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NEWS LETTER

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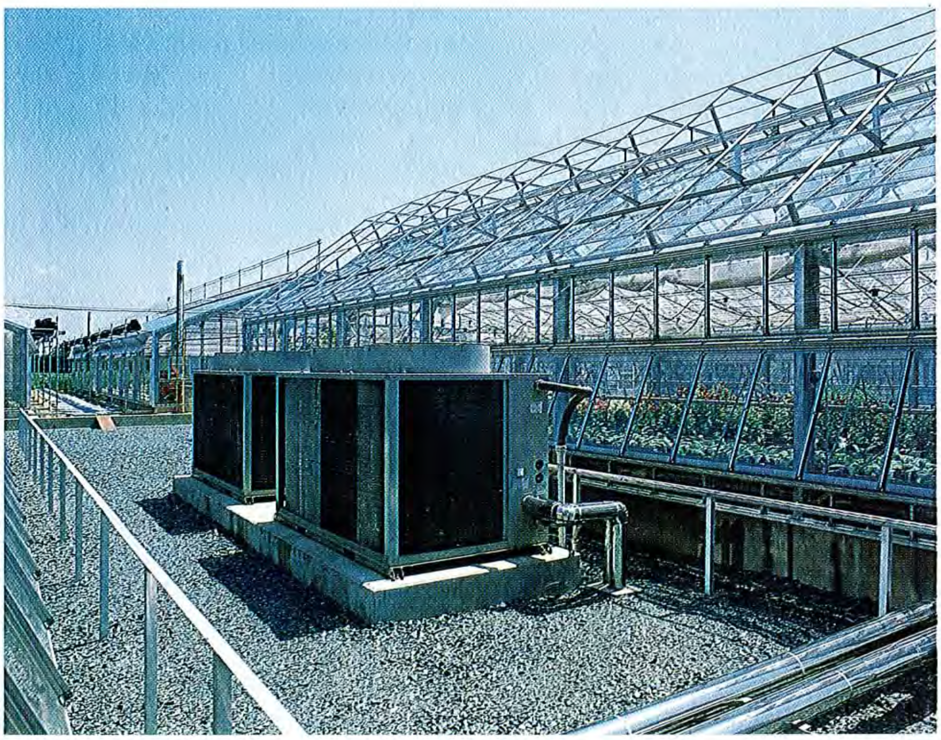

Greenhouse using air-source heat pumps (see page 24)

Contents **Page**

This issue: Heat pumps -- heating and cooling

Heat pumps for heating and cooling are of increasing importance for reasons of economics, energy conservation, and environmental protection. This application involves combining such things as space heating, hot water or process heat production with refrigeration or industrial, commercial, and residential cooling. Depending on the heating and cooling load requirements, both simultaneous and intermittent or seasonal modes of heating and cooling can be utilized.

Currently heat pumps are of primary interest as heating devices. Nevertheless the cooling effect available at the evaporator should be used for an economic benefit whenever possible to make the heat pump a more attractive investment. In this context more attention should be paid to the recovery of waste heat from refrigerants, chillers and air-conditioners using heat pumps. For example, at a university in Germany, use is made of refrigeration condenser heat for the preheating of fresh air for air-conditioning resulting in energy savings of around 1400 MWh per year.

The economic benefit of the heat pump's cooling effect is often called a consequential profit and can easily be determined for such benefits as reduced cooling water consumption, reduced refrigeration plant power requirements (this could be the case when using refrigeration plant cooling water as a heat source for the heat pump) or air conditioning (see the article by Eggen). The profit may be more difficult to determine for such things as reduced temperature of waste water, reduced pollution levels, and improved product quality (see article by Kondo).

In industrial processes proper heat pump integration will insure the most profitable use of the heat pump's ability to transfer heat from a place where cooling is needed to one where heating is required. The article by Naka et al discusses this for a heat pump application in a distillation process. Pinch technology can also be used as an analysis tool to identify heat pump integration opportunities as described in the article by McMullan.

Periodic or seasonal heating and cooling heat pump units have become quite common. The economic benefits of utilizing the same equipment for cooling purposes during the summer and heating during the winter make the heat pump an attractive system for air conditioning in appropriate climates. Prime examples are North America and Japan where large markets exist for this type of heat pump.

New developments will allow heat pumps to meet heating and cooling requirements more effectively. This includes the ability to adjust the cooling and heating capacity according to the load requirements. The article by Ziegler describes a compression absorption system for simultaneous cooling and process hot water production. The article by Kern describes a double-lift heat pump capable of simultaneous cooling and high temperature output. The unit can also be operated as a high efficiency chiller in summer and in winter as a heat pump.

To supplement the articles mentioned previously a bibliographic review has been prepared which presents recent work on heat pump heating and cooling applications.

Two additional articles which are not directly related to the main topic of this issue have also been included for general interest. An article by Kita discusses the application of a heat pump in a greenhouse in Japan. The author identifies improved product quality and cost savings as some of the benefits of using the heat pump. An article by Gerdsmeyer and Kruse describes the results of some experimental work aimed at evaluating a method for changing the refrigerant composition in a heat pump using a refrigerant mixture. Improved capacity control is the ultimate goal of this work.

In the extended News Briefs section, information on heat pumps and related topics is given. Of particular interest is the information on CFC issue developments in the United States and Japan.

Finally, we would also like to direct our readers to page 31 where information on the 3rd IEA Heat Pump Conference in Tokyo and related activities can be found. This conference is an excellent opportunity to learn about the most recent heat pump activities going on worldwide. Please note that the registration deadline for the conference is January 31, 1990.

Errata

1. In Volume 7, Number 2, the article entitled "Recommended Thermodynamic Data for NH_3-H_2O in the Temperature Range of -50° to 316°C and Pressure Range of 0.05 to 170 Bar" (pages 48-53) was co-authored by Dr. Thomas S. Zawacki of Phillips Engineering Company, St. Joseph, Michigan, USA.

2. In Volume 7, Number 2, in the article entitled "Latest Developments and Future Trends in the Field of CFC Substitutes," page 14, second column, last sentence, the reference to Figure 2 on page 23 should read "...only the compounds in the black area are left over."

G. Eggen*

Heat Pump for Heating and Cooling of a Hotel

In 1985 the old hotel building of Hotel Maritim in Haugesund, Norway, was expanded. A new congress hall was made in the annex. For cooling purposes, a cooling plant had to be installed. Besides the demand for air cooling, the hotel also requires heating. Throughout the year there is a great demand for sanitary hot water. This means long operation times and favorable conditions for heat pumps. For this reason, a heat pump for combined heating and cooling purposes was installed.

Dimensioning data

The heat pump is mainly dimensioned to cover the cooling demand in the new hotel building. The cooling capacity is 225 kW at 6°C cooling water temperature out of the evaporator, and 50°C forward water temperature out of the condenser. At these operating conditions, the condenser heat production is 310 kW. This will cover most of the heat demand in both the new and the existing building of the hotel (Figure 1).

Plant design

A principal piping diagram of the heat pump for heating and cooling is shown in Figure 2. The heat pump is operating with R12 as working fluid. On the low temperature side, the heat pump extracts heat from a water/glycol pipeline, which is connected both to the air cooling heat exchangers in the hotel as well as to a sea water plate heat exchanger.

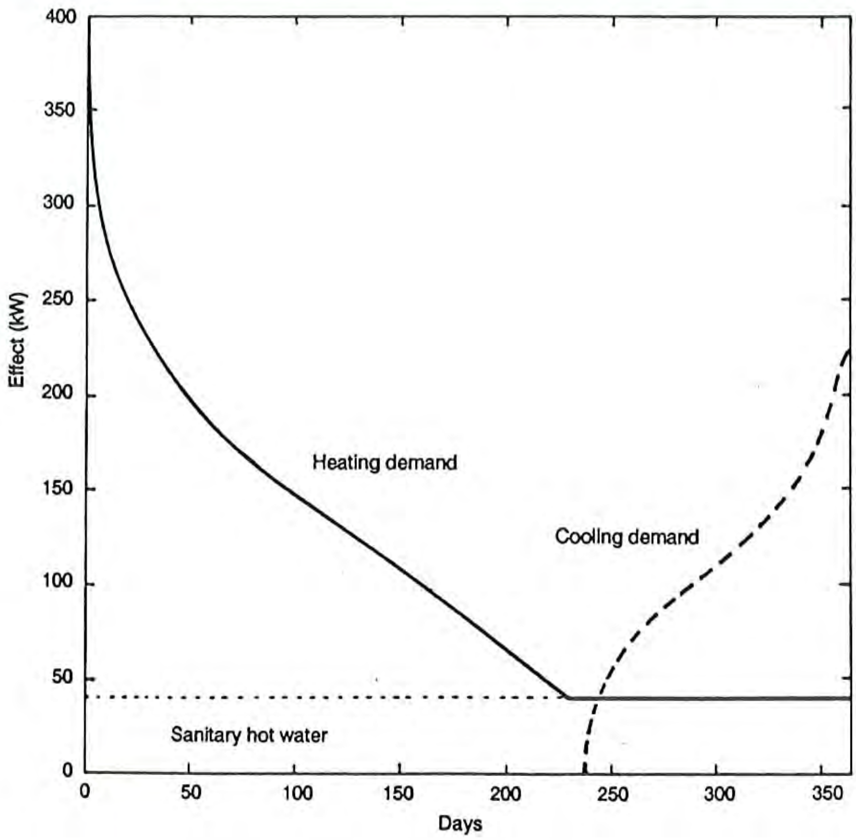

Figure 1. Capacity duration of heating and cooling demand

The heat pump plant is located on the quay just outside the hotel (see Figure 3). The sea water system is quite simple. It consists of a 4m pipeline with a submersible pump in a basin underneath the heat pump. The pump basin is directly connected to the sea through a perforated plate acting as strainer. After passing the plate heat exchanger, the sea water is drained back to the sea.

The heat pump is connected to the heating central in the old building by means of insulated water tubes. The return water is preheated by the heat pump. At low ambient temperatures, the heat distribution water will be heated after exiting the heat pump's condenser by an electric or an oil-fired boiler. The heat distribution system consists of radiators and of heat exchangers in the ventilation system. Sanitary hot water is heated by the heat pump in separate hot water tanks. To produce high temperature hot water for kitchen purposes, a part of the pre-

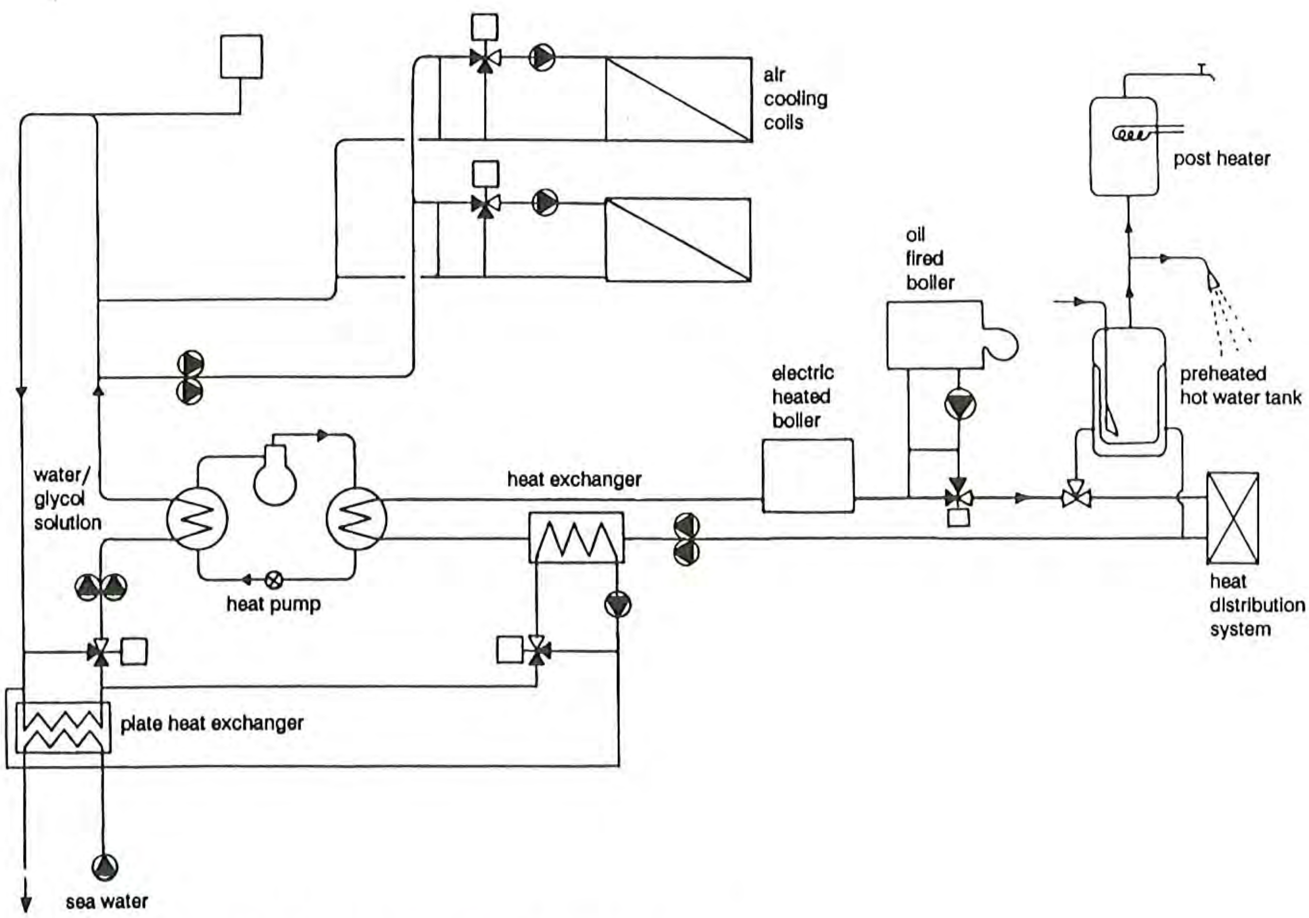

Figure 2. Principal piping diagram for the heat pump for both heating and cooling

heated water flows to an electrically heated water tank, where the water is heated to the desired temperature.

Operation strategies

The maximum forward water temperature for heat distribution to the old part of the hotel needs to be 70°C at the design outside temperature of -10°C. The water temperature is decreased at increasing ambient temperatures. However, the forward temperature is not decreased below 50°C in order to heat the sanitary water to at least 45°C.

When the heat requirements dominate, the heat pump transfers heat from the air cooling heat exchangers to the heat distribution system. If available heat from the air cooling is not sufficient, the heat pump will use sea water as an additional heat source. Heat is transferred from sea water to the water/glycol-solution through a titanium plate type heat exchanger.

In summer, the cooling requirements dominate. Then the sea water is used as a heat sink for the air cooling plant. The surplus of heat is transferred from the heat distribution system to the sea water via the two heat exchangers shown in Figure 2. The operation of the heat pump for heating and cooling is regulated automatically.

Figure 3. Hotel Maritim, Haugesund, Norway

Experiences

The heat pump has been in operation for more than three years. During the first year of operation, there were some problems with the automatic control equipment. These problems resulted in compressor damage. The compressor was replaced in 1986. During 1987 and 1988, the plant has been running with 100% availability most of the time.

Normally, the cooling requirements dominate from April to September. During the rest of the year, the plant operates primarily as a heat pump. Table 1 gives some results from energy measurements during one year of operation.

The extra expenses incurred in order to upgrade the refrigerating plant for heat pump operating conditions were approximately 300,000 NOK (1985). The payback period for this extra heat pump investment is less than two years.

Conclusion

Hotels usually have cooling demands which require installation of a refrigerating plant. Besides the demand for air cooling, heating is also required. By installing a heat pump for both heating and cooling purposes, the heating plant will be very efficient and economical.

At the Hotel Maritim, a heat pump was installed for heating and cooling in 1985. The payback period for the extra heat pump investment was less than two years.

Operation		Heating Sep-Apr	Cooling May-Aug	Total Sep-Sep
Heat production				
Heat pump	(MWh/year)	650	550	1200
Heat rejected during cooling operation (estimated)	(MWh/year)		400	400
Useful heat prod	(MWh/year)	650	150	800
Electric heated boiler	(MWh/year)	750	150	900
Energy consumption				
Heat pump	(MWh/year)	235	195	430
Electric heated boiler	(MWh/year)	100	0	100
Total energy consumption	(MWh/year)	335	195	530
Energy savings	(MWh/year)	415	150	565

Table 1. Energy measurements from September 1987 to September 1988

**Geir Eggen, SINTEF (The Foundation for Scientific and Industrial Research at the Norwegian Institute of Technology), Trondheim, Norway*

Y. Kondo*

Heat Pump System Featuring Efficient Heat Utilization at a Plating Plant

Although the largest advantage of a heat pump system is said to be its capability for both cooling and heating, this use is actually limited due to the characteristic relations between heat balance and utilized temperature width. Such being the case, we do not know of many actual examples of installations in industrial processes. In the plating process, however, a heat pump system allows ideal utilization of heat.

State of heat source utilization in electric plating

Electric plating uses zinc, gold, silver, copper, chromium, nickel, etc., to provide a corrosion-proof or decorative finish.

By immersing the plating substance and plated piece as two poles in a solution of cyanide, sulfate, etc., and running a certain current between the two poles, the piece to be plated is coated with the plating substance. The current efficiency at this time is not 100%, and the electrolyte in the electrolytic cell generates heat due to various factors, such as electric resistance of the electrolyte and contact resistance. In order to maintain a high plating quality, however, the liquid temperature must be kept constant.

Depending on the type of plating, the required temperature is high so that the electrolyte must be heated. In the case of zinc plating with a cyanide or chloride bath, cooling is required to maintain the temperature at about 25°C.

In addition, the pieces to be plated require pretreatment (defatting and pickling) and drying after plating, which all require heating.

In this series of processes, water or a refrigerating machine is used for cooling while steam or an electric heater is used for heating, thus consuming heat separately in most cases.

Outline of the system

The plating equipment covered is as shown in Table 1 and Figure 1. The flow diagram of the heat pump system used for the plating equipment is shown in Figure 2. In designing this system, the following points have been considered to enhance the system efficiency:

1. Effective utilization of existing equipment. Since the facilities other than new facilities are already equipped with cooling/heating devices, they shall be used effectively.

2. Economic scale. The heat pump must be of proper size so that it can exhibit high performance all year round, and be selected from those on the market on condition that it satisfies the specific working requirements (18°C for chilled water, 80°C for hot water). The peak load in mid-summer and mid-winter shall be covered by changing over part of the load to cooling/heating by water or steam.

3. The heat pump shall be controlled primarily to maintain the plating temperature. (As it is mainly operated for cooling, the heating capacity varies with the cooling load.) When there is no cooling load, such as during the rise time before starting the plating work, steam is used as the heating source.

Selection of the heat pump

Based on the design conditions mentioned above, a heat pump model satis-

Line No.			Section 5	Section 4
System			Elevator type fully automatic hooking use zinc cyanide plating	Elevator type fully automatic hooking use zinc chloride plating
Cooling side	Plating tank	Rated output of rectifier	1.2V x 5000A	5V x 5000A
		Normal output of rectifier	7.5V x 3700A	3V x 2600A
		Liquid level	8000l	10000l
		Control temperature	25±2°C	25±2°C
	Hydraulic device	Oil level	360l	450l
		Control temperature	40±10°C	40±10°C
Heating side	Defatting tank	Liquid level	2400l	
		Control temperature	65±2°C	
	Pickling tank	Liquid level	1500l	
		Control temperature	43±2°C	
	Electrolytic defatting tank	Liquid level	1000l	
		Control temperature	53±2°C	
	Dryer	Fan capacity	201m³/min	
		Control temperature	61±1°C	

Table 1. Heat related specifications

fying the cooling load of the electrolyte was selected. The cooling load was based on data obtained by actual measurements during the past year of cooling water quantity and heat exchanger inlet and outlet temperature differences.

The heating load was calculated from the amount of boiler fuel used. However, since the results of use at the subject line alone remained unknown, the heat pump was used to measure the heating load share of the subject line through actual measurements against the overall plant use. Table 2 shows the seasonal average load.

As for the heat pump capacity a system which satisfies the cooling load in summer and heating load in winter separately causes a surplus capacity in other seasons so that it is not effective in terms of investment. When the capacity is too small, it will fail to satisfy load requirements. Thus, the following selection criteria were set for the heat pump unit:

- Its capacity is mainly intended for cooling the electrolyte
- Its capacity assures year-round operation at a high load factor
- It satisfies cooling and heating temperature conditions

Based on the results of a tentative calculation showing that it is possible to cool electrolyte in Section 5 all year round, cool the hydraulic device in the same section for about six months centering around winter, and also cool electrolyte in Section 4 for about 10 months except for two months in summer, a "water-source heat pump 15.5kW x 1 unit" was selected from among those on the market. The demonstration test results were close to the manufacturer's capacity indication.

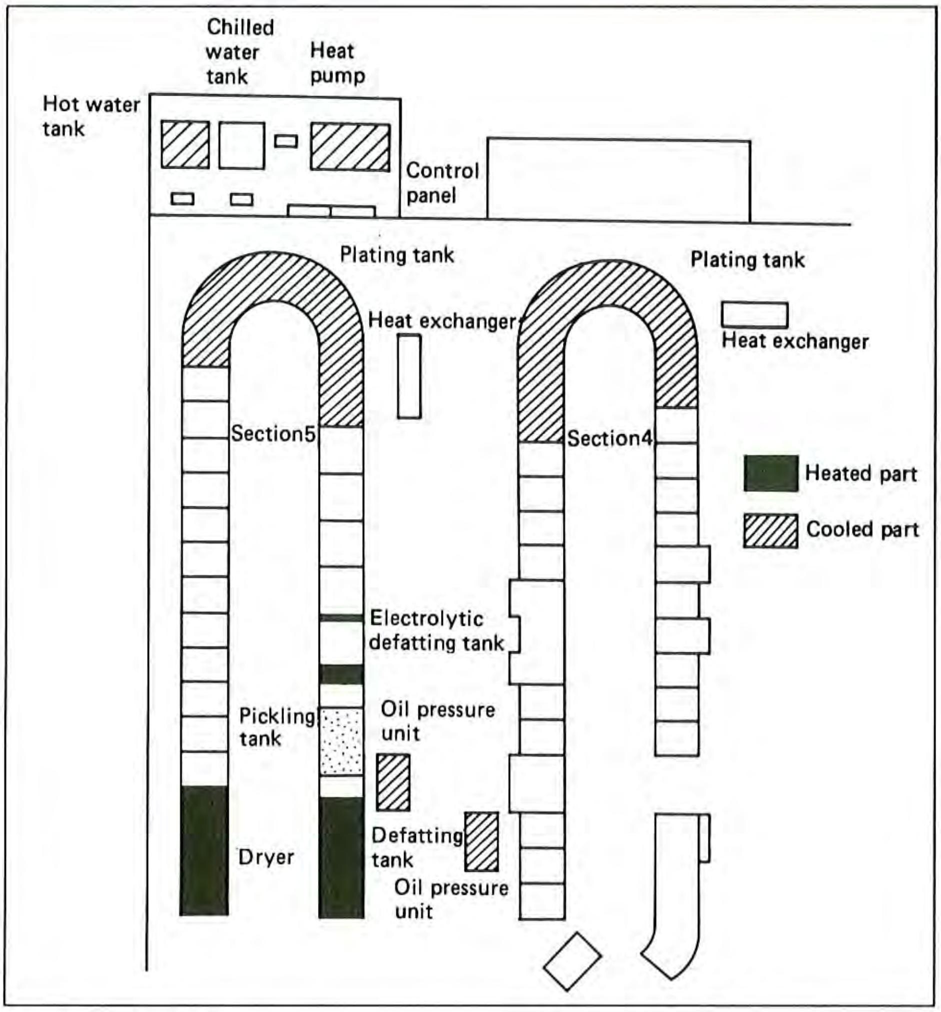

Figure 1. Plant layout

Special features of the system

The system is designed to collect waste heat from cooling the plating tank and hydraulic device, and use it as a heat source for heating the defatting tank, pickling tank, electrolytic tank and warm air dryer.

The heat pump gives priority to cooling load in its control, and supplies 18°C chilled water and 75°C hot water.

Heating for the defatting tank when there is insufficient cooling load is backed up by the existing steam boiler.

	Cooling load		Heating load		Remarks
	Average load	ratio	Average load	ratio	
Summer	32,299	1	47,002	0.73	Average for Jun to Sep
Intermediate season	30,363	0.94	56,397	0.88	Average for Apr, May, Oct, and Nov
Winter	26,544	0.82	64,412	1	Average for Dec to Mar
Annual average	29,760		55,775		

Table 2. Seasonal average load (unit: Mcal/month)

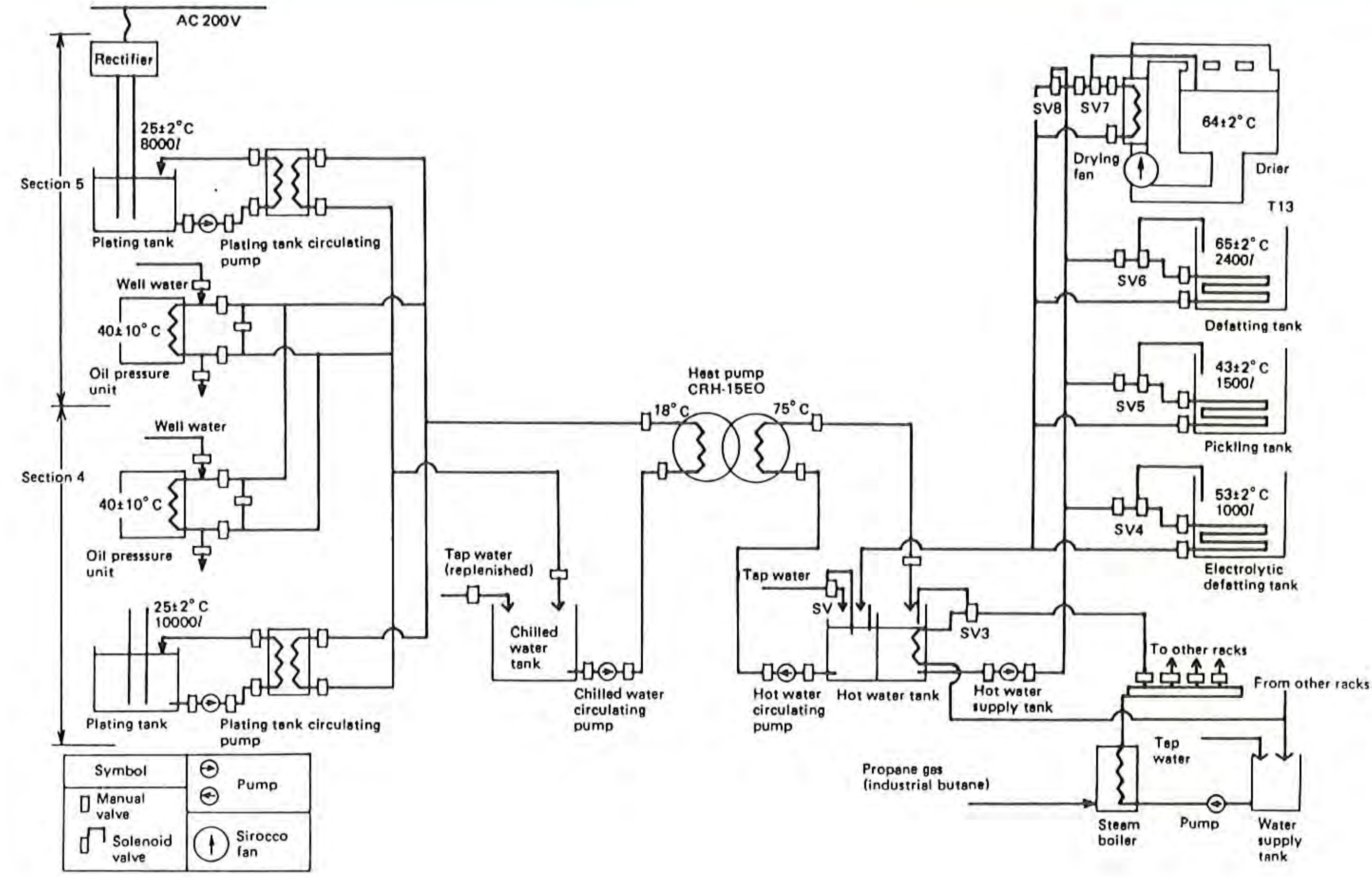

Figure 2. Plant's cooling heat recycling/utilization system flow chart

Month	Heat quantity for cooling Mcal			Heat pump's power		Heat qty total	Heat pump's COP		
	Section 4	Section 5	Total	kWh	Mcal	Mcal	Cooling	Heating	Total
Jun	1,510	3,947	5,457	4,391	3,776	9,233	1.45	2.45	3.90
Jul	514	5,427	5,941	4,621	3,978	9,915	1.50	2.51	4.01
Aug	0	4,868	4,868	3,840	3,303	8,171	1.48	2.48	3.96
Sep	438	4,874	5,312	4,217	3,627	8,939	1.47	2.47	3.94
Oct	958	4,270	5,228	4,227	3,635	8,863	1.44	2.44	3.88
Nov	511	3,878	4,389	3,695	3,178	7,567	1.38	2.38	3.76
Dec	100	3,114	3,214	2,726	2,345	5,559	1.37	2.37	3.74
Jan	58	3,842	3,900	3,312	2,848	6,748	1.37	2.37	3.74
Feb	85	3,425	3,510	3,018	2,596	6,106	1.35	2.35	3.70
Mar	266	3,816	4,082	3,540	3,044	7,126	1.34	2.34	3.68
Apr	327	4,318	4,645	3,863	3,322	7,967	1.40	2.40	3.80
May	690	3,741	4,431	3,706	3,187	7,618	1.39	2.39	3.78
Total	5,457	49,520	54,977	45,156	38,835	93,812	1.42	2.42	3.84
Auxiliary unit's power 12,739kWh When 10,956Mcal is added:				49,791		1.10	1.88	2.98	
Heat pump specifications:						1.73	2.78	4.50	
Quantity of fuel saved: 16,530kg x 10,830kcal x 0.8 x 0.8 = 114,573Mcal Heat pump's heat quantity: 93,812Mcal/114,573Mcal = 0.82									

Table 3. Heat balance of heat pump

Month	Ground water savings		Fuel savings		Capacity increase		Tap water increase		Cost saved
	m³	Value @190(¥)	kg	Value @95(¥)	kWh	Value @21(¥)	m³	Value @427(¥)	Value (¥)
Jun	1,212	230,280	932	88,540	5,315	111,615	24	10,248	196,957
Jul	870	165,300	2,087	198,265	5,747	120,687	30	12,810	230,068
Aug	708	134,520	697	66,215	4,752	99,792	39	16,653	84,290
Sep	1,043	198,170	1,570	149,150	5,243	110,103	30	12,810	224,407
Oct	1,161	220,590	1,970	187,150	5,389	113,169	19	8,113	286,458
Nov	1,052	199,880	1,246	118,370	4,810	101,010	21	8,967	208,273
Dec	955	181,450	1,203	114,285	3,742	78,582	6	2,562	214,591
Jan	669	127,110	1,998	189,810	4,372	91,812	4	1,708	223,400
Feb	1,381	262,390	1,192	113,240	4,085	85,785	3	1,281	288,564
Mar	1,202	228,380	866	82,270	4,700	98,700	7	2,989	208,961
Apr	1,088	206,720	1,301	123,595	4,979	104,559	16	6,832	218,924
May	1,257	238,830	1,468	139,460	4,761	99,981	18	7,686	270,617
Total	12,598	2,393,620	*16,530	1,570,350	57,895	1,215,795	217	92,659	2,655,513

*The figures for fuel represent results obtained by deducting the effect of heat retention between November and May (¥209,760 for 2,208kg).

Investment cost and depreciation period:

Machinery and equipment cost	¥2,580,000
Construction cost	¥2,920,000
Total equipment cost	¥5,500,000
Annual cost savings	¥2,655,513
Payback period	2.07 years

Table 4. Saving effect and cost recovery period

The special features of this system may be summarized as follows:

- Reduced yearly energy cost (about 2,200,000 yen/year as of May 1985)
- Improved product quality due to automatic control of the plating tank temperature
- Saving on underground water used as cooling water, and reduced load on drainage treatment device

Data on operation was collected by attaching measuring instruments, such as a calorimeter, to this system. The quantity of high temperature heat obtained was about 82% of the value calculated from the quantity of fuel saved, while the coefficient of performance (COP) of the heat pump was 3.84. This is lower than the COP of 4.5 set at the time of design. However, a value close to 4.5 is likely to represent a true value, if one takes into account that a counting mistake on the calorimeter is possible since the heat pump was operated with a small difference between the outgoing/return chilled/hot water temperatures. In addition it was difficult to correctly measure the boiler efficiency and the values obtained are estimates involving an error.

Table 3 summarizes the monthly heat balance of the heat pump.

Economic effect

The measured values for the year prior to the introduction were compared at the same level of area processed. Apart from atmospheric temperature differences, cutdown on underground water pumping up power, etc., the economic benefits were as shown in Table 4. An overall cost saving of 2,655,513 yen was obtained.

Summary

This system is an example of maximum utilization of heat pump characteristics. Although it has been developed for zinc plating, it can also be used for other purposes if there are processes that require cooling/heating. At present, however, the upper limit of deliverable temperature is about 80°C. Since a high temperature heat pump capable of delivering 100°C or higher is being developed the scope of applicability will expand in the near future.

In order to produce a beneficial effect when introducing a similar system in processes that require cooling/heating:

1. It is important to make preliminary investigations to confirm the actual loads. This will allow economical equipment design.
2. It is desirable to introduce this technology through plating machinery manufacturers, although design and installation can be carried out with the aid of the plant owner.
3. During operation, attentive year-round management can bring about the highest performance possible.

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Development of a Distillation System Using a Heat Pump

Operability studies on a distillation system with a heat pump are carried out in a pilot plant using digital simulation. The heat pump is not installed between a condenser and a reboiler, but between a condenser and an intermediate plate in the heating zone of a distillation column (C/SH System). This system reduces by approximately 36% (rotary compressor) and 40% (centrifugal compressor) the steam supplied to a conventional column. In addition, it assures that wide regions of feasible and stable operation can be established by using a conventional control structure.

Introduction

Distillation systems are well known as a useful separation unit in the chemical industry. But, as there are many distillation systems in a factory, they consume a large amount of heat energy. On the average, the distillation systems in a factory consume more than 60% of the total energy supply. Several technologies to reduce the energy consumption of distillation systems have been developed including design of a heat exchanger network around distillation systems and a multi-effect column system with heat integration, introduction of a heat pump system, and so on.

A general concept for installation of heat pumps to distillation systems has been developed since the 1940s and a few combined systems have been realized. This paper discusses the feasibility and operability of such a combined system, and the problems to be solved for extensive application of a combined system to a chemical plant.

Previous types of distillation systems using a heat pump

Many types of distillation systems associated with heat pumps have been developed, as shown in journals and books. Figure 1 shows the schematic flowsheets of typical examples.[1]

1. Closed cycle process. This system uses a working fluid. A condenser which removes the heat energy of the process vapor stream from the top of a column vaporizes the working fluid. The temperature is raised above the boiling point of the bottom product in the reboiler by a compressor. The pressurized working fluid supplies heat energy to a reboiler and is condensed. Then, it returns to the condenser after decompression by an expansion valve.
2. Vapor recompression. This system used a process stream as a working fluid. The process vapor from a column is directly pressurized by a compressor. Then, it supplies heat energy to a reboiler and is condensed.
3. Bottom flashing. This system also employs a process stream as a working fluid. The liquid stream from the bottom of the column is flashed by an expansion valve and its temperature decreases. The decompressed process stream receives heat energy in a condenser. Then it is pressurized by a compressor and returns to the bottom.

The above types are designed based on heat exchange between the top and bottom of a column by using a heat pump, therefore several problems must be considered. For example, a combined system must be investigated in terms of feasibility, start-up procedure, and operability.

Problems in introducing a heat pump to a distillation system

For the above mentioned types, the heat pump systems are used only between a condenser and a reboiler. In a practical sense, a compressor system cannot be used to raise the temperature of a working fluid by more than 30°C because of the cost. Types 2 and 3 have one heat exchanger in a heat pump system to reduce the temperature difference between the condenser and the reboiler. For many distillation columns the temperature difference is above the feasible temperature increment of a compressor. In such cases installation of a heat pump to a distillation system is not feasible. The following restrictions seem to be caused by the heat pump:

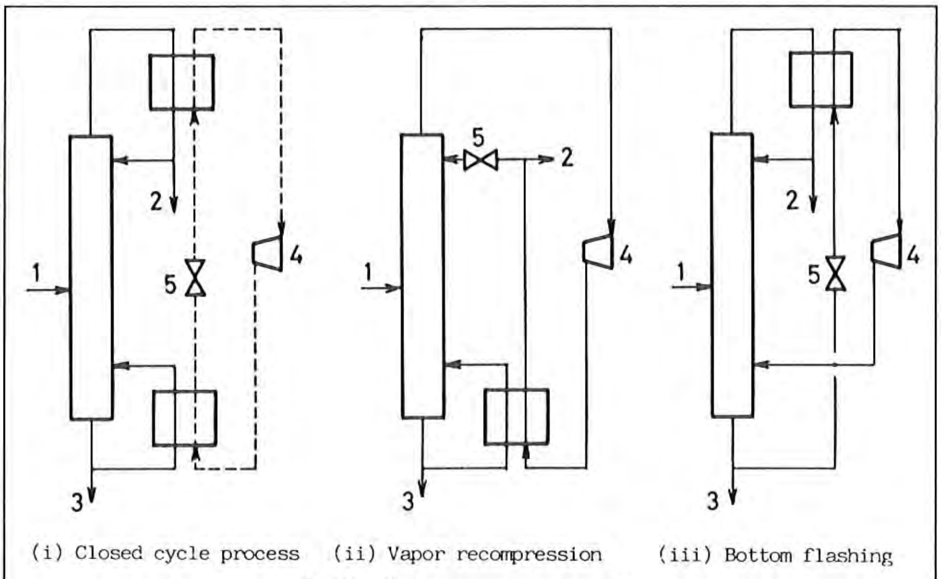

Figure 1. Types of combined systems previously proposed

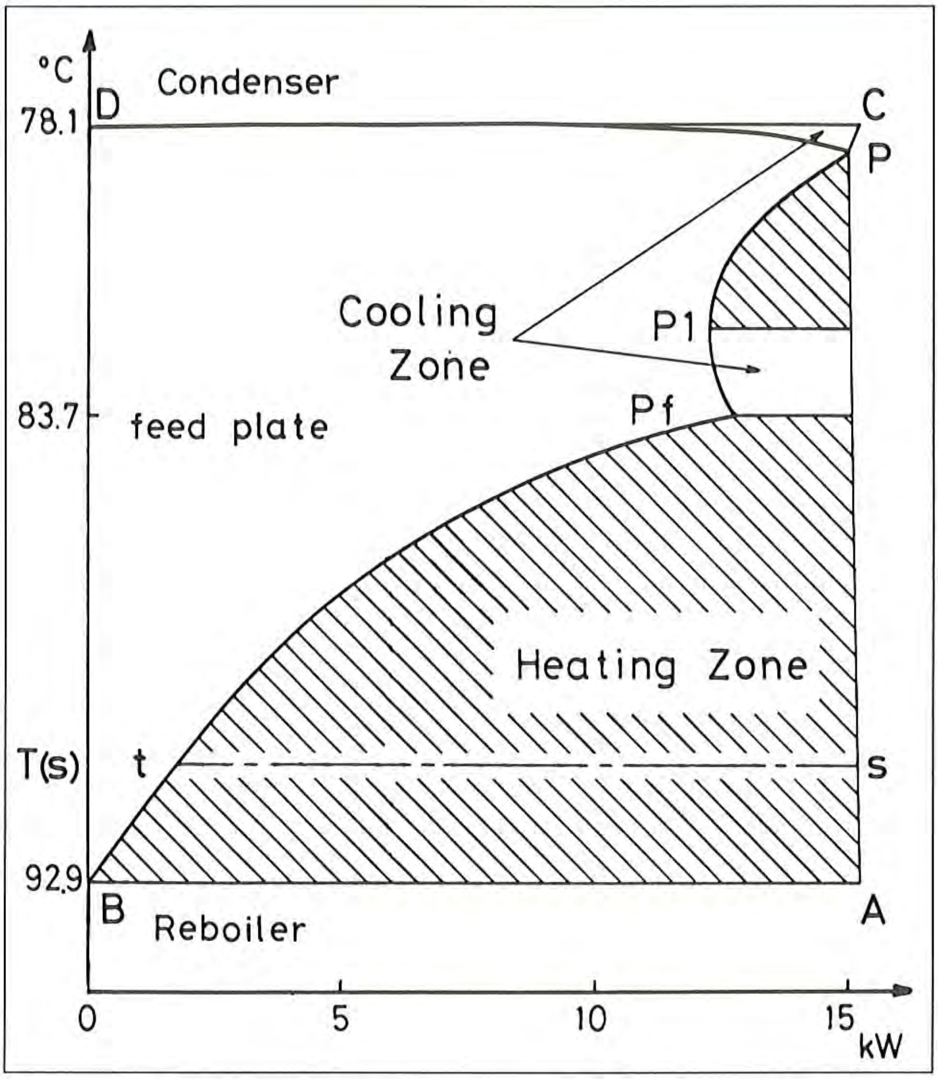

Figure 2. T-H diagram of a distillation column
(x_f = 17.7%, F = 2.48 kmol/h, q = 1)
(x_d = 82.5%, D = 0 .458 kmol/h)

1. The temperature difference between condenser and reboiler, which includes the minimum approach temperature of a heat exchanger, should be less than about 30°C, because a heat pump system with compression is expensive.
2. There are few suitable working fluids. When a combined system operates over 150°C, thermal decomposition of working fluids and super-heated conditions may occur at the outlet of a compressor. If available, it is very expensive. Due to these restrictions, a combined system cannot be widely used in a distillation system. In general, in designing a distillation system associated with a heat pump, we must investigate the following points:
   a. Where should a heat pump system be installed, taking into account restrictions due to existing heat pump technologies?
   b. How can a heat pump system save energy?
   c. If a working fluid is needed, what sorts of working fluids are suitable?
   d. How can a combined system be started up and controlled?

For the combined system, remaining significant problems center on operation, that is, start-up, dynamic behavior, operability, safety, and so on.

Thermodynamic analysis of a distillation system

A design method of a distillation system using a heat pump should evaluate the feasibility of creating a combined system with ease, and determine the design parameters and operating conditions so as to satisfy the given specifications. Recently, for design of a distillation system, the thermodynamic analysis has made advances in solving some of these problems.[2,3] With this analysis method, a distillation system has at least one heating zone and one cooling zone in its column.

A heating zone, in general, corresponds to a stripping section which is under a feed plate. The heating zone means that limited heat energy can be

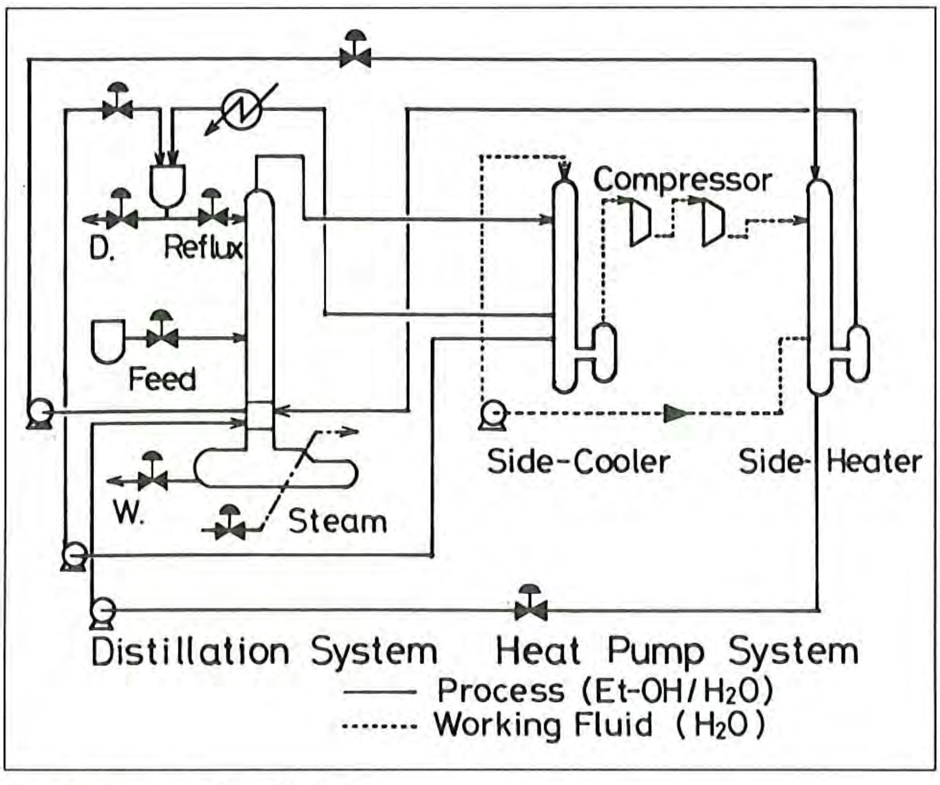

Figure 3. Flowsheet of the pilot plant

supplied to this zone for energy saving. Its maximum value is at a reboiler.

A cooling zone corresponds to an enriching section above a feed plate where limited heat energy can be removed. The maximum value appears at a condenser.

The pattern of the heating and cooling zones of an ethanol-water distillation system is shown in Figure 2 as an example. The figure is called the T-H diagram of a distillation system. This figure shows the relationship between temperature and an exchangeable heat energy in a distillation column. The shape of the pattern depends on the conditions of a feed stream, specifications of the top and bottom products. The heat energies CD and AB, respectively, correspond to heat loads of a condenser and a reboiler in a conventional distillation column. As shown in this figure, there are two cooling zones, from C to P and from P1 to Pf, and two heating zones, from P to P1 and from Pf to B. The heat energy can be supplied from the side of the column with temperature T(s) and heat energy supplied to a reboiler can be reduced from AB to (AB-st).

With the T-H diagram of a distillation system, it is possible to determine a suitable location for an intermediate plate where a heat exchanger in a heat pump should be attached, on the condition of an optimal feed position, and then easily design a heat integrated system.

General structures of a distillation system using a heat pump

A limited amount of heat energy can be removed from the cooling zone and supplied to the heating zone. Its temperature should be raised by means of heating and/or compressing. Combining two heat loads from both zones can save energy. In general, the schemes of heat exchange by use of a heat pump are classified into:

1. Supply of heat energy removed from a condenser to a side-heater, which is attached to an intermediate plate in the heating zone (C/SH, Condenser/Side-Heat type)
2. Supply of heat energy removed through a side-cooler, which is attached to an intermediate plate in the cooling zone to a reboiler (SC/R, Side-Cooler/Reboiler type).

		Case 2		Case 3	
		Exp.	Simulated	Exp.	Simulated
Feed comp.	(mol %)	6.30	---	17.71	---
flow rate	(kmol/h)	3.05	---	2.48	---
Top comp.	(mol %)	83.77	83.76	82.59	82.54
flow rate	(kmol/h)	0.14	---	0.46	---
Bottom comp.	(kmol/h)	2.69	2.66	3.02	3.03
flow rate	(kmol/h)	2.91	2.91	2.02	2.02
Total work	(kW)	0.211	(0.235) 0.809	0.244	(0.265) 0.916
W. fluid	(kmol/h)	1.29	1.29	1.47	1.48
H_E	(kW)	14.95	---	17.14	---
Side-cut temp.	(°C)	83.13	82.88	81.49	81.65
Steam	(kW)	15.08	11.17	15.70	13.70
Reflux ratio		14.61	---	4.46	---
EFF1	(%)	35.7	39.2	37.3	38.4
(centrifugal)		39.9	52.8	41.8	51.3
EFF2	(%)	43.9		45.7	
(centrifugal)		45.6		47.6	

The variables with the broken line are used for simulation. The value with the parentheses is calculated on the assumption that the vapor from the secondary compressor is saturated.

Table 1. Comparison between experimental and simulated data

Their selection can be easily carried out based on the T-H diagram. The heat exchange between a condenser and reboiler is one extreme of the above mentioned examples. The other is a type of heat exchange between two intermediate plates in the respective zones.

The C/SH (condenser/side-heater) system is discussed below. Two basic types of side-heaters can be designed, as follows:

1. A part of a liquid stream in a column is cut from an intermediate plate in the heating zone and the split stream is vaporized in a side-heater, which is denoted as C/T-SH (Condenser/Total Side-Heater) system.
2. A whole liquid stream in a column is withdrawn from an intermediate plate in the heating zone and fed into a side-heater. The process stream is partially vaporized by a working fluid. Both liquid and vapor streams are returned to the column, which is denoted as C/P-SH (Condenser/Partial Side-Heater) system.

In reference to the results of Naka, et al, we selected the C/P-SH system with ethanol-water mixture. The aim of this project is not only to develop such a

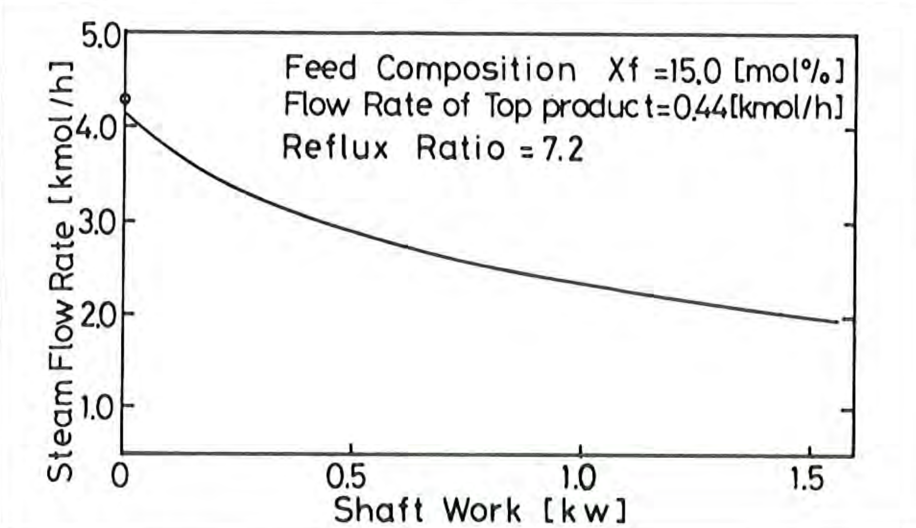

Figure 4. Electricity vs. steam supply

specific separation system but also to acquire data in order to establish a general design method of SC/H or C/SH systems and investigate their operable region.

Flowchart of C/P-SH system

The research project to investigate operability of a distillation system using a heat pump has built a pilot plant of C/P-SH system. The flowsheet of the plant is shown in Figure 3. This plant has been used to acquire basic data for design and control.

Distillation system. The packed column (the packing is CMR made by DODWELL) corresponds to a column with 11 ideal plates. The feed stream enters at the 8th plate from the top. Two types of condensers are equipped, a partial condenser with a working fluid and a total condenser with cooling water. The whole liquid process stream in a column is withdrawn from the 9th plate. The withdrawn stream is partially vaporized in the side-heater by the working fluid and both heated streams return to the same plate. The liquid stream flows down toward the reboiler. The steam is supplied to the reboiler. The reboiler can be used for start-up and shut-down operation, as well as normal steady state operation.

Heat pump system. The working fluid circulates around the partial condenser and the side-heater. Both heat exchangers are falling-film types. Water is used as a working fluid. It is vaporized in the partial condenser and is condensed in the side-heater. As the pressure in the partial condenser is lower than that in the side-heater, two rotary compressors are installed to pressurize the vapor. In general, this type of compressor is not suitable for heat pumps. But, this pilot plant has been built for investigation of its operability with a wide range of operating conditions. Therefore, rotary compressors were selected from the standpoint of feasibility and cost. A pressure reducer is added to the pipeline from the side-heater to the partial condenser.

Control system. There are 42 sensors of temperature, flow rate, pressure, liquid level, and density. All signals are transferred to a distributed control system (YEWPACK, Yokogawa Elec. Co.) and some of them are used for control of hold-up and product specifications.

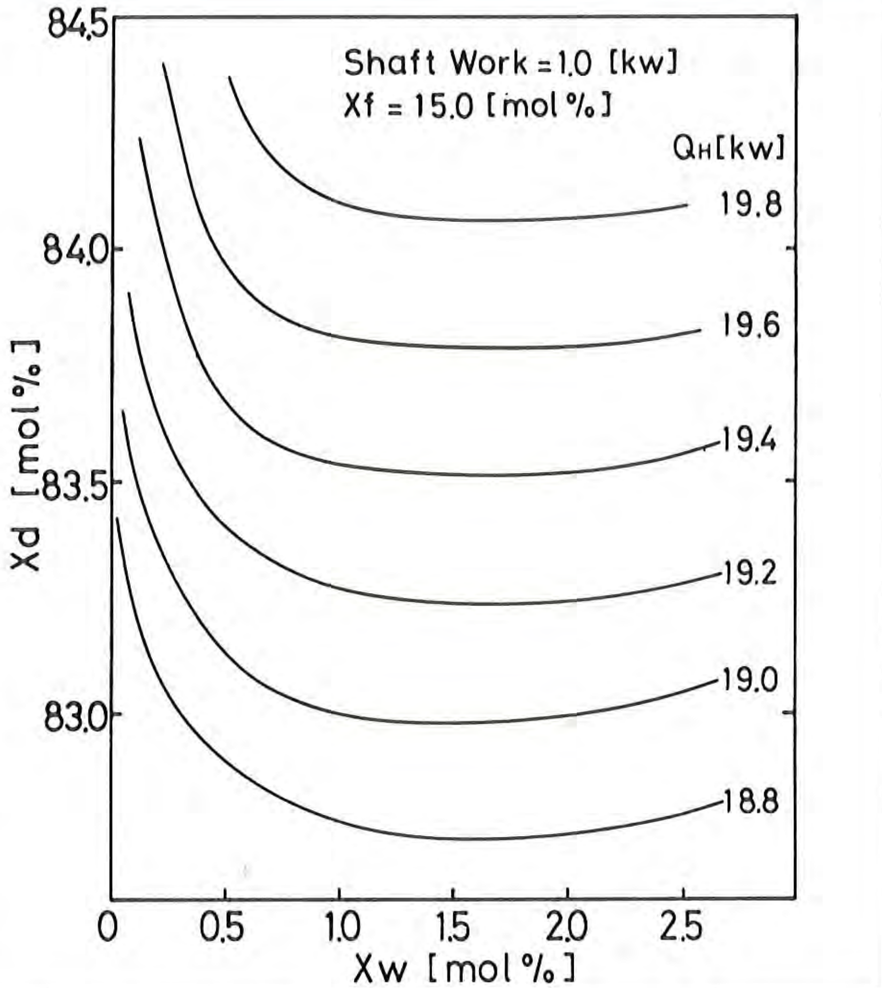

Figure 5. Exchanging heat load vs. product compositions

Operability studies on combined system (C/P-CH)

In the case of a conventional column, given the feed conditions, the top and bottom compositions and the operating pressure, no degree of freedom remains. On the other hand, in the case of introduction of a heat pump, the amount of electricity for compressor (shaft work) is added as a degree of freedom. For simplification of the problems, it is assumed that the operating pressure remains constant. Let's investigate the relationships among the exchanging heat energy in the side-heater, the steam flow rate and the product compositions based on the experimental data and simulation.

Table 1 denotes the comparison between experimental data and simulated values for the pilot plant. Since the consistency between both data is quite good, even though the operating conditions are changed, the following simulated results may be highly reliable.

Reduction of energy supply. The efficiencies of a heat pump system are listed in the same table. There are two different definitions, as follows:

1. Reduction of energy supplied to a conventional column, EFF_1

$$EFF_1 = 1 - (H_W + H_r)/H_{co}$$

H_w = Total electricity supply to a heat pump
H_r = Heat energy supplied to the reboiler
H_{co} = Heat energy supplied to a conventional column

2. Contribution of a heat pump to the total energy supply, EFF_2

$$EFF_2 = H_E/(H_E + H_r + W_L)$$

H_E = Exchanging heat energy in the side-heater
W_L = H_W - net shaft work

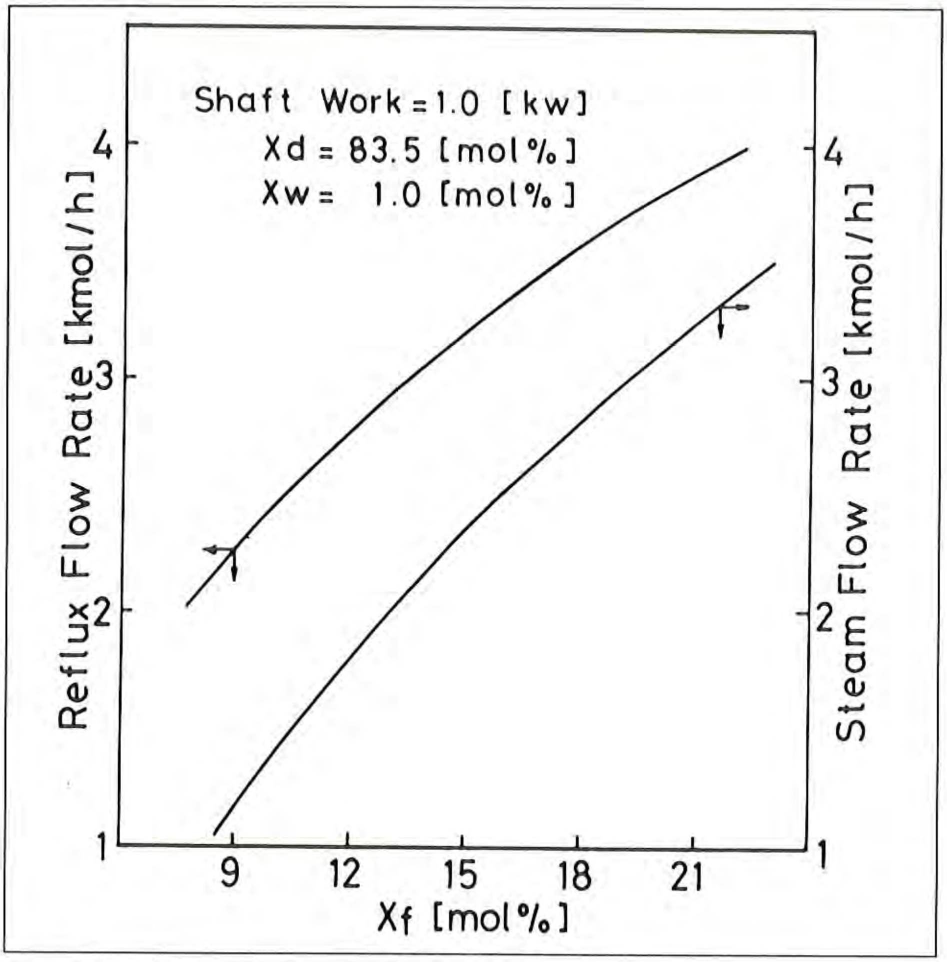

Figure 6. Feed composition vs. manipulative variables

EFF_1 is over 35%. The efficiency of a rotary compressor employed in the pilot plant is not higher than that of a centrifugal compressor. If a centrifugal compressor with an efficiency of 70% is available, EFF_1 may be about 40%.

Electricity vs. steam supply. As the quantity of electricity supply to the compressors increases on the condition that the reflux ratio (reflux flow rate/top product flow rate) is fixed at 7.2, the steam load gradually decreases, as shown in Figure 4. But the compositions of the top and bottom do not change.

Exchanging heat load vs. product compositions. Figure 5 shows that when the quantity of electricity is fixed at 1 kW, the product compositions affect the exchanging heat load in the side-heater. As the top composition of ethanol increases, the exchanging heat load increases. This is because when the top composition increases, the temperature at the side-cut plate becomes lower and the temperature difference between the partial condenser and the side-cut is smaller.

Change in feed composition. When the feed composition is changed, the manipulative variables of the steam flow rate and the reflux ratio should be operated so that the composition of the products remain at their specific values. Figure 6 shows the relation between the feed composition and manipulative variables. As the ethanol composition of the feed increases, the temperature at the side-cut plate trends downward and the exchanging heat load in the side-heater increases. Consequently, the steam load should be reduced as the ethanol concentration increases.

Control performance. A conventional control structure is installed in the plant and the rotation speeds of the compressors are kept constant. There are 9 control loops: 7 level controls, a top flow rate control, and a steam flow rate control. The control structure can realize very stable performance when the feed conditions change. But the settling time seems to be longer than that of a conventional distillation system.

Conclusion

The static characteristics of distillation using a heat pump were investigated based on simulation. The C/P-SH system can be operated within a broad range of feed composition changes and as it has a reboiler, it is easily started up. Though this article has not mentioned dynamics, its dynamic behavior was very similar to a conventional column in primary experiments.

The possible energy reduction can reach more than 35.7% (rotary compressor) and 40% (centrifugal compressor). The characteristics of the C/P-SH system were investigated and the experiments were carried out on the condition that the electricity supply is kept at a very low value. According to Figure 4, it may be possible to reduce by 60% (centrifugal compressor) the energy load of a conventional column.

This project will continue by applying the proposed systems to multi-component separation systems and developing an automatic start-up system.

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A. McMullan*

Heat Pump Integration - Pinch Technology's Role

A key factor for a successful heat pump installation is selection of the right application. It is important to ensure that a heat pump is the technically and economically correct energy cost reduction measure for the host process. Pinch Technology plays an important role in systematically identifying heat pumping opportunities and other competing options. This paper briefly reviews the Pinch Technology concepts relevant to heat pump application selection and highlights the use of Pinch Technology in generating other energy cost reduction options.

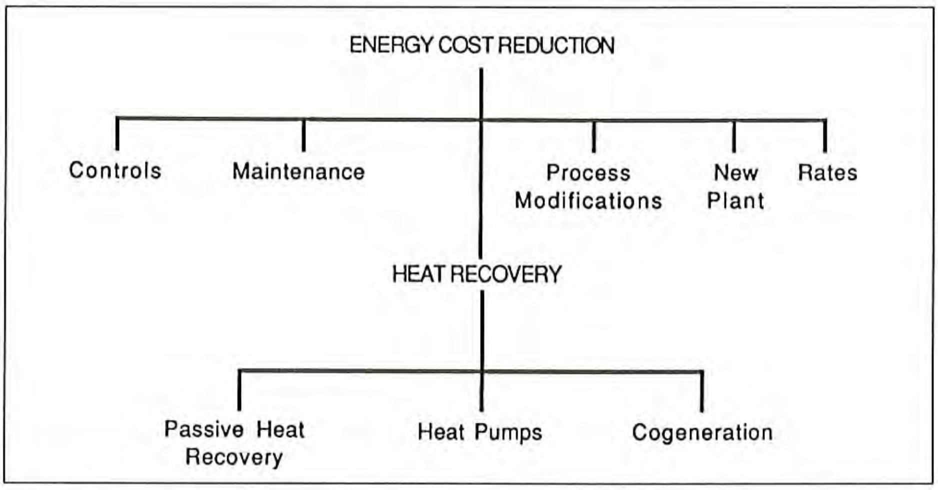

Figure 1

Introduction

There are many ways in which the energy efficiency of a process can be improved; the "Option Tree" in Figure 1 gives a broad overview. More specifically, the options are:

-- Improve maintenance, housekeeping, insulation, etc.

-- Modify the process to be inherently more energy efficient.

-- Install passive heat recovery equipment (e.g., heat exchangers).

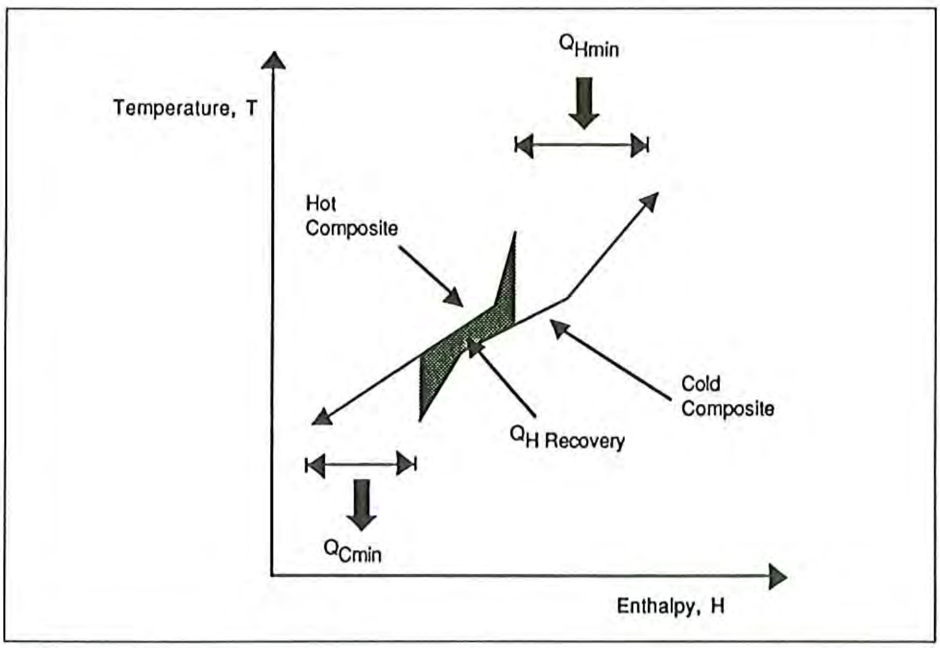

Figure 2

-- Install a heat pump.

-- Install a lower cost utility system (e.g., Cogeneration).

Pinch Technology[1] provides a thermodynamically consistent approach to evaluating the technical and economic merit of these options. It enables the effect of individual heat recovery measures to be understood in the context of the overall process rather than on a "local" basis. This is a very important feature because it ensures that an optimum combination of mutually compatible projects can be developed.

Pinch technology

Pinch Technology has developed a long way from its origins as an energy targeting and heat exchanger network design tool. It now provides an approach to process design which results in more energy efficient and lower capital cost designs[2]. Despite its wide ranging applicability, the technology is based on relatively few key concepts. These include:

-- Setting energy consumption and heat exchanger network capital cost targets for the process using "composite curves";

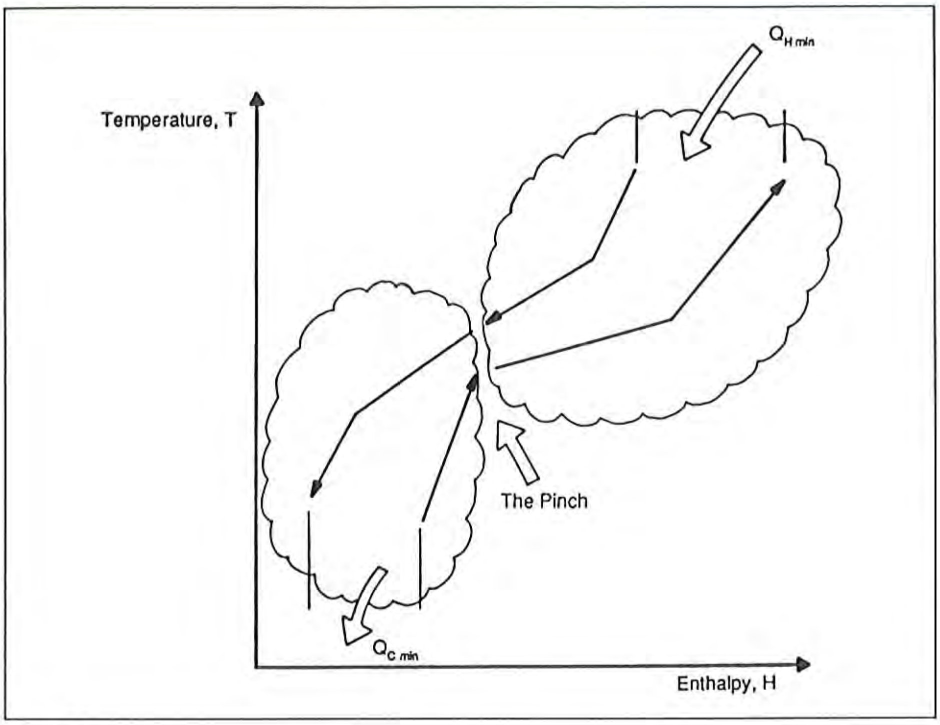

Figure 3

-- Modifying the process to make it inherently more energy and capital efficient using the "Plus Minus" principle;

-- Designing heat recovery networks to meet these targets using the "Pinch Principle" and "Pinch Design Methodology";

-- Integrating unit operations such as distillation and heat pumping in the process using the "Appropriate Placement Principle"; and

-- Selecting site utility systems using the "grand composite curve".

The Appropriate Placement Principle is of particular relevance to heat pump application selection. This, and the way in which Pinch Technology can be used to identify other energy cost reduction options is described in this paper.[3]

Energy targets and the pinch

Pinch Technology allows rigorous thermodynamic targets for passive heat recovery to be set. The first step in setting targets is collecting information about the process energy flows. Process streams are categorized as either cold (requiring heat) or hot (rejecting heat), their start and finish temperatures identified, and the associated enthalpy change determined. This information is compiled to form the composite heating and cooling curves shown on the temperature-enthalpy axis in Figure 2. The cold composite curve represents the cumulative heat demand of the process and the hot composite curve represents the cumulative heat supply of the process. The overlap between the hot and cold composite curves represents the maximum scope for heat recovery between process streams. At the hot end of the composite curves there is an "overshoot" of the cold composite curve where no corresponding hot streams are available for heat exchange. This overshoot is the minimum amount of utility heating required, Q_{Hmin}. Similarly, the overshoot of the hot composite curve at the cold end represents the minimum amount of utility cooling required, Q_{Cmin}. The difference between the actual energy consumption and the targets represents the scope to save through improved heat recovery.

The point at which the two curves comes closest together are known as the "Pinch." Recognizing the significance of the pinch is of prime importance when identifying projects that will reduce energy consumption to the targeted level.

Figure 3 shows how the pinch divides the process into two thermodynamically separate sub-systems. The subsystem above the pinch has a deficit of

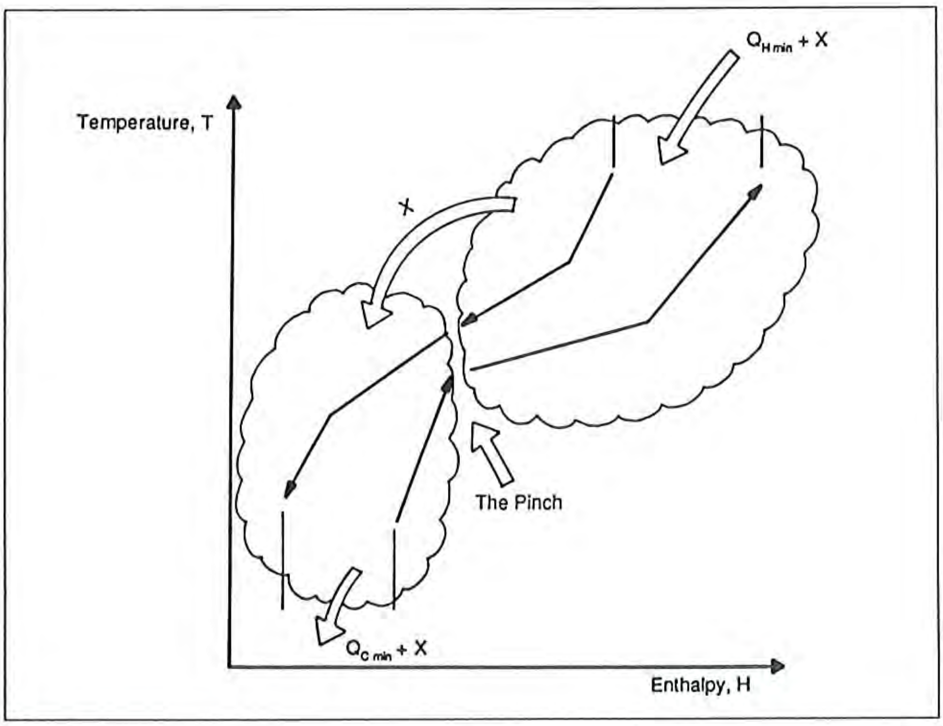

Figure 4

heat while that below the pinch has a surplus of heat. When the target hot and cold utilities are supplied, the two systems are in heat balance.

In general, processes consume more than the targeted amounts of energy. Why does this happen? Figure 4 shows a process consuming X more units of heat than necessary. To maintain energy balance above the pinch, these X units of heat must be transferred from the above pinch to the below pinch system. Similarly, to maintain energy balance below the pinch, the X units of additional heat must be rejected into cold utility. It follows from this argu-

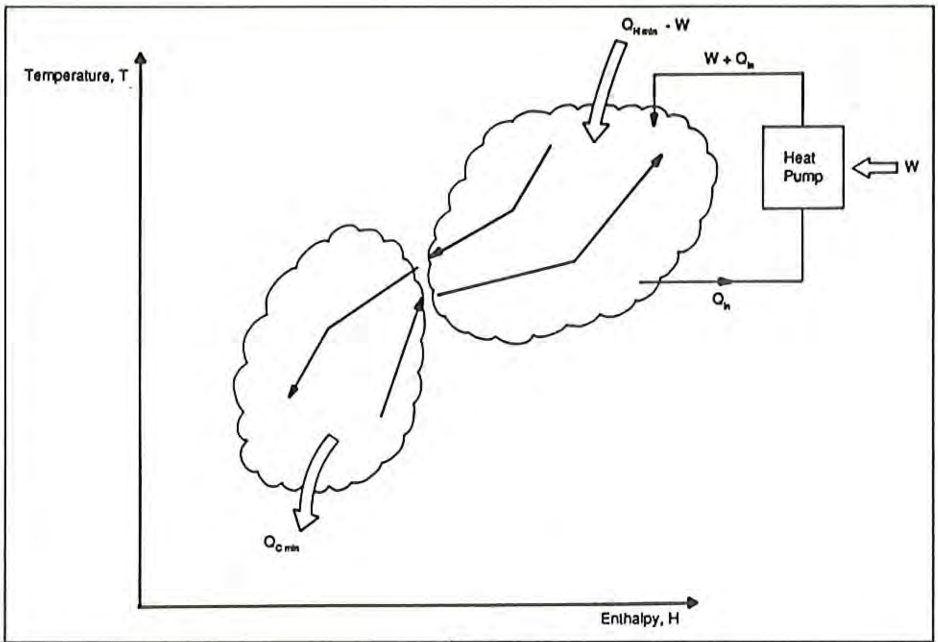

Figure 5

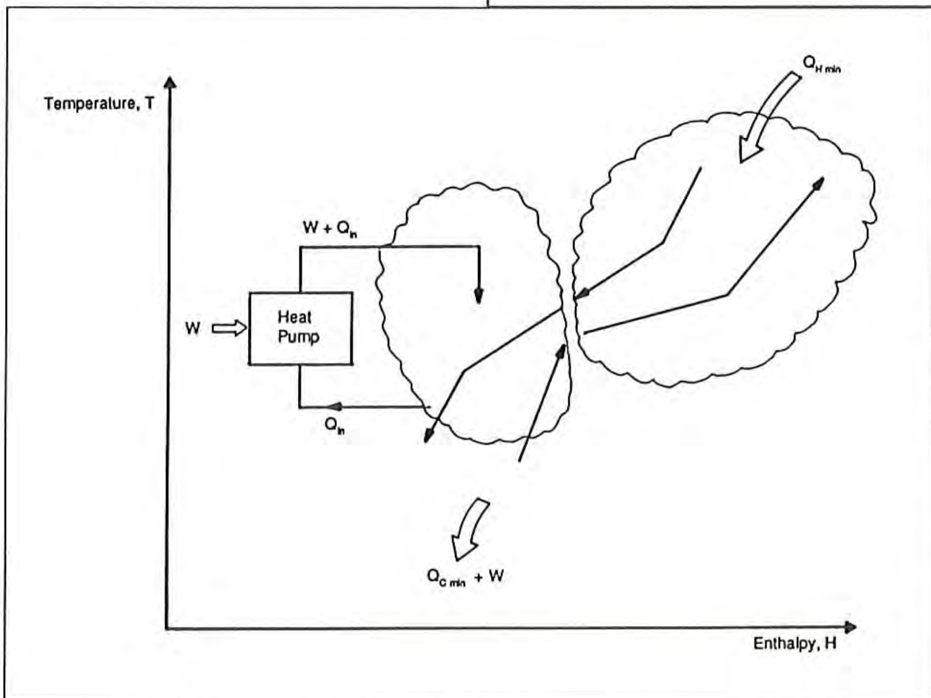

Figure 6

ment that to meet the energy targets, heat must not be transferred across the pinch. This conclusion is known as the "Pinch Principle."

Improved heat recovery

To design a heat recovery system which meets the energy targets , the above and below pinch systems are kept separate. The procedure for determining which streams to match is known as the "Pinch Design Method." It provides specific rules and tools and guarantees that the targets can be met.

Applying the Pinch Design Method to a process will generate a consistent set of heat recovery projects which can then be assessed for their technical and economic merit.

Heat pump integration

The Pinch Principle provides guidelines on how to integrate heat pumps within a process. Tools for determining the appropriate heat sources and sinks are also available.

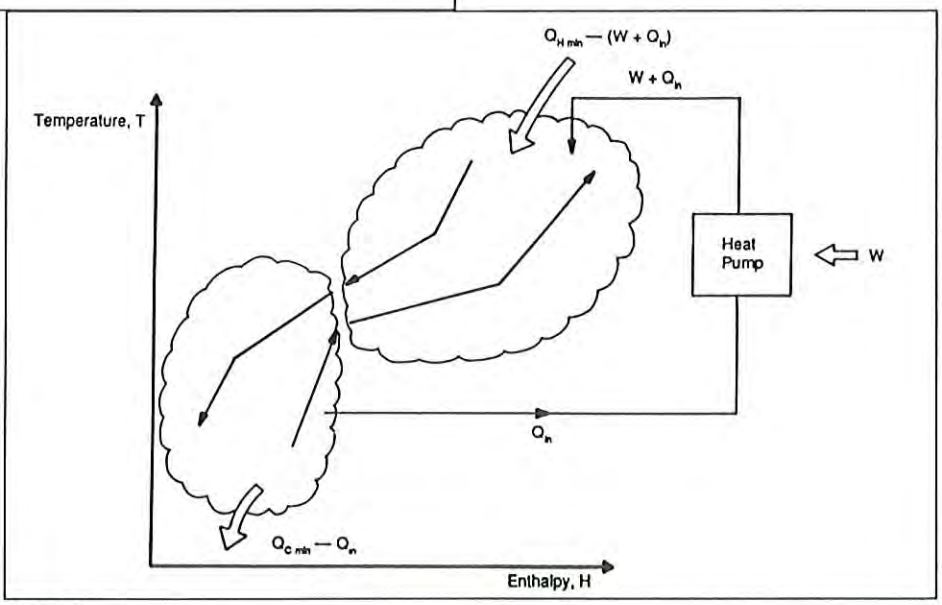

Figure 7

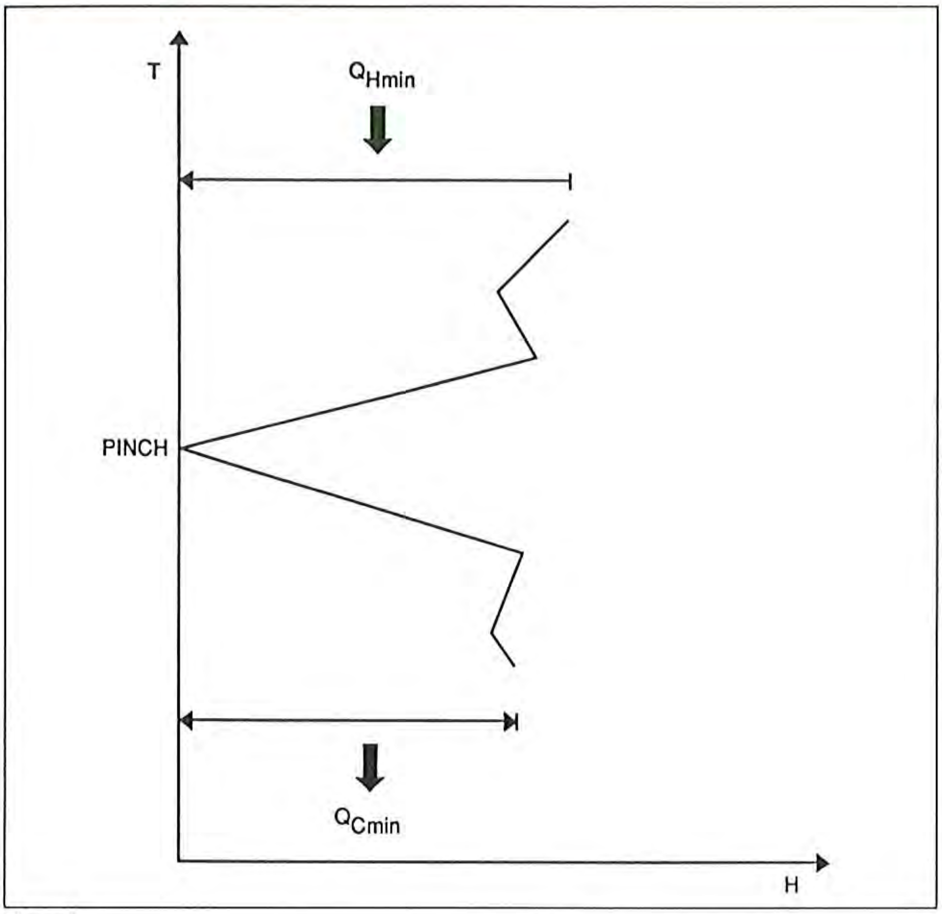

Figure 8

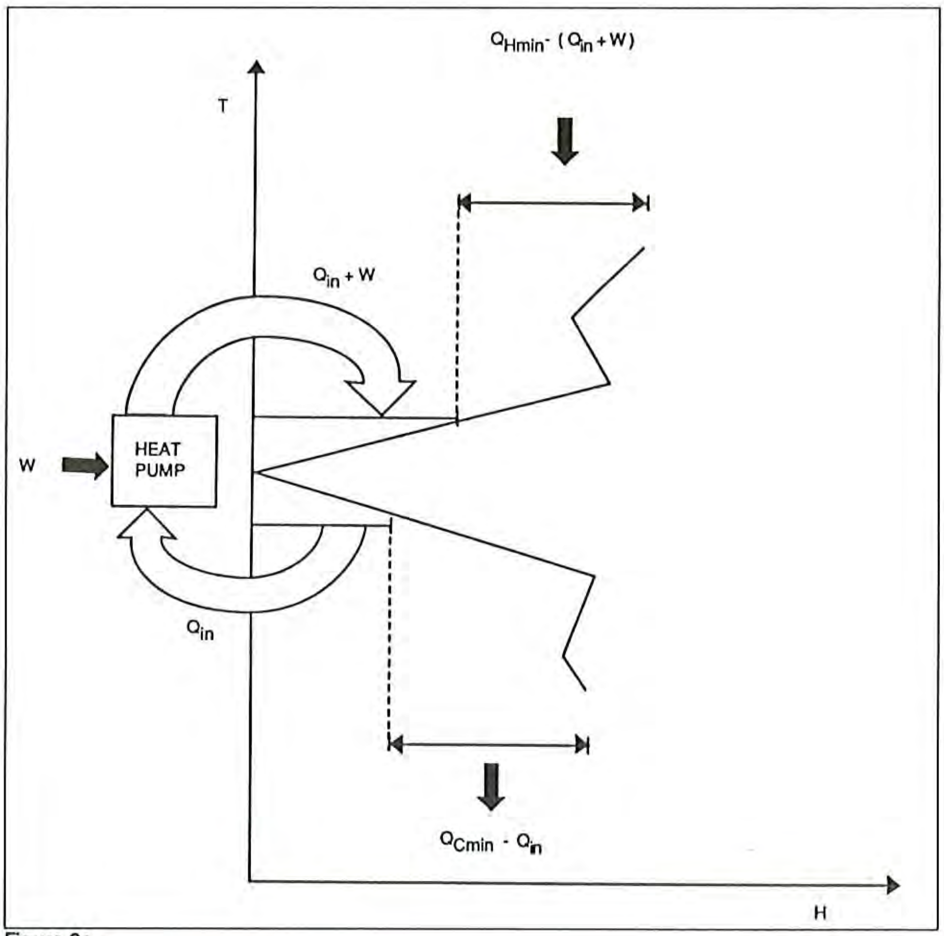

Figure 9a

A heat pump accepts heat at a low temperature level and delivers it at a higher temperature. For maximum thermodynamic benefit the source and sink must be correctly placed relative to the pinch temperature. Figure 5 shows a heat pump accepting Q_{in} units of heat above the pinch and using W units of work to deliver Q_{in}+W units at a higher temperature above the pinch. The effect of this is to reduce the hot utility target from Q_{Hmin} to (Q_{Hmin}-W); work has been degraded to heat. This is not usually cost effective because work is seldom less expensive than steam, so the conclusion is that a heat pump is not appropriately placed if both the heat source and user are above the pinch.

A similar argument for the heat pump in Figure 6 operating below the pinch shows that hot utility use is not reduced at all. In fact, the shaft work required to operate the heat pump is degraded into heat and actually increases the cold utility requirement. This configuration is also inappropriate.

An "appropriately placed" heat pump is shown in Figure 7. In this case, Q_{in} units of heat are supplied to the heat pump below the pinch and Q_{in}+W units of heat are delivered above the pinch. The effect of this is to reduce the hot utility requirement by Q_{in}+W, and the cold utility requirement by Q_{in}. Under these circumstances, the heat pump has a chance of being economically feasible.

This appropriate placement rule is of fundamental importance when selecting heat pump applications. When used with a tool called the "grand composite curve," a systematic approach to finding good heat pumping opportunities emerges.

The grand composite curve represents the heat demand and rejection profile of the process on a single curve. Figure 8 shows a typical grand composite curve. It clearly shows the minimum utility requirements and the location of the pinch. In addition, it shows the quantity of heat that must be supplied or rejected at each temperature level. This allows the operating temperatures and heat loads of heat pump evaporators and condensers to be determined.

Figures 9a and b illustrate two different heat pump possibilities that are shown by the grand composite curve. Each has different economics which can be screened quickly. It is important to realize that the grand composite curve already has the scope for passive heat recovery "built-in." Thus, the heat pump opportunities identified are compatible with passive heat recovery projects.

This example was based on a mechanical heat pump, but absorption systems can be analyzed just as easily.

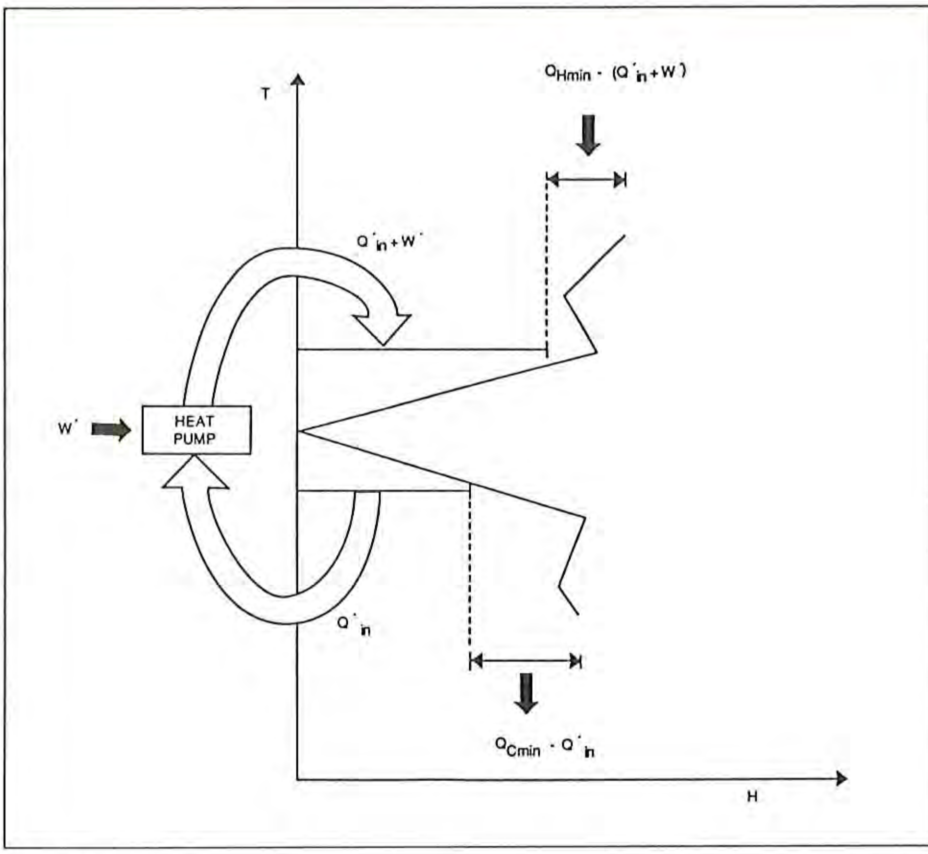

Figure 9b

Modifying the process

In addition to giving a graphical representation of the energy targets, the composite curves can reveal modifications to the process conditions (i.e., temperatures, flows, etc.) which will increase the scope for heat recovery. In Figure 10, the process is shown divided at the Pinch. To reduce the hot utility target, Q_{Hmin}, the only options are to i) reduce the heat duty of cold streams above the pinch (-) or ii) increase the heat content of hot streams above the pinch (+). Similarly, the only way in which the cold utility target, Q_{Cmin} can be reduced is by either reducing the heat content of hot streams below the pinch (-) or by increasing the heat duty of cold streams below the pinch (+). This is known as the plus/minus principle. Modifications suggested by the composite curves can then be translated into changes in equipment design or process operations. Examples of such changes include increased pumparound rates and increased condenser pressure (temperature).

Utility system design

The configuration of a site utility system can have a significant impact on the cost of energy supply which, in turn, affects the savings available from heat recovery or heat pumping projects.

The grand composite curve can be used to identify utility systems which most clearly match the process heating requirements. Figures 11a and b illustrate this. In Figure 11a, a high pressure steam is used to satisfy the total process heating demand. However, the grand composite curve shows the presence of a low level heat sink. Figure 11b shows how this can be exploited for power generation, resulting in a lower utility cost than in the previous case. It is possible to explore many

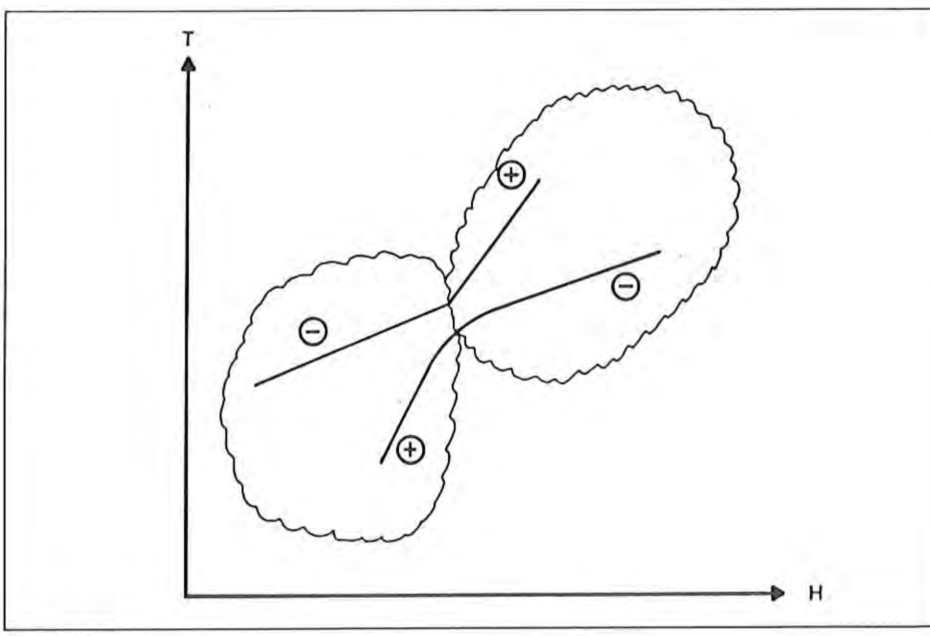

Figure 10

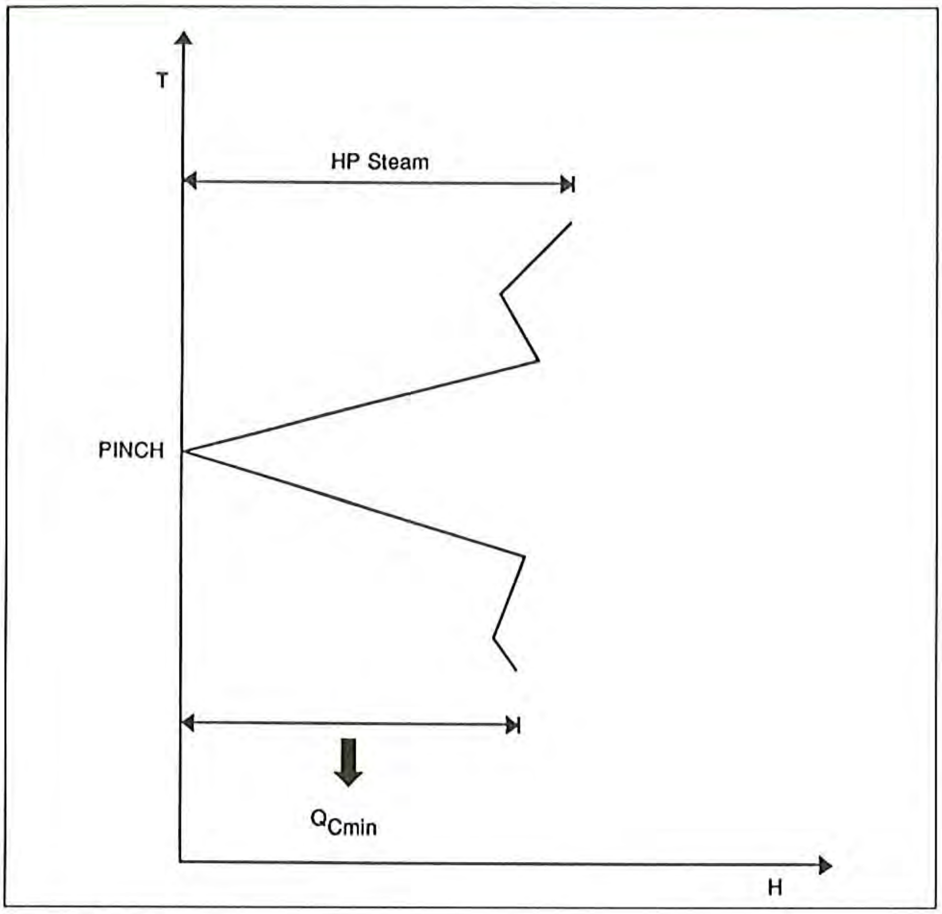

Figure 11a

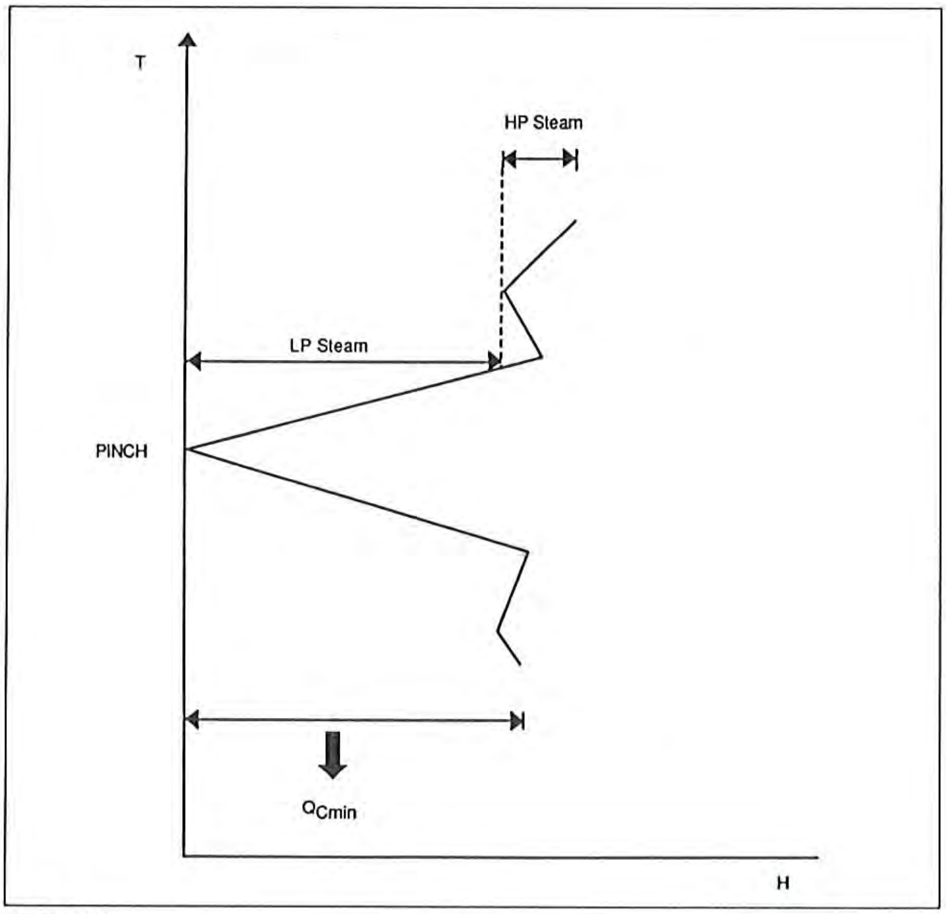

Figure 11b

different types of utility systems using this approach.

Conclusions

Pinch Technology provides a thermodynamically consistent approach for identifying energy cost reduction options. The insights into heat pump placement and sizing are particularly important for application selection. Without a knowledge of the pinch location, it is impossible to ensure that a heat pump is appropriately placed.

Acknowledgement

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More details can be found in EPRI publication EM-6057, "Industrial Heat Pump Manual", available from the Research Reports Center (RRC), Box 50490, Palo Alto, California 94303.

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Double-Lift Compression-Absorption Heat Pump

An R22-compression refrigeration system with an integrated R22-DMETEG-absorption cycle is described. This compression-absorption heat pump is capable of simultaneous production of refrigeration at, for example, -20°C and hot process water at 80°C. Even temperatures of 120°C for producing steam can be attained. The test system has a capacity of 5 kW for refrigeration. Projected load curves for flexibility of operation are discussed.

Introduction

Up to now, to provide simultaneous refrigeration and process water with a compression heat pump, cascaded systems with two compressors and two working fluids have been used. Working fluids like R114 for the high temperature stage must eventually be phased out due to the ozone depletion problem. As an alternative a high temperature heat pump with only one compressor, but an additional absorption stage, has been proposed.[1,2,3] This compression-absorption heat pump is capable of a temperature lift of more than 100°C. It is adjustable to varying demand with regard to capacity and temperature lift, and can be operated with non-ozone depleting working fluids like ammonia. A prototype with a 5 kW refrigeration capacity built of standard refrigeration equipment is being tested (Figure 1). The working pair is R22-DMETEG.

The double-lift compression-absorption cycle

Figure 2 shows the principle of the cycle on a pressure/temperature scale. At the evaporator at about -20°C R22 is evaporated producing refrigeration Q_o. The vapor is compressed from about 2.5 to about 20 bar consuming work W_1. The vapor is condensed at about 50°C releasing heat $Q_1 + Q_x$. The portion Q_x of this heat is not rejected but is fed into the generator G. The heat is used to generate R22 vapor from an R22-rich R22/DMETEG solution at a low pressure of 2.5 bar. This vapor is sucked off by the compressor as well and delivered at 20 bar consuming work W_2. The R22-poor solution is pumped to 20 bar consuming work W_3 which is small compared to $W_1 + W_2$.

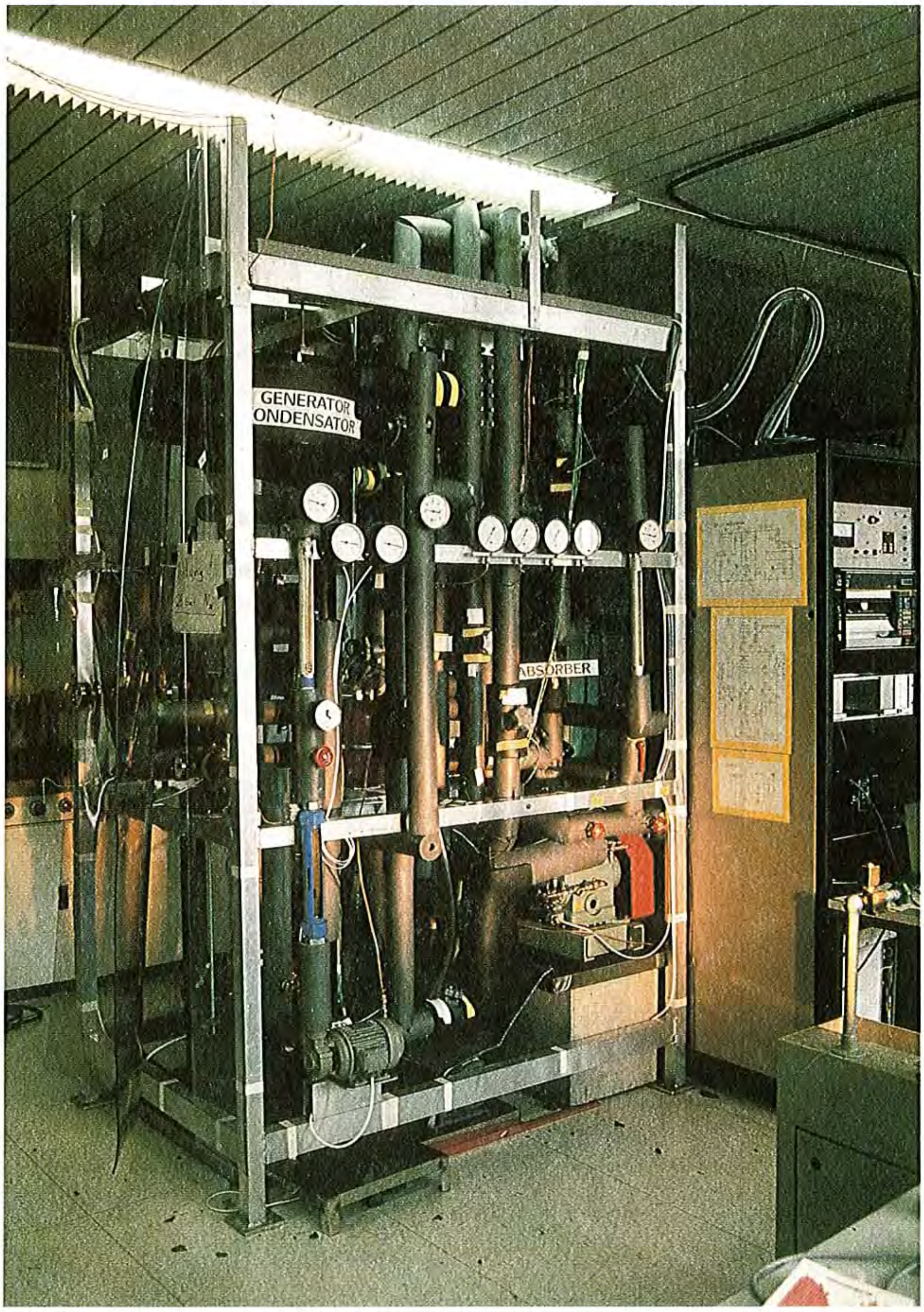

Figure 1. Prototype of the double-lift compression-absorption heat pump with 5 kW capacity for refrigeration

The solution is preheated in the solution heat exchanger. It then absorbs R22 vapor in the absorber, rejecting heat at a high temperature of for example 85°C. The R22-rich solution is cooled in the solution heat exchanger and then throttled to generator pressure, where the R22 vapor is desorbed again. So the vapor leaving the compressor is partly condensed and partly absorbed. The ratio of the vapor condensed to the vapor absorbed can be varied.

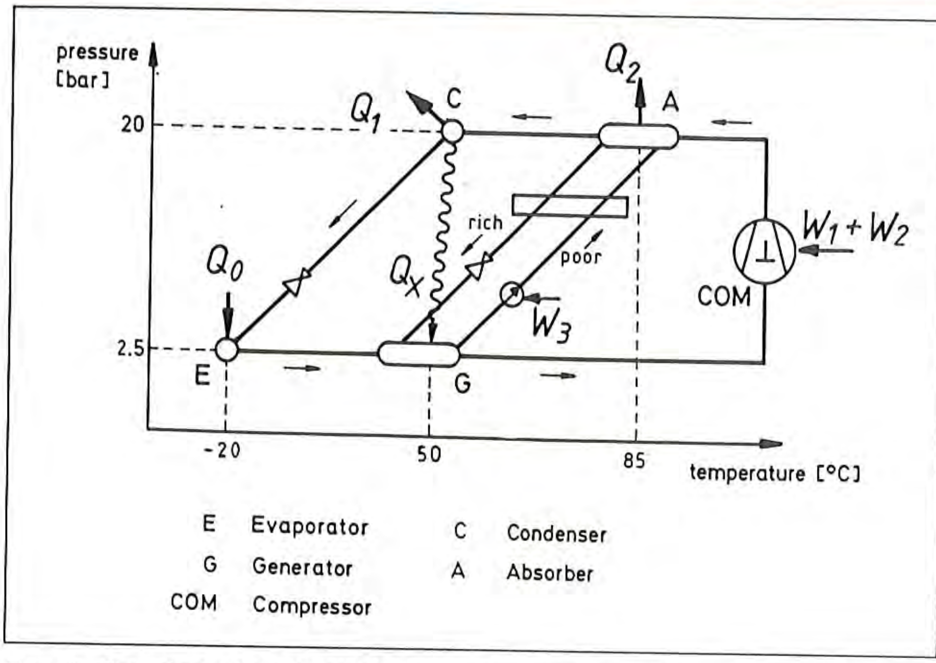

Figure 2. Flow scheme of the double-lift compression-absorption cycle

If no heat Q_1 is rejected at the condenser but all of the condensation heat is recovered as Q_x at the generator, the machine produces a temperature lift of about 100 K with a single compressor, a moderate pressure ratio of about 8 and a conventional condenser pressure of 20 bar.

Discussion

The main purpose of the cycle is to produce a high temperature lift in order to provide refrigeration and process water with one machine. But since the required heating and cooling capacity will not always match with the output of the machine, and the required temperature of the process water will vary, a certain flexibility is of importance. This flexibility is given by the possibility of changing the ratio of the amount of vapor condensed to the amount of vapor absorbed. By this means it is possible to adjust the amount of heat Q_1, which is rejected from the condenser to the cooling water. We define a heat recovery factor R, which denotes the ratio of the heat Q_2 rejected from the absorber and the total heat rejected from both the condenser Q_1 and the absorber Q_2:

$$R = \frac{Q_2}{Q_1 + Q_2}$$

To get rough figures for load management and efficiency of the system quickly, it is possible to model the compression-absorption cycle as a superposition of a compression cycle and a resorption-compression cycle similar to the modeling of advanced absorption cycles.[4] Doing this in a conservative way we arrive at the curves of Figure 3. Here the ratios of source heat Q_o, condenser reject heat Q_1 and absorber reject heat Q_2 to the required work $W = W_1 + W_2 + W_3$ are plotted against the recovery factor R. The ratio $Q_o/W = COP_{Refr}$ is the coefficient of performance of the refrigerator. It starts off with its highest value of about 1.7 with all heat rejected from the condenser (R = 0) and falls to 0.5 if all condenser heat is recovered and boosted to the absorber (R = 1)

The heat rejected from the absorber Q_2 and the condenser Q_1 divided by the required work is of course equal to $COP_{Refr} + 1$, and can be split into the part Q_1/W rejected from the condenser, which falls with R, and Q_2/W rejected from the absorber, which is, of course, increasing with R.

Since the work input at a given displacement is almost independent of the heat recovery factor R the curves in Figure 3 represent the absolute amount of heat flow as well as the ratios of heat to the work input. They thus can be used for load management. As an example we consider the case of process water, which has to be heated up from tap temperature of 15°C to 80°C. We do

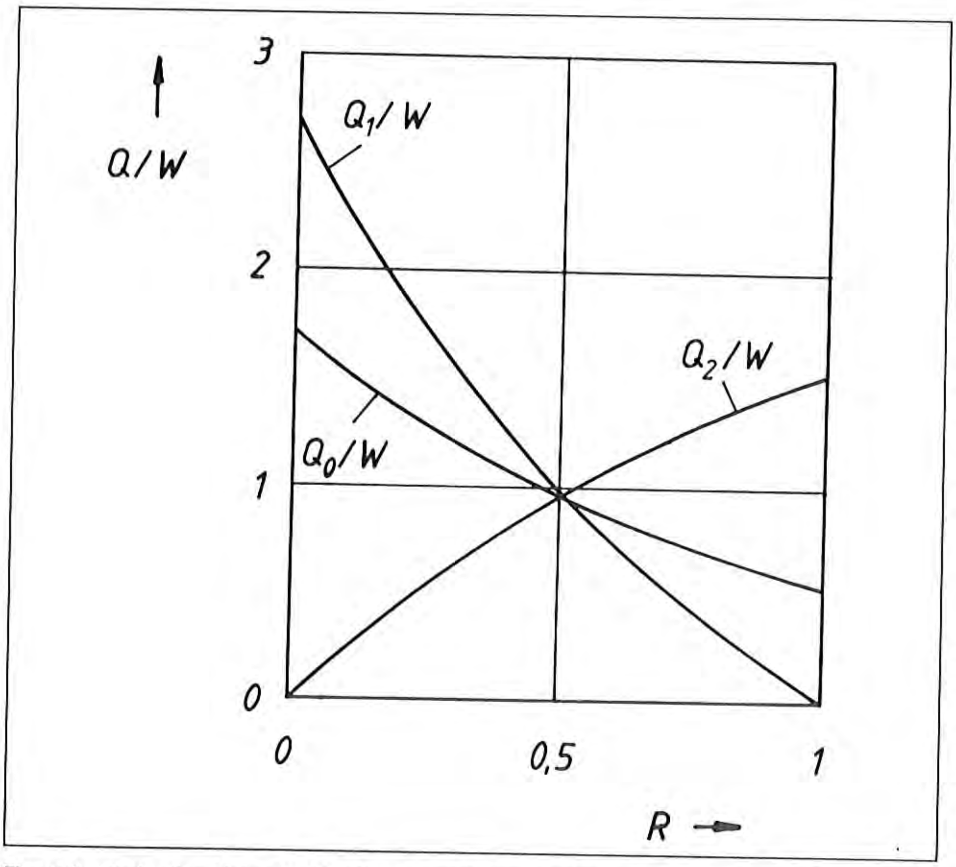

Figure 3. Heat ratios of the double-lift compression-absorption cycle plotted against the recovery factor R = Q_2 (Q_1 + Q_2)

this in two steps. In the condenser the water is preheated with heat Q_1 to 45°C, and in the absorber it is heated from 45°C to 80°C. So the recovery factor R has to be adjusted to:

$$R = \frac{80-45}{80-15} = \frac{35}{65} = 0.54$$

From Figure 3 we find for the COP for refrigeration $Q_o/W = 0.9$. The ratio between refrigeration capacity and heating capacity is $Q_o/(Q_1 + Q_2) = 0.48$ for this application.

Thus we have a refrigeration system working at -20°C which is easily capable of producing process water of 80°C with a fairly good COP using conventional copper components and "safety working fluids." Temperatures of 120°C for producing steam should also be possible.

If instead of the working pair R22-DMETEG the thermodynamically favorable pair NH_3-H_2O would be chosen, the COP of the system could be improved by at least 10%.

Conclusion

An absorption-compression heat pump working with R22-DMETEG is being tested at the Physics-Department in Munich. Its features are a refrigeration capacity of 5 kW at -20°C with simultaneous heat output for producing process water of at least 80°C. The COP is estimated to be about 1 for refrigeration under these conditions. Temperatures of 120°C should be attainable.

The test facility consists totally of commercially available refrigeration equipment with the exception of the solution pump, which is a membrane type. So high temperature heat pumping with simultaneous refrigeration using compression machinery is possible with conventional components and safe low ozone depleting working fluids.

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W. Kern*

A Two-Stage Absorption Heat Pump for Heating and Cooling

The "double-effect" two-stage heat pump with the working pair water/LiBr is very reliable and widely used in Japan as a highly efficient chiller. With small modifications it is possible to operate this device as a "double-lift" heat pump. In this mode it provides simultaneous heating and cooling with an improved efficiency compared to a conventional chiller and furnace system.

The two-stage heat pump

The high efficiency "double-effect" chiller (Figure 1) provides cooling capacity at a temperature of about 4°C. The COP for cooling reaches values of about 1.25,[1] which is a great improvement compared to the value of 0.7 for the single-stage chiller.

Due to the limits set by crystallization of the salt the "double-effect" chiller has the same temperature lift as the single-stage machine. The heat is rejected at a temperature of about 30°C which is too low for heating purposes. If higher temperature heat is required the conventional "double-effect" machine is switched into a pure heating mode with a COP for heating of about 0.9.

In Figure 2 the "double-lift" cycle is shown, which has the same number of heat exchangers but different connections. The advantage of this cycle is the simultaneous production of cooling capacity at 4°C and heat output at high temperatures, for example 70 to 100°C. For this "double-lift" two-stage heat pump a COP for cooling of about 0.3 and for heating of about 1.3 can be expected.[2]

So, for example, if 300 kW of cooling capacity and 1300 kW of heating capacity is required, the total energy consumption is 1000 kW by using a "double-lift" heat pump. By using the "double-effect" chiller and a conventional burner for the heat, 240 kW for the chiller and 1440 kW for the heater is required, which is more than 1.6 times the energy consumption of the "double-lift" heat pump.

Comparing Figures 1 and 2, one notices that both types of heat pumps have the same number of heat exchangers. So the two cycles can be looked at as two modes of operation of one heat pump. Changing the mode is accomplished by switching valves. Therefore, only one device is necessary to meet both cooling and heating demand; this device is slightly more complicated than the cooling device alone.

In many applications a temporary need for hot water is added to a basic cooling load. So this device can operate as a highly efficient chiller to meet the basic load and can be switched into a high

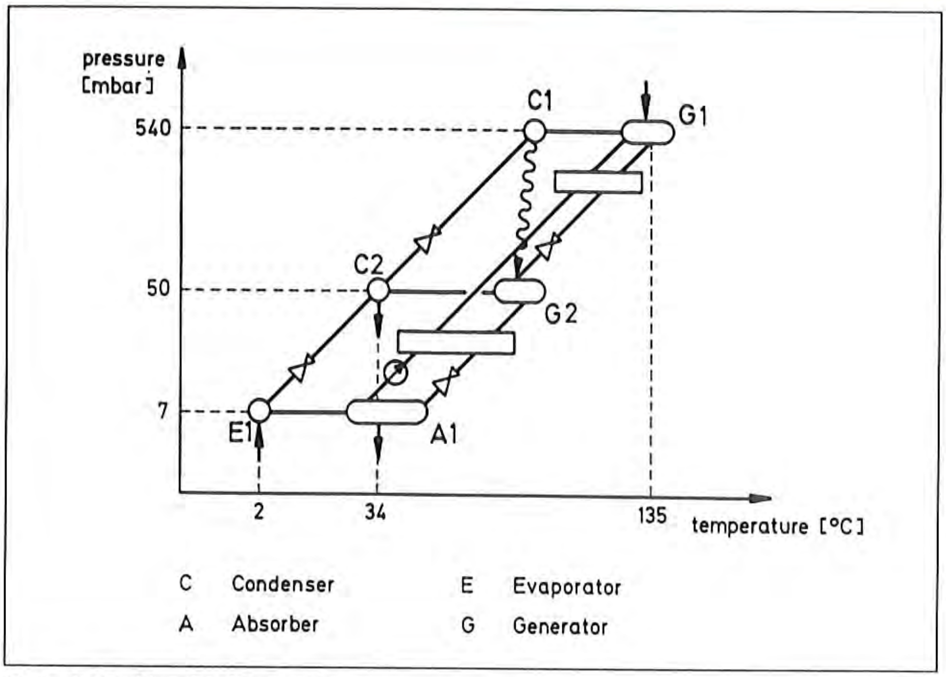

Figure 1. Double-effect heat pump

temperature lift mode, using the heat pump effect, when hot water is demanded.

This switchable heat pump may be used for air conditioning and hot water supply, e.g., in hotels or other large buildings. In other applications the two-stage heat pump can be used in summer as a chiller and in the heating season as a heat pump. The running time is increased and consequently the economics are improved.

Experimental set-up

To acquire information on the stability of operation and the performance of a two-stage heat pump we have built an experimental set-up. In both modes of operation the generator 1 at the highest pressure level is driven by steam up to 170°C and the cooling capacity of the evaporator is used to produce chilled water of about 4°C. In the "double-effect" mode there is a heat exchange between the condenser 1 working at the highest pressure level and the generator 2 at the intermediate pressure. Because of the low temperature of the heat output a heat sink at 30°C, normally cooling water, is required. In the "double-lift" mode the internal heat exchange works between the absorber 1 at the low and the evaporator 2 at the intermediate pressure level. Now the heat is rejected from the condenser and the absorber 2 at temperatures high enough for heating purposes. Switching between the two modes of operation is realized by manipulating heat transfer fluids only. The solution circuit is not changed at all. The maximum input power is 18kW. So we can reach about 6kW cooling capacity and 24kW heat output in the "double-lift" mode and about 25kW cooling capacity in the "double-effect" mode.

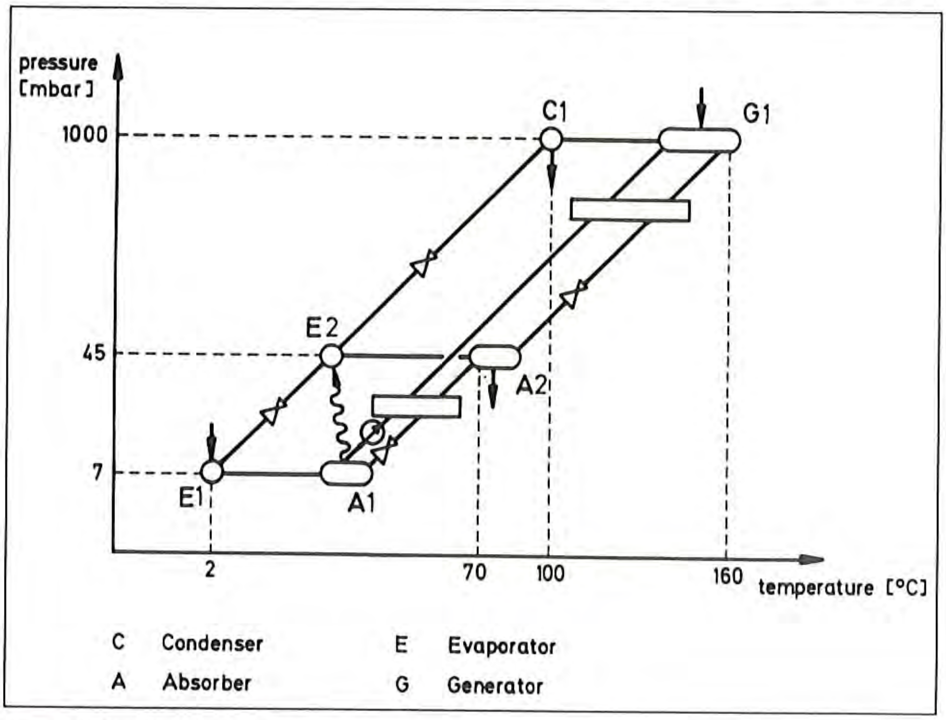

Figure 2. Double-lift heat pump

The measured COP for cooling of the "double-lift" cycle increases from 0.26 to 0.34 with decreasing temperature lift.[3] Also a very good partload behavior has been found.

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H. Kita*

Heat Pumps in Hothouse Horticulture

In the field of hothouse horticulture, an increasing number of heat pumps have been installed because (a) they are more energy saving than conventional hot water boilers and space heaters; (b) they are more fire resistant; and (c) they can satisfy various requirements for timely shipments along with the upgrading of items cultured. In view of this, the author has surveyed actual installation examples to date and presents the results of evaluation experiments.

General situation

Reflecting the growing requirement for high quality products, an increasing number of horticulture hothouses have been installed in recent years, showing an average yearly growth of 4%. About 40% of all these hothouses come with heating equipment, such as hot water boilers and space heaters.

In recent years, heat pumps have been used for hothouse horticulture, since they are energy saving, fire resistant, and can perform various functions including cooling and dehumidifying, making them suitable for maintaining growing conditions for high-quality produce and decorative plants.

In this report, the state of this heat pump application in Japan is outlined with special reference to an actual installation example of an air-source heat pump in Miyazaki Prefecture.

Hothouses using heat pumps in Japan

Table 1 lists representative hothouses utilizing heat pumps that are currently in commercial operation in Japan. These hothouses are either covered with glass or vinyl. The vinyl-covered hothouses are mainly installed in mild districts, such as Hochi and Miyazaki, and for vegetable culture.

Water-source heat pumps utilize underground water or river water as their low-temperature heat source so that operation is possible even when outdoor temperatures are low. Air-source heat pumps are used in mild districts only when water resources are not available.

In addition to the examples shown in Table 1, heat pumps are being utilized to maintain hothouse temperatures at an adequate level to adjust the flowering time of orchids (introduced from the West) and also the fruiting time of strawberries.

An example of hothouse using an air-source heat pump[5]

Described here are the details of the hothouse (see cover photo) in Miyazaki Prefecture shown in Table 1 and various measured values. Its specifications are as shown in Table 2. Here cucumbers are cultured. Since the optimum night-time temperature for cucumbers is 10 to 15°C, the target control temperature is set at 13°C (Table 3).

In Miyazaki Prefecture, the average night-time temperature in winter is 7.4°C, the average temperature in January (lowest temperature month) is 4.4°C, and the lowest temperature is -

Area of hothouse m²	Type of hothouse	Items cultured	Heat pump system	Compressor output (kW)	Number of unit(s)	Prefecture	Remarks
2,670	Vinyl	Green pepper	Water-source heat pump chiller	30	1	Kochi(1)	
300	Glass	Orchid introduced from the West	Water-source chiller	30	1	Okayama(2)	
670	Glass	Mandarin orange	Water-source heat pump package	7.5	2	Tokushima(3)	Energy saving 30%
520	Glass	Flowering plant	Water-source heat pump package	11	1	Tokyo(3)	Energy saving 400 k¥/year
1,000	Glass	Melon	Water-source heat pump package	7.5	2	Chiba(3)	Year-round culture
3,300	Vinyl	Green pepper	Water-source heat pump chiller	15	2	Kochi(3)	Energy saving 35%
670	Vinyl	Green pepper	Air-source heat pump chiller	3.7	1	Kochi(4)	Heat storage system
830	Glass	Green pepper	Water-source heat pump chiller	7.5	1	Kochi(4)	
500	Vinyl	Flowering plant	Water-source heat pump chiller	15	1	Kochi(4)	
2,670	Vinyl	Cucumbers	Air-source heat pump chiller	11	2	Miyazaki(5)	Energy saving 10%; heat storage system

Table 1. Examples of hothouses using heat pumps

Type	Joined pipe house
Frontage x depth x number of joined houses	5.4m x 5.4m x 9 houses
Eaves (height of ridge)	2m (3.5m)
Structure cover material	Agricultural PVC film 0.1mm
Heat retaining material	Agricultural PVC film 0.1mm (1 layer)
Air conditioning heat source unit	Heating by air-source heat pump chiller (auxiliary boiler also available)
Air conditioning system	Vertical fan-coil duct connected

Table 2. Hothouse specifications (in the heating mode)

3°C. Due to these conditions and poor availability of water resources, an air-source heat pump chiller and a heat storage tank are provided to perform heat storage and heating by the heat pump down to outdoor temperatures of 4°C. When the outdoor temperature is below 4°C and heat storage becomes deficient, an auxiliary boiler is used for heating.

Figure 1 shows the plan view and Figure 2 shows the flow sheet of this equipment.

Results of operation and examination

At the temperatures mentioned in the preceding section, cucumbers were cultured and various values were measured between November 15, 1986 and February 28, 1987. Although the outdoor temperature dropped to -2.5°C at its lowest, the temperature in the hothouse could be maintained at 13°C. The hours of operation in January, the lowest temperature month, were 221 for the heat pump and 102 for the auxiliary boiler. During the experimental period, the heat pump was operated for a total of 730 hours.

Figure 3 shows the relation between outdoor temperature and the heat pump's COP, based on the experimental results. From these results, the cost of operation incurred when a boiler is used instead of a heat pump was tentatively calculated and compared to the actual cost of operation. Table 4 summarizes the results. As shown, a 10% reduction in cost has been achieved.

As for the influences upon the crop during culture, the following findings were obtained:

1. Since the exit temperature of the heated air was low and the temperature difference was small, there was less trouble due to high temperature and downy mildew due to wetting leaves.

Crop		Daytime temp.		Nighttime temp.	
		Highest limit	Suitable temp.	Suitable temp.	Lowest limit
Eggplant family	Tomato	35	25-20	13-8	5
	Eggplant	35	28-23	18-13	10
	Green pepper	35	30-25	20-15	12
Melon family	Cucumber	35	28-23	15-10	8
	Watermelon	35	28-23	18-13	10
	Hothouse melon	35	30-25	23-18	15
	Muskmelon	35	25-20	15-10	8
	Pumpkin	35	25-20	15-10	8
Strawberry		30	23-18	10-5	3

Table 3. Temperatures suitable for vegetable growth and limit temperatures (°C) (Takahashi 1977)

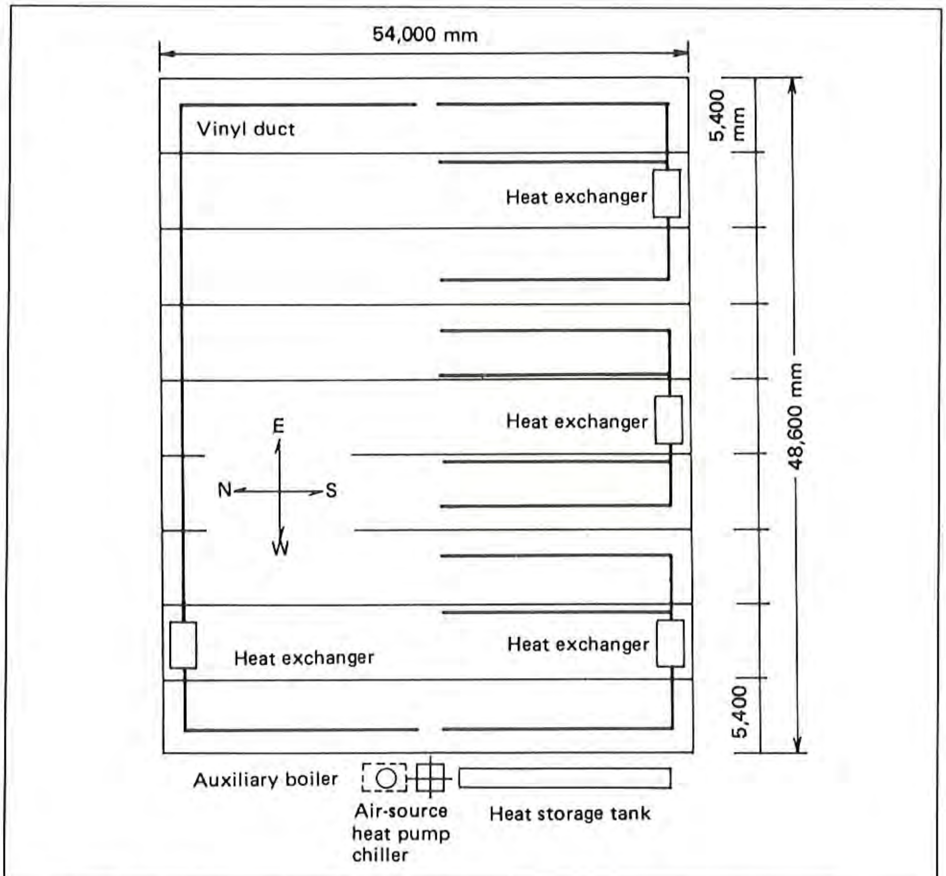

Figure 1. Plan of the equipment

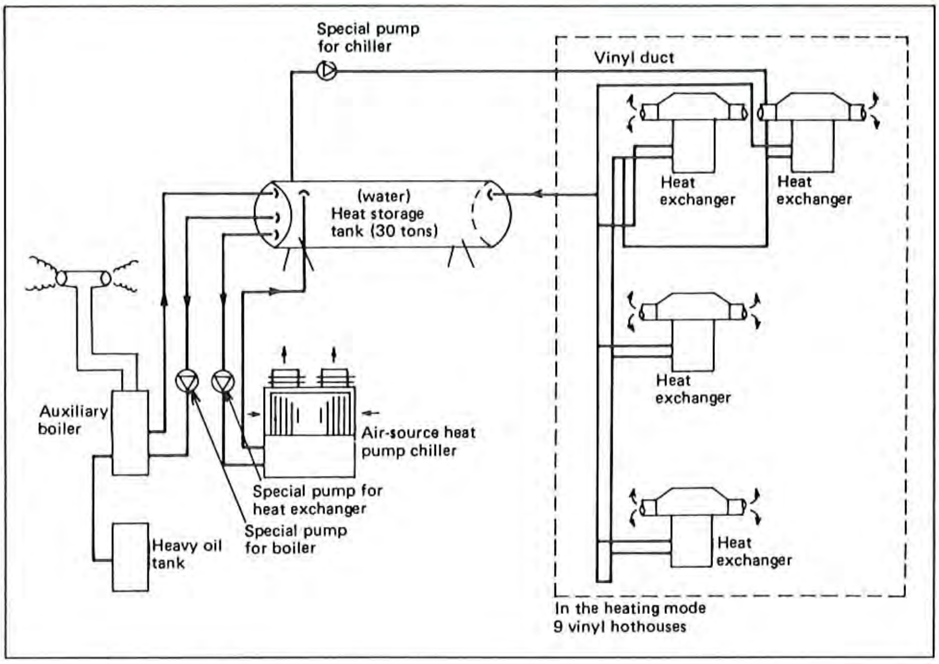

Figure 2. Flow sheet of the equipment

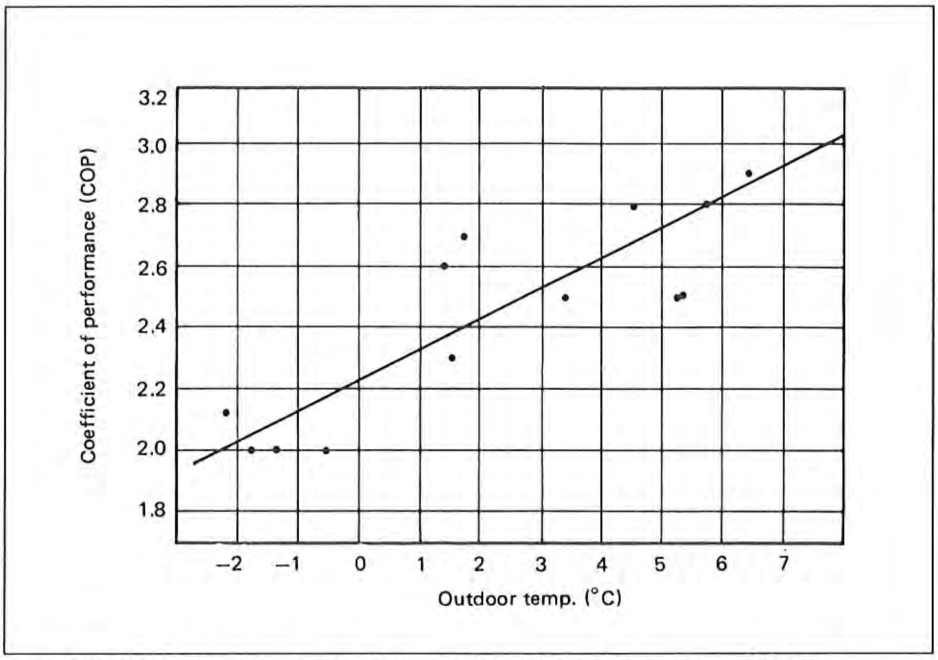

Figure 3. Influence of outdoor temperature upon the coefficient of performance (COP)

	Air-source heat pump chiller (1) (midnight power contract 30kW)	Boiler (2) (Tentative calculation by conventional system)
Power consumption	23,826kWh	9,801 liter
Power cost	313,074 yen	402,210 yen
Basic cost	27,600 yen	--
Electricity tax (5%)	17,034 yen	--
Total	357,798 yen	402,210 yen
Energy saving rate	((2) - (1))/(2) = 0.11	

Table 4. Comparison of working cost during heating

2. Since the temperatures in the hothouse were uniform, uniform results were obtained for crop growth and quality.

Conclusion

Compared to hothouse heating by a conventional oil boiler, hothouse heating by means of the heat pump achieved a 10% cost saving and improved crop growth.

From now on, along with the steady sophistication of hothouse horticulture and a shift to year-round culture, heat pumps will play an increasingly important role.

In closing, it should be noted that this experiment was undertaken jointly with Kyushu Electric Power Co., Inc. and many thanks go to all those concerned.

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Capacity Modulation of Heat Pumps by the Means of Partial Condensation of Refrigerant Mixtures

Continuous capacity control of compression heat pumps can be achieved by the use of nonazeotropic refrigerant mixtures with a composition shift during operation. The separation of the liquid and the vapor phase during condensation is one way to obtain a composition shift. This has been investigated in a laboratory heat pump set-up. The test rig is described in this paper and experimental results are presented.

Introduction

Since the capacity of, for example, residential heat pumps increases with increasing heat source temperature, these heating systems are not able to match the building load which increases with decreasing ambient temperature. Ever since nonazeotropic refrigerant mixtures have been investigated for heat pumps, the possibility of continuous capacity modulation by shifting the composition and thereby reducing the described mismatch has always been mentioned as one of the merits of these working fluids. Various methods have been proposed in the literature[1-5] to achieve this composition shift, but only a few of them have also been investigated experimentally.

At the Institute of Refrigeration at the University of Hannover a research program for fundamental investigations of different means for composition shifting using computer simulations as well as experimental tests has been started. The first experimental results concerning a separation of the mixture during condensation will be presented here.

Principle of partial condensation

During phase changes of mixtures the prevailing concentrations of both phases, liquid and vapor, differ from each other, which is the basis for thermal separation processes. In the case of condensation, which is illustrated in a temperature-concentration diagram (Figure 1), the first liquid drop at the dew point has the highest concentration x_a of the less volatile component b.

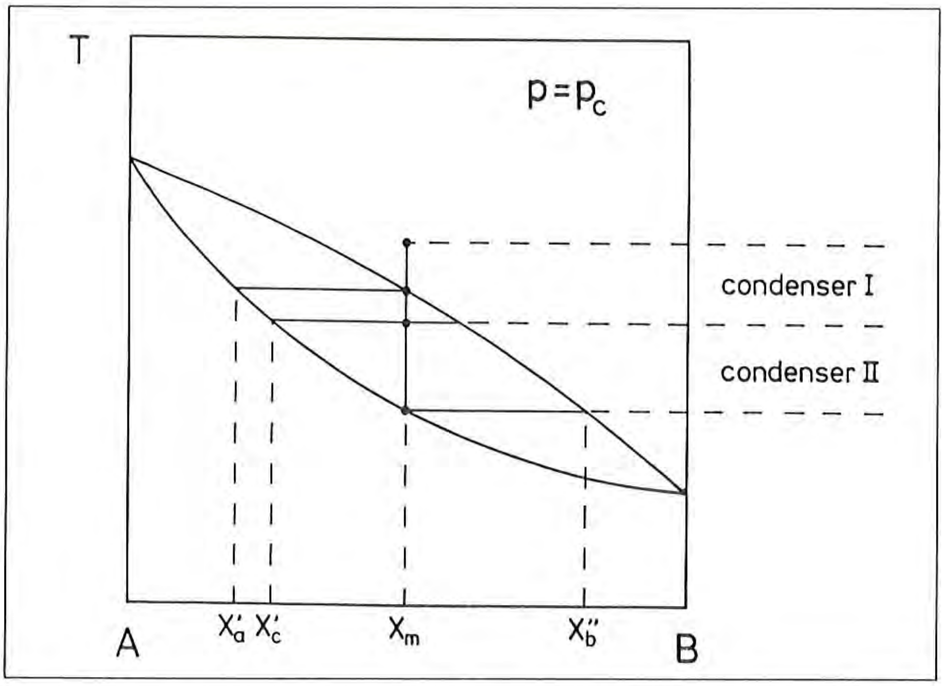

Figure 1. Temperature-concentration diagram for a binary mixture, and illustration of the condensation process

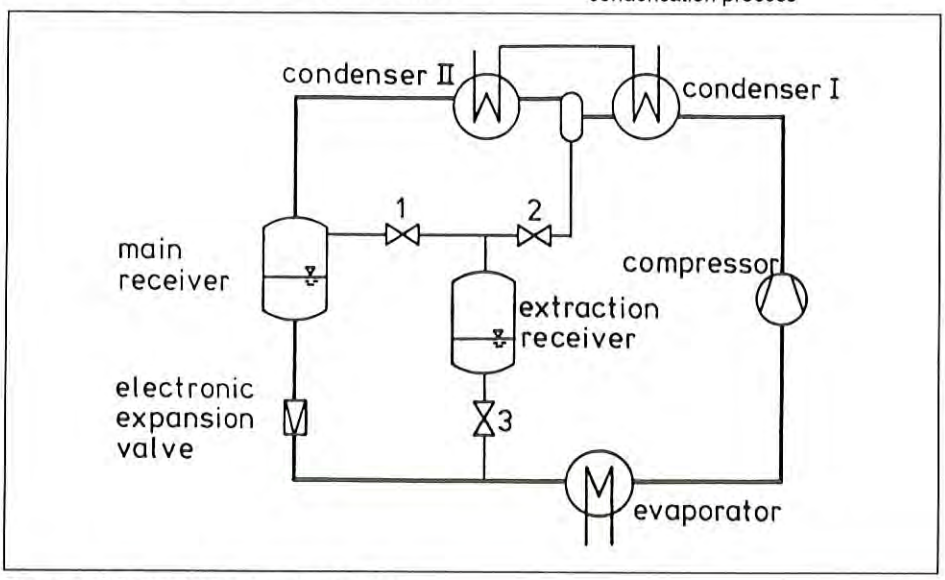

Figure 2. Schematic diagram of the test rig

In the case of complete condensation (without subcooling) a refrigerant receiver connected to the condenser exit contains liquid of the concentration x_m of the circulating mixture and saturated vapor of the concentration x_b, which can easily be separated by condensing it into another vessel. On the other hand, it is hard to achieve a separation of the first condensing fraction x_a, since the location of beginning condensation after desuperheating the discharge gas must be known. This is the theoretical location for extracting this liquid from the circulating mixture. At our laboratory heat pump, however, the condenser has been divided into two units (see Figure 1 and Figure 2), where condenser I represents one third of the total condensation area, which is always a sufficient heat exchange area for complete desuperheating and the beginning of condensation. This arrangement enables an extraction of liquid after condenser I at a concentration x_c (Figure 1) which is somewhat closer to the composition x_m of the circulating mixture than the composition x_a at the optimum location for extraction.

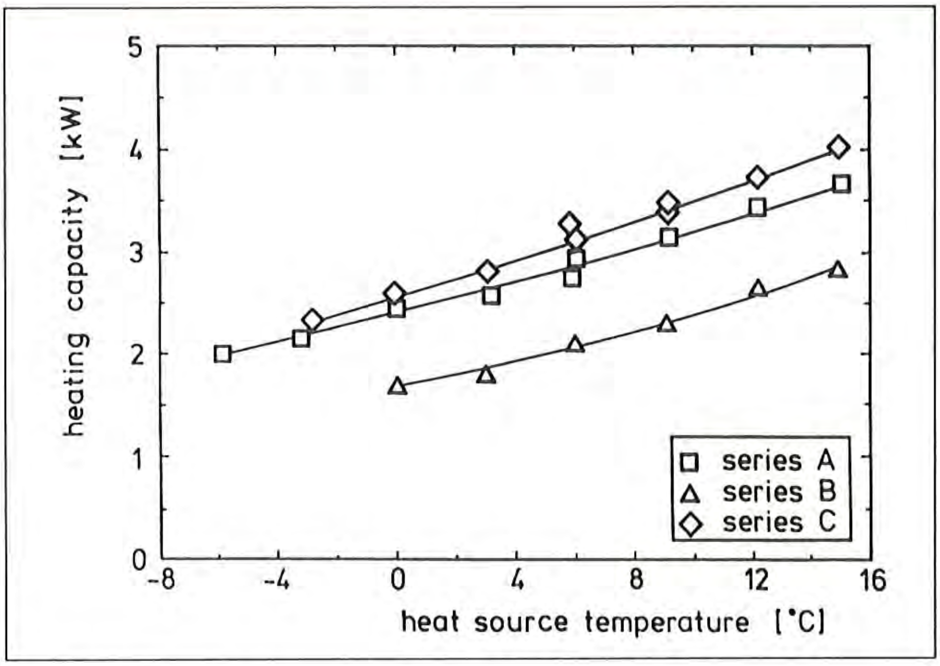

Figure 3. Heating capacity vs. heat source temperature

Test rig

A schematic diagram of the main components of the laboratory heat pump is shown in Figure 2. The compressor is of reciprocating type (Bock F2). Coaxial heat exchangers are chosen for the evaporator and the two condensers (Wieland WKV1 and KWG1, respectively, condenser II consists of two KWG1's). They are operated in counterflow with brine and water, respectively. The main receiver as well as the extraction receiver are cylindrical vessels with a diameter of 130mm and a volume of 7.5 liters. The extraction receiver is equipped with an optical fluid level indicator consisting of a pressure resistant glass tube, and it can be cooled by tap water for the purpose of condensing saturated vapor from the main received into it. An electronic expansion valve (Egelhof MPS 20) is used as the throttling device. Its design is comparable to a conventional thermostatic expansion valve, but the needle of the valve is positioned by a stepping motor. The difference of the refrigerant inlet and outlet temperature at the evaporator is the control variable for this device, which enables controlling independent from mixture compositions and temperature glides at the evaporator, but the ideal value for this temperature difference must be determined by a trial-and-error procedure for each composition.

For separation processes, either valve 1 must be opened while valve 2 is closed to extract saturated vapor of the more volatile fraction from the main receiver into the water cooled extraction receiver or valve 2 has to be opened (valve 1 closed) to feed the less volatile fraction, which is already liquified after condenser I, into the extraction receiver, which needs no cooling in this case. Valve 3 is normally closed and has to be opened only to feed the extracted mixture back to the plant.

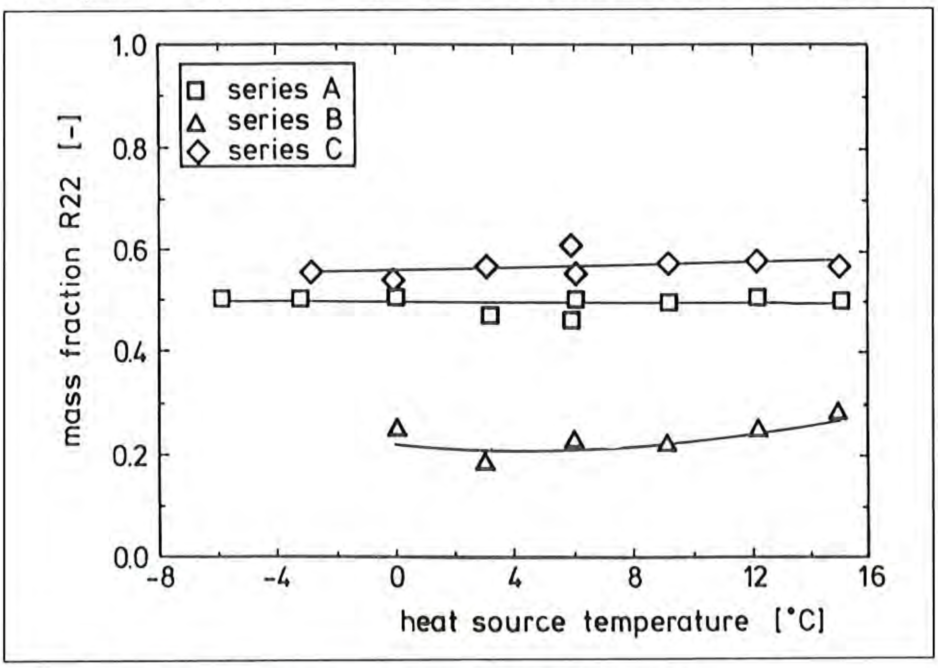

Figure 4. Composition of circulating mixture vs. heat source temperature

The measuring equipment is not included in the schematic of the test rig. The refrigerant cycle temperatures are measured inside the tube at the inlet and outlet of each component with calibrated thermocouples, as well as the temperatures of the water and the brine. Calibrated pressure transducers are located at the inlet of condenser I, the outlet of condenser II and at the inlet and outlet of the evaporator. The concentration of the circulating mixture is evaluated by taking small samples from the suction line, and analyzing them with a gaschromatograph. The compressor power is determined by measuring its speed and the reaction force of the driving motor. Furthermore, the mass flow rates of the brine and the water are measured manually.

Experimental procedure

The mixture R22/R114 has been selected for the experiments presented here due to the following reasons. First, this mixture proved to have a favorable

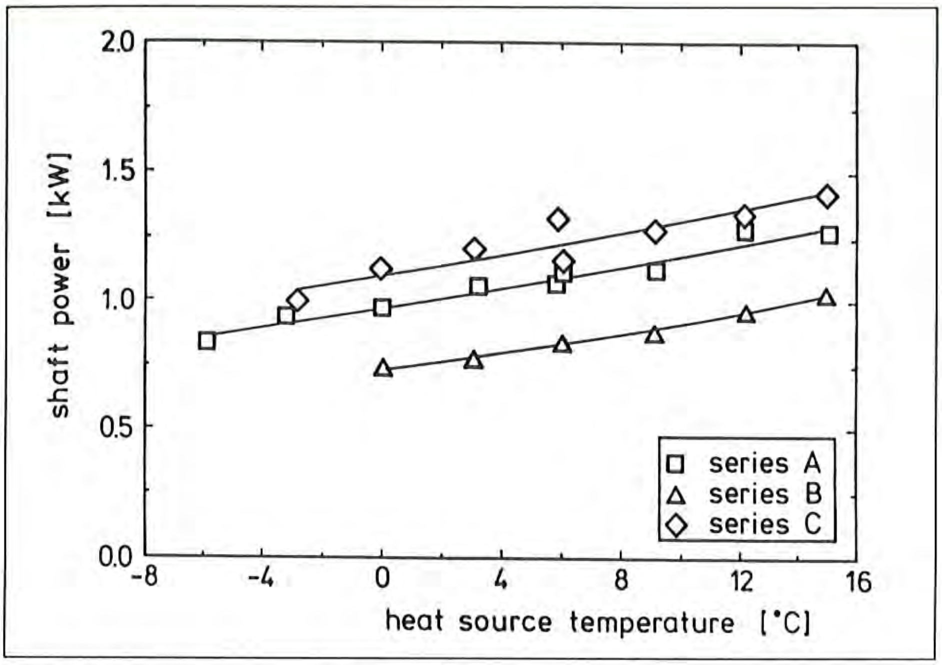

Figure 5. Measured compressor shaft power vs. heat source temperature

energetic behavior for heat pump applications.[6,7] Second, extensive measurements of state properties are available for this mixture[8], which supply the basis for a reliable determination of other thermophysical data by the means of a suitable equation of state. For data reduction of the experiments carried out here (e.g., calculation of enthalpy values), the Lee-Kesler-Plöcker equation of state is selected.[9] Furthermore, these experiments will be used to verify a computer model, which has been developed for further investigations of capacity control besides these measurements. Future investigations will focus on mixtures with ozone-safe components only.

The test rig has been charged with 13.1 kg of the mixture R22/R114 at a mass fraction of 50% R22. The following conditions were adjusted for all tests:

-- Heat sink temperature, i.e., water temperature at the inlet condenser II: t_{w1} = +40°C

-- Mass flow rate of the water at the condenser: m_w = 0.12 kg/s

-- Mass flow rate of the brine at the evaporator: m_b = 0.15 kg/s

-- Compressor speed: n_{comp} = 1500 rpm

The measurements were carried out at follows: The heat source temperature t_{b1} (temperature of the brine at the evaporator inlet) was reduced by steps of 3K from one test to the next one beginning at the highest considered value of t_{b1} = 15°C down to at least -6°C. In this way three test series were carried out, series A with the composition of the initial charge, i.e., without any separation, series B with separating as much as possible of the high volatile fraction and series C with separating as much as possible of the less volatile fraction. The separations of series B were limited either by reaching the minimum charge of the circulating refrigerant or by reaching pressure equivalence in the main receiver and the extraction receiver; the separations of series C were limited either by reaching the minimum charge, too, or by a totally filled extraction receiver.

Besides these test series, another series D was carried out to analyze the dependence of the concentration of the circulating mixture upon the mass and volume of the liquid, which was fed into the extraction receiver. Regarding this analysis, the mass within the extraction receiver had to be calculated with the equation of state. The calculations are based on the extracted volume, which is given directly by the visual liquid level indicator.

Results

The measured heating capacities of the test series A, B, and C are plotted against the heat source temperature in Figure 3. As expected, the heating capacity of series A with the constant mixture composition is decreasing with decreasing heat source temperature, which is in principle the same behavior as obtained for heat pumps working with a pure refrigerant. The heating capacities of series B and C are also decreasing with decreasing heat source temperature, actually with about the same gradient as series A. It can be concluded from this characteristic that the achievable composition shift, i.e., the concentration difference between

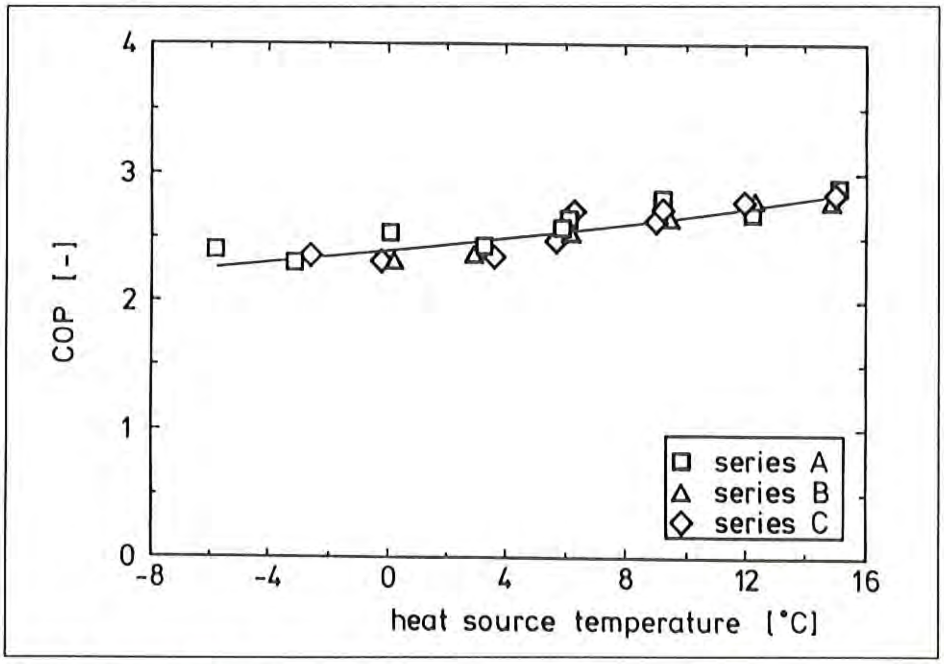

Figure 6. Measured COP vs. heat source temperature

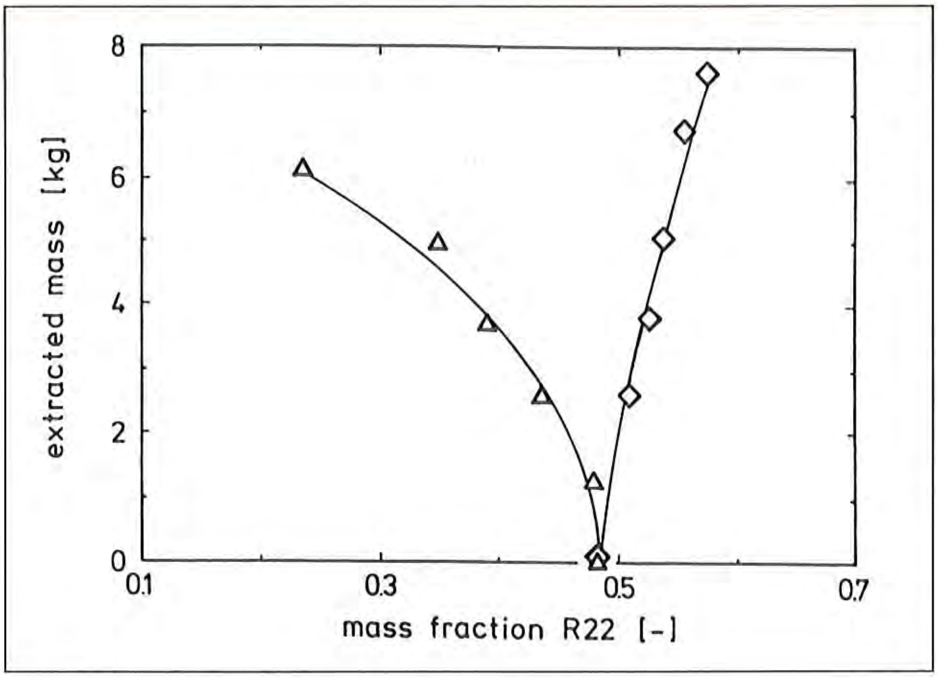

Figure 7. Extracted mass vs. composition of the circulating mixture

series A and B and between series A and C, respectively, is nearly independent of the heat source temperature. This conclusion is confirmed by Figure 4, where the mixture concentrations of the three test series are plotted against the heat source temperature.

Both Figures 3 and 4 also show that the separation quality is distinctly better for the extraction of the high volatile fraction, i.e., the saturated vapor from the main receiver. The main reason for the poor performance of the separation of the less volatile fraction is the location for its extraction. The exit of condenser I is not located close enough to the dew point of the fluid. An appropriate reduction of the heat exchange area of condenser I and an equivalent extension of condenser II would increase the mixture concentration values at series C and thereby extend the possible capacity shift.

Furthermore, according to Figure 3 the heating capacity of series B at a heat source temperature of t_{b1} = +15°C is almost identical with the heating capacity of series C at t_{b1} = +3°C. This can be observed for the capacities at t_{b1} = +9°C (series B) and t_{b1} = -3°C (series C), too. It indicates that this simple arrangement for composition shifting without any optimization efforts is already able to supply a constant heating capacity over a heat source temperature range of about 12°C. The corresponding capacity shifts at constant heat source temperatures amount to a factor of approximately 1.5 (= Q_{max}/Q_{min}). Both effects are considerable improvements according to the well-known mismatch between the performance characteristic of a conventional residential heat pump with a pure refrigerant and the building load characteristic and controlling requirements, respectively.

Apart from the fact that the compressor shaft power, which is plotted in Figure 5 against the heat source temperature, shows a somewhat higher uncertainty in measurement caused by torque pulsations, its curves of the series A, B, and C and the distances between them are in accordance with those of the heating capacities. Therefore the COP shown in Figure 6 is virtually independent of the mixture composition. Its increase with increasing heat source temperature is not significant because of the small evaporator size, which causes distinctly increasing temperature differences between brine and refrigerant at higher capacities.

Finally the above-mentioned test series D is illustrated by Figure 7, which shows the relation between the mass separated in the extraction receiver and the concentration of the mixture circulating in the plant. For both separation methods considerable amounts of refrigerant compared to the initial charge of 13.1 kg must be fed into the extraction receiver to achieve a significant composition shift. Fortunately for the less effective enhancement of the high volatile fraction the extractable amount of refrigerant is higher because the minimum charge of the circulating refrigerant is decreasing with increasing concentration of R22.

Conclusion

An experimental investigation of a means for composition shifting by partial condensation in a heat pump working with the binary mixture R22/R114 yielded a considerable potential for capacity control with the described arrangement, although some possible effective improvements have not yet been done. These are:

-- Better location for the extraction point of the less volatile fraction

-- More effective cooling of the extraction receiver during separation of the high volatile fraction by using the brine leaving the evaporator instead of tap water

-- Selecting other mixture components with higher boiling point differences (if this does not harm the COP values too much).

The research program will proceed mainly with computer simulations to analyze other possible improvements for this method, to compare other suggested means for composition shifts and to apply the results to mixtures with ozone-safe components.

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K.D. Gerdsmeyer and H. Kruse, Universität Hannover, Institut für Kältetechnik und Angewandte Wärmetechnik, Hannover, FR Germany

JAPAN 1990

"Heat Pumps – Solving Energy and Environmental Challenges"

THE 3RD INTERNATIONAL ENERGY AGENCY HEAT PUMP CONFERENCE 1990 – TOKYO

Meiji Kinenkan (Meiji Memorial Hall), Minato-ku, Tokyo
March 12—15, 1990

Introduction

The purpose of this conference is to share among the IEA member nations information that can lead to increased heat pump use and to the attendant improvements in the efficiency of our energy resources utilization.

The conference will take place March 12-15, 1990 in Tokyo. English is the official conference language. Simultaneous Japanese/English translation will be provided. Invited speakers not only from IEA member nations will discuss important sub-topics in plenary and poster sessions. Other events of special interest to conference attendees are the pre- and post-conference site visits and the exposition "HVAC & R '90 Japan."

Registration deadline:
January 31, 1990

The registration fee for the conference includes a copy of the proceedings, copies of papers not included in the proceedings, four lunches, a welcome reception, and an evening banquet. Costs for the conference and the optional associated events are summarized in Table 1.

The IEA Heat Pump Conferences

The IEA Heat Pump Conference in Tokyo is the third in a series of conferences organized by IEA nations who are members of the Implementing Agreement for a Programme of R&D on Advanced Heat Pumps. The first conference was held in Graz, Austria in 1984 and the second in Orlando, Florida, USA in 1987. (Proceedings from these conferences are still available; contact the HPC for more information.) The IEA, recognizing the value of these events in enabling experts from various countries to meet together and exchange information, is encouraging this third conference organized by the Heat Pump Technology Center of Japan.

Event	Fee (per person)
3rd IEA Heat Pump Conference, March 12-15, 1990	30,000 Yen (approx. U.S.$ 210)
Pre-conference site visits, March 4-9, 1990	96,500 yen (approx. U.S.$ 675)
Post-conference site visits, March 16, 1990	9000 yen (approx. U.S.$ 63)
HVAC & R '90 Exposition, March 13-17, 1990	Free of charge

Table 1. Cost for conference and related events

The 3rd IEA Heat Pump Conference, March 12-15, 1990

The conference will be a four-day event and is divided into eight sessions as shown in Table 2. Each session will consist of seven or eight presentations from invited speakers and a time for questions from the audience.

Date Day	March 12 Monday	March 13 Tuesday	March 14 Wednesday	March 15 Thursday
Morning Session	I Opening Plenary Session (1000-1200)	III Advances in Thermally Activated Heat Pumps (0900-1200)	V Environmental Aspects (0900-1200)	VII National R&D Programs, Governmental and Utility Roles (0900-1200)
Afternoon Session	II Advances in Electric Heat Pumps (1330-1700)	IV Industrial and District Heating Applications (1330-1700)	VI Operating Experi- ences, Economics, and Marketing (1330-1700)	VIII Closing Plenary Session (1330-1600)

Table 2. 3rd IEA Heat Pump Conference Sessions

Session 1, the opening plenary session, will include addresses from officials of MITI and the IEA. A keynote address will emphasize the environmental aspects of heat pump use. Day 1 of the conference continues with Session 2 on advances in electric heat pumps. The presentations in this session will deal with prospects for technological improvements in electric heat pumps leading to enhanced performance, improved economics, and increased user satisfaction.

For Session 3, advances in thermally activated heat pumps, speakers from Japan, USA, France and Germany will present recent activities in this area. Session 4 will deal with industrial and district heating applications. A broad range of topics including process integration, industrial applications, heat pump driers, and district heating and cooling uses will be covered.

Session 5 starts the third day of the conference and deals with the environmental aspects related to heat pumps. Speakers will present information on heat pumps and their role in environmental protection. Environmentally safe refrigerants will also be discussed. Session 6 on operating experience, economics, and marketing of heat pumps will conclude Day 3 of the conference.

The final day of the conference begins with Session 7 on national research and development programs and the government, utility role on advanced heat pumps and protection of the environment. Session 8 will close the conference with the chairmen of the previous sessions summarizing in a panel discussion the results, taking into consideration future prospects, possible improvements, and the need for international cooperation.

The Japanese hosts will also show a special movie on the status of Japan's "Super heat pump energy accumulation system." A general discussion will follow and the final conclusions of the conference will be presented by the chairman of the organizing committee.

Poster Sessions

Poster sessions will be held during the second and third day of the conference. These sessions will allow conference attendees to get a quick overview of the type of work being carried out worldwide on the subject of heat pumps.

Table 3 gives an overview of the conference and related events.

Pre- and Post-Conference Site Visits

Everyone attending this conference from overseas is encouraged to take part in the site visits to important heat pump related industries and organizations. The pre-conference tour event will begin on Sunday, March 4, in Osaka and will end March 9 in Tokyo. The itinerary for the pre-conference tour is shown in Table 4. The sites listed include heat pump laboratories, manufacturing plants, district heating and cooling systems, a thermal power station, commercial heat pump installations, and a residential heat pump installation. The five-day tour begins on Monday and ends on Friday. The fee for this tour includes hotel costs for five nights and transportation.

Day / Date	Event
Sunday-Friday / March 4-9, 1990	Pre-conference site visits
Monday-Thursday / March 12-15, 1990	IEA Heat Pump Conference
Friday / March 16, 1990	Post-conference site visits
Tuesday-Saturday / March 13-17, 1990	HVAC & R '90 JAPAN

Table 3. Schedule of conference and related events

Day/Date	Sites	Travel	Hotel Location
Sunday, March 4	Rendezvous and briefing at hotel at 1900	Participants arrive in Osaka	Osaka
Monday, March 5	Kansai Electric Co. Tsurumi Greenery site	Osaka to Nagoya	Nagoya
Tuesday, March 6	Toho Gas Co. Howa Sports Land (bring shorts) Chubu Electric Power Co.	Nagoya to Shizuoka	Shizuoka
Wednesday, March 7	Hitachi Ltd. Hakone National Park	Shizuoka	Hakone
Thursday, March 8	Tokyo Electric Power Co. New City Higashi Totsuka	Hakone to Tokyo	Tokyo
Friday, March 9	Nishi-Shinjuku District heating and cooling systems Shibaura district heating cooling systems	Tokyo	Free

Table 4. Pre-conference site visits

The one day post-conference tour will be in the Tokyo suburb area and will take place on Friday, March 16. The sites included on this tour are the Tsukuba Research Laboratories of MITI's Agency for Industrial Science and Technology and Mayekawa Mfg. Co., Ltd.

HVAC & R, '90 JAPAN

The trade exhibition HVAC&R '90 JAPAN will be held March 13-17, 1990 at the Tokyo international fair ground. This event is organized by the Japan Refrigeration and Air Conditioning Industry Association, the Japan Heating Industrial Association, and the Solar System Development Association. The exhibition is free of charge.

The Meiji Memorial Hall, founded in 1881, was originally used for Imperial Conferences and Banquets. In 1888, conferences on the Imperial Constitution of Japan were held in this hall in the presence of the Meiji Emperor. In November 1947, the hall was opened as a wedding hall belonging to the Meiji Shrine. It is now reconstructed and widely used for a variety of conferences and parties.

Where to get more information

The 3rd IEA Heat Pump Conference is a rare opportunity to learn about international heat pump activities, the Japanese heat pump industry, and Japan. Those interested in receiving a registration form, more detailed descriptions of the conference, and travel and hotel information are asked to contact immediately the secretariat of the conference or the appropriate contact person as listed below.

Japan

Contact: Y. Igarashi
Address: Heat Pump Technology Center of Japan
Azuma Shurui Building
9-11, Kanda-Awaji-cho 2-chome
Chiyoda-ku, Tokyo 101
Japan
Telephone: Country Code + 3/258-1035
Telefax: Country Code + 3/258-1037
Telex: 222-4601 hptc j

Europe

Contact: W. Hochegger
Address: Energiesparhaus Graz
Petersgasse 45
A-8010 Graz
Austria
Telephone: Country Code + 316/822045
Telefax: Country Code + 316/826371
Telex: 3-12305

North America

Contact: R.L. Douglas Cane
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6981 Millcreek Dr., Unit 28
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Canada
Telephone: Country Code + 416/542-2890
Telefax: Country Code + 416/542-3160

Bibliographic Review

Residential and commercial heating and cooling heat pumps

Steady-state refrigerant flow and airflow control experiments for a continuously variable speed air-to-air heat pump. Miller, W.A. (Efficiency and Renewables Research Section of the Energy Div. of Oak Ridge National Lab., TN, US) ASHRAE Trans. (Jun 1987) v. 93(2) p. 1191-1204. (English)

A continuously variable speed, air-to-air split-system residential heat pump of nominal 2 3/4-ton (9.7-kW) capacity was instrumented and tested in the laboratory. The coefficient of performance (COP), capacity of the system, and component efficiencies were measured during steady-state heating- and cooling-mode operation. The indoor blower speed and the refrigerant subcooling at the condenser exit were varied at discrete compressor speeds to determine the best COP as constrained by residential comfort requirements. Heating- and cooling-mode test results showed a 10% improvement in COP for operation with a variable area throttle as compared with a capillary tube throttle. Improved control of refrigerant flow and reduction of the indoor blower speed improved humidity control. As compared with results for capillary tube flow control, the sensible heat ratio was reduced from 0.95 to 0.77 at 15-Hz compressor speed.

Reasonable utilization of electric power in HVAC installations. Rationeller Einsatz von elektrischer Energie in lufttechnischen Anlagen. Carl, Horst. ETA Elektrowärme Tech Ausbau v 46 n 3 May 1988 p A71-A73. (German)

At enterprises or business offices in which there is a simultaneous demand for heating and cooling, it is possible to save energy by utilizing the waste heat rejected by refrigeration units for space and water heating. Known examples are slaughterhouses, dairies, breweries, etc. This contribution gives a brief survey of other possibilities for energy saving, e.g., utilization of waste heat in telecommunication installations, heating and cooling with small heat pumps and utilization of control technology in HVAC installations. (Translated abstract)

New utilization for small air/air heat pumps. Une nouvelle utilisation de petites pompes a chaleur air/air. Le Vaguerese, Patrick (CETIAT, Villeurbanne, Fr). Rev Gen Electr, 10 Nov 1987 p 35-38. (French)

Thirteen individual houses and one office building have been equipped with split-type heat pumps. When the geometry of the houses is favorable to a good heat distribution, the heating energy supplied by the heat pumps has been often more than 75% of the total energy demand and the seasonal COP about 2. The comfort has been judged favorably by the occupants. The air conditioning in the office building was to be tested during the summer '87. (Edited author abstract)

Development of small air/air reversible heat pumps. L'evolution des petites pompes a chaleur reversibles air/air. Keiflin, Jacques (L'Air Conditionne Entreprises, Fr). Rev Gen Electr, 10 Nov 1987 p 32-34. (French)

The author reviews the general characteristics and recent development of small and medium power outside air/air heat pumps intended for heating and cooling in individual applications in the residential and, the tertiary sectors. These equipment units are intailed directly on the premises to be treated and ensure at lower cost the winter heating function and the cooling one during the warm weather. (Author abstract)

Packaged terminal air conditioning heat pumps. Poole, G.W. (Henrico County Public Sch, Richmond, VA, USA). Heat Piping Air Cond v 59 n 7 Jul 1987 p 45-50. (English)

Maybeury Elementary School is one of 50 facilities in the Henrico County Public School System of suburban Richmond, Virginia. The hydronic heating system consisted of two 50 hp fire tube scotch marine boilers, a circulating pump, and direct buried black steel piping looping throughout the campus. Packaged terminal air conditioning heat pump units replace central boiler system in school rehab. Author describes the replacement of the central boiler system with air conditioning heat pump units.

Commercial heat pump water heaters. Kohloss, Frederick H. (Frederick H. Kohloss & Associates, Honolulu, Hawaii, USA), Plumbing Eng v 15 n 6, Jul-Aug 1987 p. 26-29. (English)

For centrally heated and cooled buildings, the heat pump principle is widely used for space heating. The commercial space-heating heat pump can provide adequate heat to air-handling unit coils with water no hotter than 105 to 115°F. A stable electricity supply system with voltage regulation is desirable, particularly with large, factory-packaged heat pumps. These units frequently use hermetic motor compressors, which are not as forgiving of unstable power supply as open, drop-proof electric motors. For this reason, some field-assembled heat pumps have tended to use open motors to drive compressors. Open motors are more easily cooled, the compressor is not affected by motor failures, and there are more available choices of refrigerant and of compressor speed.

Applications tests of commercial heat pump water heaters. Oshinski, John N. (D.W. Abrams P.E. & Associates, Atlanta, Georgia, USA); Abrams, Donald W. Energy Eng v 85 n 6, Oct-Nov 1988, p 49-71. (English)

Field application tests have been conducted on three 4- to 6-ton commercial heat pump water heater systems in a restaurant, a coin-operated laundry, and an office building cafeteria in Atlanta. The units provide space cooling while rejecting heat to a water heating load. The tests, conducted for Georgia Power Company, examined both quantitative and qualitative aspects of the heat pumps and the overall water heating systems. The results provide valuable insight into the actual operating characteristics of heat pump water heaters and useful guidelines for system design and operation.

Multi-source hydronic heat pump application study and analysis. Meckler, Milton (Meckler Group, Encino, Calif, USA), Energy Eng v 85 n 6, Oct-Nov 1988, p 32-48. (English)

A multi-source hydronic heat pump (MSHHP) system was recently evaluated for a proposed Valley Oaks Village senior housing project containing 258 units arranged within two- and three-story buildings, which with a two-story recreation building, comprise approximately 200,000 gross square feet. The project was planned for construction in Southern California in 1988. The project involved determination of the engineering-economic feasibility of incorporating a cost-effective off-peak cooling system. The serving utility, Southern California Edison (SCE) Company, offered an attractive cool storage inducement program. It was decided to evaluate the use of a MSHHP system integrated with an ice builder thermal energy storage so that the water source heat pump unit compressors could all be shut down during on-peak hours.

District heating and cooling with heat pumps outside the United States. Calm, James M., ASHRAE Trans. 1988, v 94, pt. 1, 1988. (English)

Companion papers in this session address district heating and cooling with heat pump systems in the United States. Many such systems are in operation in other countries; for example, several systems incorporating cooling are in use in Germany, Italy, Japan, and Norway. For district heating alone, the history is much older, the technologies more diverse, and the aggregate installed capacities - primarily in Europe - much greater. To provide a broader perspective of district heating and cooling with heat pump systems, the international status is reviewed and selected systems are described. The summaries provided indicate that such systems are reliable, economical, and environmentally attractive.

Industrial heating and cooling heat pumps

Demonstration projects industrial heat pumps. Demonstratieprojecten industriele warmtepompen. Bouma, J.W. (NEOM, Sittard, Netherlands). Koeltechniek (May 1988) v. 81(5) p. 6-11. (Dutch)

Seven industrial heat pumps are described, which are realized through the NEOM (Dutch Energy Development Company) demonstration program in this field. Layouts, operation, economy and experiences are briefly treated. A sensitivity analysis demonstrates how economic feasibility is influenced by prices of gas and electricity. Industrial heat pumps can clearly be attractive for processes where small temperature differences between heat source and sink exist, where heating and cooling are needed at the same time and for drying applications.

Heat recovery by means of heat pumps in a dairy. Wärmerückgewinnung mit wärmepumpen in einem milchwerk. Wissel, R. (Siemens AG, Erlangen, West Ger), Elektrowärme Tech Ausbau v 43 n 3 May 1985 p 99-101. (German)

A combined heating and cooling installation with electric heat pumps, installed in a dairy, has been planned for heat recovery. Details about the installation at an Oberallgau dairy in Sonthofen are presented. The investments of about DM 200,000 will be recovered within 2-3 years.

A vacuum-freeze evaporator for large heat pump installations. Collet, P.J. (Hoofdgroep Maatschappelijke Technologie TNO, Apeldoorn, Netherlands), Koeltechniek (Oct 1987) v. 80(10) p. 17-20. (Dutch)

The application of a vacuum-freeze evaporator as part of large heat pump installations provides the opportunity for combining heating, cooling and (sea)water desalination in a cost-advantageous way. Besides acting as a heat source for the heat pump, the freezing operation also yields ice which can be used for long- or short-term storage of cooling capacity. Because ice is formed in a pure crystalline form, potable water can be obtained after washing and melting of the ice crystals. The contents of this article mainly concern the design and operation of a novel type of vacuum-freeze evaporator, and the experimental investigations which were carried out with regard to this particular vacuum-freeze process. Brief mention is also made of possible applications for which the economic prospects seem particularly promising.

Agricultural Heating and Cooling Heat Pumps

Heat pumps for greenhouses. Camporeale, S. (ENEA, Rome, Italy. Agric. Innovazione (Sep 1987) (2-3) p. 66-73. (Italian)

Using for example, the experimental greenhouse installed at ENEA's Casaccia (Italy) facility, the report illustrates the advantages of heat pumps. They are easy to operate, pollution free, multi-purpose (heating, dehumidification and air conditioning) and permit precise control of temperature and humidity, thus resulting in reduced energy consumption and overall economization. In addition, heat pumps avoid problems associated with conventional equipment such as fire protection and fuel storage.

Heat pump air conditioning for mushroom growing. Dean, C. (3CL, Telford, Engl), H&V Eng v 61 n 687 p 10-11. (English)

A major air conditioning company has developed a heat pump based heating and cooling package with double cost savings for the booming mushroom growing industry. 3CL (Closed Circuit Cooling Ltd.) has launched a consultancy and supply service combining a totally integrated set of hardware and all installation drawings to enable mushroom growers to install a full heat-pump system themselves. (Author abstract)

New technologies for heating and cooling

Cascading two-stage sorption chiller system consisting of a water-zeolite

high temperature stage and a water-LiBr low-temperature stage. Ziegler, F.; Brandl, F.; Voelkl, J.; Alefeld, G. (München Univ., Garching, Germany, F.R.) Absorption heat pumps congress. Proceedings of the congress. 1985. pp. 231-238. (English)

An adsorption heat pump based on the system zeolite-water has been built and tested. A temperature lift of 80°C and more can be reached with a COP of 1.2 to 1.3. If no heating is required the temperature of the output heat is high enough to drive an absorption chiller. The coupled system could reach a COP for cooling of over 1.2.

Analysis of a solid adsorbent heat driven heat pump. Miles, D.J.; Shelton, S.V. (THERMAX, Inc., Atlanta, GA). American Society of Mechanical Engineers. 1986. pp. 55-60 . (English)

An analysis of the performance of a solid adsorbent based heat driven heat pump is presented. The model as sumes a uniform heating and cooling o two adsorbent beds by a fluid circula ing through embedded tubes. A corre lation for the level of refrigerant adsor tion in each bed as a function of the be temperature and pressure has bee utilized. Results of the analysis for system utilizing zeolite as the adsorb ent and ammonia as the refrigerant ar presented.

News Briefs

Second IEA Heat Pump Center Workshop 1989

The Second IEA Heat Pump Center Workshop 1989 will take place in Hannover, FR Germany, on November 21-22, 1989. The topic of this workshop is "High temperature heat pumps -- requirements and solutions for future refrigerants, processes, and applications." The workshop is organized by the IEA Heat Pump Center in cooperation with the DKV (Deutscher Kälte- und Klimatechnischer Verein e.V.). It will take place in connection with the 1989 annual meeting of the DKV (Kälte-Klima Tagung 1989, November 22-24, 1989: Newest Developments of Components and Materials for Refrigeration and Air-Conditioning Technology).

Purpose of the workshop

The overall goal of the workshop is to discuss needs and problems of high temperature compression heat pumps for wider application of this energy-saving, economical, and environmentally sound technology. To ensure an efficient meeting, discussions will concentrate on closed cycle vapor compression heat pumps with specific consideration for applications in the temperature range of 70-120°C. This type of heat pump is of special interest since it mainly uses ozone-depleting refrigerants (R12, R114).

Persons interested in participating should contact the Heat Pump Center (see back cover for the address).

General

Electricity to fuel cost ratios for industry

The ratio of the electricity to the fuel price is the most important parameter for an economic use of electric heat pumps. For **heating only** heat pumps driven by **electrical energy** this ratio will dictate the minimum COP needed to achieve operating costs which ar equal to conventional oil or gas fire heating systems. For the calculation shown in the figures energy costs i industry published by the IEA in 198 were used and an 85% conversion eff

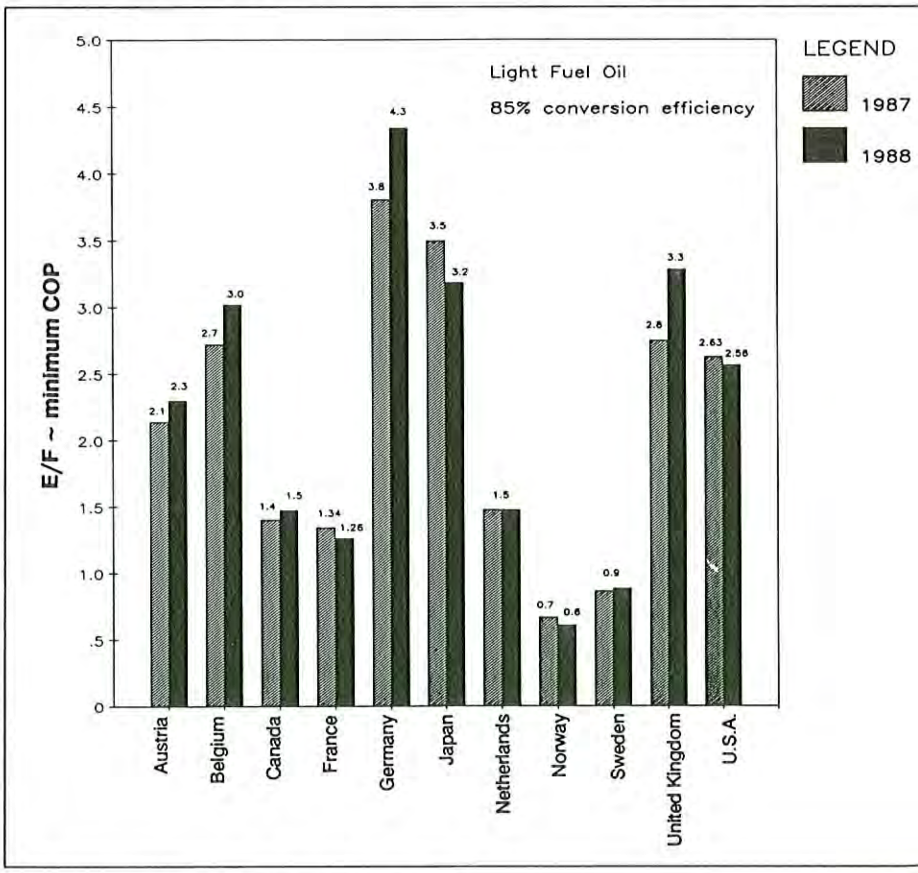

Figure 1. E/F ratio for industry: light fuel oil

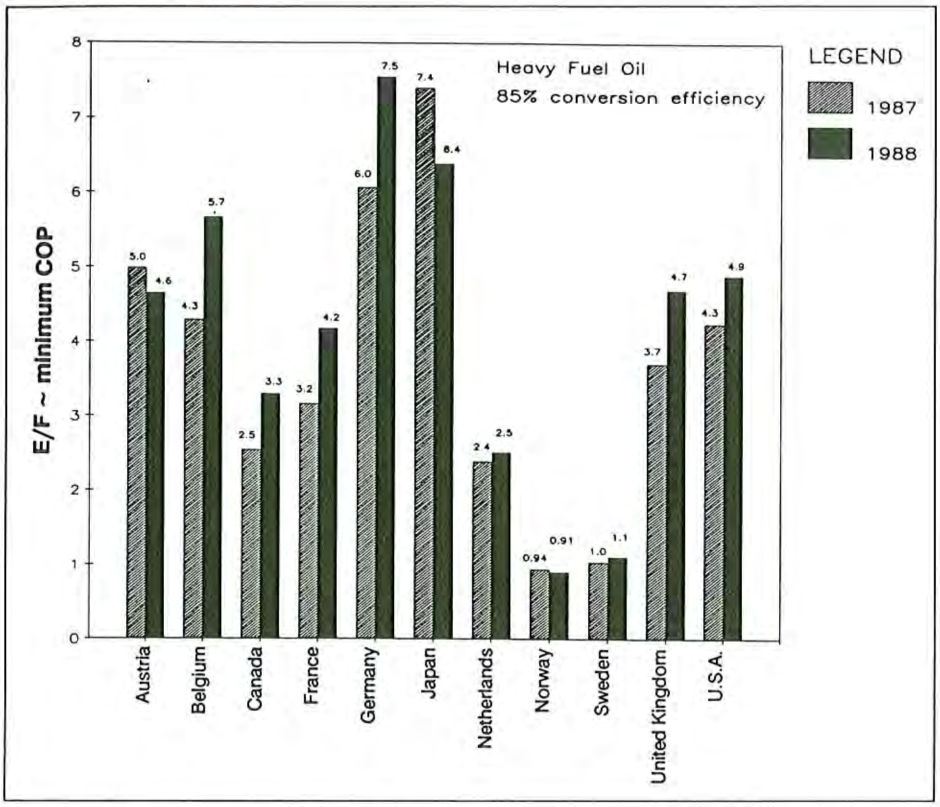

Figure 2. E/F ratio for industry: heavy fuel oil

ciency was assumed for the fuel based system.

Figures 1, 2, and 3 show the results of such a calculation for light fuel oil, heavy fuel oil, and natural gas, respectively. The ratio E/F is the cost of electricity divided by the cost of heat from the replaced fuel. Values for 1987 and 1988 are given for 11 countries. In the case of natural gas the E/F ratio is the highest in Germany at 5.3 and is the lowest in the Netherlands at 2.5. For light fuel oil Germany again has the highest E/F ratio at 4.34 and the lowest is in Norway at 0.608. For heavy fuel oil, Germany again tops the list with an E/F value of 7.5 and Norway has the lowest value of 0.91.

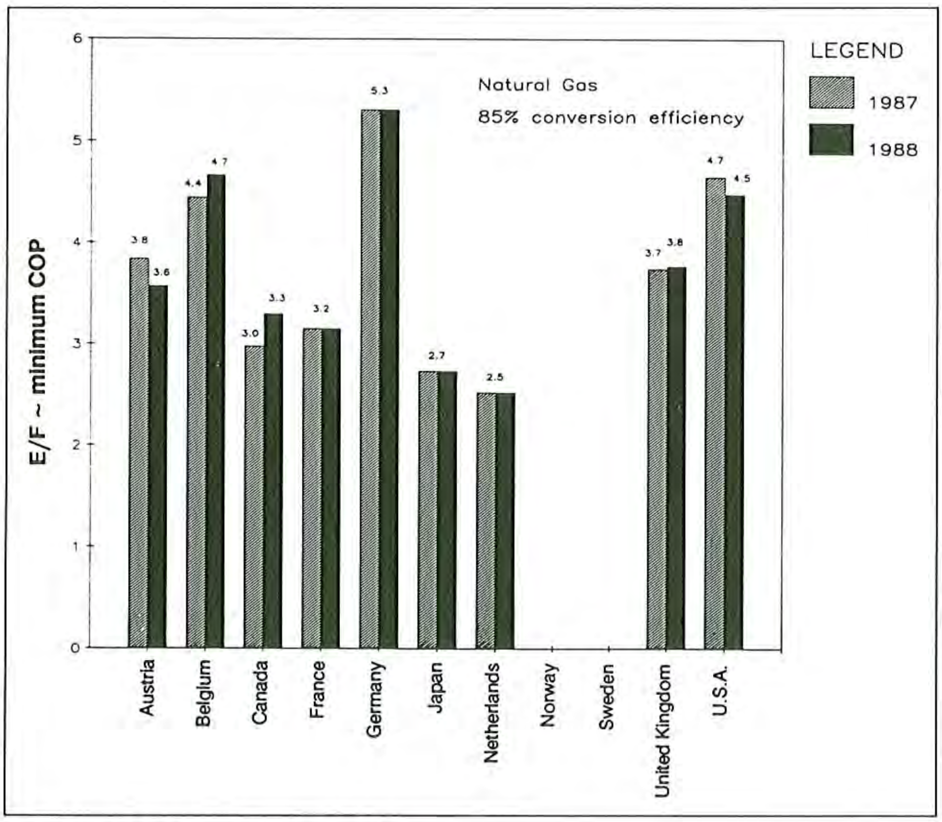

Figure 3. E/F ratio for industry: natural gas

What can be concluded from these figures?

1. The figures give the maximum COP that an electrically driven **heating only** heat pump must have in various countries to equal the cost of producing heat by conventional systems.

2. By utilizing the cooling effect of a heat pump along with the heating effect, the economics of the heat pump can be greatly improved since the monetary value of the cooling effect can be calculated (integrated heating and cooling heat recovery).

3. Thermally activated heat pumps theoretically have a good application potential considering that with a COP > 1, they are already saving on operating costs compared to conventional fuels. (Problem: capital costs are often too high.)

It should be remembered that the investment costs for a heat pump and the parameters affecting heat pump use such as temperature levels heating requirements, and control methods must also be considered. All of these factors will tend to add to the cost of a heat pump and will require that it operate even more efficiently to be economical. On the other hand, consequential profits of using a heat pump such as cooling and heat recovery, improved product quality, lower pollution, higher process output, etc., will all tend to make the heat pump more economically attractive and environmentally acceptable.

IEA Implementing Agreement on Advanced Heat Pumps: Annex Update

The Executive Committee of the Implementing Agreement for a Programme of Research and Development on Advanced Heat Pumps held its first meeting in 1989 on June 22 in Bruegge, Belgium. At this meeting the status of ongoing annexes and new proposals for annexes were discussed. Annexes are projects carried out under the guidance of the Executive Committee by two or more member countries of the implementing agreement. Each annex is funded by the particular participating countries and at least one country takes on the role of operating agent for the project. Currently there are eight annexes which are active and a number which have been recently completed. Table 1 shows the annexes currently in progress or recently completed, the participating countries, and the operating agent. The last detailed review of all annexes took place in the September 1988 Newsletter. A short update on each annex is presented below.

COUNTRY	ANNEX									
	IV	VII	VIII	IX	X	XI	XII	XIII	XIV	XV
Austria	X						X	X		X
Belgium				A			X		X	
Canada	X	X	A					X		A
Denmark		X								
Finland		X		X				X		
Germany	A	X		X			X		X	
Italy	X									
Japan	X			X		X	X	X	A	X
Netherlands	X			X						
Norway	X							X		
Sweden	X	A		X	X	X		A	X	
Switzerland			X	X			X			
U.S.A	X		X		A	A	A	X	X	X

X - Participant
A - Operating agent

Table 1. Annexes currently active or recently completed

Annex IV -- IEA Heat Pump Center. This annex will be completed at the end of 1989 after seven years of activities. The decision to close this annex was based on a review by a special committee established by the member countries of Annex IV. Their final report indicated that all work had been carried out in a satisfactory manner and that the present operating agent was qualified to continue as such.

Annex VII -- New developments of the evaporator part of heat pump systems. The final report for this annex is now available to all participating countries. For more information contact Professor T. Berntsson, Chalmers University of Technology, Dept. of Heat and Power Technology, S-41296 Göteborg, Sweden.

Annex VIII -- Advanced in-ground heat exchange technology for heat pump systems. The project is essentially complete and a final report is being prepared. The operating agent indicated that the scope of the final report will be more extensive than originally planned. For more information contact Mr. O.J. Svec, National Research Council Canada, Institute for Research in Construction, Montreal Road, Ottawa, Ontario K1A0R6, Canada.

Annex IX -- High temperature industrial heat pumps. A final report will be available to all participants by the end of 1989. For more information contact Professor J. Berghmans, Katholieke Universiteit Leuven, Celestijnenlaan 300 A, B-3030 Heverlee, Belgium.

Annex X -- Technical and marketing analysis of advanced heat pumps. This annex will continue for an additional year. A study will be undertaken concerning the market possibilities of heat pumps in heating and cooling applications. This study will be available to all member countries of the Implementing Agreement and should provide a basis for future work by the Implementing Agreement.

Annex XI -- Stirling engine technology for application in buildings. This annex is officially closed. The final report can be obtained from the operating agent at the following address: Mr. P. Fairchild, Oak Ridge National Laboratory, Building 3147, P.O. Box 2008, Oak Ridge, TN 37831, USA.

Annex XII -- Modelling techniques for simulation and design of compression heat pumps. This annex will be extended to April 1990 for purposes of completion.

Annex XIII -- State and transport properties of high temperature working fluids and nonazeotropic mixtures. This annex is still in progress.

Annex XIV -- Working fluids and transport phenomena in advanced absorption heat pumps. A second working meeting was held in Stockholm in May 1989. During this meeting discussions and presentations of the literature survey on absorption fluid data, advanced cycles, and transport phenomena took place. A seminar on absorption technology was also held during the meeting. This annex is scheduled for completion in June 1990. A final meeting will be held in Tokyo on March 11, 1990.

Annex XV -- Heat pump systems with direct expansion ground coils. This Annex was officially started with the approval of the Executive Committee on June 22, 1989. Research work is already well under way in all of the participating countries.

Heat Pump Center Internal

Japanese experts visit the HPC

On May 31, 1989 two experts from Japan visited the IEA HPC offices in Karlsruhe. Mr. T. Kashiwagi (of Tokyo University and the Japanese Association of Refrigeration) and Mr. S. Kurosawa of Tokyo Gas Company accompanied by Mr. N. Yamanouchi of Honeywell Europe S.A. were on a fact finding tour of Europe. They had already visited various important institutions in Europe to obtain more information about work being carried out in the use of ammonia as a working fluid in heat pump and refrigeration. At the HPC the topic of discussion was ammonia and its promotion as a refrigerant in Japan and in other countries. Mr. Kashiwagi explained that presently in Japan ammonia is not widely used due to the strict laws governing its use. This is in spite of the fact that ammonia is an environmentally safe refrigerant and one with excellent thermal properties. One of the goals of the visit was to gather more information about laws, codes, and standards governing the use of ammonia in Europe.

Europe

Heat pumps and district heating: report of a Japanese-Dutch information exchange

In the first week of July, a Japanese Technical Mission, headed by Mr. T. Sugimoto (NEDO), visited several European countries in order to gather information on district heating (DH) plants with heat pumps. The survey had been organized by the Heat Pump Technology Center of Japan, while the Dutch National Team made arrangements for the visit to The Netherlands. The program contained presentations and a site visit for the 26 Japanese and 13 Dutch attendees.

Presentations by NOVEM and VESTIN on district heating in The Netherlands showed that, at present, 21 DH projects (125,000 connections, heating only) have been realized, but that economical feasibility is disappointing. In three of these DH projects a heat pump has been installed: Deventer, Enschede and Bergen op Zoom. Separate presentations were given on these projects.

In the Deventer project, because of severe evaporator corrosion problems as well as a bad economical performance, the two heat pumps (gas-engine HP, 1500 kW) will be dismantled. In Enschede the heat pump (gas-engine HP, 820 kW) has been taken out of operation because of inadmissible fluctuations of the hot running water temperature. The DH project in Bergen op Zoom is operated by the Energy and Water Board ENWA, that acted as a host for the Mission. The DH plant uses industrial waste heat that formerly was available at 65°C. As a result of changes in the industrial process in November 1988, the waste heat temperature dropped to 42°C. The ENWA then decided to install an electrically driven screw compressor heat pump (1750 kW) to provide the necessary 20°C temperature rise. Operating experiences have been quite good since then. A visit was made to the heat pump installation as well as to a 15th century church where DH heat is used for floor heating.

Presentations were also given by three members of the Japanese mission on heat pumps in DH plants in Japan, the Super Heat Pump Project, and heat pumps for cold districts.

From both sides, this forum of information exchange was considered as very effective and fruitful, thanks to all those who contributed and to the contacts resulting from the IEA Heat Pump Center project.

Editor's note: Newsletters for distribution in the Netherlands contain a special insert (in Dutch) detailing the visit of the Japanese delegation.

Ir A.J. Meijnen, IEA HPC National Team, MT - TNO, Apeldoorn, The Netherlands

Cogeneration/heat pump systems for local and district heating

Factory assembled cogeneration heat pump units are being sold on the European market for use in local and district heating applications. These units consist of a natural gas fueled engine, an electric generator/motor and a heat pump screw compressor. All components are mounted on a common frame and are isolated from each other by magnetic couplings. The electric generator/motor is mounted between the gas engine and the heat pump compressor. The design of the system allows it to operate in various modes. It can operate for example as an electric generator, a gas engine driven heat pump, an electric driven heat pump, or as a combination heat pump electric generator. This flexibility allows the most economic operating configuration to be chosen for the given conditions. In addition during heat pump operation the gas engine can be driven at its optimum speed since the excess power output can be taken up by the electric generator. BBC-York GmbH, Germany is manufacturer of these systems. (ref: "Einsatz von Wärmepumpen in der Nah- und Fernwärmeversorgung" by H. Jacobowsky, Fernwärme International Vol 8 (1989) No. 1.)

Closed loop water source heat pump systems in Germany

Systems using small size water-to-air heat pumps for air conditioning shopping centers and office buildings are beginning to be used in Germany. In these systems, which are already popular in such areas as North America and the United Kingdom, each heat pump is connected to a closed water loop typically having a cooling tower and a supplemental source of heat. The heat pump serves a specific area providing heating or cooling as required. In Germany this concept is still considered "new" and is being cautiously accepted by HVAC design engineers. Benefits of this technology include long operating lifetimes with low maintenance, energy savings, and individual tenant control of energy use. Will these systems gain wide acceptance in Germany? It seems that building owners and operators are most interested in these systems and are getting their design engineers to consider the feasibility of their use. German heat pump manufacturers are also working on improving their products in order to gain a wider market. (CCI 9/88)

Japan

Countermeasures and future problems regarding total abolishment of CFCs in Japan

Based on the Montreal Protocol Treaty Nations Conference held in Helsinki in May of this year, the Helsinki Declaration has been adopted as an international agreement that proposed to totally abolish CFCs as early as possible within this century. Described below are countermeasures and problems regarding the total abolishment of CFCs, centering around those in the field of refrigeration where many technical problems remain unsolved.

Japan's basic approach to CFC control

In May 1988, as Japan's legal measure for observing the "Montreal Protocol Concerning Substances Depleting the Ozone Layer" (hereinafter referred to as "Montreal Protocol"), the Japanese government took the initiative in the world by promulgating and implementing the "Bill Concerning the Protection of the Ozone Layer Through Control of Specific Substances" (hereinafter referred to as "Bill for Controlling CFCs").

The Bill for Controlling CFCs stipulates not merely the control on the quantity of production and consumption for observing the Montreal Protocol, but also steps to be taken for a smooth implementation of CFC control through various emission control/rationalized utilization measures (i.e., seal-off, reclaim/recycling, introduction of alternatives, etc.).

This Bill is intended to reduce CFC emissions that affect deeply human living conditions without throwing into confusion industrial activities. This will prevent any disturbance to national life.

At present, the public in every nation is not fully aware of the fact that almost all alternative substances known so far are inferior to CFCs in performance. Therefore, to achieve a total abolishment of CFCs, it is important to take the utmost care so as not to cause any confusion to the national life.

Countermeasures and problems for different fields of use

Centrifugal refrigerating machines. R&D for transfer from CFC-11 to HCFC-123 are underway, and it is expected that beginning in 1995, centrifugal refrigerating machines using the alternative CFC will be commercialized.

To fill the demand for refrigerant for existing equipment it will be necessary to depend on recycling.

According to the examination by the technical panel that is revising the control level under the Treaty Nations Conference, the time for total abolishment is likely to be delayed to the year 2015. Technically, however, it seems quite possible to shorten this time.

Automotive air conditioners. Although R&D for transfer from CFC-12 to HFC-134a is underway, drop-in performance is not possible due to reasons such as low mutual solubility between HFC-134a and oil. Thus, further research is necessary.

Meanwhile, from the point of view of the possible greenhouse effect on the earth, the Environmental Protection Agency (EPA) in the United States is beginning to assert that a tertiary mixture (HCFC-22, HFC-152a, and HCFC-124, pseudo-azeotropic mixture) developed by DuPont should be adopted rather than HFC-134a. However, since HCFC-22 features a large selective permeability to rubber hose, it will be impossible to use it in automotive air conditioners.

As for filling in the existing automobiles, it is expected for the time being that HFC-134a and CFC-12 will remain coexistent. However, since a performance deterioration will occur when they are mixed, it is necessary to adopt a strictly separated supply system. Also, by reclaim/recycling, improvement of rubber hose, etc., losses must be minimized while efforts to develop drop-in type refrigerants continue.

Incidentally, since international compatibility is required for these products, it is vital that automotive manufacturers in the nations which signed the treaty take appropriate steps while cooperatively adjusting the basic orientation towards the total abolishment of CFCs.

It seems that at the technical panel for reviewing the control level, discussions are being held towards delaying the time for total abolishment to around the year 2015. However, it is possible technically to advance the total elimination of CFCs in this application to before 2015 through collection and recycling of the refrigerant.

Domestic refrigerators. Although R&D efforts for transfer from CFC-12 to HFC-134a are being carried out, they are faced with a low mutual oil solubility, as in the case of automotive air conditioners. What's more, when a polyglycolic oil is used, a deterioration of electric insulation at the driving part causes some problems. A continued effort is thus necessary to solve these problems.

Incidentally, the latest recommendation to adopt tertiary mixture system featuring a high cooling efficiency and a low greenhouse effect upon the earth must be carefully examined by securing a large amount of samples to confirm its availability, performance, etc., in the consumer industries.

Although it is possible to examine a case in which HCFC-22 can be adopted partially as a short-term substitute prior to industrial production of HFC-134a, an effort should be made to realize it while taking into account the effect of increased discharge temperature on the living environment.

However, since the use of HCFC-123 and 141b (substitute candidates for CFC-11) for hard urethane foam involved problems, such as a drastic drop in heat insulating performance and a deterioration of foaming performance, it is expected that other measures will be essential such as the introduction of vacuum heat insulating technology. Thus, it is necessary to carry out overall research on durability, etc., as soon as possible.

Commercial refrigerating machines. The research on changing from CFC-12 to HFC-134a and from CFC-115 to HFC-125 as an ingredient of CFC-502, etc., is underway. The use of HFC-134a in commercial refrigeration presents problems similar to those for domestic refrigerators. Thus research on the feasibility of units using HCFC-22 is being examined on a small scale, but its use is limited.

At present, no proper refrigerant for replenishment is found. However, it is necessary to secure an ample amount of the tertiary mixture (HCFC-22, HFC-152a, and HCFC-124) to examine its availability since it may be used simply by changing the molecular sieve.

Smooth realization of total abolishment

In order to realize the total abolishment within this century, it is imperative that (1) substitutes be developed and supplied by CFC manufacturers and (2) technologies to utilize substitutes and also technologies for CFC-less systems be introduced by consumer industries. The major task is to facilitate steps for the total abolishment in smaller enterprises, by, for example, (1) stopping the use of CFCs through changes in the specifications ordered from parent enterprises and extending technical/economic support to enable such stoppage; and (2) improving the political environment by supplying information to facilitate the diffusion of substitute CFC utilization technologies and technologies for CFC-less systems.

In view of these circumstances, the industries concerned will have to make a concerted effort towards the total abolishment of CFCs. Administrative guidance and advice will have to shift its importance to aspects of CFC reduction to assist industry in this transition.

Hisashi Shingai, Basic Industries Bureau, Ministry of International Trade and Industry, Tokyo, Japan

North America

Recent activities concerning CFC reductions in the U.S.

In January 1989 the Montreal Protocol came into effect after being ratified by 37 nations representing 90% of the total world production of CFCs and halons. Since then the President of the United States has called for a complete phase out of all fully halogenated CFCs and halons covered by the Protocol by the year 2000, providing that safe alternatives are available. Recently 80 nations including parties to the Vienna Convention and the Montreal Protocol met in Helsinki and adopted the so called "Helsinki Declaration" to support additional ozone protection measures. The declaration also calls for a CFC phase out on or before the year 2000.

The U.S. Environmental Protection Agency (EPA) is of the opinion "that a complete phase out of the fully halogenated CFCs and halons by the year 2000 is feasible." Questions on safety, toxicity, energy efficiency, national security, and environmental impacts need to be resolved over the next few years. The U.S. EPA does not consider a complete phase out of CFCs by the mid 1990s, as called for in some proposed laws, to be technologically nor economically feasible.

At the Helsinki Conference the U.S. EPA presented the results of an analysis of atmospheric chlorine levels and their impact on the stratosphere. The contribution to chlorine levels in the stratosphere from HCFCs was also considered in the EPA analysis. It was assumed that HCFCs would take over 20 to 50% of the market created by a complete CFC phase out. "In summary, the analysis suggests that in the absence of further action, chlorine levels would significantly increase. The actual data in the EPA analysis presented in Helsinki states that to stabilize world wide chlorine at 1985 levels (2.7 ppb), a phase-out of CFCs, methyl chloroform, and carbon tetrachloride are necessary, and HCFCs could be substituted for only about 20% of the market left by the currently planned phase-out of CFCs."

To assist industry in meeting the goal of reducing CFC use, the EPA is working with U.S. industries to identify alternatives. This involves organizing various industry groups so that they can decide how to best reduce and eventually eliminate their use of CFCs. For example a number of manufacturers have begun testing of alternative CFCs or have modified their processes to reduce or eliminate the use of CFCs. (The above is taken from a document titled "Testimony of William Rosenberg, Assistant Administrator for Air and Radiation, U.S. Environmental Protection Agency, before the Subcommittee on Environmental Protection, Committee on Environment and Public Works, U.S. Senate May 19, 1989".)

GRI gas fired cooling based on heat pump technology

Introduction

In the United States, recent changes in the price of energy--oil, natural gas, and electricity--have provided an opportunity for the natural gas industry to develop and market gas-fired cooling systems based on previously developed heat pump technology. Competitive pressure from oil pushed natural gas prices up in the 1970's, generating interest in high-efficiency appliances. Condensing furnaces and heat pumps were considered cost-effective, despite their high first-cost premium, due to the fuel savings and lower operating costs. The subsequent erosion in worldwide oil prices impacted natural gas costs and diminished the attractiveness of high-efficiency heating equipment.

527-kW (150-RT) engine-driven cooling program

In 1984, the Gas Research Institute (GRI) initiated a program to develop a high-performance gas cooling system based on the integration of a gas engine and an absorption chiller. In this concept, the gas engine would drive a conventional refrigeration compressor, and waste heat from the engine would drive the absorption chiller, supplementing the cooling capacity. A system coefficient of performance (COP) was targeted in excess of 2.0, effectively doubling the performance of state-of-the-art gas cooling systems. In addition, the integrated engine chiller would achieve competitive first-cost targets.

A prototype chiller system was fabricated and installed in a nearby hospital to evaluate in-the-field performance and installation. The cost of the absorption module (which was affected by currency exchange rates), the complexity of the piping, and the floor space requirements were unable to warrant the incremental fuel savings. A decision was made to focus the development activities solely on the engine-driven chiller.

The engine chiller was equipped with a microprocessor-based controller that

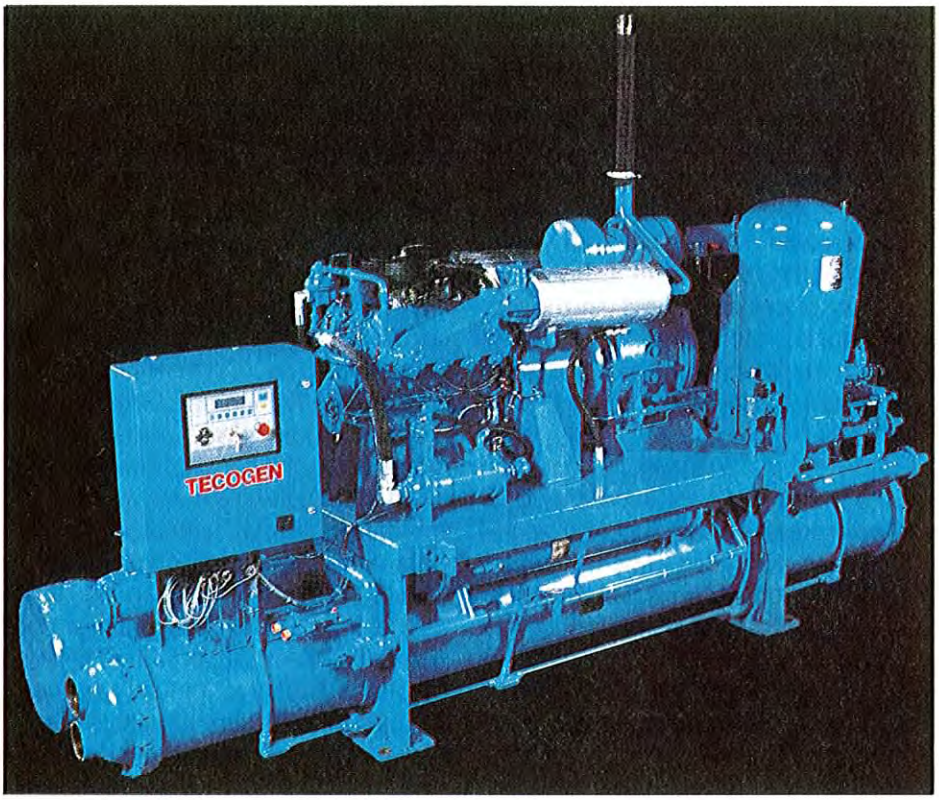

Figure 1

monitored engine and compressor functions. The controller permitted the engine to warm up, shut down slowly, and vary speed for optimal performance and longevity. The controller was also capable of remote monitoring, which enabled technicians to diagnose problems without the expense of a field visit. The enhanced capabilities of the controller were critical to meet GRI's targets for maintenance cost and reliability.

In 1987, a seven-unit field test was initiated. Figure 1 shows the advanced design. The 527-kW water chillers (nominally rated at 457 kW) were installed in a variety of commercial buildings and in several configurations.

Fleet operating hours:	23,055
Average COP:	1.42
Average availability:	96%

In the course of the field test, numerous flaws in the control strategies, compo-

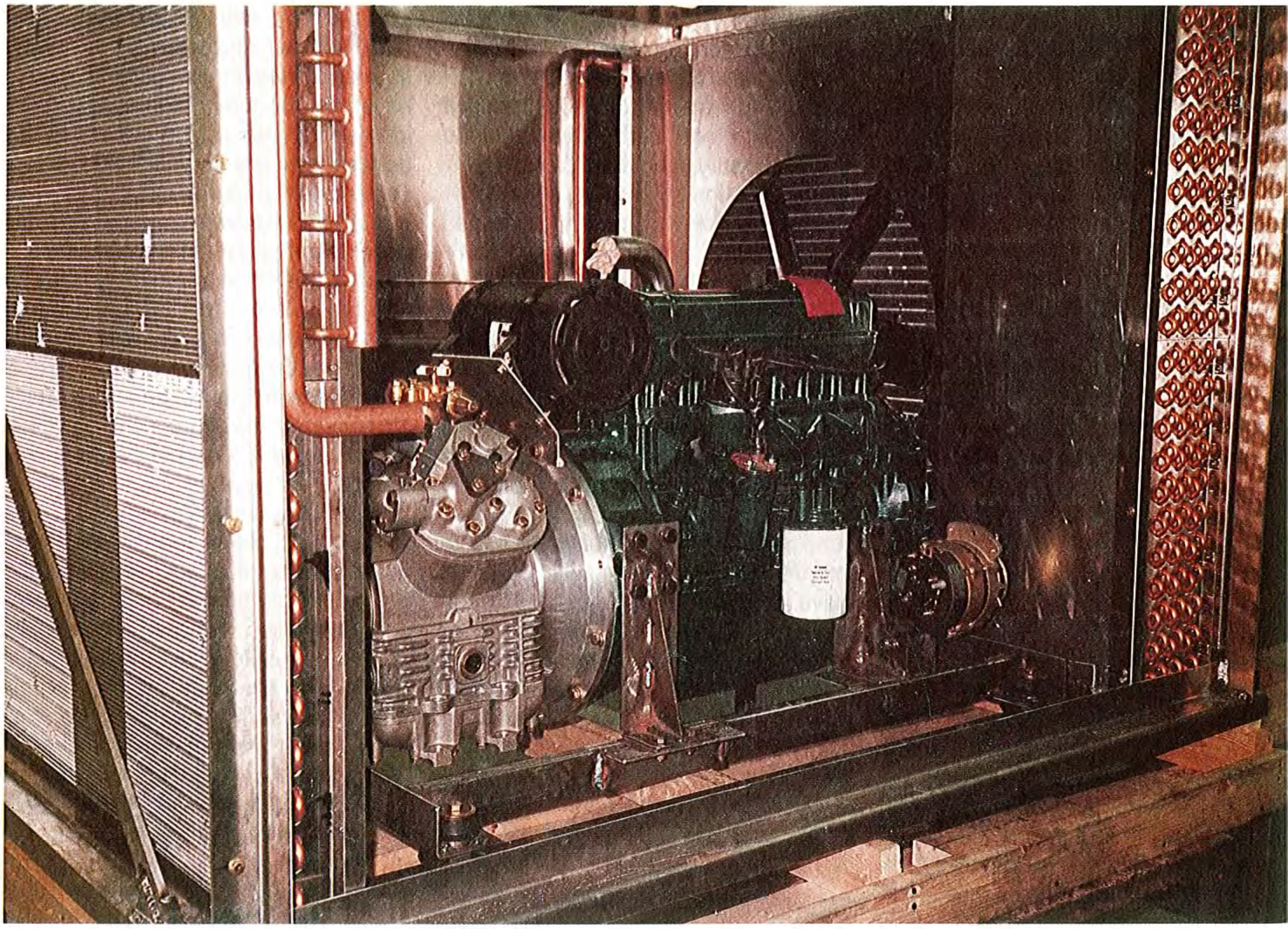

Figure 2

nent design, and installation procedures were identified and corrected. The improvements enhanced users' confidence in the system and demonstrated the benefits of the advanced controls. Engine performance and durability during the field test was satisfactory throughout the field test, with only one site experiencing an engine-related problem. The benefits of certain design options such as economizer circuits, cycling at low loads, and extended engine oil systems were evaluated.

The enthusiasm of the participating users and utilities, the outstanding reliability and performance of the prototypes, and the favorable market were sufficient for Tecogen to commercialize the unit in 1988. As of March 30, 1989, 45 commercial units have been shipped and 70 units were on order. The units are also being modified for industrial and low-temperature applications by Tecogen and cooperating gas utilities. Slight variations in heat exchanger selection and engine speed will permit Tecogen to offer the units in 422, 527, and 632-kW capacities. A second-generation design, using a different compressor, will be available in 1989 and should be very competitive with conventional electric chillers.

Gas engine-driven unitary cooling packages

The 527-kw engine chiller program targeted large commercial buildings with central heating, ventilation, and air conditioning (HVAC) systems. Unfortunately, 57% of the commercial buildings in the United States are one-story construction and typically heated and cooled by unitary equipment. In order to effectively impact the potential gas cooling market, gas-fired unitary systems were required.

Unitary packages are generally electric-driven, direct-expansion (DX), vapor-compression cooling systems with or without heating capability. Commercial packages range from 17.6 to 176 kW in capacity. Units may be configured as fully self-contained rooftop packages, split systems (interior evaporator and exterior condenser), or heat pumps. Unitary packages are widely specified due to:

-- Simple design requirements,
-- Ease of installation,
-- Single-source responsibility for service,
-- Minimal floor space requirements, and
-- Most of all, low cost.

Efforts to develop gas engine-driven unitary heat pumps over the past twenty years have generated an extensive data base on refrigerant properties, compressor performance, engine design, and control strategies. The commercial success of these ventures has been limited, however, due to the inherent high cost of these packages,

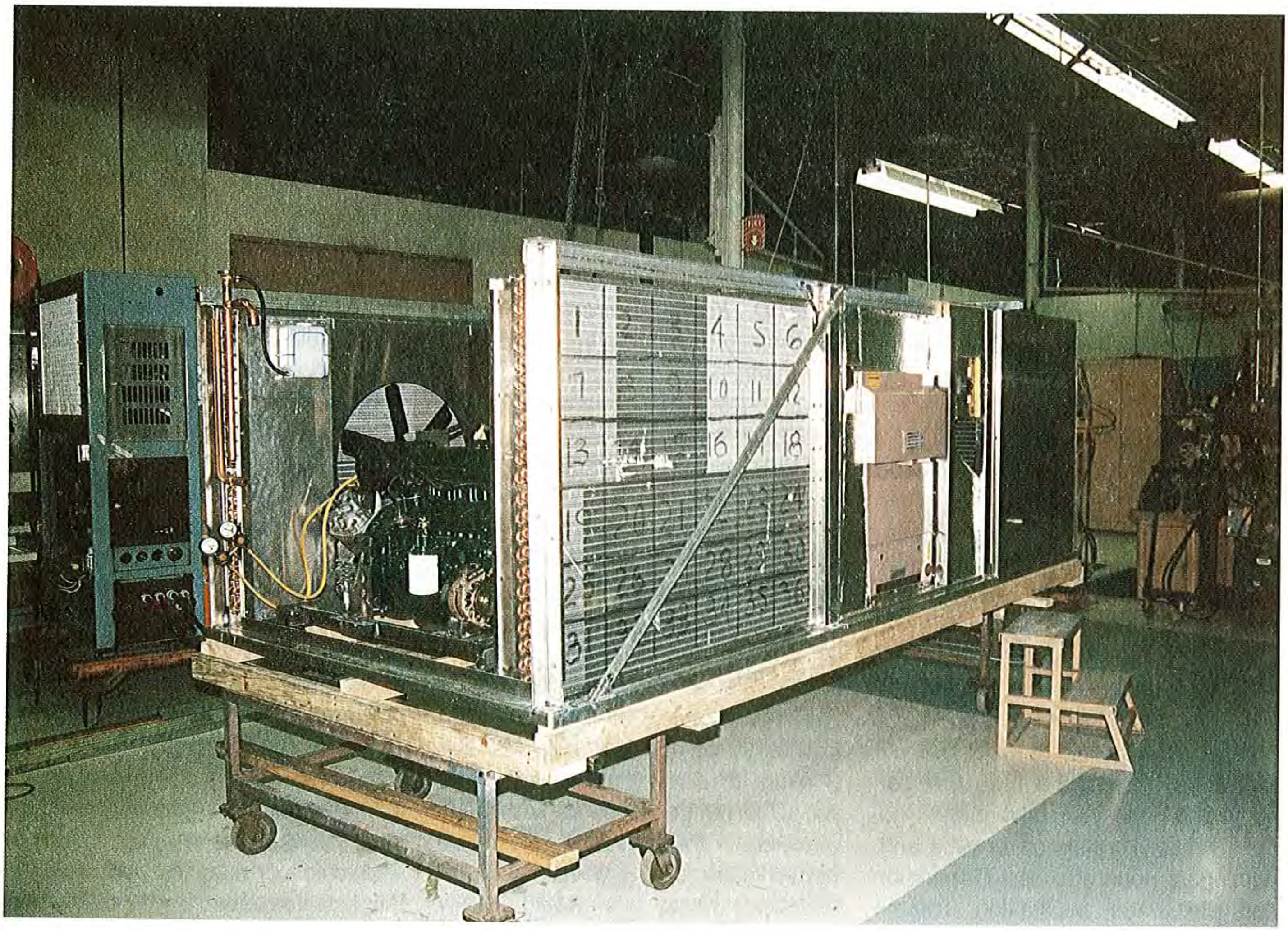

Figure 3

the lack of qualified service, and manufacturer reluctance to enter and support the market. The marketing limitations greatly outweighed the technical challenges.

In 1986, GRI embarked on a program to develop gas engine-driven rooftop unitary systems for cooling-only applications. Heating, if required, would be provided by a conventional gas furnace. The availability of moderately priced condensing furnaces and the low cost of natural gas did not warrant the additional maintenance and engine costs associated with a heat pump. The economics of the package would be based solely on cooling costs.

American Gas Association Laboratories (AGAL) was awarded a contract in 1987 to work with Thermo King to develop a 53-kW gas engine-driven rooftop package. The package would conform to conventional HVAC specifications and codes. AGAL and Thermo King have completed the design, fabrication, and testing of the laboratory prototype. The refrigeration module consists of a Thermo King TK430 reciprocating compressor, a Hercules NG-1600 gas engine, and an engine-driven condenser/radiator-fan assembly. The engine operates at three speeds: 2400, 1600, and 1100 r/min. The full load COP of 0.653 appears to be low, but the improvement in compressor performance at the lower speeds, where the unit will primarily operate, enables the system to achieve a seasonal COP of 1.16. A maintenance interval of 3000 hours is targeted, consistent with Thermo King's diesel package maintenance requirements. The laboratory prototype is shown in Figures 2 and 3.

Thermo King is very enthusiastic about the new gas engine rooftop product line. Economic analyses conducted by AGAL have identified 16 major U.S. cities where the Thermo King unit can achieve a three year payback. A national field test is planned for 1989, and, contingent upon successful operation and site owner satisfaction, Thermo King will introduce commercial units in 1990-91 in selected cities. Thermo King may also consider sales outside of the United States. AGAL and Thermo King are also examining the feasibility of producing a split system and heat pump version of the 53-kW unit, as well as the practicality of smaller systems.

Other development activities

In addition to the 527-kW engine chiller and the 53-kW gas engine unitary rooftop package, several additional gas engine-driven systems are under development by GRI and others. These include a 1760-kW (500-RT) engine-driven centrifugal chiller, a 1054-kW (300-RT) engine chiller with ice storage, and an 89-kW (25-RT) rooftop package. Hercules Engine, Gemini Engine, American Utility Control, and Econo-chill have developed units, ranging from 105 to 704 kW capacity. These new systems will greatly expand the ability of the natural gas industry to pursue commercial cooling and secure a large segment of the commercial market. In addition, these non-electric units will assist the electric industry's efforts to manage its peak demands and operate more efficiently.

Bruce B. Lindsay, Senior Project Manager, HVAC Systems, Gas Research Institute, Chicago, Illinois, USA; Michael D. Koplow, Director, Engine Technology, Tecogen Inc., Waltham, Massachusetts, USA; James Hatfield, Director, Research and Development, American Gas Association Laboratories, Cleveland, Ohio, USA.

U.S. EPA strengthens industry recovery and recycling efforts

The U.S. Environmental Protection Agency (EPA) has reaffirmed its decision not to classify used refrigerants and used refrigerant oil as hazardous waste under authority by the Resource Conservation and Recovery Act. A hazardous waste label would greatly hinder efforts to recycle chlorofluorocarbon (CFC) refrigerants, according to representatives of the air-conditioning and refrigeration industry. With the Montreal Protocol requiring a 50% reduction in CFC production by 1998, recovery and recycling of CFCs are viewed as necessary to continue some refrigeration and air-conditioning applications until CFC substitutes can be put into production and distribution.

The EPA action followed a roundtable meeting sponsored by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) and the Air-Conditioning and Refrigeration Institute (ARI) at which leaders of industry associations reviewed technological developments related to the ozone depletion issue, progress toward conserving CFCs and developing alternatives.

In addition to the hazardous waste issue, industry concerns for making the transition to replacement refrigerants and the impact of CFC substitutes were discussed by industry representatives. A brief summary from their discussions about individual industry concerns and policies follows:

-- Representatives agreed that individual industries should continue to develop specific data to test acceptability of reclaimed refrigerants to supplement ARI's Standard 700-88, a generic standard for refrigerant quality reuse.

-- Also, individual industries should prepare contingency plans to supplement ARI's overall industry vintaging study of what percent reduction of harmful CFCs can be met by using HCFC-22 and R-502 in the event that such replacement refrigerants as HCFC-123 and HFC-134a are not forthcoming.

-- In preparation for the reassessment of the Montreal Protocol, industries should consider the economic impacts that the revised document will have through the year 2000.

- ASHRAE will continue its efforts to develop Guideline 3P covering the reduction of CFC emissions during installation and servicing of air-conditioning and refrigerating systems.

- Industry information on the status of replacement refrigerant development and other relevant data should be shared with established information gathering groups.

- Overall, when possible for both retrofit and new construction, systems that do not use CFCs should be installed.

- Leadership in promoting the responsible use of refrigerants should continue. This includes reducing leakage and minimizing emissions through recycling and reclamation. The industry should also apply these practices to HCFC-22.

ASHRAE-ARI CFC Roundtables provide a forum for the assessment of scientific research and allow for dialogue on technological solutions to reverse ozone depletion. The first roundtable took place in June 1988.

ASHRAE calls for comments on guidelines for reducing CFC refrigerants

ASHRAE announced a sixty-day public review for proposed ASHRAE Guideline 3P "Guideline for Reducing Emission of Fully Halogenated Chlorofluorocarbon (CFC) Refrigerants in Refrigeration and Air-Conditioning Equipment and Applications." The public review began on July 15, 1989, and ends on September 14, 1989.

The purpose of this proposed guideline is to recommend practices and procedures that will reduce the inadvertent release of fully halogenated CFC refrigerants during the manufacture, installation, testing, operation, maintenance, and disposal of refrigeration and air-conditioning equipment and systems. It covers all refrigerating and air-conditioning equipment and systems that use CFC refrigerants.

The proposed guideline will be submitted to the American National Standards Institute's Board of Standards Review for approval as an American National Standard. Copies of the draft guideline may be ordered from Manager of Standards, ASHRAE, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA.

ASHRAE publishes resource book

ASHRAE has announced the publication of "CFCs: Time of Transition." Compiled from a variety of ASHRAE literature and other worldwide technical sources, this new 262-page publication is a comprehensive information source on chlorofluorocarbons (CFCs), covering emerging technology and the reduction of CFC emissions. It includes 32 papers written by more than 60 recognized authorities in the field and arranged into four parts: The CFC Issue, Alternatives, Applications, and Recycling/Recovery.

"CFCs: Time of Transition" reviews the history of CFC development along with the chemical make-up and properties of CFCs and other refrigerants. Use of alternatives to CFCs, such as HFCs and HCFCs, are covered in 13 papers. Refrigerator-freezer testing with alternative CFCs, materials compatibility of R134a, standards for acceptable levels of contaminants in refrigerants, and the impact of the Montreal Protocol on automotive air conditioning are among papers included in the Applications Section. The last section covers the true cost of refrigerant leaks, automated leak detection, reclaiming refrigerant in OEM plants, and recovery by filtration methods. ASHRAE has assembled the collection of papers to provide its members with a single source of current CFC information.

"CFCs: Time of Transition" is available from ASHRAE Publication Sales, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA.

Energy conservation standards for U.S. industry

ASHRAE announces the publication of Standard 100.4-1984, "Energy Conservation in Existing Facilities -- Industrial." The ASHRAE 100 standards series provides requirements for the conservation of nonrenewable energy resources in different types of existing buildings.

Use of Standard 100.4-1984 will improve thermal performance of the building envelope; increase the efficiency of the energy-using systems and components; and provide procedures and programs essential to the energy-conserving operation, maintenance, and monitoring of the existing building's systems. This standard applies to all existing industrial facilities. It is intended to encourage analysis of manufacturing processes contained within the following: industrial facilities; manufacturing facilities; utility facilities; rolling mills and foundries; storage and warehouse structures; and facilities for reclaiming waste energy from processes contained within industrial buildings.

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

Schedule of Conferences

October 10-11, 1989

Washington, D.C. (USA); **International Conference on CFC & Halon Alternatives.** Sponsored by the U.S. Environmental Protection Agency, the Alliance for Responsible CFC Policy, and the National Institute for Emerging Technologies. Contact: International Conference on CFC and Halon Alternatives, P.O. Box 868, Frederick, Maryland 21701, USA.

October 26-28, 1989

Kuala Lumpur (Malaysia); **2nd Far East Conference on Air Conditioning in Hot Climates.** Contact: ASHRAE International Headquarters, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA, telephone 01-404-636-8400, telex 705343.

November 7-11, 1989

Paris (France); **International Heating, Refrigerating and Air-Conditioning Exhibition (INTERCLIMA '89).** Contact: CEP 7, Rue Copernic, F-75782 Paris, Cedex 16, France.

November 21-22, 1989

Hannover (FR Germany); **2nd IEA Heat Pump Center Workshop 1989.** Organized by the IEA Heat Pump Center in cooperation with the Deutscher Kälte- und Klimatechnischer Verein (DKV). Contact: IEA Heat Pump Center, Fachinformationszentrum Karlsruhe, D-7514 Eggenstein-Leopolds-hafen 2, telephone 49-7247-808351, telefax 49-7247-808666. **NOTE:** See page 36 for more details.

November 22-24, 1989

Hannover (FR Germany); **DKV - Kälte-Klima - Tagung 1989.** Contact: Deutscher Kälte- und Klimatechnischer Verein (DKV), Pfaffenwaldring 10, D-7000 Stuttgart 80, Federal Republic of Germany.

Nov 28 - Dec 1, 1989

Atlanta, Georgia (USA); **Meeting Customer Needs With Heat Pumps - 1989: A Conference and Equipment Show.** Sponsored by American Public Power Association, Edison Electric Institute, Electric Power Research Institute, International Ground Source Heat Pump Association, and the National Rural Electric Cooperative Association. Contact: David P. Ross, Policy Research Associates, Inc., 12121 Basset Lane, Reston, Virginia 22091, USA, tel 703/620-1008.

March 12-15, 1990

Tokyo (Japan); **The 3rd International Energy Agency Heat Pump Conference.** Contact in Japan: Secretariat, Heat Pump Technology Center of Japan, Azuma Shurui Bldg., 9-11 Kanda Awaji-cho, 2-chome, Chiyoda-ku, Tokyo 101, Japan, telephone 03-258-1035, telefax 03-258-1037, telex 222-4601 hptcj. Contact in North America: R.L.D. Cane, Energy Systems Centre, ORTECH International, 2395 Speakman Drive, Mississauga, Ontario, Canada L5K1B3, telephone 416/822-4111 x238, telefax 416/823-1446. Contact in Europe: W. Hochegger, Energiesparhaus Graz, Petersgasse 45, A-8010 Graz, Austria, tel 316-822045, telefax 316-826371, telex 31-2305.

September 24-26, 1990

Graz (Austria); **3rd International Workshop on Research Activities on Advanced Heat Pumps.** Organized by the Technical University of Graz. Contact: Doz. Dr. H. Schnitzer, Organization Committee, Workshop on Research Activities on Advanced Heat Pumps, Institut für Verfahrenstechnik, Inffeldgasse 25, A-8010 Graz, Austria, telephone 0316/7061, telex 311221, telefax 0316/77685.

Services & Publications

Inquiries

Have a specific question about heat pumps?

Contact the Heat Pump Center directly with your questions on all non-commercial aspects related to heat pump topics. HPC staff members will do their best to answer directly or point you to the right expert.

IEA Heat Pump Center
c/o Fachinformationszentrum Karlsruhe
D-7514 Eggenstein-Leopoldshafen 2
Fed. Rep. of Germany

Telephone: (0)7247-808-351
Telex: 17724710+
Telefax: (0)7247-808-666
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Reports

The following reports are published by the HPC:

Report No.	Report Title	
HPC-WR2	Proceedings of the Workshop on Ground-Source Heat Pumps (1987), 245 pages	DM 50,--/U.S. $30
HPC-WR3	National Reports on the Status of Heat Pumps (1987), 105 pages	DM 40,--/U.S. $25
HPC-R3-1	Comparison of National Standards Testing and Rating Procedures for Heat Pumps (December 1986), 162 pages	DM 50,--/U.S. $30
HPC-R-4	Inverter-Driven Heat Pumps (September 1988), 109 pages	DM 50,--/U.S. $30
HPC-WR-1/1-12	Workshop: Electric Heat Pumps for Retrofit in Existing Small Residential Buildings (1985), 13 separate reports, 506 pages	DM 70,--/U.S. $36
HPC-R-5	Report on the Application of Heat Pumps in Industry (1989), 62 pages	DM 50,--/U.S. $30
HPC-WR-4	Workshop Proceedings: The IEA Heat Pump Center's Future Activities and Organization (1989), 145 pages	Free of charge
HPC-HB-1	User's Handbook for Heat Pumps in Dairies (June 1989), 106 pages	DM 50,--/U.S. $30
----------	Industrial Heat Pumps, Summary Report on Research and Development Work (December 1988)	Free of charge

Write to the HPC for a complete listing of HPC publications.

The IEA Heat Pump Center Newsletter is published quarterly.

Subscribers from countries participating in the IEA Heat Pump Center (Austria, Canada, Fed. Rep. of Germany, Italy, Japan, the Netherlands, Norway, Sweden, and the USA) will receive the newsletter free of charge. Subscribers from other countries will be billed DM 80,-- per year.

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

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Future Issues:

Vol/No	Topic	Deadline for Contributions
7/4	Planning for the future: national R,D&D programs, incentives, codes, and standards	October 7, 1989

If you would like to contribute an article on any of these topics, please contact the Heat Pump Center to obtain a copy of our author guidelines. Our regular features (feedback, bibliographic review, news briefs, and schedule of conferences) are included in each issue.

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 Program. 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 program of collaborative RD&D consisting of 42 Implementing Agreements, containing a total of over 80 separate energy RD&D projects. IEA's address is 2 Rue André-Pascal, 75775 Paris Cedex 16, France.

Heat Pump Center

The IEA Heat Pump Center (HPC) was established in 1982 as Annex IV of the "Implementing Agreement for a Programme of Research and Development on Advanced Heat Pump Systems." Operating agent for the center is the Fachinformationszentrum Karlsruhe GmbH. Presently, nine countries are members of the HPC. These are: Austria, Canada, Federal Republic of Germany, Italy, Japan, the Netherlands, Norway, Sweden, and the USA. The language used in all material published by the HPC is English. Please write us at the address below if you would like to receive more information on the activities of the center.

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Address
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