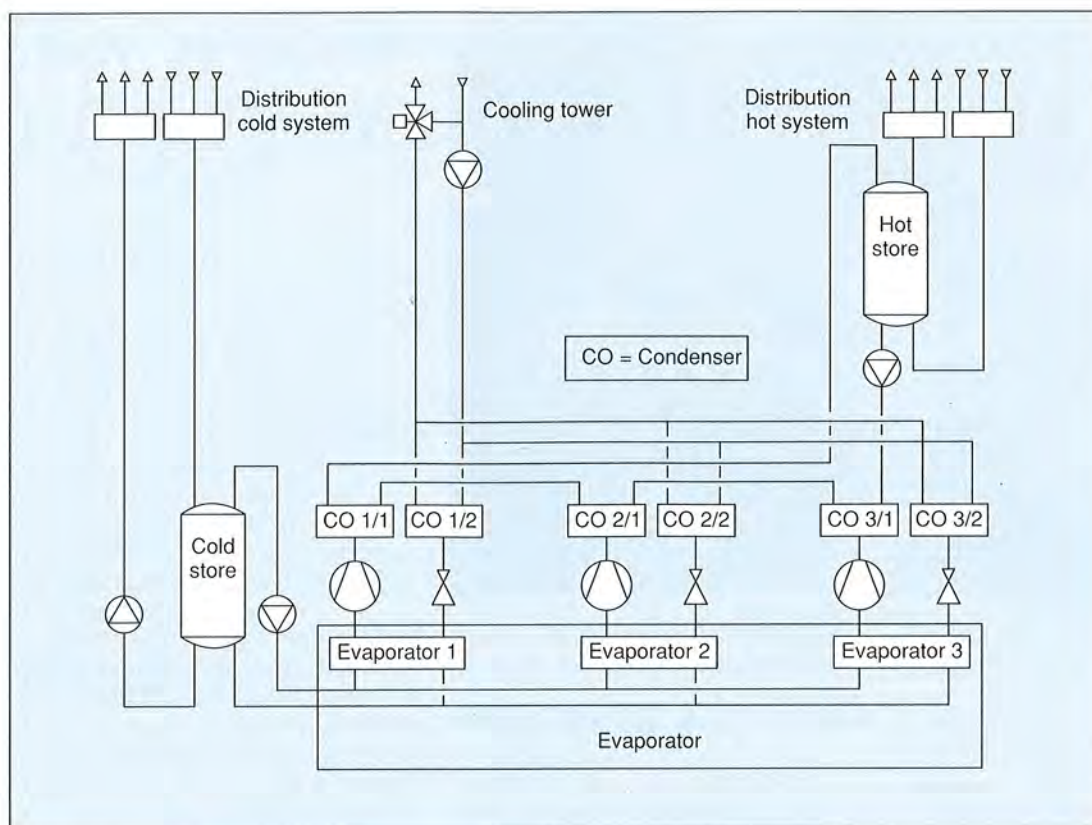


Figure 2: Flow Diagram of the Dry Powder Metal Factory Cooling/Heat Recovery System.

condensers (CO1/1, 1/2, 1/3) in each refrigerant circuit is transferred to the hot store. The temperature in the hot water tank floats, up to a maximum of 70°C. The total heating demand of the factory can be covered by the heat pump even if the ambient temperature is -4°C.

- at lower ambient temperatures, and during production shut-down at weekends, the heating demand is covered by conventional gas-fired boilers which are switched on automatically. If there is no heat requirement, the condensing heat of the second condensers in each refrigerant circuit is dissipated via dry-type air coolers.

Between these two operation modes - maximum heat demand and no heat demand - operation of the system can be varied in small steps. The electronic control system always ensures efficient operation.

RUNNING COSTS

A high reliability is achieved due to the six separate cooling circuits,

the six identical compressors, the hydraulic separation between the heating circuit (water) and the cooling tower circuit (anti-freeze solution) by two separate groups of condensers. Even so, economic advantages, with respect to the power supply costs, are still achievable. At the same annual cooling output, the generously dimensioned cooling tower plant, in conjunction with a very low minimum anti-freeze fluid temperature of 25°C, result in a reduction of the annual power supply costs by about 180,000 Austrian shillings (US\$ 14,130). To this must be added a lower refrigerant pressure at the compressor, which in turn, results in a longer service period (due to reduced wear), and a longer operating life. The higher cooling effect, resulting from the low temperatures in the cooling tower circuit also implies a higher output yield at reduced operating hours, resulting once again in a longer service period. The series connection on the water side of the condensers for heating purposes makes it possible to achieve an improved coefficient of performance at a water temperature of

maximum 70°C, ensuring the optimum energy utilisation in the heating system. As a result, the annual heating requirements are covered by more than 80%. The volume of natural gas substituted in 1000 operating hours amounts to roughly 114,000 m³.

CONCLUSIONS

The design engineers have proved convincingly that the concept of using a heat pump is attractive, not only from the point of view of conserving ground water and reducing emissions, but also from an economic point of view.

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BREHM, 1988) 12 samples of five different materials were exposed to ground water in a 50 m deep well. In this article the mass loss due to corrosion, as well as the chemistry of the ground water, are presented.

The investigation shows that pure iron, when in contact with ground water, will lose half of its mass in approx. 30 years. Only Inox steel or plastic (e.g., High-Density-Polyethylene, HDPE) are durable enough to justify being used as a ground heat exchanger.

SITE DESCRIPTION

The test site is located on a plateau in the Rhenish Massif, about 290 m above sea level. The ground consists of devonian/carboniferous shales and greywackes (greywackes consist of sandstone with a substantial amount of clay and rock debris). Precipitation averages 720 mm/year, the ground water table varies between 12 and 15 m below ground level and the annual mean temperature is 7-9°C. Under these conditions, enrichment of ground water with specific ions (e.g., Cl^-) is not to be expected.

For heat extraction tests, 12 boreholes were drilled to a depth of 50 m each. Eleven of these holes had a diameter of 3.5" (89 mm) and one had a diameter of 5.25" (133 mm). In one of the 3.5" holes, samples of materials commonly used in earth heat exchange were installed at 30-40 m depth, well below the ground water table. The samples were in contact with the

Sodium	Na^+	11.2 mg/l	Chloride	Cl^-	13.0 mg/l
Potassium	K^+	0.8 mg/l	Sulphate	SO_4^{2-}	7.0 mg/l
Calcium	Ca^{2+}	17.7 mg/l	Hydr. Carbon	HCO_3^-	61.0 mg/l
Magnesium	Mg^+	3.3 mg/l	Nitrate	NO_3^-	27.0 mg/l
Ammonium	NH_4^+	0.6 mg/l	Phosphate	PO_4^{3-}	0.3 mg/l

ground water in situ for approximately two years. The 5.25" hole was used as an open well for sampling ground water.

GROUND WATER CHEMISTRY

The Schwalbach ground water is a typical, low-salinity ground water of the Rhenish Massif (SANNER et al., 1986). Ca^{2+} and HCO_3^- are predominant, followed by Na^+ , Mg^{2+} and Cl^- . The concentration levels of K^+ , NO_3^- , SO_4^{2-} , and PO_4^{3-} vary due to the use of fertilizers and to the agricultural character of the region. The result of a chemical analysis at the beginning of the exposure is given in Table 1.

Between June 1985 and August 1986, before exposure of the material samples, ground water samples were analysed at regular intervals. The evolution is shown in Figure 1; while Na^+ remained between narrow limits, SO_4^{2-} and NO_3^- varied widely as did Cl^- . For Cl^- the

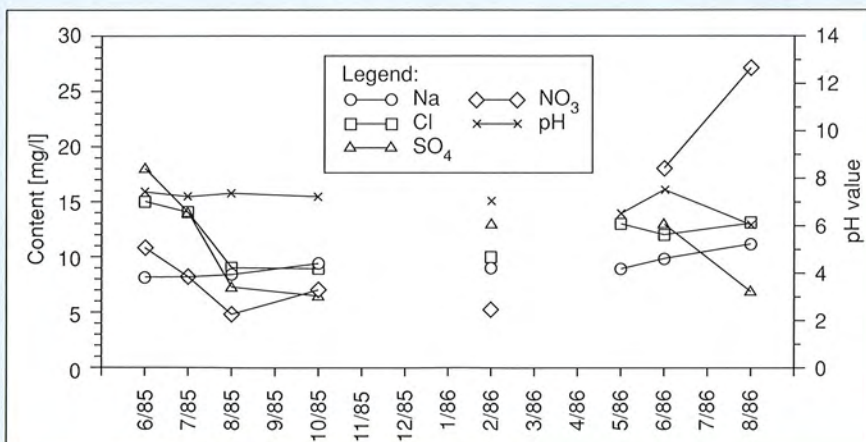


Figure 1: Analysis of Ground Samples between 1985 and August 1986.

Photograph 1: Comparison of Steel Tube and Copper Tube Samples Exposed to Ground Water (Left) and Stored in a Dry Place in the Laboratory for the Same Period of Time (Right).

average value was 11.9 mg/l with 16 mg/l maximum and 8 mg/l minimum; for SO_4^{2-} the average value was 11.3 mg/l. During the exposure period (1986-88) no ground water samples could be taken, but an extrapolation of the values from 1985/86 is supposedly possible, giving rise only to minor errors.

MATERIAL SAMPLES

Twelve samples of five different materials were attached to a nylon string and placed into the ground water: three types of iron and steel and two types of copper tube were used. HDPE used as the sole material, had not been applied widely in Germany when the tests began, hence no HDPE tube was included in the test. A comparison of the results for pure copper and Polyethylene (PE)-coated copper

allows an assessment of the corrosion resistivity for PE. HDPE is used for water supply lines in the ground as it is known to be corrosion resistant.

CORROSION AFTER 2 YEARS OF EXPOSURE TO GROUND WATER

The samples were weighed using a precision balance before exposure. After recovery, mud and loose rust were removed from the samples with paper napkins and a soft brush, and the samples were weighed again. The mass loss of the samples is given in Table 2.

The actual processes in ground water could neither be controlled nor monitored. Thus, the interpretation of the values may not be very accurate (e.g., high scattering for the three steel tube samples), but on

Table 2: Mass of Samples for In-situ Corrosion Test, Before and After Exposure.

Sample	Mass before exposure (grms)	Mass after exposure (grms)	Mass loss (absolute) (grms)	Mass loss (specific) (%)
Iron, test body	17.94	17.18	0.76	4.24
Iron, test body	17.83	17.02	0.81	4.54
Steel, tube 1 3/4"	126.77	121.41	5.36	4.23
Steel, tube 1 3/4"	127.47	119.89	7.58	5.95
Steel, tube 1 3/4"	124.76	121.41	3.36	2.69
Inox, corrug. tube 2"	28.06	28.06	0.0	0.0
Inox, corrug. tube 2"	28.37	28.37	0.0	0.0
Copper, tube 1"	27.26	26.38	0.88	3.23
Copper, tube 1"	27.02	26.02	1.00	3.70
Cu, corrug. tube 2", PE	51.30	50.31	0.99	1.93
Cu, corrug. tube 2", PE	51.15	50.18	0.97	1.90
Cu, corrug. tube 2", PE	52.76	51.72	1.04	1.97

Table 3: Summary of In-Situ Corrosion Test.

Material	Mass loss per year (average) %	Time for loss of half of the sample mass (years)
Iron, test body	2.2	32
Steel St37, tube	2.15	32
Inox steel, corrugated tube	0.0	—
Copper, tube	1.74	40
Copper, corrugated tube, PE-coated	0.97	72

average, a good picture of the corrosion of the different materials has been obtained.

While the Inox steel did not exhibit any visible corrosion, the iron and plain steel samples were obviously severely corroded, showing development of a loose layer of reddish iron hydroxides up to 1 mm thick. Corrosion on the copper samples could not be detected with the naked eye, but mass loss was nevertheless substantial. A summary of the test is shown in Table 3; the time for loss of half of the sample weight has been calculated by reference to the average mass loss per year. Photo 1 shows a comparison of corroded and uncorroded samples.

Pure iron, as well as simple steel with qualities such as St37, cannot be recommended for use in applications in ground water. During the two year test period, Inox steel proved it has good properties for long-term use in the ground, but this is very strongly dependent upon water chemistry, and in particular, on the content of chloride Cl^- . In the presence of chloride, very small corrosion nuclei can develop on the material surface and lead to pitting corrosion. In general, and regardless of the price, Inox steel is a good choice for ground heat applications in low-salinity ground water.

Pure copper is marginally better than iron in corrosion resistance. The corrugated tube was lined with PE from the outside, thus ground water could only make direct contact with copper from within the tube. About half of the surface was protected in this way, and the relative mass loss was slightly more than half the value for pure copper. The PE-liner appears to work well.

Corrosion of metals in ground water is very strongly dependent upon the chemical composition of the water. High or very low oxygen content, a low pH value, or increased content in NH_4^+ , Cl^- , SO_4^{2-} , S^{2-} foster corrosion. Therefore, ground water chemistry should be investigated carefully before installation to ensure the correct material choice for long-term resistance.

CONCLUSION

In general, three materials can be recommended for use in the ground: Inox steel (in ground water with "normal" chemistry); any metal protected from ground water and ground contact by PE-coating; and HDPE used as the sole material for all tubing.

Acknowledgements:

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News and Views

Market News

First Japanese/European A/C Joint Venture on the Continent

Sanyo Electric Co., Ltd., ELFI S.P.A., and Sumitomo & Co., Ltd. have announced plans to form a joint venture company to produce air-conditioners for the European Market. Sanyo Electric Co., Ltd. of Japan produces over 1.4 million A/C units worldwide annually and has nine overseas production points. ELFI S.P.A. is a major Italian manufacturing group. It's major business sectors include commercial refrigeration, heating and A/C, and home appliances. Sumitomo & Co., Ltd. is a leading Japanese general trading company. The new joint venture company, Sanyo Argo Clima S.P.A., will be located in Gallarate, Varese, Italy. Of the capital of 2.3 billion lire (approx US\$ 1.8 million) ELFI S.P.A. will contribute 51%, Sanyo 39% and Sumitomo 10%. Production is scheduled to begin in late 1991 with an estimated initial production volume of 100,000 units per year. The new company has been established in response to the increasing demand for A/C equipment in the European market.

(Source: JARN, April 25, 1991.)

Market Trends

Shipments Japan

The shipment figures for the first half year (Oct.'90 - Mar.'91) and for the second quarter (Jan.'91 - Mar.'91) of Ref. year 91 are shown in Table 1.

Both for room and packaged A/Cs, the heat pump share in the first half of ref. year '91 was 87%. Room A/Cs are small household and window types; packaged A/Cs include all other types. The exports for the period Oct.'90 - Mar.'91

Table 1: Units of A/C and Heat Pump Domestic Shipments in Japan and Percentage Increase Relative to Same Period of Ref. Year 1990.

Item	Jan'91-Mar'91		Oct.'90-Mar'91	
	Number	Increase %	Number	Increase %
Room A/C	1,521,790	25.0	2,464,717	36.4
Cooling only	301,407	- 2.3	326,288	2.3
Heat pump	1,220,383	34.2	2,138,249	43.8
Packaged A/C	202,841	17.6	421,175	20.6
Cooling only	29,295	9.9	54,279	13.6
Heat pump	173,546	19.0	366,896	21.7

were for room A/Cs 670,598 units, an increase of 6.6 % and for commercial A/Cs 98,778 units an increase of 21.5 %.

(Source: JARN, May 25, 1991.)

Diffusion of Room A/Cs in 2000 in Japan

The Japan Refrigeration & Air-conditioning Industry Association (JRAIA) predicts that the diffusion of room A/Cs will increase from 69.5% in 1990 to 78% in 1995 and 82% in 2000. The average number of units per family will increase from 1.85 in 1990 to 2.25 in 1995 and 2.74 in 2000. Installation of housing A/C systems comprising 2 or 3 concealed indoor units against one outdoor unit is also rising. Along with the increasing floor space, the demand for multi-type A/Cs will also steadily rise.

(Source: Jarn, May 25, 1991.)

Shipments US

The shipments in the US of unitary A/C, including heat pumps for the period Jan.'91 - Apr. '91 were 1,169,106 units, which is 13% below the shipments in the same period of 1990.

(Source: A/C, Heating & Refrigeration NEWS, June 17, 1991.)

Honeywell Predicts Growth in US after 1991

According to a forecast by Honeywell, after a dip of 14% in 1990 and a decrease by another 11% in 1991, shipments of unitary products will gradually rise to a near record level of 4.2 million units by 1996 plus exports of 400,000. In addition to a moderate growth in housing, the biggest push will come from the replacement-modernisations sector. Honeywell sees the electric heat pump share of unitary products growing from 800,000 units to 1 million by 1994, and staying at that level. (Remark HPC: In 1990 total heat pump shipments were 808,665 air/air type and 117,900 water(ground)source type, see Newsletter Vol. 9 No.2) Saturation of central A/C in new single family homes will hit 80% in five years, four percentage points higher than the present level.

(Source: A/C, Heating & Refrigeration NEWS, June 17, 1991.)

Refrigerants/CFCs

Production of CFC alternatives

Alternative for R-502

Rhone-Poulenc Chemicals, the English daughter of the French chemical group, has brought a direct replacement for R 502 on the market under the name of Isceon 69-S. The new refrigerant, having been developed jointly with the Scottish refrigeration firm, Star Refrigeration, is a mixture of R-22, propane, and octafluoropropane. It can be used as a drop-in fluid in existing refrigeration units and in the deep-cooling sector. The thermodynamic properties almost equal those of R-502. Isceon 69-S is being field tested in an English supermarket. Rhone Poulenc Chemicals manufactures Isceon 69-C in its plant in Avonmouth near Bristol. The sales price is estimated to be three times that of R-502.

(Source: Die Kälte und Klimatechnik, 4/91.)

Atochem (France) Launches Production of R-141b, 142b and 134a

At its production site near Lyon, Atochem is investing close to FF 1 billion (approx. US\$ 170 million) in two new units to produce 40,000 tonnes R-141b and 142b per year from the beginning of 1992, and 9,000 tonnes R-134a later in 1992. (Source: *Revue Generale du Froid*, May 1991.)

ICI to Increase Production of R-134a and Compatible Lubricants

In addition to its present production of R-134a (Klea 134a) in its Runcorn plant (UK) and an increase of the production from its Gabriel, La, (USA) plant by early 1993 to 20 million pounds (about 9070 tons) ICI will be building additional plants in the UK and Japan. Production of the compatible lubricants Emkarate RLE, an ester-based substance, and Emkarox RL, a polyalkylene glycol-based substance in ICI's Baleyourd plant in France now exceeds 2.5 million pounds. Ultimately ICI will be able to produce 12.5 million pounds. (Source: *A/C, Heating & Refrigeration NEWS*, June 24, 1991.)

Declining CFC Production in the US

In the US, CFC production in 1990 declined 30% to a total production volume of 61,200 kg for R-11; and 46% to 94,400 kg for R-12. The sales of HCFC R-22 declined 7% to a volume short of 140,000 kg. The decline is attributed to recycling and the massive price increase due to CFC taxes. (Source: *CCI*, May 6, 1991.)

R-32: A Likely Refrigerant Substitute

Methylene fluoride (R-32) and its mixtures are likely candidates as replacements for R-12, R-502 and R-22 in applications such as heat pumps, supermarket units, and home appliances. The benefits of R-32 were acclaimed at a conference on March 19 and 20, 1991 in Washington. The conference was sponsored by the Environmental Protection Agency Global Change Division in co-operation with the US Department of Energy and the German Ministry of the Environment. Research indicates the positive environmental features of R-32, such as low stratospheric ozone and atmospheric warming impacts. Presentations featured modelling results with R-32; the history and environmental characteristics; research needs to analyse and qualify R-32; and strategies for government-private sector co-operation. (Source: *Air-Conditioning, Heating & Refrigeration NEWS*, April 22, 1991.)

Cost for France to Abandon CFCs

Table 2 shows the results of a cost estimate by the French Association for Cold (AFF) for phasing out CFCs in the A/C field for two cases, i.e. assuming that R-22 is used as a substitute and, alternatively, assuming that R-22 must also be considered as a refrigerant to be phased out. Adding refrigeration and mobile A/C, the total costs are 45.64 and 73.38 billion FF respectively. It is concluded that an accelerated replacement of CFCs makes sense only if R-22 is assured of a reasonable future. All other choices lead to increased costs and a concentration of the period of adaption into a shorter timespan, with, as a result, a much higher inflationary effect, without yet taking into account

the impact on energy use or the actual availability of alternatives.

(Source: *Revue Generale de Froid*, May, 1991.)

Table 2: Cost of Phasing Out CFCs Excluding and Including R-22 phase out (note: FF 1 is approx. US\$ 0.17).

	Indiv. A/C	Comm. A/C	Build. A/C
Number of units	1,000,000	100,000	10,000
Charge per unit (kg)	2	10	500
Total charge (kg)			
R-11			500,000
R-12		500,000	1,000,000
R-22	2,000,000	500,000	3,500,000
R-500			10,000
Cost per unit in FF			
• replacement	10,000		
• modification		40,000	1,000,000
Total costs in billion FF			
• excluding R-22 phase-out		2	3
• including R-22 phase-out	10	4	10

Systems/Installations

Ground-source Heat Pump News

Canada

In Brampton, Ontario, a system using 88 ground-source heat pumps (GSHPs) is being installed to provide heating and cooling for the staff and more than 1,500 students of the new 165,000 ft² (appr. 15,000 m²) St. Augustine school. The manufacturer is Florida Heat Pump, Fort Lauderdale, Florida, USA. The GHSPs use closed loops. Ontario Hydro, the utility, announced cash incentives of Can \$ 600 per ton of cooling. St. Augustine school will qualify for approx. Can \$ 120,000, and the payback period will be cut to 3 1/2 years. The project is claimed to rank as one of the largest GHSP installations in Canadian schools. For more information contact: J.A.T. Ventilation Inc. Mississauga, Ontario, Canada; Tel.: +1-416-890-2004. (Source: *The Air-Conditioning & Refrigeration NEWS*, April 22, 1991.)

USA

The largest ground-source heat pump system in the state of Oklahoma began operation at the state capitol Oklahoma City in February, 1991. Presently 277 Climate Master heat pumps, ranging in size from 0.5 to 23.5 tons, have been installed, serving 172,535 sq ft (approx. 16,000 m²) from

the basement through to the third floor. 90 of the building's 138 existing high maintenance HVAC systems were replaced.

The project is estimated to save US \$ 160,000 a year in energy costs and sizable savings in maintenance costs are expected as well. Project costs are approx. US\$ 3,025,000. Conversion of the fourth and fifth floor to the system is pending legislative approval of US\$ 1.7 million in additional funds. When serving the entire 398,548 ft² (37,000 m²) building, energy cost savings are projected at US \$ 200,000 a year. For more information, contact: Ray Roberts, State of Oklahoma Office of Public Affairs; Tel.: +1-405-521-3395.
(Source: EPRI Heat Pump News Exchange, summer 1991.)

Water Loop Heat Pump (WLHP) News

Pilot Survey Shows Long Service Life

A WLHP system consists of: a water pipe loop; heat pumps, one for each thermal zone, which use the water in the piping as a heat source or sink; and some means of rejecting heat from and adding heat to the water, usually a cooling tower and a boiler. One example indicates how a WLHP system operates: in the winter heat pump units on the perimeter of a building will be heating, extracting heat from the water loop, while units in the core of the building will be cooling, adding heat to the loop.

The Electric Power Research Institute (EPRI) sponsored a pilot survey of service records of WLHPs in commercial office buildings, using large numbers of small capacity (0.75-ton to 1-ton) perimeter WLHPs and smaller numbers of 7.5-ton to 20-ton core WLHPs. The records showed that very few of the heat pumps in the sample had been replaced in their entirety. The researchers went on to examine the service records of the unit's compressors, as a key component, costly to replace. Of the original core-unit compressors, 73% were still operating nine years after being installed.

Extrapolation techniques applied yielded an estimated median service life (age at which 50% of the original units have been replaced and 50% remain in service) of 12 years for the core-unit compressors. Of the original perimeter unit compressors 94% were still operating, yielding an estimated median service life of 47 years. Explaining the difference requires further investigation. For the WLHP population covered in the pilot study the median service life of complete units is clearly much longer than the 19 years reported for water-source HP units in the ASHRAE Systems and Applications Handbook (1987).

It is concluded that the median service life of WLHPs is as long as, or longer than, those of competitive options.
(Source: EPRI Journal April/May, 1991 pp. 50-52.)

Field Test of WLHP

Results of a three year field test of a WLHP system in a typical commercial building (72,000 sq ft = approx. 6,700 m²) indicate that WLHPs can provide efficient space heating and cooling with low installed costs. However, care must be given to system design and operating strategy. The system consisted of 140 one-ton console-type perimeter heat

pumps, 12 vertical dual-compressor core heat pumps, 11 special task heat pumps, a 300 kW electric boiler and a closed-loop cooling tower.

Findings included:

- More accurate controls for electric boilers which dominate demand in colder climates would help limit demand and excess energy use. Control of boiler stages during the morning warm up is the key to managing building demand. Gas-fired boilers can provide an economic option in some cases.
- Closed-loop cooling towers that cannot be isolated from the main loop cause large thermal losses in colder weather and can severely degrade building thermal performance. If these cannot be isolated, inclusion of an isolation heat exchanger is a leading choice and may be cost-effective as a retrofit.
- In cold weather, excessive outside air to the building core can cause high heating use of core heat pumps and thermal drain on the loop. An appropriate level of outdoor air can be attained that delivers sufficient fresh air to the building while retaining core cooling that permits heat recovery.
- Staggered start-up of the heat pumps during the morning warm up can minimise electricity demand during this period.
- Buildings in colder climates should not have open-to-air ambient parking below a conditioned floor, and if unavoidable, the interface must be heavily insulated.

For more information contact: Mukesh Khattar, EPRI Commercial Program, Tel.: +1-415-855-2699.
(Source: EPRI Heat Pump News Exchange, summer 1991.)

Codes and Guidelines

Joint US/Canada Safety Standard

A bi-national safety standard covering heating, ventilation, A/C and refrigeration equipment for commercial and light industrial applications has been published. The new standard replaces six Canadian and six US standards and allows these appliances to be sold in either country. The standard becomes effective on new and revised products on January 2, 1993, and for all A/C and refrigeration products on January 1, 1998. Manufacturers can submit products for certification to the Canadian Standards Association (CSA) or Underwriters Laboratories (UL). The standard is published in the US as UL 1995 and in Canada as CAN/CSA-C22.2, No. 236-M90 (Heating and Cooling Equipment).
(Source: ASHRAE Journal, April 1991.)

Meetings - Reports

Conference "Cogeneration Power Plants and Heat Pumps"

This conference, organised by the Association of German Engineers - Society for Energy Technology (VDI-GET) was held in Essen (Germany) on June 4 and 5, 1991. The following topics of particular interest to the heat pump community are excerpted from the proceedings (see also the Bibliography section):

HFC 227 a Possible Alternative to CFC 114.

Starting with the requirement that the alternatives to the CFCs 12, -114 and -502 should be chlorine-free and non-flammable, E. Preisegger of Hoechst AG (Germany), notes that the choice is limited. As a replacement for CFC-12, only HFC-134a will be available in sufficient quantities in the coming years. CFC-502 will be replaced by HCFC-22, but, because of its chlorine content this is considered to be a temporary solution. For condensing temperatures up to 90°C, a possible alternative to CFC-114 would be HFC-227 (CF₃-CHF-CF₃). As is the case for HFC-134a, the Ozone Depletion Potential (ODP) is zero. The Atmospheric life of HFC-227 is approx. 30 years (15 years for HFC-134a) and the Greenhouse Warming Potential (GWP) is approx. 0.7 (0.3 for HFC-134a). The research results have, up till now, indicated no toxicological effects of HFC-227. During corrosion tests, no material problems have occurred. A tested ester oil showed itself to be completely miscible with HFC-227 in the temperature range of -40 to 80°C. (Source: Paper E. Preisegger, Hoechst AG, Germany; paper in German, graphs in English.)

Status of Gas Engine-driven Heat Pumps in Japan

In Japan, small gas engine heat pumps (GHPs) were first marketed in 1987. Sales have steadily increased since to approx. 25,000 units in 1990. The market share in the Tokyo Gas area is approx. 3%. Currently, there are 17 models, ranging in capacity from 1.3 RT to 23.5 RT. The main applications of GHPs sold by Tokyo Gas up to the end of 1990 are: offices (43.4 %); schools (22.0%) and restaurants (8.4%).

Short-term issues are creating a greater assortment of products, including, in particular, direct expansion multi-type systems for mid-size commercial buildings (<10,000 m²); and reducing investment costs. At present a GHP is 20 to 30% more expensive than a comparable electric heat pump. Long-term issues are:

- greater compactness to bring the size down to, or at least close to, that of electric heat pumps;
- reduction of maintenance: present GHPs require an annual inspection (each 2,000 hours); it is necessary to lengthen the interval between inspections and the number of items to be inspected;
- reduction of NO_x emissions.

(Source: Paper H. Kobayashi, Tokyo Gas CO. Ltd., in English.)

Standards for Gas Engine NO_x Emission

Starting from February 1994, the NO_x emissions from gas engines used for heat pumps and cogeneration must be lower or equal to 600 ppm (at 0% O₂). For major urban regions more stringent standards apply. For instance, from April, 1992 standards for Tokyo are 300 ppm for gas engines with a gas consumption equivalent to less than 50 l/h heavy oil, and 200 ppm for larger gas engines, both levels converted to 0% O₂.

(Source: Paper K. Hattori, Japan Gas Association, in English.)

IEA and Member Countries

Purchasing Incentives for Heat Pumps in Italy

The Italian Government has promulgated two laws concerning the norms to accomplish the new Italian Energy Plan. One law (No. 9, January, 1991) regards institutional aspects, electric plants, hydrocarbons, geothermics and fiscal regulations. The other (No. 10, January, 1991) concerns the rational use of energy, energy saving, and development of renewable energy sources. The latter law grants a contribution between 20 and 40% of the costs for an energy saving system. In the case of heat pump installations, the grant applies to hot water or ambient air heating systems, being able to cover at least 30% of the heat requirement. Industrial heat pump applications are also considered, though with lower contributions than in the residential/commercial sector.

A brochure on heat pumps will be published, this year, by ENEA, the Italian Commission for Nuclear and Alternative Energy Sources. The brochure explains what a heat pump is, how it works, its applications, how to select and install it, and how to evaluate its economics. The brochure, to be printed in approx. 100,000 copies, aims at making the heat pump system known to the general public in order to diffuse its use for energy saving and environmental protection purposes.

(Source: Italian National Team.)

Start of a New National Project in Japan

The Japanese national Super Heat Pump System project is now in the last stage of its nine years course and will be completed in fiscal year 1992. This huge project, with 100% government financing of about 10 billion yen (approximately US\$ 70 million) is expected to mark a new era in heat pump history. As a follow-up, the Japanese government has started a new R&D project this year concerning the "Utilisation of Unused Energy". The project will cost about 15 billion yen (US\$ 110 million) over a seven year period. Of this cost, 50% will be subsidised by the government; that is 7.5 billion yen will be appropriated to the Heat Pump Technology Center of Japan (HPTCJ). The member companies of the HPTCJ, interested in this project, are to share in all the work involved, following their own interests and specialities.

At present, about twenty companies are expected to participate. The HPTCJ will co-ordinate the project. Unused energy means waste heat from artificial sources such as incinerators, and natural sources such as river water. District heating and cooling is considered to be the main application area. See also the article on pp. 4-7 in this Newsletter.

(Source: HPTCJ.)

New Annex

A new Annex is being discussed, titled "Global Environmental Benefits of Industrial Heat Pumps". The purpose is to develop a range of estimates on the impact that industrial heat pumps could have on reducing greenhouse gas emissions along with the associated capital costs and energy savings. An expert meeting to formulate the proposal to be submitted to the Executive Committee was held on June 18, 1991, at the offices of the HPC. It was attended by experts from the USA, Sweden, Netherlands, Germany, Norway, and Belgium. Other countries have also expressed interest. For more information contact the proposed operating agent: US DOE, Paul E. Scheihing, Programme Manager, Office of Industrial Technologies, CE-211, 1000 Independence Avenue, S.W. Washington DC 20585, USA; Tel.: +1-202-586-7234, or the HPC.

It's "Heat Pumping or Drowning" in the Netherlands

On May 22 and 23, 1991, the Netherlands National Team, together with Novem, organised a workshop with the above

title. The title reflected the philosophy that without application of heat pumps no significant reduction of greenhouse gas emissions will be achieved in a society confronted with a constant growth of its production. Below sea-level countries, such as the Netherlands, might therefore be flooded because of the predicted rise in sea level.

The goal of the workshop aimed at having a searching discussion with all market parties involved. A complete retreat out of the field of heat pumps, was taken into account as a possible outcome of the discussions due to the poor results of demonstration projects. The results should lay out the course for measures to be taken by the Ministry of Economic Affairs, in the short-term, in order to achieve a maximum penetration of heat pumps in the year 2000.

The first day was devoted to setting out the goals for the industrial sector, the second day for the building environment. The response was high; 80 out of 120 invited experts attended the workshop: consultants, equipment manufacturers, installers, users, investors, and research technologists. Following presentations, questions were discussed such as: main options for heat pumps, pitfalls, impediments to overcome, competitive options, process integration, measures to be expected from government. The initial evaluation led to the following conclusions:

- For the building environment, the long-term goal should be the use of electric heat pumps in standard low temperature distribution systems, in combination with high efficient electric power generation or combined heat

Summary of Technical Annexes per April 1991

Annex	XII	XIII	XV	XVI	XVII	XVIII	XIX	XX	Explanation of Annexes:
Austria	+		+	+		(+)		(+)	XII Modelling techniques for simulation and design of compression heat pumps.
Belgium	+							OA	
Canada		+	OA	+	+	+	(+)	(+)	
Denmark							(+)		
FRG	+	+				+	OA		
Finland									XIII State and transport properties of high temperature working fluids and non-azeotropic mixtures.
Italy	OA			+				(+)	
Japan	+	+	+	+		+	(+)	(+)	
Netherlands				OA	+	(+)	+	+	
Norway		+		+	+	(+)	(+)	(+)	
Sweden		OA		+	OA	+	(+)	(+)	XV Heat pump systems with direct expansion ground coils.
Switzerland	+				+		(+)	+	
UK						(+)			
USA	+	+	+	+		OA	(+)	(+)	
Status	cr	cr	cr	cr	cr	cr	p	cr	
Start	'86	'87	'89	'90	'91	'90	'91	'91	XVII Experiences with new refrigerants in evaporators.
Completion	'88	'89	'90			'92		'91	
Extended until	'91	'91	'91		'92				
* country is not involved in extension (+) = participation not yet confirmed cr = current p = proposed OA = Operating Agent									XVIII Thermophysical properties of the environmentally acceptable refrigerants.
									XIX Advanced second-law analysis for heat pump systems.
									XX Working fluid safety.

and power generation. Gas motor-driven and absorption heat pumps are seen as alternative options.

- For the industrial sector, the attention was fully focussed on mechanical vapour recompression and heat transformers, and transportation of energy across the borders of industrial complexes.
- There is a need for increased dissemination of knowledge.

(Source The Netherlands National Team.)

Workshop With East and Central European Countries

On request of the IEA, a workshop on Industrial Heat Pumps will be organised by the HPC and the Austrian National Team and will be held on October 24-25, 1991, in Budapest, Hungary. Speakers will come from IEA and from east and central European countries. Participation will be by invitation.

European Community

European Community (EC) Activities on Heat Pumps

In view of the benign environmental prospects of heat pumps, in 1990 a European Heat Pump Working Group was established, and chaired by the EC (Directorat General for Science, Research & Development, DG XII), with the participation of France, Germany, the UK, Italy and the Netherlands.

As a first action, a Workshop on "A Strategy for Heat Pump Development in Europe" will be held on September 24 and 25, 1991 in Paris. The objective is to discuss the state-of-the-art concerning development and use of heat pumps, including the transfer of obtained results of R&D; measures to improve the market situation; and recommendations for an EC strategy. Around 60 people, mainly from industry, have been invited to participate. The Working Group has prepared an EC brochure, providing a total picture of heat pumps which is also available to non-EC countries (contact IEA HPC). Furthermore, within the context of the research programme "JOULE", the EC will issue a call for tender, in the period September - December 1991. R&D on environmentally benign refrigeration systems, and heat pumps are two topics anticipated to be included in the call for tender. Areas for heat pump R&D are: cost reduction, improved efficiency, and environmental benign systems such as Brayton Cycles and ab/adsorption systems.

Finally the EC has commissioned a study to assess the possibilities for heat pumps. The study should lead to recommendations for the European heat pump industry and also for future R&D activities of the EC. The study is expected to be completed early 1992. (Source: EC.)

Bibliography

Refrigerants/CFC's/Environment

The Global Business Outlook for CFC Alternatives

Papers of the conference March 12-13, 1991, London, UK. (English).

Twenty-three papers covered the environmental, manufacturing, toxicological, economic and business issues. They discussed the importance of the alternatives in such markets and applications as household refrigeration, industrial chilling, electronics cleaning, soft and rigid foam blowing, fire extinguishing and mobile air-conditioning. The speakers were invited from the major manufacturers and users of alternatives, as well as from trade associations. They emphasised the economic aspects of the new compounds and of the transition. The papers are available as a set for US\$ 775. Contact Conference Services: Falmouth Associates Inc., 170 U.S. Route One, Falmouth, Maine 04105, USA. Tel.: +1-207-781-3632; Fax: +1-207-781-4177.

Greenhouse Gas Emissions, the Energy Dimension

IEA, 199 pages ISBN 92-64-13444-1 (English).

The book brings together, in one source, a comprehensive discussion of the relationship between energy use and greenhouse gas emissions. It provides an outlook for world energy demand through 2005, and an analysis of the greenhouse gas emissions related to that energy use, by sector and by geographical region. It also examines technologies and policies for limiting those emissions and provides a foundation for further analysis.

The chapters include: an Overview; Global energy outlook; Greenhouse gas emissions factors and emission projections through 2005; Energy technology options for limiting emissions of greenhouse gases; Policy instruments to implement short-term greenhouse gas limitation strategies in the energy sector. The book is available from OECD publication and information centres.

A Global Assessment of Opportunities in CFC Alternatives

Falmouth Associates, 200 pages (English)

The report details the opportunities and problems in developing, manufacturing, testing and introducing new CFC alternatives to the market. It provides relative cost estimates for the manufacture of alternatives versus the present CFC compounds. It also considers the availability of precursor chlorocarbons and HF and the effect on the Cl₂/HCl balance. It also contains advice to CFC and Halon users on how to evaluate and use the substitutes.

For a programme prospectus and more details contact Dr. Hugh Olmstead of Falmouth Associates Inc.,

170 Route One, Falmouth, Maine 04105 USA.
Tel.: +1-207-781-3632; Fax: +1-207-781-4177.

Marketing

Heat Pump Marketing for Electric Utilities

Swiss Commission for Rational Use of Electricity, 39 pages (German).

This handbook is a guideline prepared for Swiss electric utilities to assess the market possibilities of heat pumps relative to the competing systems of oil and gas heating, and how to minimise the risks. It deals with the following application areas (business fields): small capacity heat pumps, which are of particular interest to utilities entering the field, district heating heat pumps, and specific applications. A marketing assessment method, the Portfolio method, is discussed, and guidelines are given on how to implement and control a market strategy. Publisher: INFEL, Informationsstelle für Elektrizitätsanwendung, Lagerstrasse 1, 8021 Zürich, Switzerland; Tel.: +41-1-291-01-02; Fax: +41-1-291-09-03.

Application Aspects

Water Source Heat Pumps: Precautions for Application

M. Delandre et al, Electricite de France (EDF), Revue General du Froid, May, 1991, pp. 27-33 (French).

The article discusses the water side corrosion aspects of water/air and water/water heat pumps. It lists and discusses water properties (such as hardness, alkalinity, pH, electric resistivity, dissolved oxygen, chemical elements, temperature effects, existence of micro organisms), the causes of corrosion, the risks involved, and the parameters that can be modified. Furthermore, the article discusses guidelines for optimum solutions during design, manufacturing, installation, and operation. The article focusses on copper as the construction material.

Anti-freeze Fluids and Their Application Fields

S. Verney, A. Malek, CETIM, Revue Generale de Froid, May, 1991, pp. 21-25 (French).

The article gives a comparative overview for 11 anti-freeze fluids of Reynolds numbers, heat transfer coefficients, pressure losses, and the ratio of pumping power to thermal capacity based on flow with a velocity of 1.5 m/s in a 4 mm long 21 mm diameter tube. The temperature range covered is -90 to 0°C. It also lists the properties such as toxicity, inflammability, and corrosion.

Proceedings

Thermastock '91 - Proceedings of the 5th International Conference on Thermal Energy Storage

Held at 13-16 May, 1991 at Scheveningen, The Netherlands (English).

At the Conference, both long and short-term heat storage were discussed in 58 and 31 papers respectively. For both fields separate sections dealt with: Applications and field experiments; Engineering tools; Recent R&D results and new prospects. The following papers deal specifically with heat pumps:

- Improved performance of heat pumps using seasonal storage of solar thermal energy; U. Henriksen et al. (Denmark)
- Heat pumps using Ates: a feasibility study for a shopping mall; J. van Hove, A.L. Snijders (Netherlands)
- Sub-surface cold storage in ground-coupled heat pump systems; B. Sanner, K. Knoblich (Germany)
- Design evaluation of the ground-source heat pump with horizontal spiral heat exchangers; O.J. Svec (Canada)
- SWEAT thermochemical heat pump storage system; H.A. de Beijer, J.W. Klein-Horsman (Netherlands).

The proceedings are available from Novem, P.O. Box 8242, 3503 RE Utrecht, The Netherlands. Price US\$ 50. For copies of an individual paper contact the IEA HPC.

Cogeneration Power Plants and Heat Pumps - combined heat, cold and power production in industry, trade and service companies

Proceedings of the 3rd international conference of the VDI-GET, June 4 and 5, 1991 Essen, Germany. 273 pages (partly in English, partly in German).

The papers on heat pumps concern, amongst others, technical standards, rules and regulations in Japan; the status of gas engine heat pumps in Japan; new refrigerants (see abstracts in the column "News and Views" in this Newsletter); the possibilities of heat pump/cogeneration systems in former East Germany; and 10 years operating experience with a gas engine-driven heat pump in a building in Germany. With regard to cogeneration, the proceedings include papers on environmental aspects; the status of cogeneration technology in Japan; and a number of field cases. The proceedings are available from VDI-Verlag GmbH Postfach 8228, 4000 Düsseldorf 1, Germany, VDI Bericht 887 (ISBN 3-18-090887-4).

Heat Pumps and Energy Recovery

IIR Proceedings of the meeting of Commission E2 in Stockholm, August 29-31, 1990; pages 296, Price FF 170 (English).

These proceedings contain 21 papers divided into three categories: Basics and Principles; Energy Efficient

Applications - Cases for Heat Recovery - Field Experiences; New Development and Pilot Plants. Examples of topics are: parameters influencing energy demand in heat pumps and refrigeration; new fluid mixtures; energy aspects in CFC developments; industrial heat pumps in Norway; heat recovery and heat pump applications; control of liquid feed to evaporators; Lysholm screw expanders; the Stirling cycle optimised for heat pump use; performance of a 50 kW chemical heat pump test plant; absorption technology and heat transformers; the Japanese Super Heat Pump Energy Accumulation System. The proceedings are available from IIR 177, bd Malesherbes, F 75017, Paris, France; Tel.: +33-1-4227-3235; Fax: +33-1-4763-1798.

Progress in the Science and Technology of Refrigeration in Food Engineering

IIR Proceedings of the meetings of: Commissions B2, C2, D1, D2/3, September 24-28, 1990 pages 871 (English).

These proceedings contain the following papers on heat pumps:

- Plate type evaporator of heat pump plant, structural features; B.T. Marinuk, B.A. Ivanov (USSR) pp 173-175
- Thermodynamic analysis of the closed type absorption transformer; Y.A. Gogus, O.E. Ataer (Turkey) pp 201-205
- Working fluid pairs of refrigerants and of NMC for compression heat pumps with solution circuit, H. Najork, A. Thies (Germany) pp 265-273
- Environmental impact of different refrigerants used in heat pumps; G. Heinrich, J. Peters (Germany) pp 289-296.

The proceedings are available from IIR, 177, Boulevard Malesherbes, F-15017, Paris, France; Tel.: +33-1-4227-3235; Fax: +33-1-4763-1798.

Research Activities on Advanced Heat Pumps

Proceedings of the 3rd international workshop, held in September 1990 at the University of Technology of Graz, Edited by F. Moser, pages 462 (English).

The proceedings include 44 papers. Thirteen of these papers concern integration of heat pumps to the technical and commercial environment; another 13 concern compression heat pumps and CFC replacement; and 12 deal with sorption heat pump/transformer related research. The proceedings are available from dbv-Verlag für die Technische Universität Graz, Uhlandgasse 8, A-8010 Graz, Austria; Tel.: +43-316-825-166.

ASHRAE Transactions

Order from ASHRAE, 1791 Tullie Circle, NE, Atlanta GA 30329-2305, USA; Tel.: +1-404-636-8400; Fax: +1-404-321-5478. Quoted prices are for non-ASHRAE members (English).

1991 Winter Meeting - New York, NY

ASHRAE Transactions 1991, Vol. 97, Part 1, 117 papers
Code 82911. Price US\$ 170.

1991 Annual Meeting - Indianapolis, IN

ASHRAE Transactions 1991, Vol. 97, Part 2, 136 papers
Available January 1992.
Code 82912. Price US\$ 170.

Text Books

Modern Refrigerant Machines

I. Cerepalkovski, 1991, 319 pages (English)

The objective of the book is to explain the various current modern refrigeration systems, as well as how to improve and develop estimation methods and design procedures. It is intended for researchers, engineers, and technicians, and may also be used as a reference book for lecturers and as a textbook for students.

Contents: Fundamental theory; Single stage and complex vapour compression cycles and systems; Air compression, absorption, steam jet refrigeration systems; Low temperature cryogenic systems for liquefaction of gases; Compressors, condensers, evaporators; Performance of a vapour compression refrigerant system; Automatic control; Refrigerants; Properties for E-717, R-22, R-502. Available from Elsevier Science Publishers, P.O. Box 1991, 1000 BZ Amsterdam, the Netherlands or P.O. Box 882, Madison Square, New York 10159, USA. Price US\$ 125.50/DFL 220.00.

Research & Development

Development of an Absorption Heat Pump Water Heater Using an Aqueous Ternary Hydroxide Working Fluid

K.E. Herold, R. Rademacher, Dept. of Mech. Engineering, University of Maryland; L. Howe, D.C. Erickson, Energy Concepts Co.; Int. Journal of Refrigeration 1991, Vol. 14, May, pp. 156-167 (English).

A novel absorption cycle, with a significantly higher temperature lift than current in technology, is under development. The cycle is based on an aqueous ternary hydroxide working fluid consisting of NaOH, KOH, CsOH in the proportions 40:36:24. The ternary hydroxide blend is used to shift the crystallisation curve to provide the high temperature lift. Water heating is one application where the high lift feature is necessary. Development efforts to date have included thermophysical property measurements on the ternary hydroxide system, heat and mass transfer measurements in absorption and vapour generation, and cycle design calculations.



Conferences

* Call for Papers

Meeting Customer Needs with Heat Pumps

October 15-18, 1991 / Dallas, Texas (USA)
Conference and exhibition.
Contact: Pam Turner, EPRI Conference and Travel
Tel.: +1-415-855-2010, or:
David P. Ross, Policy Research Associates, Inc.
Tel.: +1-703-620-1008.

European Refrigeration and Air-Conditioning Conference

October 16-19, 1991 / Essen (Germany)
Contact: Verband Deutscher, Kälte-Klima-Fachbetriebe e.V.
Tel.: +49-911-86-06-315.

1991 DKV Refrigeration and Air-Conditioning Conference

November 20-22, 1991 / Berlin (Germany)
Organised by German Assoc. of Refrigeration & A/C (DKV).
Contact: DKV, Pfaffenwaldring 10, 7000 Stuttgart 80, Germany.
Tel.: +49-711-685-3200; Fax: +49-711-685-3242.

1991 ASME Winter Annual Meeting, Symposium on Heat Pump Design, Analysis and Application

December 1-6, 1991 / Atlanta, Georgia (USA)
Contact: Keith E. Herold, Department of Mechanical Engineering, University of Maryland, College Park, MD 20742, USA.
Tel.: +1-301-405-5268; Fax: +1-301-314-9477.

The International CFC and Halon Alternatives Conference

December 3 to 5, 1991 / Baltimore, Maryland (USA)
Scholarly papers about new technology and regulatory activity.
Contact: Heather White, P.O. Box 236, Frederick, Maryland 21701, USA.
Tel.: +1-301-695-3762; Fax: +1-301-695-0175.

1992 ASHRAE Winter Meeting

January 25-29, 1992 / Anaheim, California (USA)
Contact: ASHRAE Meetings Department, 1791 Tullie Circle NE, Atlanta, GA 30329, USA.
Tel.: +1-404-636-8400; Fax: +1-404-321-5478.

CEX '92 (Canadian Environmental Exposition)

March 13-15, 1992 / Toronto, Ontario (Canada)
Sponsored by the Canadian Institute of Plumbing and Heating, together with the Heating, Refrigeration & Air-Conditioning Institute of Canada.
Contact: H.P. Shield and Ass. Ltd., 25 Brigade Road, Don Mills, Ontario, Canada M3B 1J6.
Tel.: +1-416-444-5225; Fax: +1-416-444-8268.

*Next Generation Technologies for Efficient Energy End Uses and Fuel Switching

April 7-9, 1992 / Dortmund (Germany)
Organised by the IEA and the German Federal Ministry for Research and Technology (BMFT).
Contact: GVD Gesellschaft für Veranstaltungs-Dienste mbH, Degerstrasse 38, D-4000 Düsseldorf 1, Germany.
Tel.: +49-211-680-1736; Fax: +49-211-681-462.

ASHRAE Annual Meeting

June 27-July 1, 1992 / Baltimore, Maryland (USA)
Contact: Meetings Coordinator ASHRAE, 1791 Tullie Circle, NE, Atlanta, GA 30329, USA.
Tel.: +1-404-636-8400; Fax: +1-404-321-5478.

Energy Economy 1992 - European Exhibition and Conference on Energy Efficiency and the Environment

September 15-17, 1992 / Maastricht (The Netherlands)
Organised by, among others, Novem b.v.
Contact: RAI Gebouw B.V., Europaplein, 1078 GZ Amsterdam, The Netherlands.
Tel.: +31-20-549-1212; Fax: +31-20-646-4469.

*Solid Sorption Refrigeration and Heat Pump

November 18-20, 1992 / Paris (France)
Symposium organised by Commission B1 of the IIR.
Contact: Mr. Francis Meunier, LIMSI-CNRS, P.O. Box 123, F 91403 Orsay Cedex, France.
Tel.: +33-1-69-85-80-56; Fax: +33-1-69-85-80-88.
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