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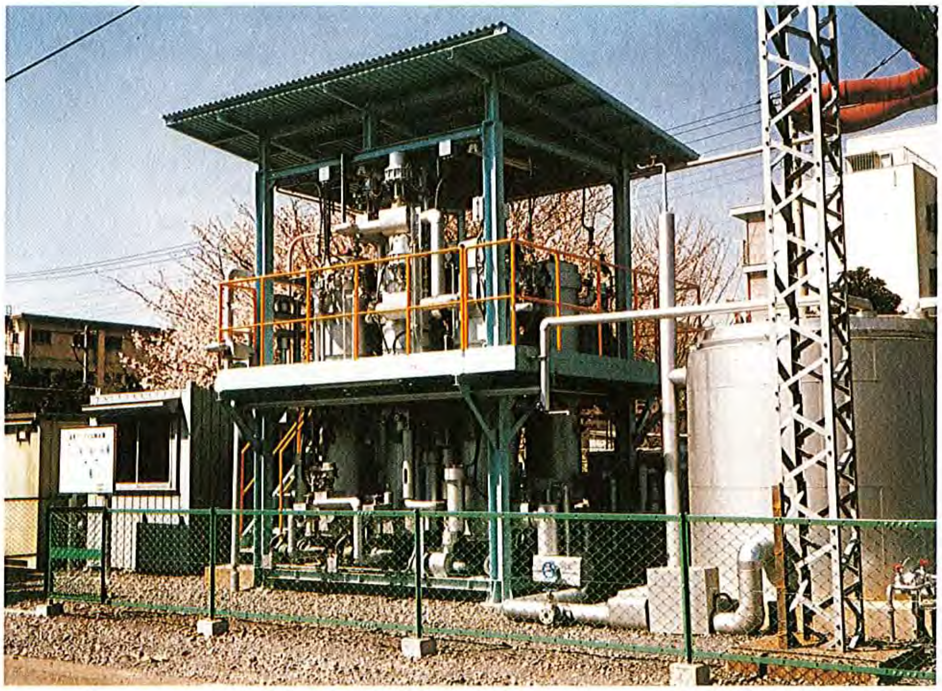

Prototype of two-stage heat transformer (see page 6)

This issue: Prospects in industrial heat pumps - process integration

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Editorial*

Industrial heat pumps is the main topic of the first Newsletter issue in 1989. It is a broad topic of interest to all member countries since the use of heat pumps in industrial applications is largely independent of climatic conditions. This topic also involves many heat pump specific aspects as well as industrial process design and integration. Heat pumps are playing an increasingly important role in helping reduce energy consumption in specific industries, thereby helping reduce operating costs and environmental impact. In areas such as drying and evaporation, industrial heat pumps are already widely used and have become a proven technology.

In other industries successful use is also being made of heat pumps after a careful evaluation of all energy saving options. The heat pump must always be considered as one of many energy saving options such as improved heat exchanger networks, process modification, or cogeneration to ensure an economical installation. For this purpose pinch technology is being used as a powerful analysis method.

What is an industrial heat pump and how is it different from an industrial refrigeration system? We asked this question to help define the scope of our work at the Heat Pump Center on this subject. As is the case with domestic heat pumps, historically industrial heat pumps were preceded by industrial refrigeration systems where the production of cold was the most important effect. Theoretically refrigeration systems and heat pumps use essentially the same thermodynamic cycle. From an engineering point of view to apply the term heat pump, the heating effect, though, must be of primary interest. For this reason we prefer to define an industrial heat pump as one where the heat produced is used in industry (excluding comfort space heating for industrial plant offices). This definition allows any type of heat source to be used by an industrial heat pump. Nevertheless in practice the industrial heat pump is defined differently by different authors.

Due to space limitations the articles in this issue of the Newsletter can only give a cross section of the industrial heat pump topic. An overview in *Application of Heat Pumps In Industry in IEA Countries* describes the results of a Heat Pump Center project investigating the market situation and economics of industrial heat pumps. Four additional articles are concerned with absorption heat pumps and heat transformers and their application in industry. This is an emerging technology especially in the case of heat transformers and is not yet widely used in industry. It is also a promising technology in that the possibility exists for high temperature lifting, reliable operation, and energy savings. One way of achieving higher temperature lifts is by the use of a two-stage heat transformer. The article *Research on a Two-Stage Absorption Heat Transformer* shows the kind of work being done in Japan to improve performance and to reduce capital costs and space requirements. A prototype unit which has been built and tested is described. A photograph of the unit is shown on the front cover. In order for heat transformers to be applied effectively in industry one must know the performance characteristics and limitations of this technology such as maximum COP and temperature lift. The article *Some Aspects on Heat Transformer Optimization* deals with these and other questions and presents the results of an optimization study. Interest in absorption heat pumps for use in industrial applications has also developed in China due to scarce energy supplies. The article *Absorption Heat Pump for Industrial Waste Heat Recovery* discusses this trend in China and describes a prototype heat recovery unit using Lithium Bromide and water as the working fluid pair. In Europe recovery of industrial waste heat is also of interest even though current energy prices are low and supplies are ample. The article *Resorption Compression Heat Pump with Solution Circuit for Steam Generation Using Waste Heat of Industry as a Heat Source* is one example of work being done in the Federal Republic of Germany.

As stated previously heat pumps have become common in drying applications in industry. The two articles *The Use of Heat Pumps for Drying Fish in Norway* and *Drying: An Important Heat Pump Application in China* show how heat pumps are being used in drying processes and describe the benefits derived. Higher temperatures and performance improvements over conventional heat pump drying systems are possible by combining solid desiccant dehumidifiers and vapor compression heat pumps. The article *Hybrid Solid-Desiccant Heat Pump System* discusses the advantages of such systems in air-conditioning and drying applications.

An example of the use of pinch technology to integrate heat pumps into an industrial process is shown in *Heat Pump Integration in Ethanol Separation*.

The three remaining articles in this issue are not strictly related to the topic of industrial heat pumps. These articles discuss unique analysis methods, new systems, and research and application trends. Of special interest is the article titled *Heat Pump Research and Application in Norway* from one of the member countries of the Heat Pump Center. The article gives an overview of the current market, statistics on the number of installed units, and information on the boundary conditions affecting the use of heat pumps in Norway. Current research and government support is also described.

Finally our bibliographic review presents recent papers and reports dealing with industrial heat pumps and heat pumps in general. A special review on the topic of pinch technology can be requested from the Heat Pump Center. In addition, the just-released Heat Pump Center report *Application of Heat Pumps in Industry* is also available.

**Editorial by IEA Heat Pump Center Staff, Karlsruhe, Federal Republic of Germany*

J. Berghmans*

Application of Heat Pumps in Industry in IEA Countries

The market potential and present situation of heat pumps in industry of member countries of the International Energy Agency Heat Pump Center is investigated by identifying sectors and processes which lend themselves to heat pump applications. Editor's note: The complete report entitled "Report on the Application of Heat Pumps in Industry" (HPC-R-5) is available at the Heat Pump Center (see page 43.)

Introduction

This paper contains the results of a study regarding the application of heat pumps in industry. It is limited to countries which are members of the Heat Pump Center of the International Energy Agency (IEA).

The study was undertaken in order to identify those sectors and processes which lend themselves for heat pump applications. The preparation of this report relied heavily upon collaboration with the IEA Heat Pump Center staff and with the National Teams from its mem-

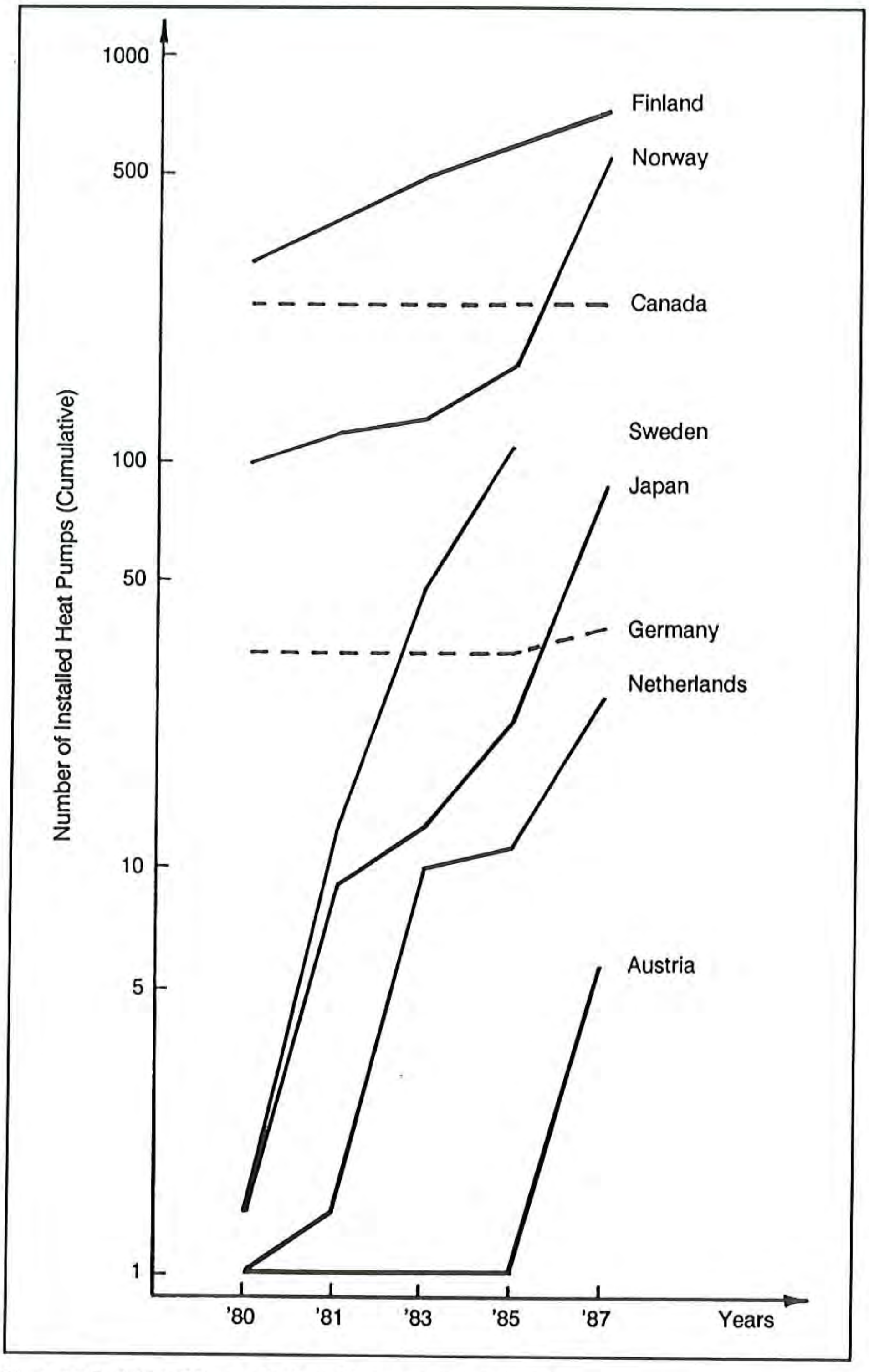

Figure 1. Evolution of the application of heat pumps in industry

ber countries. This fruitful collaboration is gratefully acknowledged.

Present market situation

Survey

During the fall of 1987 a survey was made of heat pump applications in IEA Heat Pump Center member countries. The survey was based upon a questionnaire which was sent to the national teams of these countries. Through use of the responses, it was possible to sketch the evolution of the application of heat pumps in industry. Figure 1 gives a representation of the result.

It should be pointed out that the data regarding Canada and the Federal Republic of Germany is not adequate enough to draw a time evolution for these countries. IEA Heat Pump Center countries not listed are considered to have very few heat pumps in operation.

The detailed survey data clearly indicates that heat pumps applied in industry consist mainly of the compression type. It should be mentioned that the data does not include mechanical vapor recompression systems.

Type 1 absorption heat pumps are applied to a limited extent in Japan and the Netherlands. Also, only a small number of heat transformers are applied in Japan, the Federal Republic of Germany, and the Netherlands.

Heat pumps are applied in large numbers in Finland, Norway, and Canada. In each of these countries, most of the heat pumps are used for wood drying. In Norway, a few hundred heat pumps are applied also in the fish industry (fish culture and drying).

The application of heat pumps in Sweden, Japan, and Germany is distributed over many industrial processes.

With respect to heat pump output, it is found that all the wood drying heat pumps and almost all of the fish culture heat pumps have thermal outputs of 100 kW or less. Therefore, the thermal output of the majority of industrial heat pumps is rather small. Heat transformers have heat outputs in the MW range and are applied in the chemical industry only.

It can be seen from Figure 1 that industrial heat pumps were applied before 1980 in Finland and Norway. A very steep increase in heat pump applications took place in the early eighties in the Netherlands, Japan, and Sweden. It is also estimated that in these countries, in addition to Canada and the Federal Republic of Germany, that heat pump application has stagnated in the last two years.

Economics of heat pumps

A simulation model was developed aimed at the determination of the payback period of a heat pump investment. By means of this model the combination of electricity price (E) to fuel oil cost (L.F.) giving rise to a payback of 2 years can be determined. Parameters in this model are of course heat pump capacity, useful temperature level, temperature lift and yearly hours of operation of the heat pump. Figure 2 gives a typical

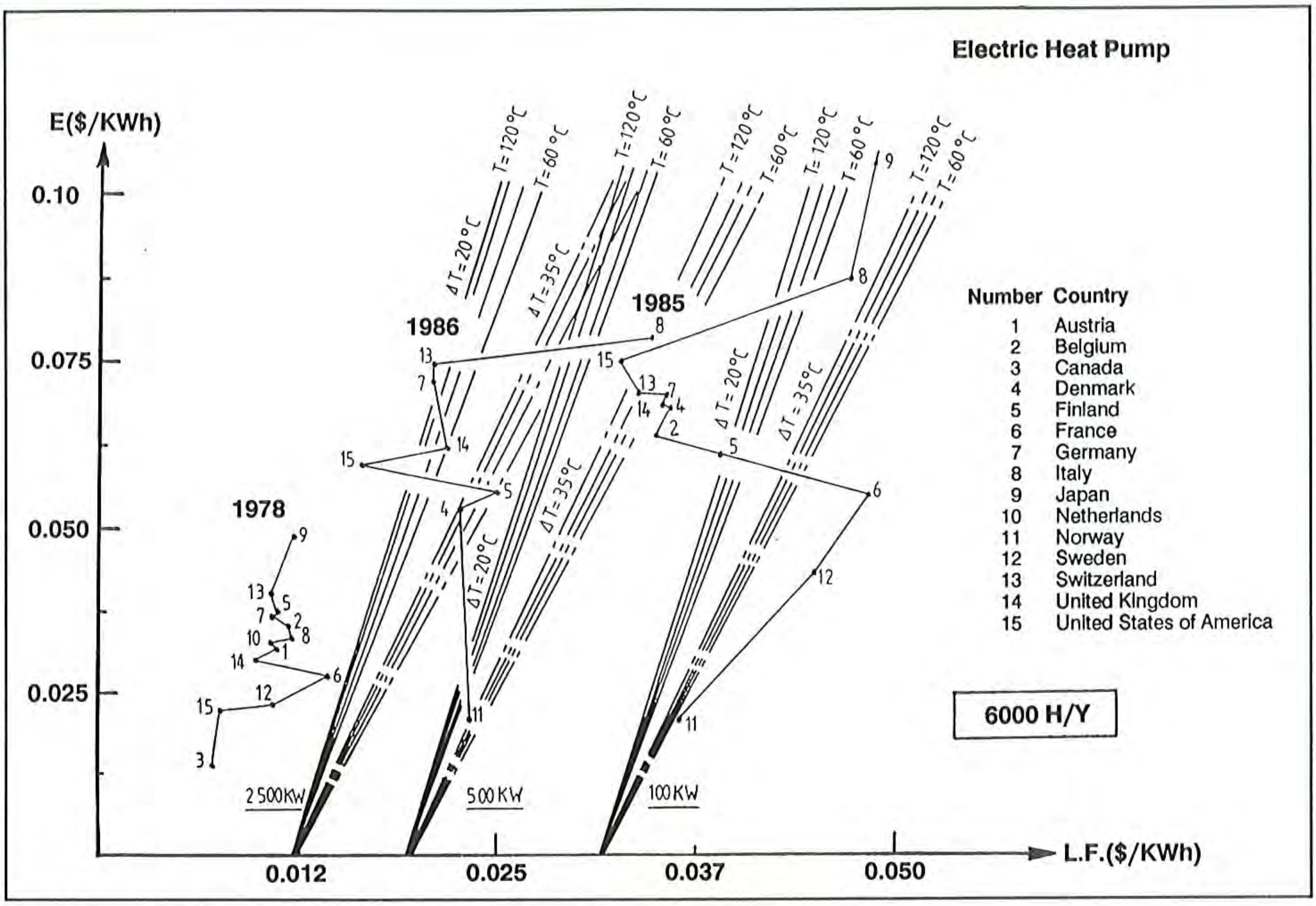

Figure 2. Electric heat pump economics at 6000 operating hours/year

example of the results which can be obtained by means of the simulation model for a heat pump running 6000 hours per year. This figure also shows the cost of electricity and fuel oil over the years in the countries identified in the figure.

From these figures it can be decided whether a heat pump application is economically justifiable. For the heat pump application considered, the electricity cost in the country considered should be smaller than the one derived from the straight lines of the figures (given capacity, output temperature, temperature lift, hours of operation, and fuel cost). As the correlations for the cost of heat pumps obviously related to 1987, only comparison with the national 1985-86 data for electricity and fuel costs is possible.

From Figure 2 and from the figures obtained for other run times of the heat pumps it is clearly seen that the present cost conditions are much less favorable than those existing in 1978, for example. A high run time, high installed capacity, low temperature lift, and high delivery temperature are the major elements in making a successful application possible. In industrial sectors where run time is about 4000 hours per year, it appears that only Norway, Sweden, and France provide good opportunities for heat pumps with a capacity of 500 kW and higher (based on the energy cost of 1985). A somewhat lower potential for application is found in Finland and Japan. In these countries, possibilities are limited to capacities of about 500 kW and temperature lifts below 20°C. The other countries form a cluster and do not appear to offer good prospects for successful use of heat pumps. Likewise, when the energy cost changes to the level in 1986 the situation does not look good either. Only Sweden, Norway, and France become marginal in the megawatt range.

It is obvious that industrial sectors with a high runtime are more profitable in view of energy savings by use of heat pumps. For sectors giving rise to 600 hours per year and for capacities above 100 kW for Sweden, Norway, France, Japan, and Finland offer interesting prospects. The other countries provide good perspectives for heat pumps with a capacity of 500 kW and a low temperature lift. A decrease in light fuel price in 1986 results in a reduced potential for most countries. Only Sweden, Norway, and France show marginal potential for heat pump application. Most industrial sectors provide good opportunities for capacities around 100 kW and higher. One can conclude that for the light fuel costs of 1985, Sweden, Norway, France, and to a somewhat lesser extent Japan and Finland, provide good opportunities for a run time close to 8000 hours. A decrease in fuel price (such as in 1986) results in lowered prospects for a successful heat pump application. It is clear that heat pumps have a higher potential for application when replacing light fuel oil as compared to heavy fuel oil.

Finally, Table 1 presents in a condensed form the ,conclusions which can be drawn from the economic simulation model. In the case of electrically driven heat pumps, Norway, Sweden, France, and to a lesser extent Finland, Japan, Italy, and Denmark are countries where it is possible to achieve successful results concerning energy recovery where heat pumps are used. Only in Japan and Switzerland is the natural gas price high enough to assure realistic heat pump applications when gas motor driven heat pumps are used.

Conclusions

It can be concluded from the study that the market potential for heat pumps depends very much on the country and the industrial sector considered. Presently, economic conditions are not favorable for heat pumps in most IEA countries. On the average the potential for heat recovery of industrial heat pumps is found to be about 15% of industrial heat consumption.

Based upon the cost of electricity, Norway and Canada offer the best prospects for electrically driven heat pumps. Irrespective of the local cost of electricity, heat pump applications are most suited to low temperature processes with large annual run times, such as:

- Drying recovery for production of boiler feed water, washing, pasteurization, fish culture, etc.
- Simultaneous heating and cooling in chemical and other processes

It is to these processes and industrial sectors that efforts to promote heat pumps should be directed.

More details about the analysis upon which this article is based can be obtained from the IEA Heat Pump Center.

References

1. Perez Blanco, H. (1981). *Heat pump concepts for industrial use of waste heat.* (Oak Ridge National Laboratory: May).
2. Adolfson, W.F. (1979). Waste heat as an alternate fuel - its opportunities and potential. *Proceeding of the American Chemical Society 14th Intersociety Energy Conversion Engineering Conference.*
3. Yoshii, Takeshi (1987). Future prospects for industrial heat pumps in Japan. *Proceedings of the 1987 International Energy Agency on prospects in heat pumps technology and marketing,* April 28-30, pp. 25-335.
4. Berghmans, J.B., Gelders, Van Looveren, and Verelst (1982). *Ekonomische aspekten van de toepassing van de warmtepomp in de Belgische industrie.* (Katholieke Universiteit Leuven, Belgium: September).
5. Van der Woerd Wouterse, K. (1986). *Toepassing orieentatie absorptie-warmtetransmissiesystemen in de industrie.* (Stichting Projectbeheerbureau Energie-onderzoek: Utrecht, Netherland).
6. Mattson, C. (1986). *Gruppcentraler i industriesektorn, Alternativa vaermeproduktionssystem och oljeersaettningspotential.* (Swedish Council for Building Research Report R27).
7. *Energy balances of OECD.* (1987). (OECD/IEA Newsletter).
8. Knobbout, J.A. (1982). Heat pump systems. *E.C.C. Contractors meeting held on April 28.*

RUN TIME (h/yr)	LIGHT FUEL		HEAVY FUEL		NATURAL GAS	
	1985	1986	1985	1986	1984	1986
4000	Norway Sweden France (Q > 500 kW) Finland Japan (Q > 500 kW and $\Delta T < 20°C$)	Norway Sweden France (MW range)	Norway Sweden (Q > 500 kW)		Japan $\Delta T = 20°C : Q > 1MW$ $\Delta T = 35°C : Q > 2MW$ Switzerland $\Delta T = 20°C : Q > 3MW$	
6000	Norway Sweden France Finland Japan (Q > 100 kW)	Norway Sweden France (Q > 500 kW)	Norway Sweden (Q > 500 kW) France Finland U.K. Netherlands (MW range)	Canada (Q > 500 kW)	Japan $\Delta T = 20°C : Q > 300$ kW $\Delta T = 35°C : Q > 600$ kW Switzerland $\Delta T = 20°C : Q > 800$ kW $\Delta T = 35°C : Q > 1$ MW	Switzerland $\Delta T = 20°C : Q > 1.1$ MW $\Delta T = 35°C : Q > 2.5$ MW
8000	All countries (Q > 100 kW)	Norway (Q > 100 kw) Italy (Q > 100 kW and $\Delta T < 20°C$) Denmark Finland (Q > 500 kW and $\Delta T < 20°C$) Germany Switzerland (MW range)	Norway Sweden (Q > 100 kw) Finland France U.K. (Q > 500 kW)	Canada (Q > 500 kW) Norway (MW range)	Japan $\Delta T = 20°C : Q > 400$ kW $\Delta T = 35°C : Q > 300$ kW Switzerland $\Delta T = 20°C : Q > 400$ kW $\Delta T = 35°C : Q > 700$ kW	Switzerland $\Delta T = 20°C : Q > 400$ kW $\Delta T = 35°C : Q > 800$ kW

Table 1. Review of the most promising countries for gas motor and electrically driven heat pumps

9. Patwardhan, V.R., Devotta, and Patwardhan, V.S. (1987). Performance characteristics of a water-water heat pump using nonazeotropic mixtures. *Third International Symposium on the large scale applications of heat pumps*. (Oxford, England: March 25-27).

10. Berghmans, J. (1985). *Heat pump systems applied in industry*. (International Energy Agency Annex III Final Report: Katholieke Universiteit Leuven, Belgium).

11. Bratt, A. (1986). Estimating the profitability of industrial heat pumps. *IEA Newsletter, 4, (2), June.*

12. *Energy prices and taxes: second quarter 1985*. (1985). (IEA: OECD, Paris).

13. Reay, D.A., and Mac Michael, D.B.A. (1979). *Heat pumps, design and applications*. (International Research and Development Col. Ltd.: Newcastle-upon-Tyne, England).

14. Deakin, W. and Gluckman, R. (1987). Review of five industrial heat pump demonstration projects. *Proceedings of the 3rd international symposium on the large scale application of heat pumps*. (Oxford, England: March 25-27), pp. 25-31.

15. Mead, K.C., and et al (1984). *Solar 85 simulation modeling*. (Report D21: Swedish Council for Building Research).

16. *Energy End-Use Prices First Quarter 1988*. (1988) IEA/OECD Caddet Newsletter, Nr. 1, September, p. 12.

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Research on Two-Stage Absorption Heat Transformer

A two-stage heat transformer, equipped with special heat and mass transfer tubes (Fir Finned Tube), was studied. In the development work, optimization of the fin configuration of this tube was obtained experimentally and theoretically. The heat and mass transfer characteristics of the tube were analyzed by modelling the absorption mechanism. Coupled with the measurements of characteristics for various kinds of tubes, the heat and mass transfer coefficients were obtained and compared. The static and dynamic performance of the heat transformer was also measured. It was found that the transformer had twice the boosting temperature (60° K) of conventional heat transformers.

Introduction

Although energy saving efforts have relaxed due to decreases in the price of crude oil, efforts in research and development of energy-saving equipment should continue for the future.

It is well known that a heat transformer is very effective in reducing oil consumption in various industrial processes. To enhance the introduction of the transformer into the processes, it is important that the performance (especially the boosting temperature) of the transformer is sufficient for reaching the required temperature level in the process. According to our survey of industrial processes, it was found that the temperature difference between the waste heat and the required heat was often much larger than the temperature boosting performance of conventional transformers.

The newly developed absorption heat pump, which is called the Mitsubishi Heat Transformer (MHT), has the highest performance for recovering low temperature industrial waste heat and generating steam at a temperature level 60°K higher than that of the waste heat. This performance is twice that of conventional units.

This high performance was realized by introduction of a two-stage cycle to the basic absorption heat transformer. In addition, a special heat and mass transfer tube was invented to provide high efficiency and compactness to the MHT.

Prototypes of two-stage MHT

The principle of the MHT equipped with a two-stage cycle is based upon an absorption process of a working pair of Lithium-bromide and water. As far as is known, use of the two-stage cycle for the heat transformer is the first attempt in the world.

A conventional single-stage heat transformer can boost the temperature of waste heat by about 30°C. By introducing the 30°C boosted output to the second evaporator, the second stage can boost the temperature by another 30°C. Therefore, the two-stage MHT has twice the boosting performance of the conventional unit. This is the most significant feature of the MHT.

Another feature is the use of a high efficiency vertical heat exchanger equipped with a newly developed heat and mass transfer tube, called the Fir Finned Tube. In the installation of the transformer the required floor area should be minimized. Therefore, vertical heat exchangers were employed and the compact heat and mass transfer tubes (Fir Finned Tubes) were used.

The flow diagram of the MHT is shown in Figure 1. The two-stage cycle is realized by adding two extra heat exchangers (a second evaporator and a second absorber) to the basic absorption heat pump cycle. Since the MHT becomes part of automated processes, its starting, running, stopping and emergency operations should also be automated. As shown in Figure 1, three solution and two refrigerant pumps and liquid level sensors in every heat exchanger allow automatic operation and make the control highly reliable.

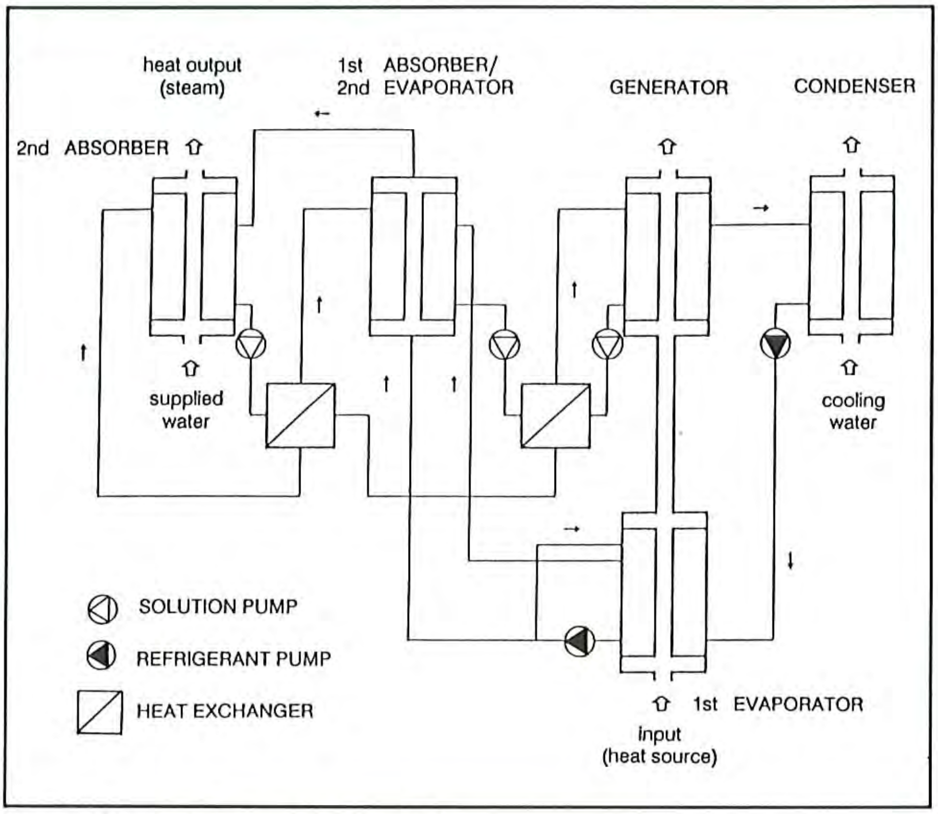

Figure 1. Construction scheme of 2-stage MHT

The two-stage MHT can also be operated as a single-stage MHT having a 30°C boosting temperature by valve control.

Analysis of heat and mass transfer in a Fir Finned Tube

Figure 2 shows a photograph of the Fir Finned Tube. The tube consists of a helical segmented fin around the tube. In the case of a vertical absorption process, LiBr solution falls down along the outer surface of the tube flowing slowly and spreading evenly over the surface area of the fins. Therefore, a highly efficient absorption process is achieved.

An analytical model of the absorption process in the falling film flow was developed. The whole tube is divided into about 10 segments along its axis. In each segment heat and mass balances are calculated by using the mass transfer coefficient in the concentration layer of the film flow, heat transfer coefficient on the outer surface of the tube, etc.

By integrating the calculated heat and mass flows from every segment, the total calculated flows are obtained and are compared with their experimental values. As a result, the coefficients for various tube geometries were obtained.

It was found that the Fir Finned Tube presents higher performance in both heat and mass transfers than either the bare tube or the corrugated tube.

Performance of the two-stage MHT prototype

A prototype of the MHT (nominal output: 77.2kW) was manufactured (see cover photo). This demonstration plant is being used to study both the static and the dynamic characteristics of the two-stage MHT and its long-term reliability. This plant is equipped with many pressure gauges, flow meters, thermo couples, and other measuring devices from which the experimental data are compiled and displayed by a computer.

The unit's heat exchanger shells and pipes are made from carbon steel. The heat transfer tubes and fins are made from copper.

Corrosion of the materials is the greatest problem. For high temperature machines operating at more than 150°C, it is said that the corrosion effect of LiBr upon the carbon steel is particularly enhanced. Anti-corrosion technology has been studied for long-term reliability of the machine. Based on these studies, it was found that LiMoO with LiOH was an effective inhibiter. For copper, some additives (for example, Benzotriazole) functioned well.

Steady and dynamic characteristics

Performance tests on the demonstration plant were carried out using steam as a simulated waste heat source to drive the MHT. Table 1 shows the static characteristics of the two-stage MHT. It is found that the MHT can generate 151.2°C output using a 91.0°C heat source and 32.5°C cooling water. Its COP in this case is 0.31. Thus the boosting temperature is 60.2°C in accordance with the design value. The obtained COP is very close to the theoretical value 0.32.

On the other hand the actual heat output was found to be as large as 90% of the design value. This seems to be due to the low thermal efficiency of the recuperator and the increased thermal resistance due to larger liquid flow rate for the longer tubes.

Figure 2. Fir finned tube

Heat source temp.	91.0 °C
Cooling water temp.	32.5 °C
Solution concentration difference	5.0 wt%
Recuperator temp. efficiency	0.86
Heat output	69.4 kW
Output temp.	151.2 °C
Output pressure	0.492 MPa
Temp. boosting	60.2 (°C) K
COP	0.31

Table 1. Characteristics of the two-stage MHT under steady state conditions

In the actual processes flow rate and temperature fluctuations of the heat source and the cooling water may occur. Dynamic response of the MHT to such fluctuations becomes important in the installation. Figure 3 shows an example of the response to an abrupt decrease of 40% of the heat source flow rate. The abrupt decrease causes the operating state of the cycle to be changed and it takes 30 minutes to return to steady state conditions.

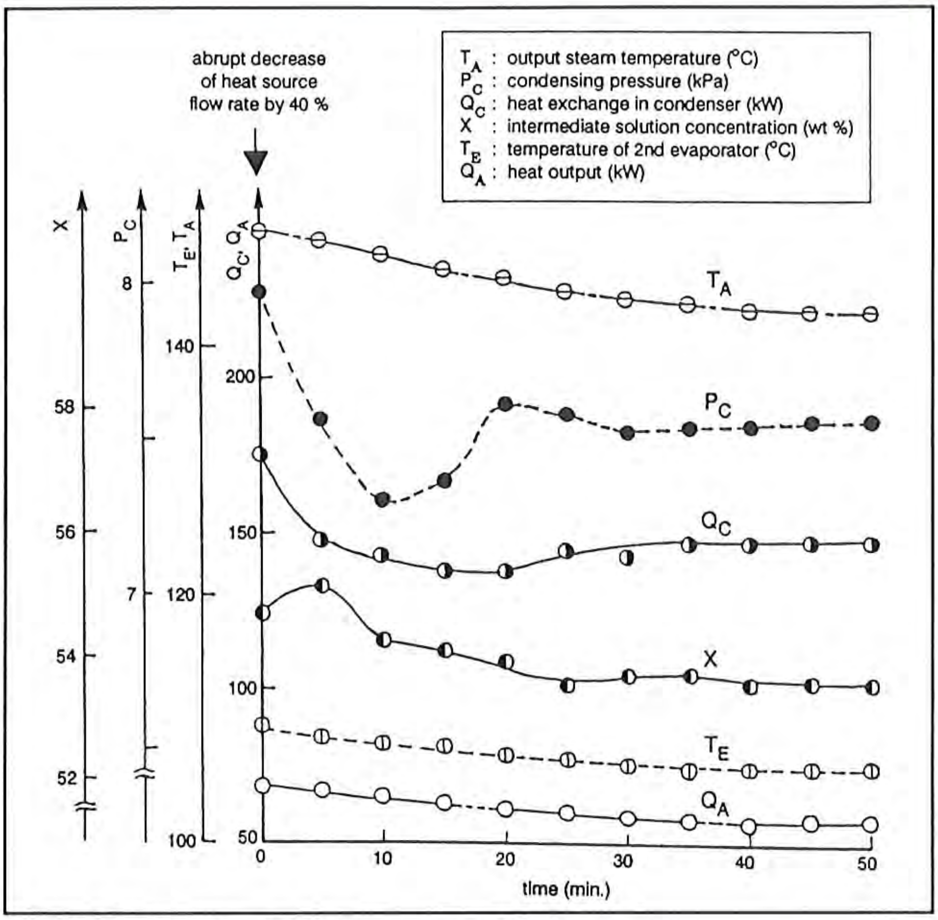

Figure 3. Experimental results of dynamic response of MHT

Economic potential

The effect of the MHT installation in industrial processes using steam was calculated. Assuming steam costs of 3000¥/ton steam and electricity costs of 20¥/kWh, it seems that the 1000kW MHT can save 35 million ¥/8000h and has a payback period of 2 to 3 years.

Conclusion

A two-stage absorption heat transformer has been manufactured for the first time. The transformer is equipped with Fir Finned heat and mass transfer tubes providing a COP of 0.31, close to the theoretical value, and a temperature boosting of 60.1°K, which is larger than that of the conventional units.

In addition heat and mass transfer phenomena in the falling film flow of the Fir Finned Tube were analyzed by using heat and mass flow models. Some coefficients in the model were determined by experiments. As a result an accurate design method was established and optimum absorber configurations were obtained by computer simulation.

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Some Aspects on Heat Transformer Optimization

A large amount of waste heat in the industry is today rejected to the surroundings. One way to recover part of this energy is to use an absorption heat transformer (AHT). In order to establish the potential of the AHT, optimization studies and sensitivity analyses, both technical and economic, have been done for the single-stage AHT cycle.[1] For these studies a computer program has been developed, which basically consists of a simulation program for the cycle combined with an optimization routine. In this article the most important results are presented.

Present status

There are today approximately 15 AHTs in operation in the industry.[2,3] The installed units are from three different manufacturers and situated in five different countries, although mainly in Japan. All are using $LiBr/H_2O$ as working fluid and are fairly large, 0.5-7 MW heat output. Another common denominator is that the heat source in nearly all cases is waste vapor while the heat sink is low pressure process steam.

Fundamentals

The AHT can be described as consisting of five heat exchangers, of which one is internal, and ancillary equipment, e.g., pumps. The cycle is shown schematically in Figure 1 with the different pressure and temperature levels indi-

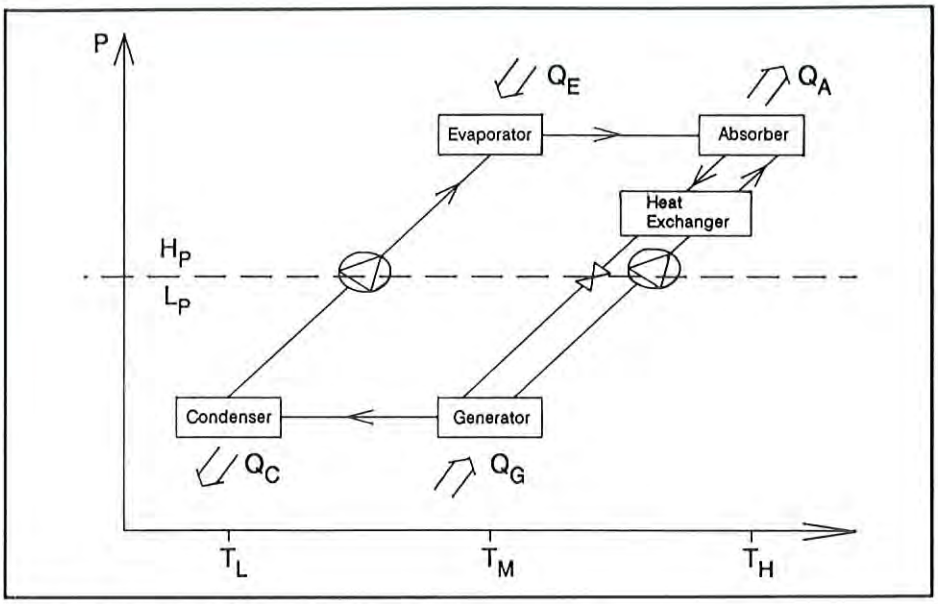

Figure 1. Schematic of the heat transformer cycle

cated. As can be seen from the figure, the heat flow to the AHT is at an intermediate temperature level, while the heat flow from the AHT is at a high and a low temperature level, respectively. The two important quantities that describe the technical performance of the AHT, the COP and the temperature lift, can now be defined:

$$COP = \frac{Q_A}{Q_G + Q_E}$$

$$\text{Temperature lift} = (T_H - T_M)$$

With $LiBr/H_2O$ as working fluid the theoretical value of the COP is over 0.5 due to the exothermic heat of mixing, though in practice the COP is lower due to internal irreversibilities. The temperature lift is in theory determined by the spread of the solution field (see Figure 2), and the temperature difference between the heat source and the low temperature heat sink. In absolute numbers a maximum temperature lift of approximately 70°C would be possible, but the need for a margin to crystallization concentration limits the temperature lift to 50-55°C maximum under the condition that the necessary temperature difference in the low end is available. Another restricting factor is that the maximum allowable temperature is approximately 150°C for the $LiBr/H_2O$ solution due to corrosion problems. The following figures can, therefore, summarize the practical limits of performance for the single-stage AHT:

$$COP \leq 0.50$$

$$(T_H - T_M) \leq 55°C$$

$$T_H \leq 150°C$$

To get a higher COP or temperature lift, multistaging would be needed, although at the expense of the other quantities. To get a higher absolute output temperature another working fluid or better inhibitors must be used.

Conditions

For the single-stage AHT with $LiBr/H_2O$ as working fluid one can, due to its specific and characteristic performance range, state some favorable conditions with regard to its surroundings. This will by necessity reduce the number of possible applications, although a large number should still remain. The most important and obvious conditions (shown in the table below) are discussed here in order to provide a well-defined starting point.

Favorable Condition	Reason
Heat source twice as large as heat demand	Maximum COP of 0.5
Large size	Nearly all cost is associated with the initial investment. A larger size should mean a lower specific cost. According to Zegers[4] the economic limit is 1MW.
Heat source in the form of latent heat (waste vapor)	This is given by the combination of large size and the fact that large temperature gradients deteriorate the possible temperature lift. The only possibility would be to have large flows and small temperature gradients which means that only a small portion of the heat content could be used.
Heat source of 60-100°C, heat sink 30-55°C higher (maximum 150°C)	Determined by the characteristics of the $LiBr/H_2O$ solution and to some extent by other alternatives (i.e., compression heat pumps) which in lower temperatures and lifts should be more favorable.

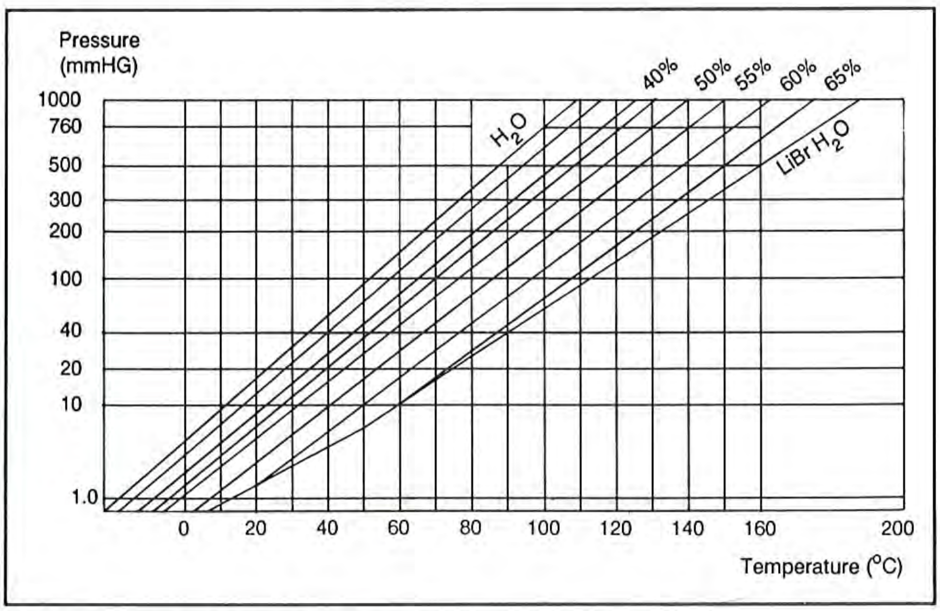

Figure 2. Solubility of $LiBr-H_2O$ as a function of pressure and temperature

Calculations

Although the area for application of the AHT is well defined, there are still some questions left to answer, namely what is not only technically possible but also economically feasible. As mentioned earlier the AHT can be seen as mainly consisting of a number of heat exchangers. The important consideration both economically and technically is the amount of heat transfer area needed to achieve a specified performance. This can also be extended to the question of what the minimum increase of heat transfer area, and, as a consequence, cost, is to achieve a better performance in terms of COP or temperature lift. For optimization of the heat transformer cycle there are basically two levels. These are:

- Optimum allocation of area between the five heat exchangers for a given total amount of area, i.e., optimal relative size of each heat exchanger in order to maximize the COP (technical optimization).
- Economically optimal total size of the heat transformer including optimal allocation of the total investment between the five heat exchangers in order to optimize the COP (technico-economic optimization).

Optimization studies have been done under two principally different conditions:

1. The hot end is large enough to receive all heat that is recovered from the waste heat.
2. There is an excess of waste heat compared with the heat sink need.

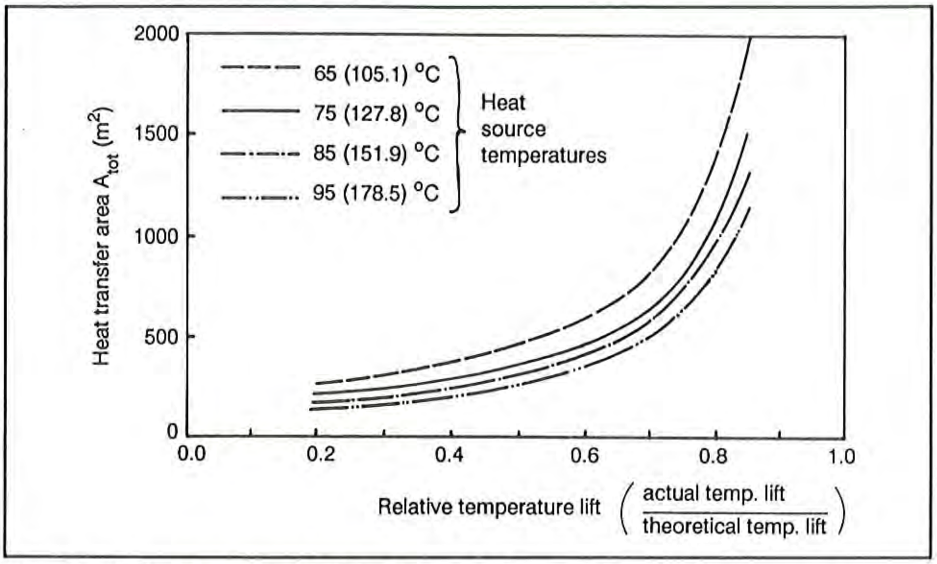

Figure 3. Relationship between heat transfer area and temperature lift ratio

These two conditions lead to different optimization results.

To be able to calculate heat transfer areas, values of the heat transfer coefficients are needed. Values found in the literature are sparse and somewhat contradictory, therefore reasonable assumed values have been used. The values used are 1500 W/m^2,K for the condenser and the evaporator, and 750 W/m^2,K for the other three heat exchangers. The external temperatures have been seen as constant for simplicity. Data for LiBr/H_2O is from McNeely.[5]

The question of whether the heat transformer is economic from an overall point of view has not been considered as this would need detailed economic data.

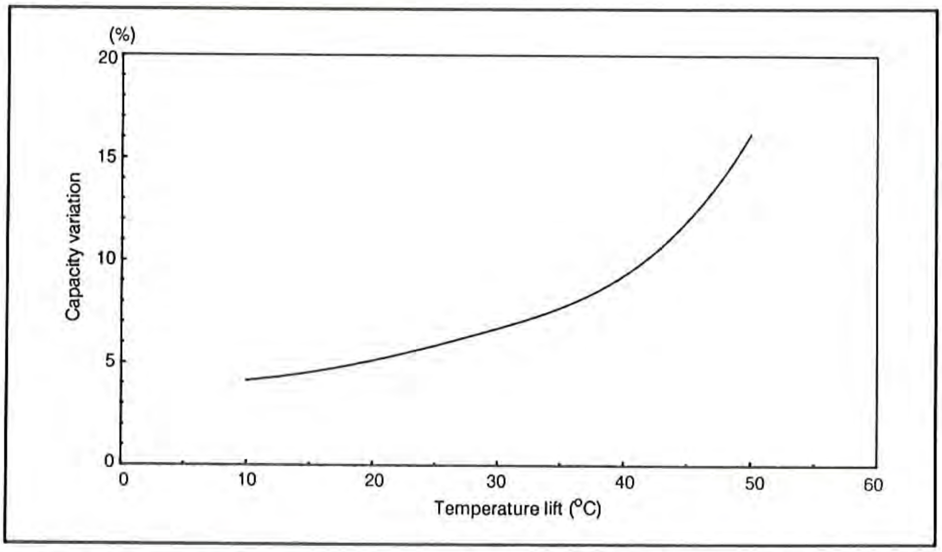

Figure 4. Capacity variation per degree of variation of the heat source temperature as a function of temperature lift

Results

COP: The optimum COP can generally be described as being fairly constant. In all calculations, for a wide variety of parameter values, the optimum COP was found to be in the range 0.46-0.49.

For the situation where the amount of waste heat is in excess of the heat demand the optimum allocation of heat transfer area is the one giving the highest ratio of high temperature heat output to heat transfer area (Q_A/A). This can be established without any specific tests, as this in all cases represents the solution where the profit (heat output) to investment (heat transfer area) has its maximum value. The Q_A/A ratio should be very useful when evaluating new working fluids although different heat transfer characteristics would have to be considered also.

If the amount of waste heat is limited, the situation changes slightly. In this case more heat transfer area can be added to reach a higher COP in spite of the fact that further increments of heat transfer area give a lower high temperature heat output to unit heat transfer area than the previous. The extent of heat transfer area increments is of course determined by the energy to heat transfer area cost relation.

The difference between these two cases is small when discussing the single-stage AHT. But there are situations where both single-stage and double-stage systems are technically possible. In such cases the waste heat to heat demand relation is important and should be considered.

Temperature lift: For each specific waste heat temperature and low temperature heat sink temperature (i.e., cooling water temperature) there exists a theoretical temperature lift determined by the characteristics of the utilized mixture. The difference between the theoretical and the wanted temperature lift provides the driving temperature differences in the heat exchangers, although a part of the overall temperature difference has to be sacrificed in order to have finite solution flows. When the wanted temperature lift is increased, the necessary heat transfer area increases accordingly as the temperature differences in the heat exchangers decrease. The extent of this can be seen in Figure 3, where heat transfer area is shown versus relative temperature lift, which is defined as the dimensionless ratio between wanted and theoretical temperature lift, for different heat source temperatures. The heat transfer areas corresponding to COPs of 0.49 for the smallest temperature lifts and 0.46 for the largest. It is found that at a ratio of wanted-to-theoretical temperature lift of approximately 0.6-0.7 there is a point where the slope becomes very steep. This indicates that the practical limit for economic reasons is around values of 0.6-0.7 for this temperature ratio.

Capacity: As the heat transformer works with small temperature differences, it is very sensitive to annual or short-term variations in the external temperatures. This is especially important when the variation is in such a way that it decreases the available temperature difference, e.g., decreasing heat source temperature. If the heat transformer is not excessively oversized, the result will be that the heat transformer is not able to utilize the available amount of heat, even with a lower COP. Instead the result will be that the capacity decreases. The relationship between external temperatures and capacity for an actual heat transformer has been shown by Furukawa.[6]

The sensitivity to changes of heat source temperatures for optimized plants has been studied in a general way by simulations.[1] The result of this study is shown in Figure 4. The diagram shows the capacity variation, as percentage of the capacity at the design point, per degree of variation of the heat source temperature, for the entire range of temperature lifts. The nominal heat source temperature is 85°C and the low temperature heat sink temperature is 30°C. The diagram clearly shows the impact of decreasing heat source temperature. It must, therefore, be emphasized that an optimized design must take into account possible variations of external temperatures, especially the heat source temperatures.

References

1. Holmberg, P. "System Studies and Optimization of Absorption Heat Transformers." Scientific paper for licentiate degree.
2. Bokelmann, H. "Advanced Absorption Systems." Preliminary Proceedings of the CEC-British Gas International Workshop on Absorption Heat Pumps, April 12-14, 1988, London, pp. 234-243.
3. Mashimo, K. "Overview of Heat Transformers in Japan." Proceedings of the 1987 IEA Heat Pump Conference, Orlando, 1987, pp. 271-285.
4. Zegers, P. "CEC Heat Pump R and D." Proceedings of the International Workshop on Heat Transformation and Storage, October 9-11, 1985, Ispra, Italy, pp. 435-444.
5. McNeely, L.A. "Thermodynamic Properties of Aqueous Solutions of Lithium Bromide." ASHRAE Transactions, Part 1, 1979, pp. 413-434.
6. Furukawa, T. "Steam Generating Absorption Heat Pumps." IEA Newsletter, Vol 4, No 1, 1986.

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Absorption Heat Pump for Industrial Waste Heat Recovery

In this article, the authors review the status of heat pump technology in China and suggest that absorption heat pumps be used for industrial waste heat recovery. This is based on the fact that the energy supply cannot meet the demand at the present time in China. Described in this paper are the operating principle and technical parameters of a 3490 kW, high temperature, LiBr-H_2O absorption heat pump designed for a chemical factory.

The scarcity of verified underground energy reserves in China per capita, underdeveloped energy mining and conversion technology, and the backwardness of energy utilization efficiency as compared with other countries have resulted in an insufficient energy supply during the recent rapid economic development in China. This has necessitated an increased energy use efficiency as one means of coping with the situation. With this background, heat pump technology has received much interest. For over ten years, Chinese refrigeration and air conditioning organizations have made efforts in promoting their technology, resulting in a relatively rapid development in the 1980s. However, heat pumps used to date are mostly electric compression type heat pump air conditioners which are limited in wider application due to an electricity shortage. Industrially, heat pump drying is popular, while in other industries heat pump applications are still rare. Non-electric heat pumps such as internal combustion engine-driven ones are still few in number. Absorption heat pumps, which are most suitable to be powered by industrial waste heat, are rarely seen.

It was only recently that a steam-powered, double effect, cooling/heating type I absorption heat pump was installed as a pilot industrial installation in a cloth weaving factory in Wusi city, Jiangsu province. In summer, the installation provides 10°C chilled water for air-conditioning; in winter, it provides 60°C hot water for space heating and domestic hot water. The installation utilizes waste water as a heat source.

At present, experimental research on type II absorption heat pumps are under way or to be undertaken in Shanghai, Tianjin, Nanjin, Guangzhou, etc. Research on gas-powered absorption heat pumps is being conducted at Beijin.

From the viewpoint of energy savings, we think that absorption heat pumps have unique advantages: the possibility of using low temperature waste heat of under 100°C which is released from industrial processes in large quantities; the energy saving possibilities when used in middle and large scale industrial thermal processes; and the low electricity consumption which is well suited to the present electricity supply situation in China. This opinion was agreed on by experts attending the Symposium on Heat Pump Development and Application in China, held in Guangzhou in March 1988 (see Newsletter, Volume 6, Number 2, News Briefs). It is expected that absorption heat pumps will continue to increase in importance.

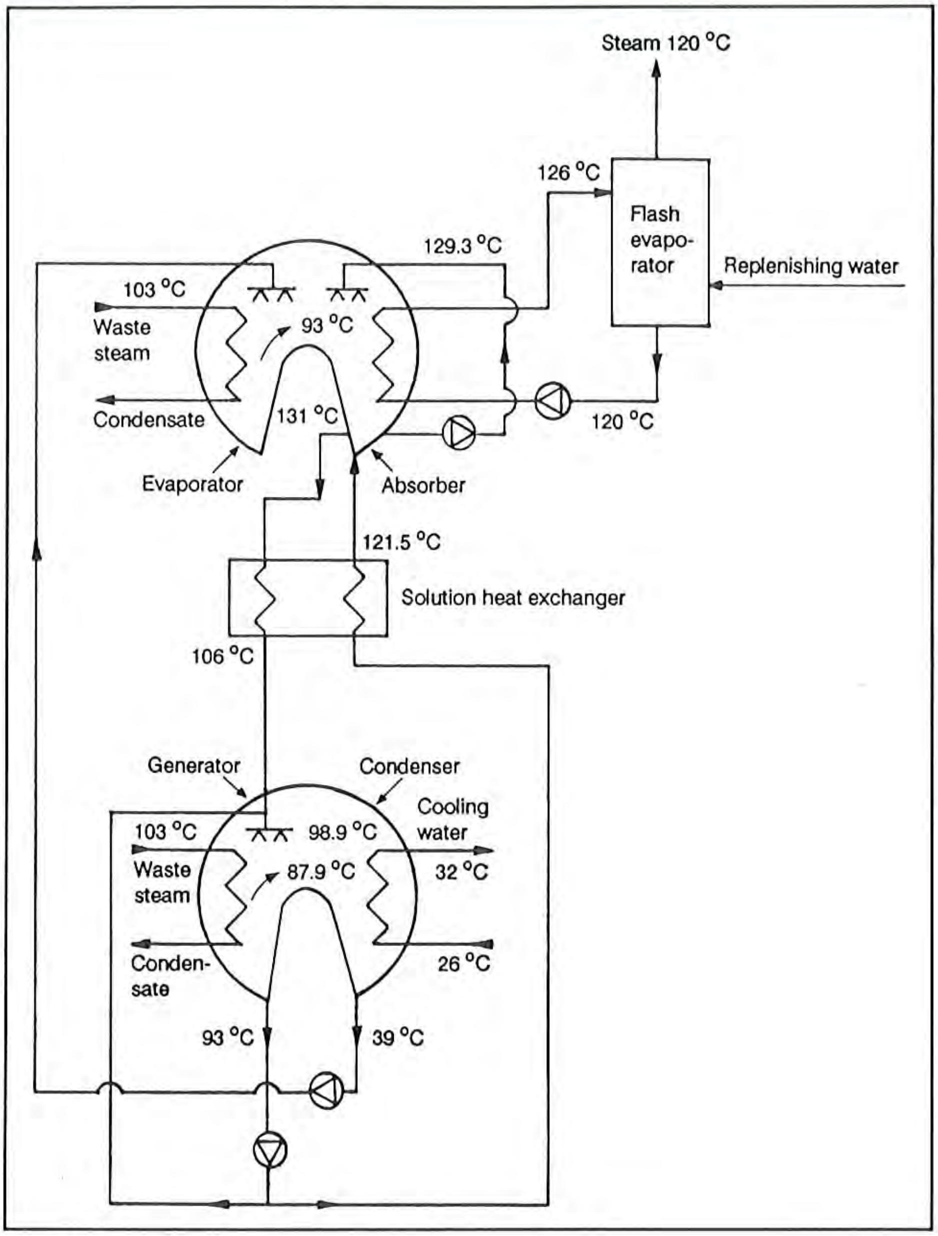

Figure 1. Operation principle

The development of the 3490 kW high temperature absorption heat pump described here was undertaken based on this viewpoint. The project was funded by the Chinese government.

For selection of a working fluid pair, a comparison was made between NH_3-H_2O and LiBr-H_2O. It was considered that a heat pump using LiBr-H_2O has the following advantages: simple process, refining distillation unnecessary, less material consumed, lighter weight of the system, the possibility of manufacturing in modules, convenient installation and operation. Furthermore, LiBr-H_2O is non-toxic, odorless, and non-flammable, with no danger of explosion under vacuum operation, thus better in operating safety, higher thermal parameters, etc. The disadvantages of LiBr-H_2O are the necessity to use copper pipes resulting in higher cost, the high price of LiBr, limitation of operation range by the solution's crystallization line, and the corrosion of steel by LiBr solution in atmospheric air which necessitates the maintenance of a high vacuum with perfect sealing. After weighing the advantages and disadvantages, especially in view of the relatively extensive experience in design and manufacture of LiBr-H_2O chillers presently in China, it was considered suitable to use LiBr-H_2O.

High temperature absorption heat pumps do not need a high temperature heat source but instead use part of the energy from a middle temperature level heat source to raise the temperature of the other part of it. This type of heat pump is preferable based on economic considerations even if its COP is low. The operation principle of the design is shown in Figure 1. Waste steam of 103°C enters the evaporator and generator, respectively. In the generator, diluted solution is heated and steam of 87.9°C and 52.4 mmHg ab. is generated. The steam enters the condenser and is condensed into water of 39°C which is pressurized by a pump to enter the evaporator where it is heated by the steam of 103°C in the tubes to become steam at 93°C. This steam enters the absorber where it is absorbed by the concentrated solution of 129.3°C and absorption heat is released thus to raise the in-tube water temperature from 120° to 126°C. This hot water is sent to the flash evaporator to generate steam of 98kPa (gauge) and about 120°C. In this way, the heat from 14.7 kPa (gauge) waste steam is upgraded to usable steam of 98 kPa (gauge).

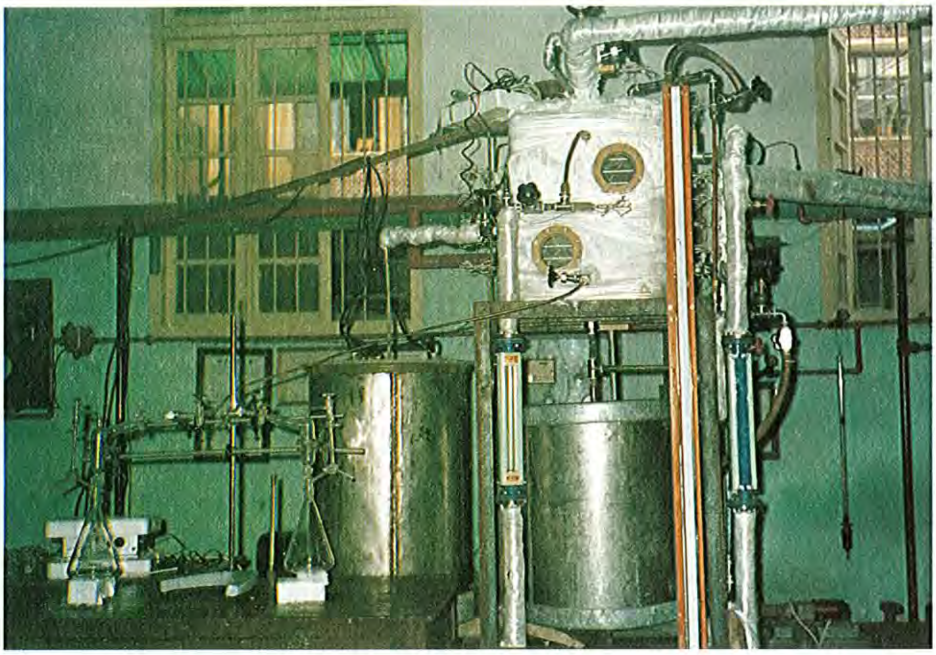

Figure 2. Experimental installation

The diluted solution from the absorber (131°C), after pre-cooling in the solution heat exchanger, is mixed with the concentrated solution (93°C) from the generator. The resulting 98.9°C mixture is sprayed onto the heat transfer tubes of the generator thus beginning the next cycle. Concentrated solution (93°C) from the generator is pressurized and preheated to 121.5°C in the solution heat exchanger and is pumped into the solution in the fluid chamber of the absorber where it is mixed with diluted solution at 129.3°C. The mixture is pumped into the spraying system and sprayed onto the heat transfer tubes of the absorber.

Principal technical data

Waste steam:

Flowrate G = 1210 kg/h
Pressure P = 14.7 kPa
Temperature t = about 103°C

Hot water:

Flowrate W = 53.15 m^3/h
Temperature t_i = 120°C
Temperature t_{out} = 126°C

Usable steam:

Flowrate G_o = about 500 kg/h
Pressure P_o = 98 kPa
Temperature t_o = about 120°C

Coolant water:

Flowrate W_c = 56.49 m^3/h
Temperature t_i = 26°C
Temperature t_{out} = 32°C

Total solution pump capacity: 8.3 kW

COP: about 0.46

Preliminary work before design was to investigate if an absorption process would be successful with the LiBr solution at high temperature (129.3°C) and what the heat transfer coefficients would be. Therefore, experimental research of the high temperature absorption process was performed for one year. First, a physical-mathematical model was set up on the basis of theoretical analysis and analog values were calculated with a computer, thus providing a basis for design of the experimental installation. Afterwards, a test rig was built (see Figure 2) on which tests were performed to obtain information on hot water temperatures achievable as well as heat transfer coefficients of the absorber's heat transfer tubes. One of the problems encountered was in designing the shielded pump for high temperature operation. For an ordinary absorption chiller using LiBr, the temperature of the fluid handled by the shielded pump does not exceed 60°C as opposed to 129.3°C in this design. The use of a shielded pump operable at such a high temperature was the key point in realizing the design.

An economic analysis indicates that the design can use waste heat of 9600 tons/year and provide 4000 tons/year of usable steam at 98 kPa. This is equivalent to almost 600 tons/year of standard coal. For chemical factories, the payback period for this absorption heat pump is estimated to be 1.5 years.

References (all in Chinese)

1. Transactions of the Symposium on Heat Pump Development and Application in China, Lin Yi-Cheng, Li Song-zhe, Guangzhou Institute of Energy Conversion, March 1988.

2. Recommendations for Absorption Heat Pump Development, Xia Wen-Hui et al, Guangzhou Institute of Energy Conversion, February 1988.

3. Feasibility Report on Recovery of Waste Heat From Mellow Wine Tower by Use of Absorption Heat Pump, Refrigeration Engineering Teaching Group of Jiaotong University, Shanghai, October 1985.

4. Analog and Cycle Analysis of High Temperature Absorption Heat Pump, Lu Zhen, Wang Chang-Ching, Yuchi Bin, Shanghai Jiaotong University.

5. Experimental Report of High Temperature Absorption Process, Research Group of High Temperature Absorption Heat Pumps, Shanghai Jiaotong University, April 1988.

6. Optimized Design of a $LiBr/H_2O$ High Temperature Absorption Heat Pump, Lu Zhen, Wang Chang-Ching, Shanghai Jiaotong University, July 1988.

7. General Description of High Temperature Absorption Heat Pump Designs, Lu Zhen, Wang Chang-Ching, Yuchi Bin, February 1988.

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V. Mučić*

Resorption Compression Heat Pump With Solution Circuit for Steam Generation Using Waste Heat of Industry as Heat Source

This article describes an industrial heat pump installation using hot brine at 100°C as its heat source. The recovered heat is raised to a temperature of 115°C by the heat pump and is used to generate steam from 115°C feed water.

In a chemical factory in the Federal Republic of Germany, sodium carbonate (NA_2CO_3) is produced among other products. It arises in large quantities during full-year continuous operation. With the production of sodium carbonate a large quantity of heat is lost to the environment in the form of hot liquid brine at 100°C. The liquid brine is essentially composed of the following:

H_2O about 80%
$CACl_2$ about 15%
NaCl about 5%

The pressure of the brine at the outlet of the production column is 1.4 bar. As the dirty brine also contains solid components, it cannot be directly led to the heat exchanger. The heat of the brine can only be used if the brine is expanded and the resulting steam is led to the waste heat recovery plant, where it condenses and the condensation heat is yielded to the waste heat recovery plant.

In order to validate this principle as the basis for the mode of operation of the waste heat recovery plant, only a small portion (66 t/h) of the total brine (260 t/h) was taken for the pilot plant, and it was expanded to only 0.85 bar. The waste heat recovery plant is a resorption compression heat pump with a solution circuit working with an ammonia-water mixture. The resorption compression heat pump or refrigeration cycle, using an ammonia-water mixture, is a good alternative to cycles which utilize conventional chlorofluorocarbon refrigerants for the following reasons:

-- Lower energy consumption
-- Higher working temperature possible (up to 180°C) since NH_3 molecules are more stable at higher temperatures than CFC molecules and the critical point of water-ammonia mixtures is higher than that of conventional refrigerants and pure ammonia.
-- Ammonia has no ozone depletion potential

The connection of the heat pump to the waste heat (steam 0.85 bar, 95°C) on the one side and to the useful heat (steam 1.7 bar, 115°C) on the other side is shown in Figure 1.

Figure 2 shows the circuit diagram of the NH_3-H_2O plant, together with the measured values for concentration, pressure, temperature, and quantities of the mixture at every point of the process, as well as the capacity of each apparatus and machine, including COP.

As the brine quantity and the production steam requirements of the factory are always greater than the brine quantity

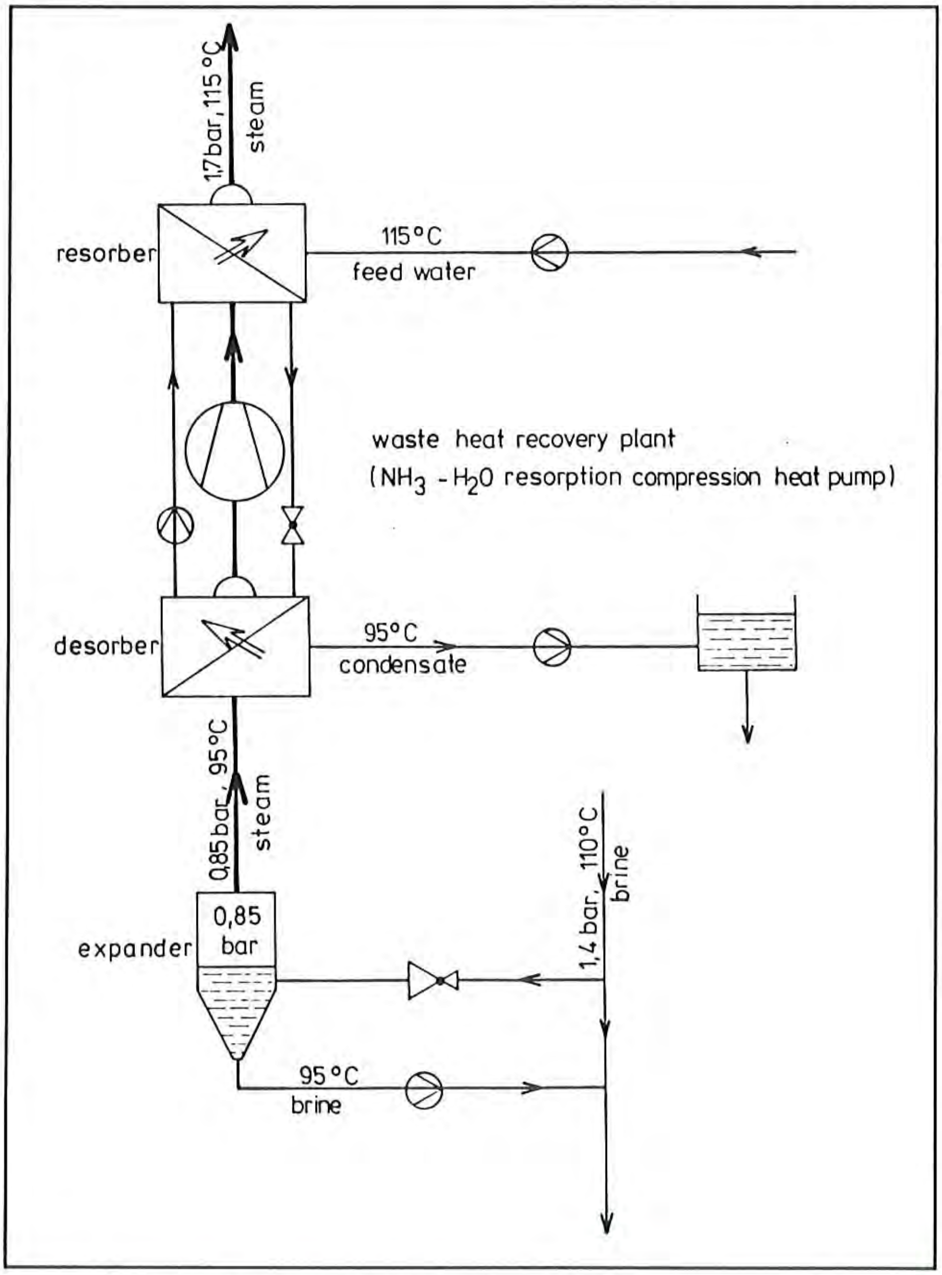

Figure 1. Connection of heat pump to heat source and heat sink

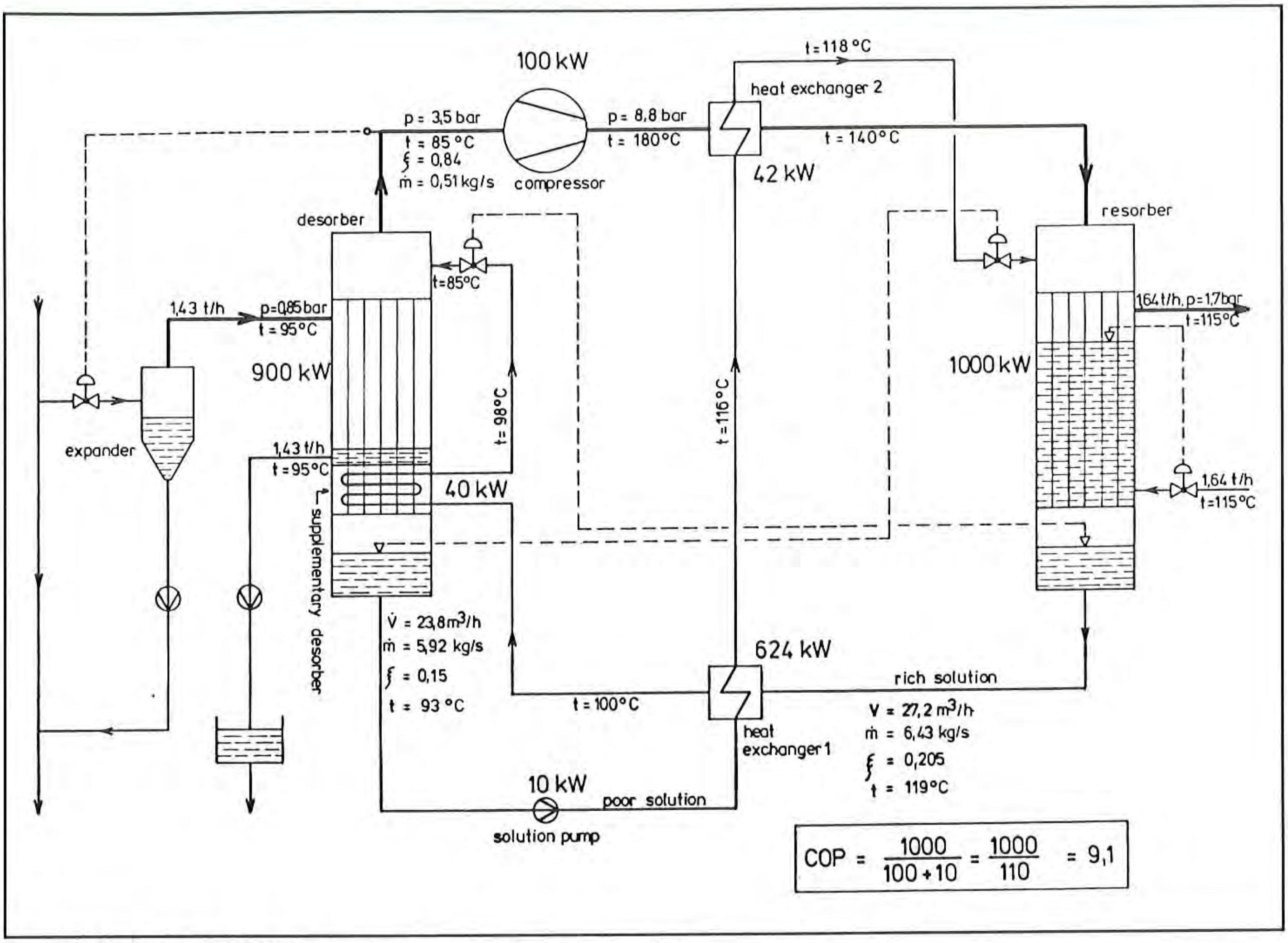

Figure 2. Circuit diagram of NH_3-H_2O resorption compression heat pump plant

and the steam generation of the heat pump, no partial load control for the compressor of the heat pump has been provided. The automatic control was installed only for:

- Maintenance of constant desorber pressure by regulating the quantity of brine
- Maintenance of constant poor solution level in the desorber by means of a control valve for poor solution at the inlet to the resorber
- Maintenance of constant rich solution level in the resorber by means of a control valve for rich solution at the inlet to the desorber
- Maintenance of constant feed-water level in the steam generator by means of a control valve for the feed water

The desorber and resorber are vertical heat exchangers with falling film desorption and resorption, respectively, inside vertical tubes. A patented construction for the intake of the poor solution into the resorber and the rich solution into the desorber guarantees that the liquid phase is evenly distributed to all tubes. At the resorber, the even distribution of the vapor phase to all tubes is also guaranteed through this construction. At the desorber, supplementary arrangements ensure that the liquid phase is not carried along with the vapor phase to the compressor.

The compressor is an oil-free screw compressor with 13000 revolutions per minute and is driven by an electric motor. Mechanical seals prevent the entry of lubrication oil into the screw element.

The poor solution is warmed up in the first heat exchanger by the heat of the rich solution and in the second heat exchanger by the heat of the superheated vapor. In order to improve the COP the rich solution is cooled down in a supplementary desorber, which is located under the main desorber.

Operating experience

After two weeks of continuous operation the heat pump was automatically turned off because the lubrication oil pressure had fallen. It was discovered that the installed mechanical seals were defective, so that the lubrication oil was entering the screw element and, therefore, also the heat pump plant. The supplier of the compressor then changed the mechanical sealing system. Once the plant had been cleaned of the oil and refilled with the ammonia-water mixture, it was again put into operation. Apart from this problem caused by the mechanical sealing system, no other difficulties have been observed during the two-year demonstration period (1986-1987). This pilot plant has been funded by the government of the state of Nordrhein-Westfalen.

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The Use of Heat Pumps for Drying Fish in Norway

Heat pumps for drying salted fish were introduced in Norway in about 1980. Since then more than 40 factories have installed heat pumps in their klipfish (salted and dried fish) production. Typical heat pump capacities are 200-400kW. This method has generally reduced the energy costs of a traditional petroleum dryer. Furthermore, with the heat pump drying system, full control of temperature and humidity can be achieved independent of the outdoor climate. This ensures a high quality of the product and continuous production throughout the year. During the last years the first industrial heat pump dryers for unsalted fish have also been introduced. In the conventional process the fish are dried on racks outdoors. The drying time and quality of the fish depend on the outdoor conditions. Flies, worms, rain and temperatures can reduce the quality or completely destroy the fish. This article presents the principle of heat pump systems for fish drying and a short survey of the dimensioning basis for unsalted fish dryers.

Introduction

Drying of fish has a very long tradition in Norway. Klipfish have been dried for 500 years and stockfish for many times as long. Very little has happened in the processing of stockfish from the old days up to modern times. A little more has happened in the klipfish production. Up to 1920, all the fish was produced by spreading the salted fish on the rocks near the sea. After 1920, more and more of the klipfish was produced in oil-heated dryers and by about 1950 all the klipfish was artificially produced in such dryers.

The years following 1980 were in fact a kind of revolution in the history of fish drying technology in Norway. Energy saving, stable production, high quality, and independence from outdoor air conditions are specific benefits of this new technology.

Heat pump drying of klipfish

The fish used for salted fish drying in Norway belong to the cod family and include cod, lang, tusk, haddock, and coalfish. The fish are laid on shelves stacked on trucks and are pushed through the tunnel in counterflow with the air. In the traditional process, the heating of the outdoor air takes place in a heat exchanger with electricity or petroleum as the heat source. The principle of heat pump drying is shown in Figure 1.

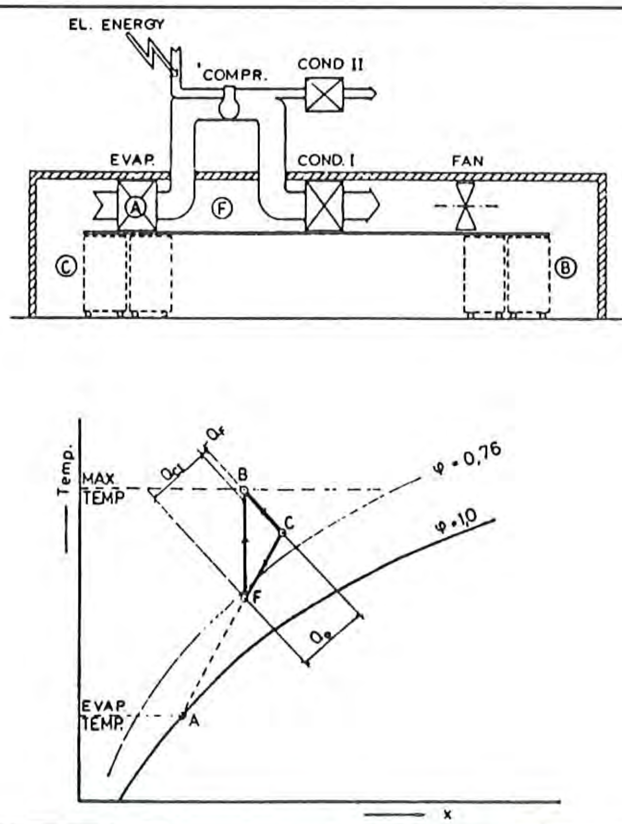


Figure 1. Principle of fish drying by heat pump

Reaching the evaporator in the refrigerating plant, the air is cooled down and water is condensed on its surface. In this process heat is preserved by the evaporation of a refrigerant. The tunnel condenser heats the air to the desired temperature and the excess heat is removed in an air-cooled condenser placed outside the tunnel. This excess energy may be used for heating purposes. For klipfish dryers, the coefficients of performance (COP) are approximately 5.0.

Dimensioning heat pump dryers

In dimensioning heat pump dryers, it is essential to take into account the drying characteristics of the material to be dried. The actual production, quality, and energy consumption will depend on many parameters, such as the size, shape and water content of the product, the air condition, the drying procedure, size and dimensions of the dryer. The final result will be given from the cross-section between the characteristics of the products and the plant. To find the optimal capacity of the heat pump, drying experiments and mathematical models are necessary to predict the behavior of the dryer.

The use of simulation in dimensioning heat pump fish dryers

To simulate an industrial drying tunnel, temperature and the relative humidity of the drying air must be known. The humidity depends on the drying rate of the fish. From drying experiments in a heat pump pilot plant at the Division of Refrigeration Engineering, the University of Trondheim, Norway, the drying rate at different temperatures, relative humidities and air velocities was determined and the quality evaluated.

Input parameters	Output parameters
Tunnel dimensions Number of racks in length Number of racks in one row Drying procedure Weight of fish on one rack Weight of one fish Fish type and preparation Air temperature at the inlet of the drying section Air speed Cooling capacity of the heat pump Water content of the fish at end of drying	Production Air temperatures in the dryer Water content of the fish in each row of racks Thermal efficiency of the dryer

Table 1. Input and output parameters of the simulation program

To simulate the production, energy consumption, etc., of a certain dryer, it is necessary to predict the drying time of the fish as a function of its type, size, preparation, water content, air condition, and air speed. The mathematical drying model is based on the assumption that during the initial drying, water evaporates from the surface. As drying proceeds, a dry layer is formed on the surface and water vapor is transported by diffusion through the layer. Relatively simple models are made for these phases and the "fish dependent" constants are found from drying experiments.

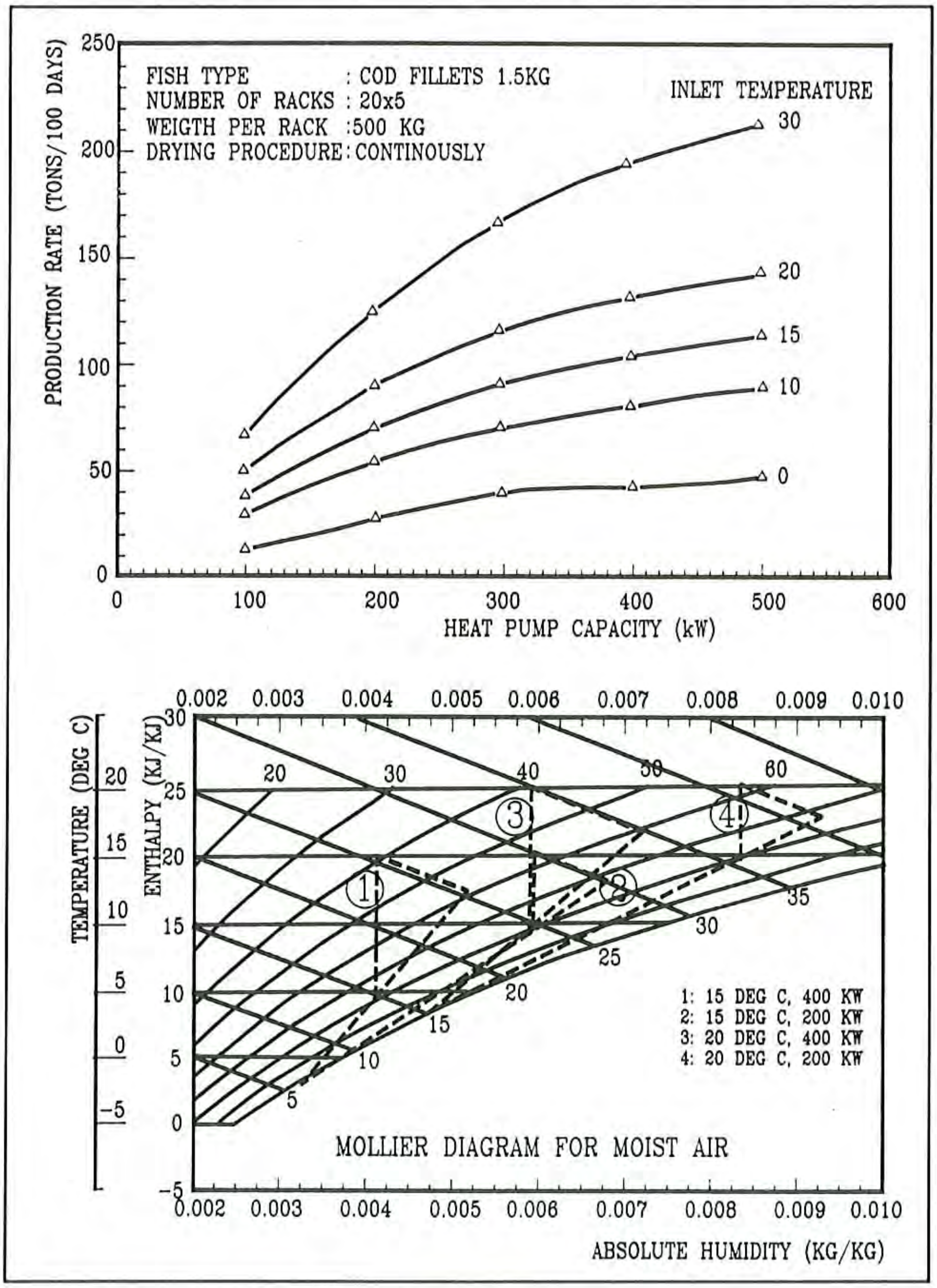


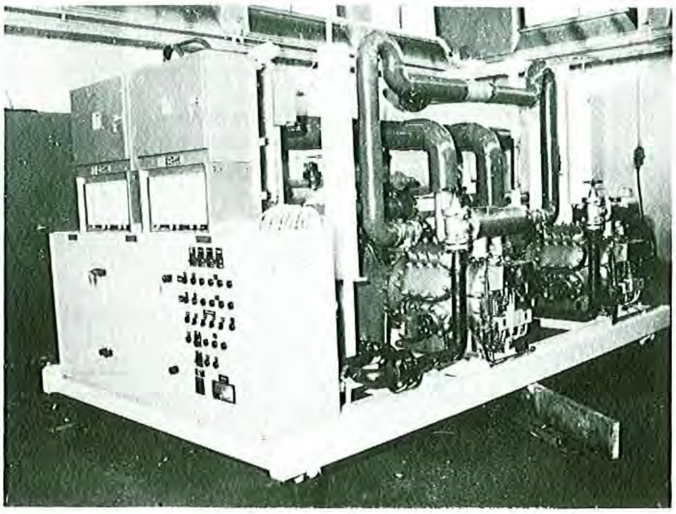
Figure 2. Production rate at different heat pump capacities and inlet temperatures

From the drying model a simulation of an industrial heat pump tunnel dryer is developed. In Figure 1, the dryer is shown schematically. The fish is placed on shelves on wheeled racks and in counterflow with the air which is assumed to follow the isenthalp through the dryer. The theoretical process is shown in Figure 1. The air temperature at the inlet of the tunnel is given from the quality aspects and a humidity is assumed. The program calculates the temperature, humidity, and water content in fish through the tunnel by stepwise iteration.

Table 1 gives a complete list of the input and output parameters for the program. With continuous drying, one row of racks with fish is taken out when it reaches the final water content. The other racks are pushed forward and new wet fish is put into the dryer. In batch drying, the whole tunnel is emptied and filled at one time. The drying velocity used is based upon the drying experiments.

The production is calculated in tons of dried product per unit time. The thermal efficiency of the dryer is defined as cooling capacity (kWh) needed to condense 1kg of water on the evaporator surface.

An example of the results of the simulation is shown in Figure 2. Here the production in tons of dried cod fillets of 1.5kg wet weight per 100 days is plotted as a function of the refrigeration capacity and the air inlet temperature in the dryer. Here, the evaporating tempera-



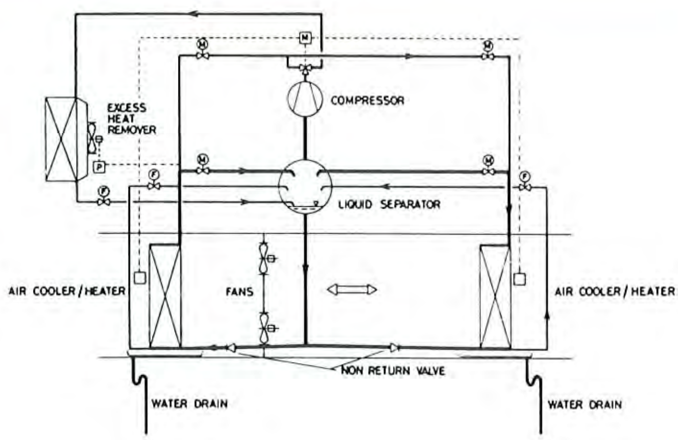


Figure 3. Factory-assembled unit at Refsnes Fiskeindustri

ture is assumed to be 17°C below the air temperature at the outlet of the dryer. On the figure we can also follow the process in the Mollier diagram. Increasing the heat pump capacity at the same inlet temperature will decrease the relative humidity and thereby increase the production. Because of a higher drying potential, the production is increased by increasing the inlet temperature. For a given product, tunnel and arrangement, the production will greatly depend on inlet air temperature and refrigeration capacity. It is important to notice the influence of air temperature. At 300kW refrigeration the production capacity can be increased by 68% by increasing the temperature from 10°C to 20°C and 145% from 10°C to 30°C. However, temperatures are restricted due to quality demands.

The model is verified by measurements at an industrial heat pump fish plant at Refsnes Fiskeindustri near Trondheim, Norway.

A picture of the heat pump unit with a system schematic is shown in Figure 3. This heat pump has two piston compressors with a total refrigeration capacity of 270kW. The COP for this plant was measured to be about 5.0. The plant made it possible to produce a very high quality ling fillet for export to Sweden and Finland.

Conclusion

The heat pump principle has been introduced for the drying of salted fish in Norway. The advantages of this equipment are energy savings, independence from outdoor air conditions, and stable production of a high quality product. The energy consumption is as low as 0.125 kWh per kg dry fish when the drying is continuous and inlet temperatures in the tunnel are 20-25°C. Today most producers of klipfish in Norway buy heat pump dryers for their plants. The first heat pump has also been introduced in stockfish production.

In dimensioning heat pump dryers, one must take into account the drying characteristics of the material to be dried. The actual production, quality, and energy consumption depend on many parameters. To determine the optimal capacity of the heat pump, drying experiments and simulation models are necessary to predict the behavior of the dryer.

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Drying--An Important Heat Pump Application in China

Heat pump drying in China can be divided into two kinds: heat pump dehumidification without the use of environmental heat, and heat pump drying using environmental heat (see Figures 1 and 2). The difference between these two systems is that the latter uses an additional evaporator to draw in environmental heat which is added to the drying chamber during the start of the drying cycle. This type of heat pump dryer is expected to find extended use in the warmer areas of China.

By the mid-1980s there were already two factories in China manufacturing heat pump dryers or dehumidifiers, with a total production of 200 to 250 units per year. In addition, some research institutes and manufacturers are developing new products. At present, more than 160 systems are in operation which (including a few imported units) account for 90% of industrial heat pumps used. This indicates that drying is an important application for heat pumps in industry and agriculture.[1]

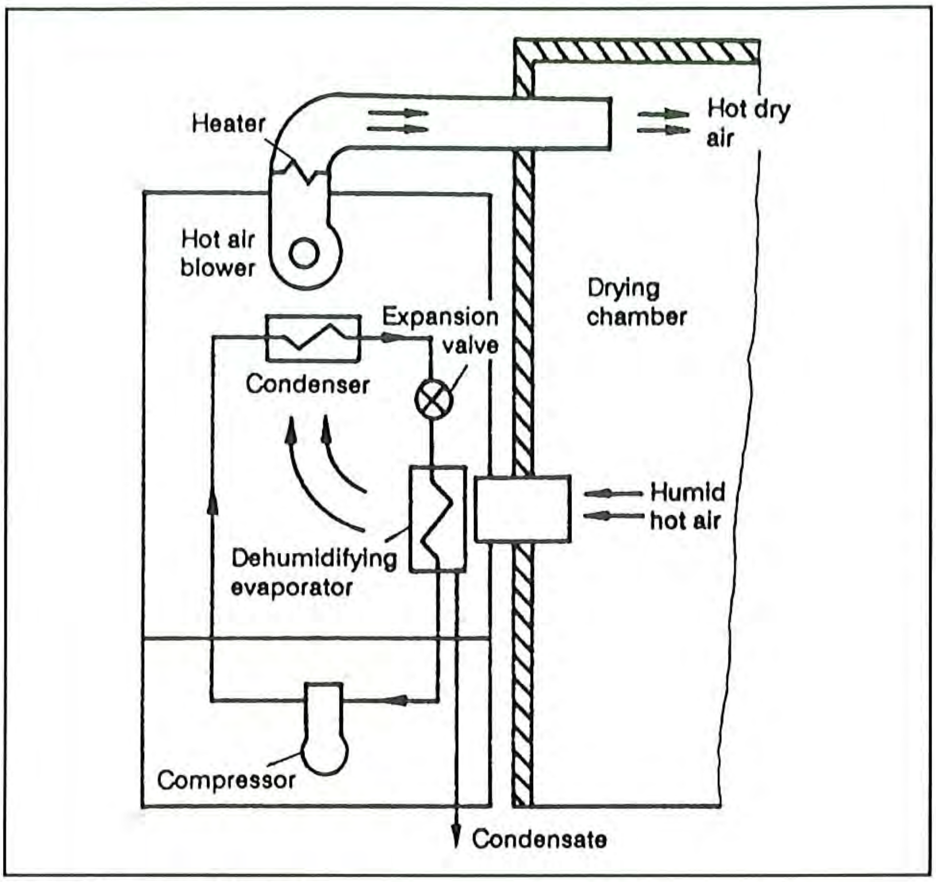

Figure 1. Heat pump dehumidifier

is decreased by a factor of 2 to 7 as compared with electric or steam drying. For example, a music instrument manufacturer in Guangzhou used a heat pump to replace an electric dryer for drying of wood. In this case, the expected energy savings will recover the heat pump investment cost in less than two years.[1] Tables 1 and 2 compare the different methods of wood drying.[4,5]

Future potential for heat pumps

The potential for heat pump application in drying is appreciable in China, especially in drying of broadleaf hard wood which is mostly used in making furniture and musical instruments, and is dried at low temperatures. The potential market is in the more than 4000 existing wooden furniture factories and over 70 musical instrument factories. There are more than 10 million cubic meters of wood to be dried each year.[4] At present, most are dried by natural or by

Economics of heat pump drying

To date, the price ratio of electricity to coal (which is the principal energy source in China) for industry is more than 10, if calculated on the basis of energy equivalent. The price ratio of coal (yuans/ton) to electricity (yuans/kwh) is about 500 to 800 while this ratio is as high as 4000 in the United States and 2000 in the Soviet Union.[1,2,3] Although heat pumps have difficulty competing with coal-fired boilers in some fields, heat pump drying is still well received among customers since the quality of the product dried by heat pumps and the rate of production are greatly improved. Furthermore, there are many advantages over drying by steam, such as improved working conditions, simplified operation, easier maintenance, and pollution reduction . The use of heat pumps also cuts down the overall cost of drying.

According to our survey, users are quite satisfied with heat pump drying. The overall production cost for wood drying

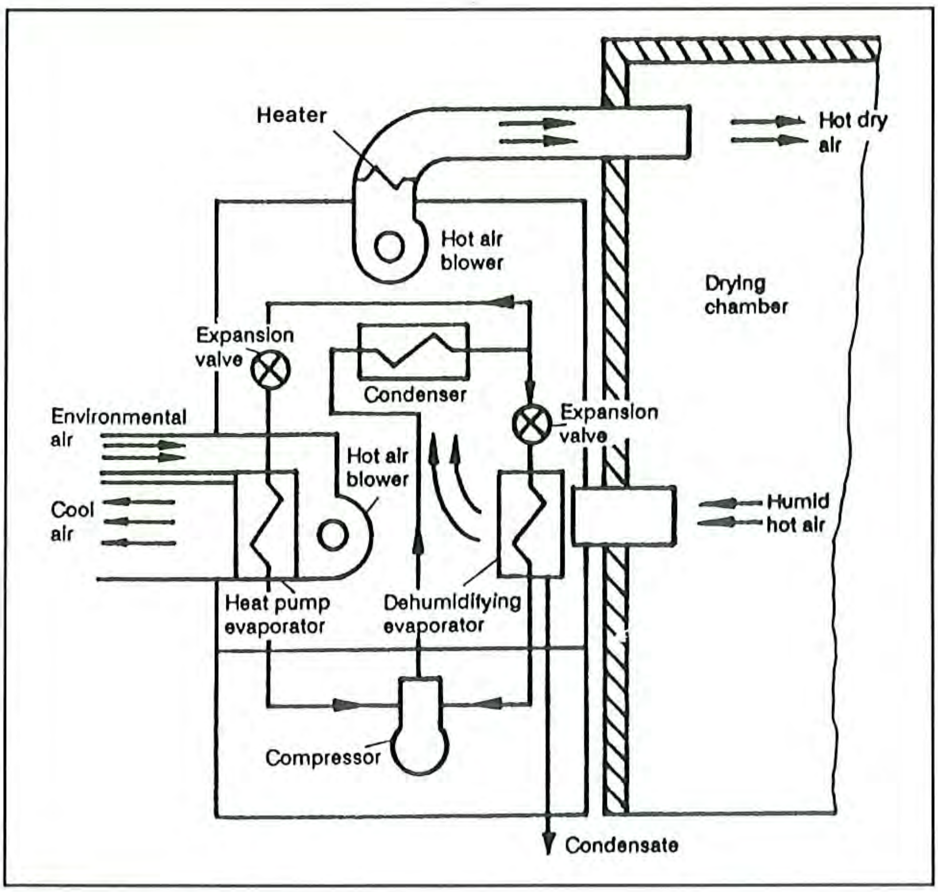

Figure 2. Heat pump dryer

Factory	Furniture		Musical Instrument		Furniture		Wood Processing	
Wood dried	Broadleaf wood		Spruce	Pine	Oregon pine	Pine	Broadleaf hard wood	Chinese fir
Plant thickness (cm)	2.2	3.0			1.7	1.7	5.5	2.5
Volume of wood per kiln (m^3)	31.8	35.7	10.9	10.9	14.5	16.8		
Water content, initial (%)	70	95	30	70	44	40	30	36
Water content, final (%)	10.2	12.1	5-7	10-12	11.8	14	10.5	11
Drying cycle (days)	8	15	5-7	11.5			4	4.5
Total elec consumption (kWh)	2531	4058	444	687	1040	996		
Elec consumption per m^3 of wood (kWh)	79.6	113.7	40.7	63.0	71.7	59.3	30	27.8
Equipment investment (thousand yuans*)	190 (US$50000)		50 (US$13000)		30 (US$8000)		30 (US$8000)	
Heat pump manufacturer	Italy		France		China		China	
Compressor capacity (kW)	5.5		2 x 2		4		5.5	
Type of compressor	Enclosed		Enclosed		Open		Semi-open	
Refrigerant	R-12		R-22		R-142		R-12	
COP of heat pump proper	--		--		4.75 (measured)		4.5 (designed)	
COP of heat pump system	--		--		3.2 (measured)		--	
Installed site	Qingyuan county Guangzhou		Guangzhou		Shanghai		Shangdong province	
Status of heat pump used	Commercialized		Prototype		Prototype		Commercialized	

* NOTE: 1US$ = 3.8 yuans. "Yuan" is the unit of Chinese dollar (also called Ren-Min-Bi or RMB).

Table 1. Typical examples of heat pump drying of wood

tunnel kilns heated by sawdust (only a small fraction is dried by vacuum drying, far infrared, steam or electricity). With continued equipment improvements, heat pumps will be a practical alternative to existing drying methods. Heat pump drying in agriculture application is also developing (for example in the drying of tea).

The popularity of heat pump drying is seen in Guangdong Province where 7 of the existing 12 musical instrument factories have introduced heat pump drying. In addition, more than 30 factories which produce wood products use heat pump drying systems; these, together with other factories using the technology in other fields, total 50 in all. These represent one third of China's total heat pump drying installations.

Drying method	Heat pump	Conventional steam drying	Infrared (by gas burning)	Infrared (by electricity)	Vacuum
Energy cost per m^3 of wood dried (yuans)	8.74	23.47	33.00	20.90	10.91
Comparison					
Savings in primary energy if HP were used (%)	--	86.93	73.46	46.25	69.79
Savings in energy cost if HP were used (yuans/m^3)	--	14.73	24.26	12.16	2.17
Annual savings if HPs are used (yuans)	--	11784	19408	9728	1736
Remarks	Prototype	China made	China made	China made	China made

Table 2. Energy costs for different drying methods

References

1. Li Song-zhe, He Rong-zhi, Lin Yi-cheng, Wu Zhi-jian, "On Application and Development of Energy Saving Heat Pump Technology in China", Guangzhou Institute of Energy Conversion, Chinese Academy of Sciences, January 1988.

2. Lin Yi-cheng, He Rong-zhi, Li Song-zhe, Wu Zhi-jian, "Status and Development of Heat Pumps in China", Guangzhou Institute of Energy Conversion, January 1988.

3. Gu Huang, Zhu Xu-di, Shi Ting-jin, "Feasibility Investigation of Heat Pump Application in China", Zhejian University, October 1987.

4. Lin Yi-cheng, Li Song-zhe, He Rong-zhi, "Potential of Heat Pump Drying in China", Guangzhou Institute of Energy Conversion, January 1988.

5. Tao Zhong-de, Kung Xian-rong, "Good Prospects for Heat Pump Drying Technology", Shanghai City Energy Institute, October 1987.

Lin Yi-cheng, Li Song-zhe, and He Rong-zhi, Guangzhou Institute of Energy Conversion, Chinese Academy of Sciences, People's Republic of China

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Hybrid Solid-Desiccant Heat Pump Systems

Hybrid systems incorporating solid-desiccant dehumidifiers and vapor-compression heat pumps have gained considerable attention over recent years. In hybrid space-conditioning systems, the air conditioner is assisted by a solid desiccant rotary dehumidifier to increase both the specific cooling capacity of the heat pump and the temperature of the evaporator. In hybrid heat-pump assisted drying systems, the dehumidifier increases the operating range of the heat pump towards high temperature applications. This paper reviews some of the systems which are currently under investigation.

Introduction

Regenerative desiccant dehumidifiers are usually configured as cylindrical wheels with core geometries allowing easy flow of air through the passages of the matrix. The walls of the flow passages are coated with a solid adsorbent. Silica gel, molecular sieves 3A and 13X, activated allumina, and hygroscopic salts like LiCl can be used for the solid desiccant.

Figure 1 shows a schematic of a rotary dehumidifier. Two streams are passed in counterflow through the regenerator. The process stream has a high relative humidity whereas the regeneration air stream has a low relative humidity. During the processing period, dry desiccant particles absorb the water vapor from the wet process air stream. As the desiccant rotates through this cycle, there is a slowly moving adsorption front passing through the matrix separating a dry-desiccant zone ahead of the front from the water-loaded zone in the wake of the front. For a tuned rotation speed of the wheel, the desiccant matrix rotates into the regeneration period when this sorption front reaches the exit face of the wheel. During the regeneration cycle, the dry air stream desorbs the wet desiccant thereby establishing a desorption front which moves slowly in the direction of the regeneration air stream, and separates the wet from the dry zone. When this desorption front reaches the exit face of the wheeı, the dry desiccant matrix rotates again into the processing cycle.

In addition to the mass exchange there is also heat exchange between the desiccant matrix and the air streams. Because the sorption processes are adiabatic, the temperatures of the matrix and the air streams vary with time and distance in the flow direction. In fact, the regeneration of the wet desiccant resembles the adiabatic saturation of an air stream with water vapor; and the change of state of the regeneration air flowing through the desiccant matrix approximates a line of constant wet-bulb temperature (or constant enthalpy) in the psychrometric chart. Similarly, the process air flow changes its state along a line of adiabatic saturation away from the wet-bulb state.

The psychrometric chart in Figure 2 illustrates the operation of the dehumidifier for two modes. In the dryer mode, the inlet process and regeneration air streams may have about the same absolute humidity; however, the regeneration air has a higher temperature than the process air. The exit process air reaches desert-like conditions and, after cooling with ambient air, can be used for air-conditioning purposes. In the heat-transformer mode, the processes are reversed. The latent energy of the water vapor stored in the humid process stream is transformed into sensible energy available at the exit of the wheels. This latter mode can be used for closed-cycle drying purposes as explained in the following sections.

Hybrid desiccant cooling systems

Hybrid desiccant cooling systems have been proposed for the air-conditioning of supermarkets, and several systems have been put into operation in the U.S. (e.g., Manley et al, 1985). Supermarkets in warm, humid climates have a characteristic air-conditioning load. The open refrigeration cases displaying various refrigerated food items provide a major part of the sensible cooling needs of the store. Furthermore, the open displays require a low indoor air humidity in order to prevent condensation on the displayed items thereby increasing the load on the refrigerator units. The central air-conditioning system is left with the dehumidification of the store air. Conventional vapor-compression systems remove the moisture in an air stream by cooling the air below its dewpoint temperature. This condensation process involves large amounts

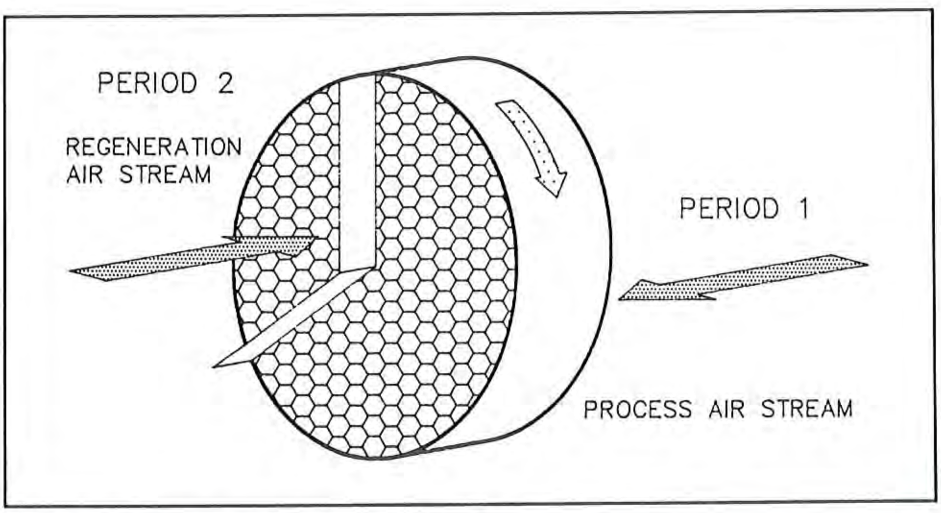

Figure 1. Schematic illustration of a rotary dehumidifier.

of heat exchanged in the dehumidifying coil of the evaporator at low temperatures. The air stream must then be reheated before being vented back into the store to maintain comfort.

In a hybrid air-conditioning system, air drying is achieved with a regenerative dehumidifier using either silica gel or LiCl as the desiccant. The central vapor-compression unit is used to cool the store air to the appropriate recirculation temperature. Figure 3 shows the schematic diagram of a possible hybrid desiccant cooling system, and Figure 4 illustrates the various processes in the psychrometric chart. Cool, dry air is vented into the supermarket store where its temperature and humidity level are increased (1 -> 2). Part of the circulated air is mixed with ambient ventilation air (6 + 2 -> 3), dehumidified (3 -> 4), cooled (4 -> 5), mixed with the recirculation air (-> 5), and finally recycled over the evaporator coil of the heat pump (5 -> 1). Ambient air is heated in a rotary heat exchanger (6 -> 7) and gas-fired heater (7 -> 8) to high temperatures, and then used to regenerate the dehumidifier (8 -> 9).

As indicated in these figures, the hybrid system reduces the load of the vapor compression system significantly. Studies have indicated that reductions in primary energy use by a factor of 50 to 70% may be achieved by proper

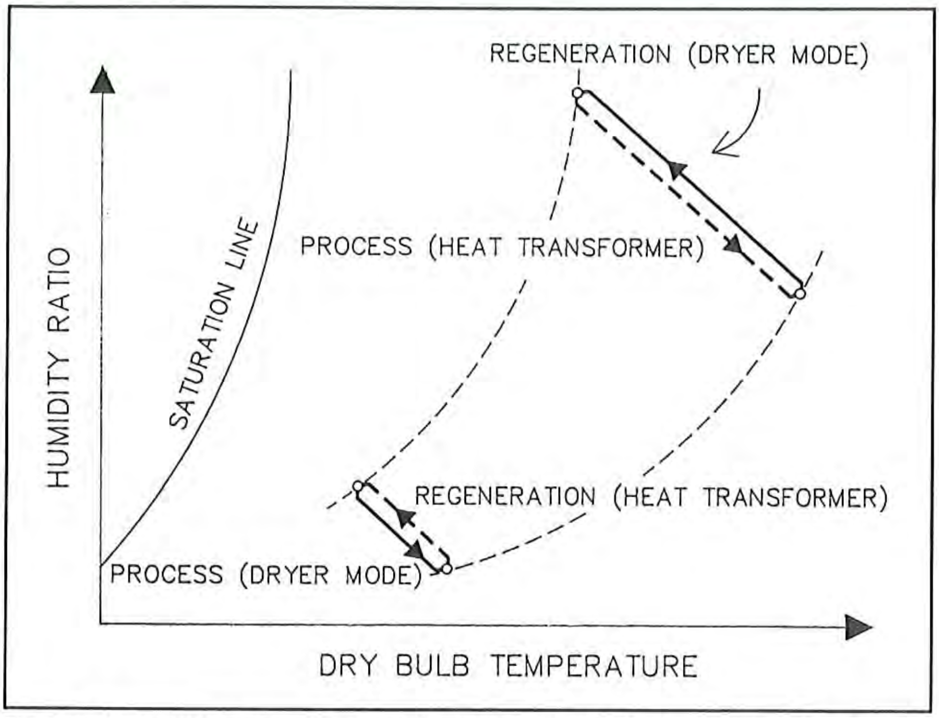

Figure 2. Psychrometric diagram showing the processes for the two operation modes of a rotary dehumidifier. The process lines approximate air-isenthalps. The dashed curved lines approximate lines of constant desiccant water load.

operation of the system (Burns et al, 1985). Furthermore, the evaporator temperature of the air-conditioner is increased substantially which leads to a higher COP of the vapor-compression unit.

Hybrid desiccant drying systems

Closed-cycle heat pump systems for intermediate temperature drying applications have been widely accepted as cost effective for reclaiming the latent heat of vaporization exhausted from the drying chamber. Primary energy savings up to 20% can be achieved depending upon the drying conditions. However, the specific drying capacity of a heat pump decreases rapidly as the drying temperature increases.

Open-cycle hybrid systems for drying applications may be used to extend the operating range of conventional heat pumps. Figure 5 shows a schematic diagram of a drying installation with a rotary dehumidifier operated as a heat transformer. The psychrometric chart in Figure 6 illustrates the operating principle. The adiabatic drying of the air stream within the chamber (2 -> 1) is reversed during the adiabatic drying in the desiccant wheel (1 -> 2). Outside air is passed over the condenser of the heat pump where the temperature is increased to the required regeneration temperature (3 -> 4). The air stream regenerates the desiccant (4 -> 5), is passed over the evaporator of the heat pump, and vented to the outside (5 -> 5).

Preliminary analyses have indicated that drying temperatures well above 100°C are possible for condenser tem-

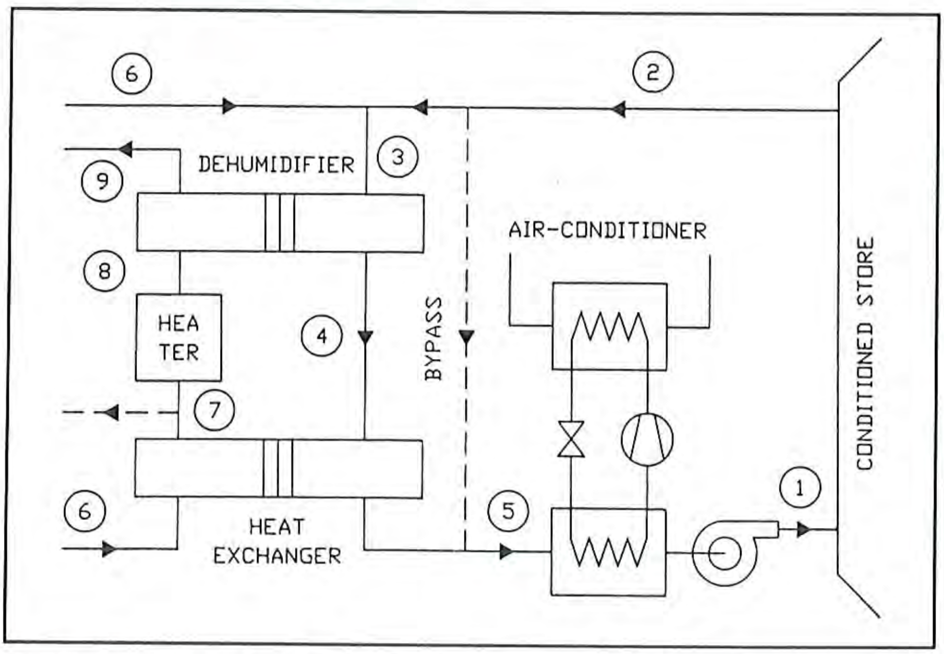

Figure 3. Hybrid cooling system with a dehumidifier-assisted air-conditioner

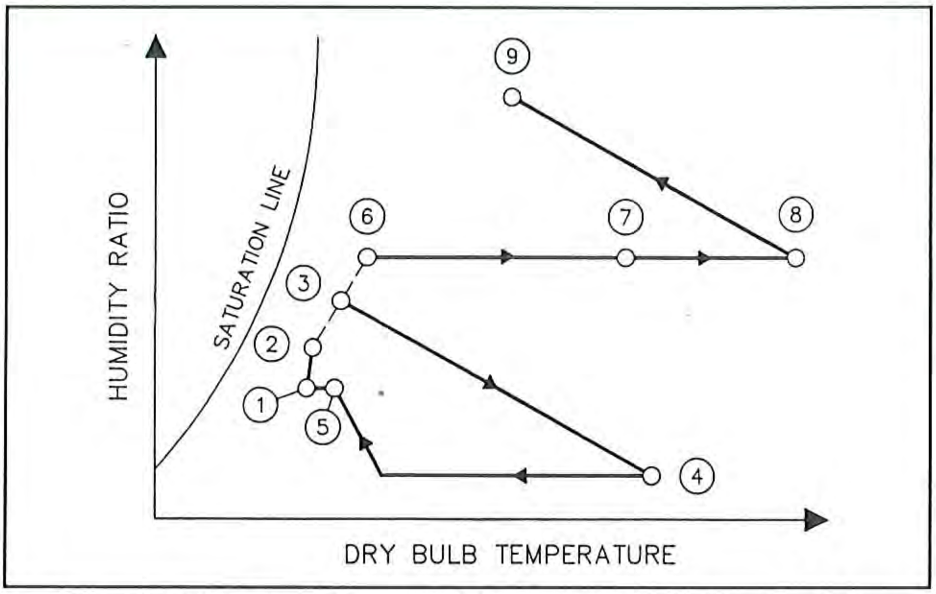

Figure 4. Psychrometric diagram for operation of the hybrid cooling installation shown in Figure 3.

ers can assist heat pumps wherever these latter devices are used for drying humid air streams. For space-conditioning applications, solid desiccant regenerators can replace part of the energy load of the vapor-compression unit, thereby increasing the specific air-conditioning capacity of the heat pump. Furthermore, the evaporator temperatures in these applications is increased significantly. The drying temperature in dehumidifier-assisted drying installations with heat pumps is raised well above the condenser temperature. In view of the current CFC problem associated with R114, R22 units can be used in the temperature ranges previously accessible only by R114 devices.

peratures ranging from 40 to 60°C. Thus, steam-air mixtures may be used for the drying process leading to a reduction in drying time. Depending upon the ambient conditions, primary energy savings with the hybrid system compared to conventional fuel-fired systems may be accomplished with a properly operated system. Quantitative estimates of energy use reduction still have to be determined experimentally.

Conclusion

Hybrid solid-desiccant heat pump systems are reviewed. Rotary dehumidifi-

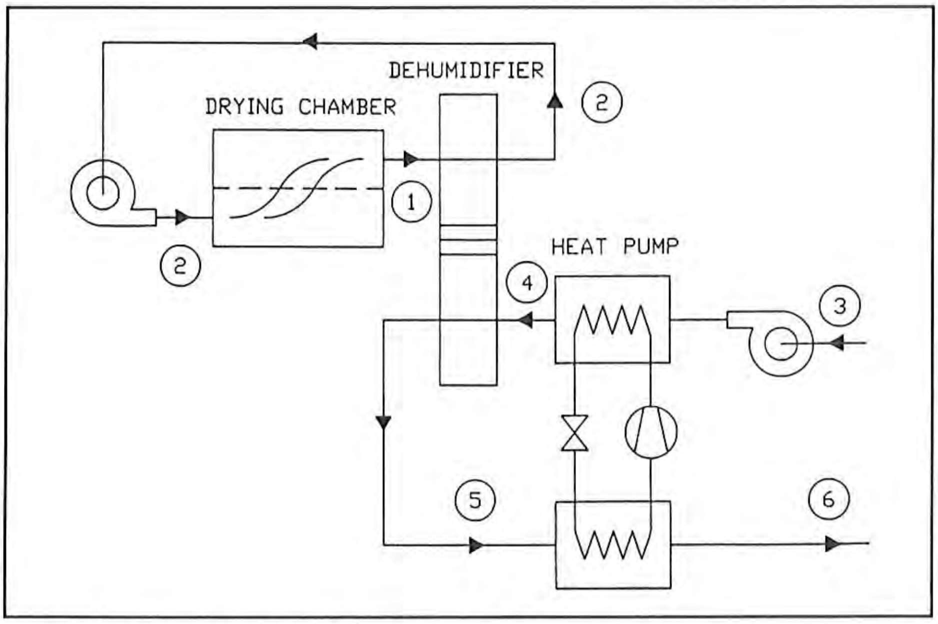

Figure 5. Hybrid drying installation with a dehumidifer-assisted heat pump

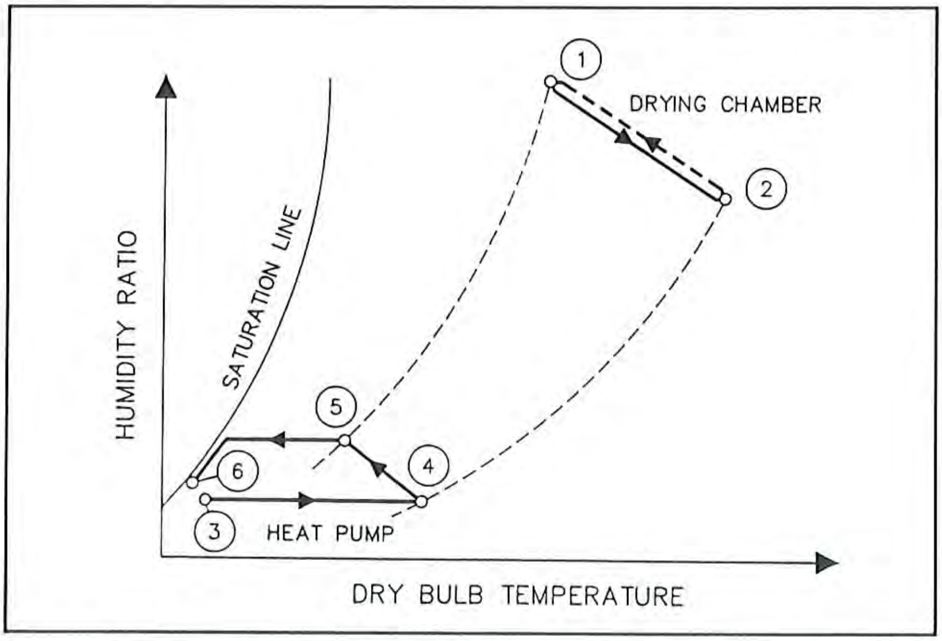

Figure 6. Psychrometric diagram for operation of the hybrid drying installation shown in Figure 5.

References

Manley, D.L., K.L. Bowen and B.M. Cohen, "Evaluation of Gas-Fired Desiccant-Based Space Conditioning for Supermarkets," ASHRAE Trans., Vol. 91, Pt. 4, pp. 447-456, 1985.

Burns, P.R., J.W. Mitchell and W.A. Beckman, "Hybrid Desiccant Cooling Systems in Supermarket Applications," ASHRAE Trans., Vol. 91, Pt. 4, pp. 457-468, 1985.

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Heat Pump Integration to Ethanol Separation

The rectification of mash to pure alcohol or an azeotropic mixture of alcohol and water is an energy intensive process. Heat pumps and process integration offer promising alternatives to reduce the energy demand. A comparison of these two technologies shows that heat integration is favorable as long as it is possible. Heat pumps may offer a further reduction of heat demand in integrated processes, as long as the remaining temperature levels are suited. In general, heat pumps will be advantageous in retrofits, where column pressures cannot be changed.

Introduction

Various technologies for the separation of mash - a mixture of ethanol and water - to produce azeotropic or pure alcohol are known.[1] The only well-established process in use is distillation. This technology requires great amounts of energy-up to 11,000 kJ/kg ethanol or almost 40% of its heating value. Heat recovery and heat pump application will reduce the energy demand and costs if applied correctly. The PINCH-technology-a proven tool for heat integration projects-can be applied in this case, too.

Ethanol separation

Ethanol and water form a mixture that shows an azeotrope at about 95% mole of ethanol. Due to this fact, distillation cannot be done directly up to pure ethanol. The exact concentration for the azeotrope depends on the pressure. At very low pressures it vanishes.

In the following discussion, two arrangements for the separation process will be considered. Figure 1 shows a three-column process for generation of an ethanol-water azeotrope. The second distillation column (V2) is installed to take off low boiling components, for example, methanol.

To achieve pure ethanol, an additional two columns for extractive distillation have to be installed. A third fluid, e.g., cyclohexane or benzene, is added to the azeotrope and recovered in a second column. The whole scheme is shown in Figure 2.

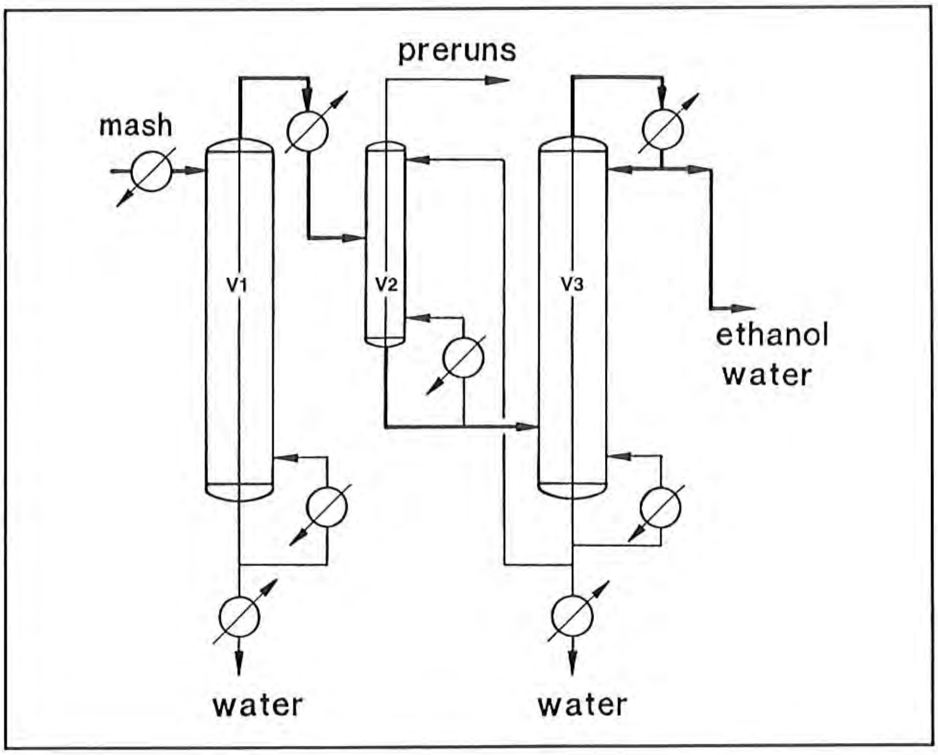

Figure 1. Three-column arrangement

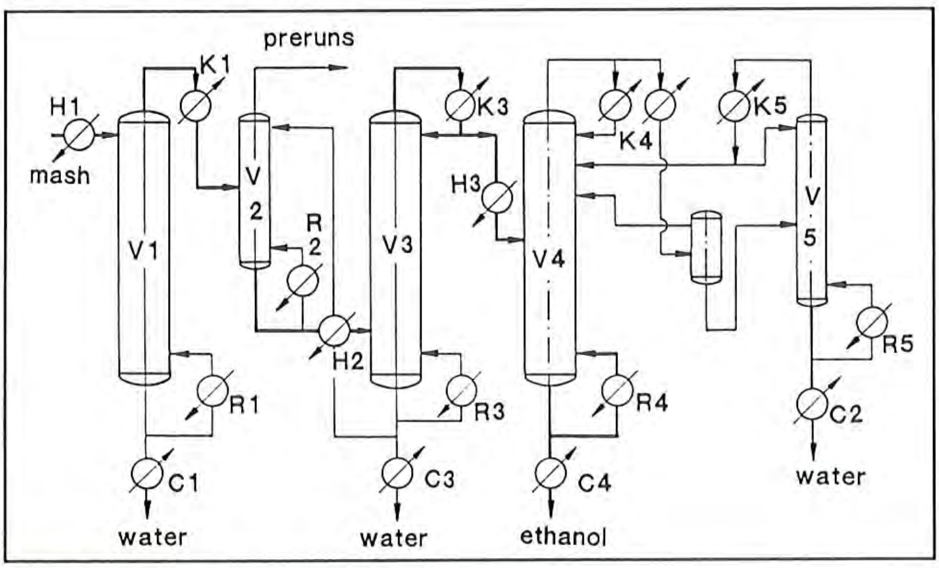

Figure 2. Five-column arrangement for production of pure ethanol

The two flow schemes do not show any heat recovery. The great number of heat exchangers [preheaters (H), reboilers (R), condensers (K), product coolers (C)] suggest heat integration studies. From the composite curves, achieved in a PINCH-point analysis, it can be seen that only a small amount of heat can be exchanged between condenser K1 and the mash preheater. For the theory of PINCH-point analysis and a discussion of composite curves, see references 2, 3, and 4. In Figure 3, the horizontal lines in the composite curves represent the reboilers and condensers. In K1 a mixture is condensing, so there is no constant temperature.

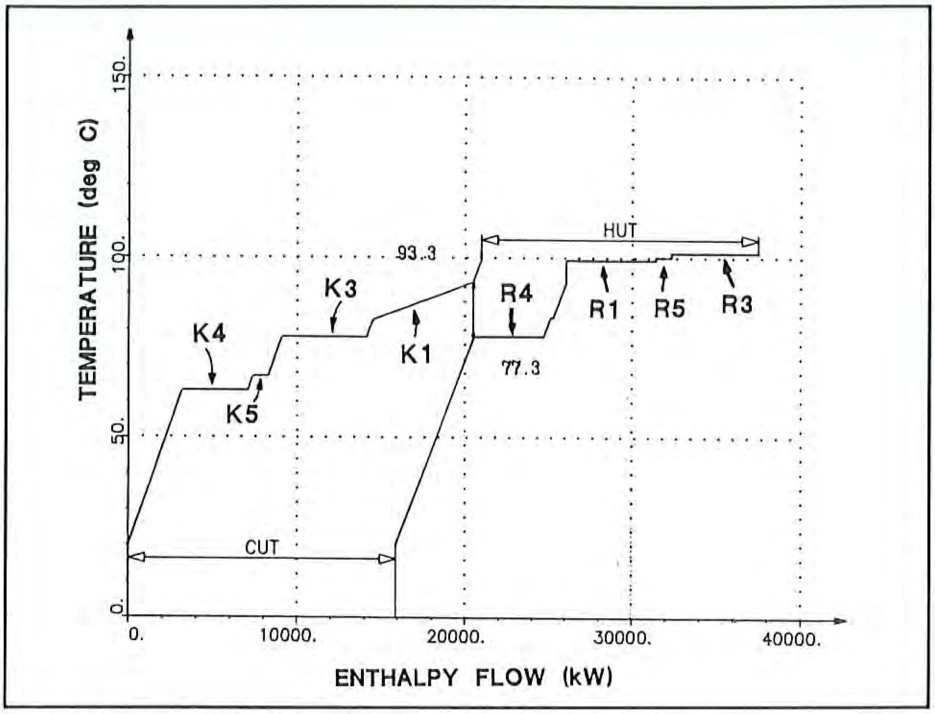

Figure 3. Composite curves for the atmospheric five-column scheme

In order to make heat recovery possible, we can use three strategies:

- Integrate heat pumps
- Shift the temperature level of reboilers and condensers by changing the column operating pressure
- Apply heat pumps to heat integrated arrangements

The result of each of these will be described in the next sections.

Application of heat pumps

According to PINCH-theory, heat pumps have to be placed across the pinch, which means that they have to take up heat from below the pinch and release it to the process above the pinch. To find an appropriate location for the heat pump, we should study the "Grand Composite Curve" (Figure 4). This graph shows the heat demand (above the pinch) and the available waste heat (below the pinch) for each temperature.

Obviously, there are great amounts of heat available between 85 and 75°C and at 54°C. Heat demand exists at 85°C and 108 to 110°C. Since only a very small temperature lift is required between 75 and 85°C, an open type of heat pump, a mechanical vapor recompressor can be chosen. With an electric power of 240kW, about 4,500kW of steam can be replaced if the vapors of V1 are compressed to reboil column V4. Figure 5 shows the Grand Composite Curve for this modification. The overall hot utility target came down from 16,560kW to 11,690kW.

For the next heat pump a lift from 68°C to 108°C is necessary (these temperatures already include the T for heat transfer). We suggest using an absorption heat pump for this. Simulations have shown a necessary steam temperature for the generator of 190°C (Figure 6).

Heat demand now comes down to 6,960kW. All the other heat requirements, except the power for the MVR, can be supplied through heat exchange.

Process integration

Process integration is another method of reducing the energy requirements of the original process. According to pinch point theory, a distillation column should never be located across the pinch. That means that reboiler and condenser should both be above or below the pinch temperature. In our case, all five columns are located incorrectly. There are a great number of possibilities to shift pressures in five columns. Out of practical considerations, only a few are worth a closer look.

One possibility is to increase the operating pressure from the first column to the last one. With this modification, the hot utility target could be brought down to 6,660kW, which is a number similar to the heat pump arrangement. Another promising modification is to keep the mash column at about atmospheric pressure and have the other columns operate below or above that pressure. The energy conservation rate is about the same as before.

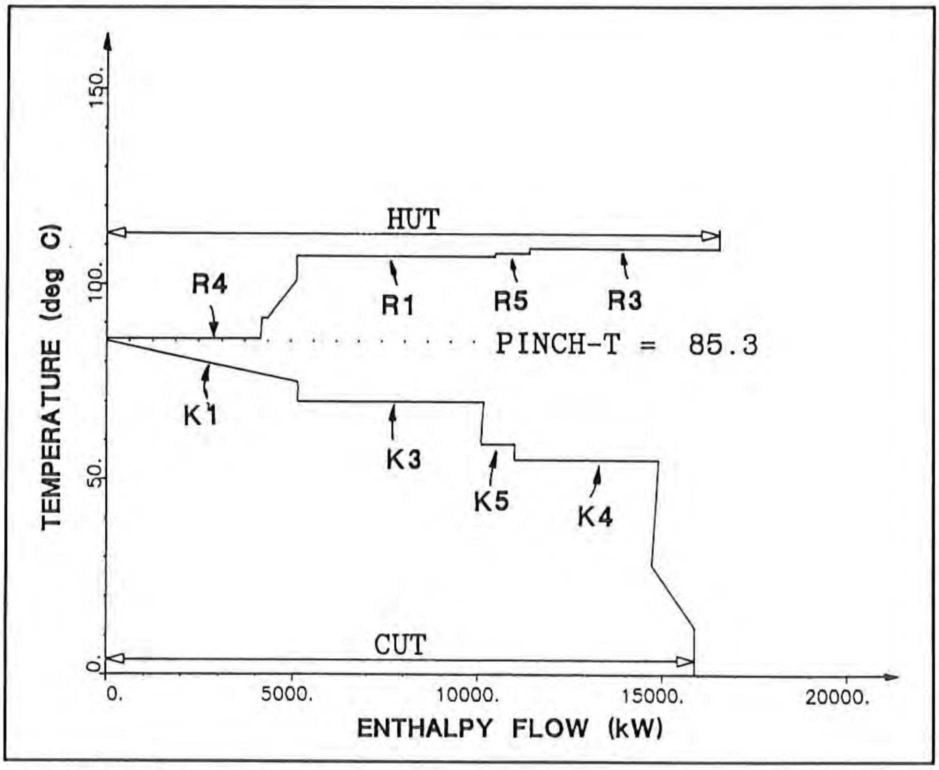

Figure 4. Grand composite curve for flow scheme of Figure 2

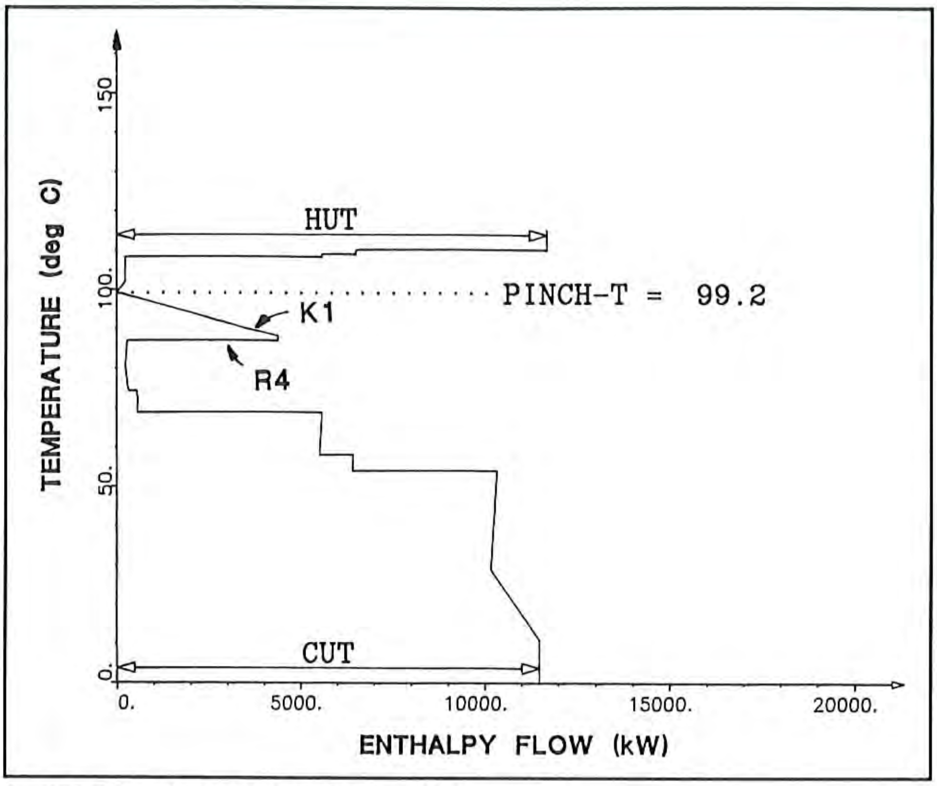

Figure 5. Grand composite curve including MVR

This analysis also showed the dehydration part, V4 and V5, does not require additional energy but can be operated merely by waste heat. Thus the three-column arrangement will need the same amount of heat utility as the five-column scheme.

A further reduction of the energy input could be achieved by parallel operation of the individual columns, but this would require splitting of all three big columns (V1, V3, and V4) to an eight-column system. Temperature and pressure levels are too high in this case to give an economically feasible layout.

Heat pumps in integrated ethanol processes

After heat integration and heat pump application have come to a limit individually, the question for a combination is still open. From the integration analysis it was determined that a heat pump applied here would have to overcome a temperature lift of 80 or even 100°C. There is no economic way to do that today. Things look different in the three-column arrangement (Figure 1). Here the hot utility target is about the same as in the five-column case but heat is used only twice. Since the temperature levels of waste heat (e.g., 95-100°C) and heat demand (e.g., 145°C) are closer to each other, a heat transformer working with H_2O-LiBr has been designed. Heat demand is then reduced to 4,520kW (see Figure 7).

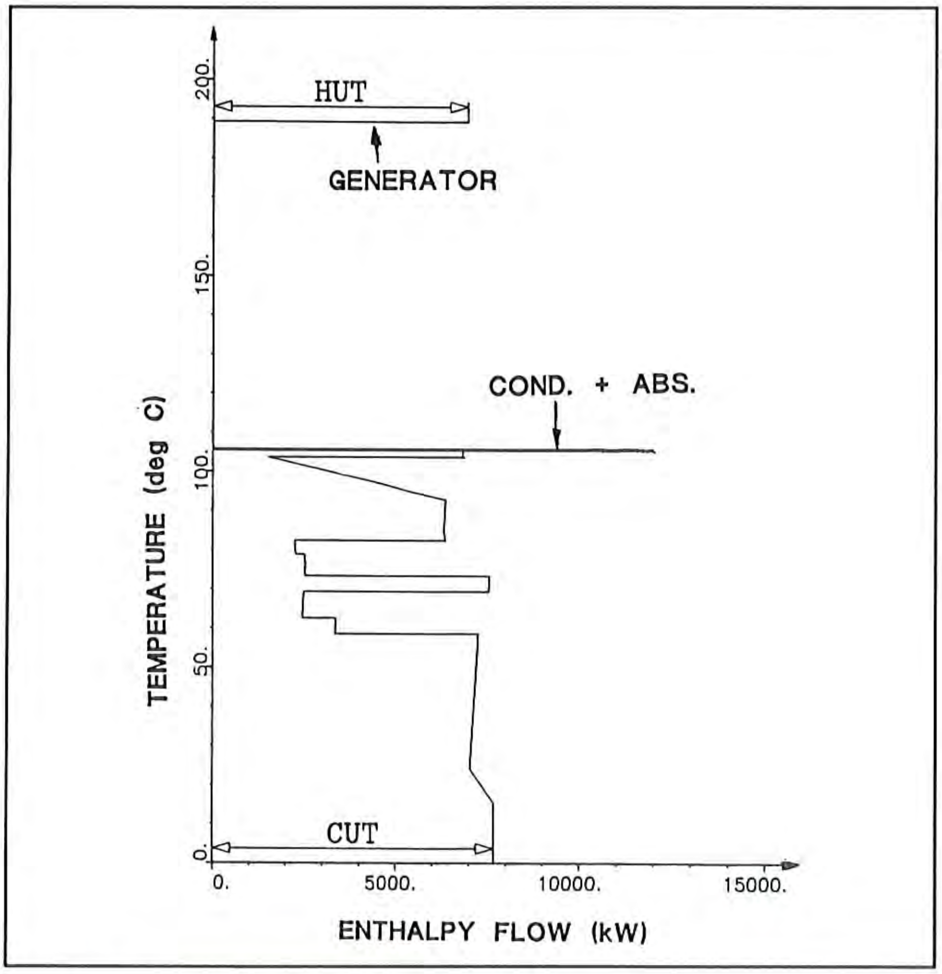

Figure 6. Grand composite curve including MVR and AHP

Conclusions

There are two ways to reduce the heat demand of distillation chains: heat integration and heat pump application. As long as heat exchange can be made possible, this will be the more economic way in general. In some cases, for example, if waste heat still is available at an appropriate temperature level after heat integration, additional heat pumping is possible. Calculations on the economy of the various flow schemes showed that the heat transformer in the three-column process gives excellent economic figures. At today's energy prices, the payback period for the heat transformer is 12,000 to 16,000 hours of operation, which should be less than two years for industrial installations.[6]

Although heat pumps might be less economic in new plant design, they offer a great potential for the modification of existing separation plants where the column pressures may not be changed.

References

1. Delago, H. and H. Schnitzer. "Eine

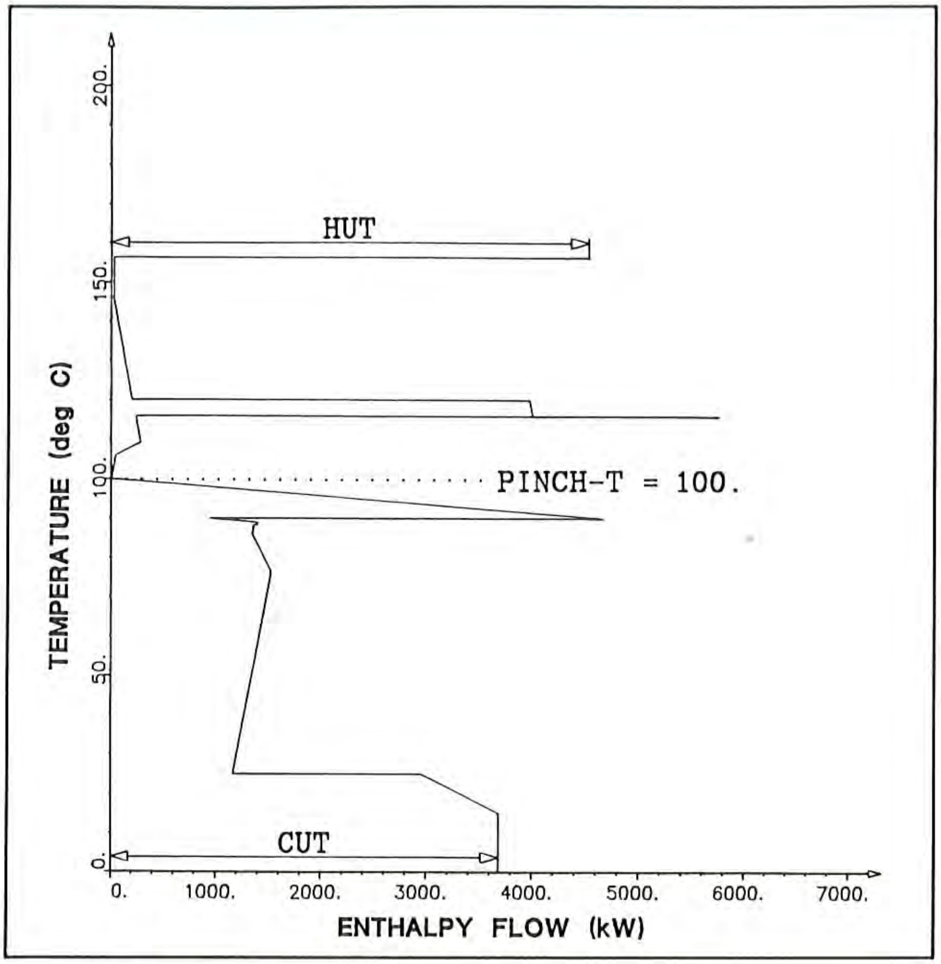

Figure 7. Heat transformer in heat integrated distillation chain

systemanalytische Untersuchung der Alkoholabsolutierung." *DECHEMA* 1988.

2. Linnhoff, B, et al. *A User Guide on Process Integration for Efficient Use of Energy*. The Institution of Chemical Engineers, Rugby, 1982.

3. Witherell, W.D., and F.A. Conneway. "Heat Pumps in Evaporation." Newsletter of the IEA Heat Pump Center, Vol 5, No 3, Sept 1987.

4. Smith, R., and B. Linnhoff. "The design of separators in the context of overall processes." *Chem Engr Res Des*, 66 (1988), p. 195.

5. Ferner, H., and H. Schnitzer. "Die Pinch-Technik - Grundlagen, Potential, Grenzen." *OEIAZ* (1989), 2.

6. Schnitzer, H., W. Hoelbling, and F. Moser. "Economic Aspects of Heat Pump Integration in Ethanol Separation Processes." Proceedings of the 2nd Workshop on Advanced Heat Pumps, Graz, Sept. 1988.

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O. Nesje*

Heat Pump Research and Application in Norway

This article gives an overview of heat pump technology in Norway. Future trends, the current heat pump market, and ongoing R&D efforts are discussed and described. Data presented shows that the largest number of heat pumps are currently used for building heating followed by applications in the fish and wood processing industries.

Introduction

In Norway, the combination of climate and availability of 4-6°C ocean water as a heat source where most of the people live provides favorable conditions for the use of heat pumps to cover heat demand in buildings. The heating season exceeds 300 days in some locations, which gives a good utilization factor for heat pump systems. The use of heat pumps in heating systems depends on the economy in the actual projects, but also on the contractor's knowledge of and attitude towards heat pump technology.

Electricity and oil/gas supply

The electricity price and the oil/electricity price ratio affect the economy of heat pump projects. The energy situation in Norway is different from many other countries. Nearly 100% of the electricity is generated from water power, and the price has until recently been very low. Norway is also an oil and gas producing country, and necessary supply of that kind of energy is ensured.

Primary energy price and heat pump application

Low electricity prices combined with presently high capital costs for investments have until now made mainly the big heat pump installations economically favorable. In a longer perspective, however, we know that oil and gas resources are limited. The price of primary energy (exergy) will, therefore, probably increase in the future. Development and improvement of other energy converting systems cost money and time. These factors indicate that prospective use of heat pumps as a part of heating systems will be more favorable both from an economic and energy point of view.

		Number of units	Installed effect (MW)
Building heating	Water as heat source	300	80
	Air as heat source	4000	80
	Heat recovering from refrigeration plants	300	50
Fish industry	Fish drying	50	10
	Fish farming	200	30
Agriculture	Hot-houses for vegetables and flowers	150	6-8
Wood-processing industry	Evaporation in pulp and paper industry	3	100
	Wood drying	250	10
Dairies	Process applications	11	6

Table 1. Installed heat pump capacity and areas of application in Norway

Government promotion

R,D&D on heat pump technology has been supported by the government during the last 12 years. The technical knowledge has increased, and courses in heat pump design and calculation are available for contractors and engineers in the industry.

Market situation

On the order of 6000 heat pumps are installed in Norway. Total capacity of these units is about 400MW, and the corresponding energy from them is approximately 1.2TWh. Most of the heat pumps are small, but 500-600 are larger than 50-100kW and about 100 are in the MW range.

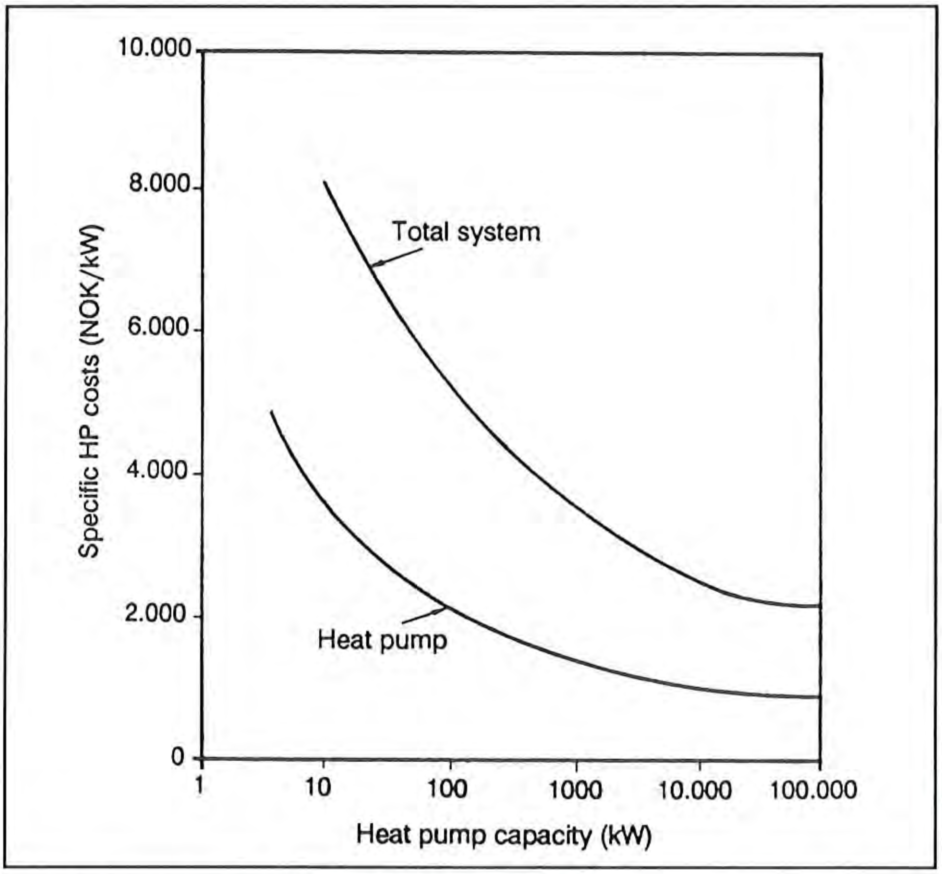

Figure 1. Typical costs per KW installed heat pump capacity

Areas of application

Table 1 indicates installed capacity and number of units within the biggest areas of application. Nearly 50% of the total heat pump capacity is installed for industrial purposes, and the other 50% is used to heat buildings.

Yearly installations

The number of heat pump units installed per year has been increasing. About 500-700 smaller combined air conditioning units and 200-300 heat recovering units for domestic use are sold per year. Approximately 200 bigger units for trade buildings, etc., are installed per year.

Heat pumps for space heating

Characteristic for the heat pumps used to heat buildings is that more than 90% of the total capacity is installed in trade and industry buildings. Less than 10% is for domestic use.

Mainly three categories of systems are used for heating buildings. Heat pumps with air and water as a heat source represent the majority of the installed capacity. A difference is, however, that the water-based systems are bigger units compared to the air source heat pumps. About the same capacity is installed within each category, but the number of air-based units is 4000 compared to only 300 water-based systems. A significant contribution to building heating using the heat pump principle is also heat recovery from refrigeration plants.

Heat pump costs

Typical costs per kW installed heat pump capacity are shown in Figure 1. The curves indicate costs for water/water heat pumps. The specific cost is significantly affected by the heat pump size.

The costs are also strongly dependent on the necessary installations and work connected to the actual situation on the heat source side. Room for the heat pump machinery, work with piping, etc., must also be included. The upper curve in Figure 1 indicates average complete prices with all these necessary additional costs included. The lower curves show the costs connected only to the heat pump components with necessary control equipment and installation. Depending on the actual plant, the total price may often be more than twice the costs connected only to heat pump components.

The potential for further utilization of heat pumps for energy saving towards the turn of the century is estimated in the range 5-15TWh. This is mainly connected to heating of buildings, but the fish farming industry also represents a potential for increased use of heat pumps.

R,D&D on heat pump technology in Norway

During the last 12 years, about 58 million NOK have been spent on heat pump research work and the related equipment and facilities. This activity is funded by government and industrial agreements. In addition, prototype projects in the industry are supported by the government, and about 25 million NOK have been spent for this purpose during the last eight years.

Research activity

Most of the research work in Norway within heat pump technology is carried out at the Division of Refrigeration Engineering at the Foundation for Scientific and Industrial Research at the Norwegian Institute of Technology (SINTEF). Subjects of priority in this work can be summarized as follows:

-- **Heat pump systems.** The fish industry is important in many regions of Norway. The heat pump principle is well suited for use in fish drying. Research work has therefore been carried out in order to develop basic data and models for fish drying. Heat pump systems for use in the drying process are developed, and the design basis for such systems is established. The use of heat pumps for this purpose has been very successful in Norway. Fish farming is a growing industry in Norway. Heat pump systems for heating the process water are being tested, and the use of the concept is increasing. A new range of application for heat pumps which may come up is drying/treating of fish fodder to make it as suitable for the purpose as possible.

-- **Computer program calculating heat pump systems.** Good economy in heat pump projects is dependent on correct design of the plants. In complex plants it is more difficult to verify the consequences of different sizing and component efficiencies. Development of a computer program for calculation of heat pump systems in general has, therefore, been done. This program, PROSIM, is used in education and courses which are available for engineers in the industry. The software is also implemented on personal computers (PCs) and is available for the industry.

-- **Optimized operation, field experiments.** To ensure optimum utilization of a heat pump, the best possible control strategies are necessary. Analysis of control algorithms and development of computer-based systems for efficient real time monitoring of process data and control of heat pump systems is being carried out. Systematic field measurements and tests on prototype plants have been an important part of this work.

Heat pump components

Optimum design and best possible efficiency is, of course, of importance for all components in a heat pump system. The research work is, however, concentrated on heat exchangers on the heat source side. Experience has shown that malfunctioning evaporators often are a reason for reduced heat pump efficiency. The most commonly used heat sources are water and air. Research on evaporators for these purposes is therefore emphasized.

-- **Development of plant evaporators.** Water as heat source represents problems such as corrosion and fouling, depending on the water type and quality. Soiled water from industry and sewage water is in many cases a good heat source. Plate evaporators have in such cases advantages because of easy removal of the fouling, and compared to a shell-and-tube design, damage caused by frost formation can hardly occur. Laboratory tests and development of basic data and theory for optimum design of plate evaporators for liquid cooling is carried out. The concept is used in large heat pump installations, and field measurements show that such evaporators have high efficiency and are well suited for cooling highly polluted water.

-- **Computer program calculating air coolers.** Air is always available and in many cases a suitable heat source for a heat pump plant. The most commonly used evaporator type for this purpose is tube-in-fin evaporators. Also in this case we have a kind of fouling problem, represented by frost formation. Optimum efficiency is highly dependent on correct piping and refrigerant distribution in the evaporators. Knowledge about subjects such as basic heat transfer, calculation of both recirculation and dry type evaporators, frost formation and refrigerant distribution is necessary if optimum design concerning piping, fin spacing, etc., shall be achieved for a defined purpose. These subjects are particularly emphasized in the research work. A computer program for detailed calculation and design of tube-in-fin evaporators has been developed. This program will also be available for the industry, and makes it possible to calculate these types of heat exchangers with high accuracy.

Also other concepts for air cooling based on the fluidized bed principle are investigated. This represents a big potential for improvement of the efficiency of air coolers for heat pump purposes

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Compressor Heat Pump for Greenhouse Heating

In the Netherlands the gas and electricity company Delfland installed a gas-motor driven air-to-water compressor heat pump of 250kW in a greenhouse. For a full year this heat pump was carefully monitored as was the conventional heating system and the greenhouse climate itself. Computer models were developed of the greenhouse and the heat pump, which were validated on the measured data and subsequently used for evaluating the heat pump project. The installed heat pump system performed with a relatively high COP and proved to be reliable. In the near future the simulation models will be used in an optimization study on heat pump application in greenhouses.

Introduction

In the Netherlands the greenhouse cultivation industry (vegetables, flowers) is extremely energy consuming. In the first place the relatively cold climate and the fairly high greenhouse temperatures lead to a large natural gas consumption per m^2 of greenhouse area: average about 40 m^3/m^2 for tomatoes and cucumbers[1] (most greenhouses are heated with natural gas). Secondly the greenhouse industry in the Netherlands is a large one: about 15350 firms[2] comprising a total surface of over 8*107 m^2.[3]

Application of gas-motor driven compressor heat pumps in greenhouse heating could, therefore, lead to important national energy savings and possibly to cost reduction of the products. It is expected that for tomatoes and cucumbers a correctly dimensioned heat pump will lead to a reduction of 3-7m^3 natural gas consumption per m^2 greenhouse area. However, in practice the energy savings generally lag behind compared to the expectations due to sub-optimal design and the occurrence of more than expected failures. Increasing attention for heat pump technology is required, focused on the choice of evaporator, the heat control, interaction with CO_2-fertilization and heat storage and on standardization. In this context the Delfland heat pump project should be seen.

The Delfland heat pump project

In 1984 the gas and electricity company Delfland installed a gas-motor driven compressor heat pump in a greenhouse. On this heat pump system a fairly extensive research project was conducted, financed in part by NOVEM on behalf of the Dutch Ministry of Economic Affairs besides the financing by the utilities firm itself.

The first phase of this project comprised the following:

-- From September 1986 up to August 1987 the TNO Institute of Applied Physics performed an extensive monitoring program in which data from the heat pump, the conventional heating system, the greenhouse climate and the ambient climate were acquired on an hourly basis. These data were analyzed, giving insight in the total system performance.

-- A computer model was developed by the TNO Institute of Applied Physics simulating the total heating system including the greenhouse climate. The Delft University of Tech-

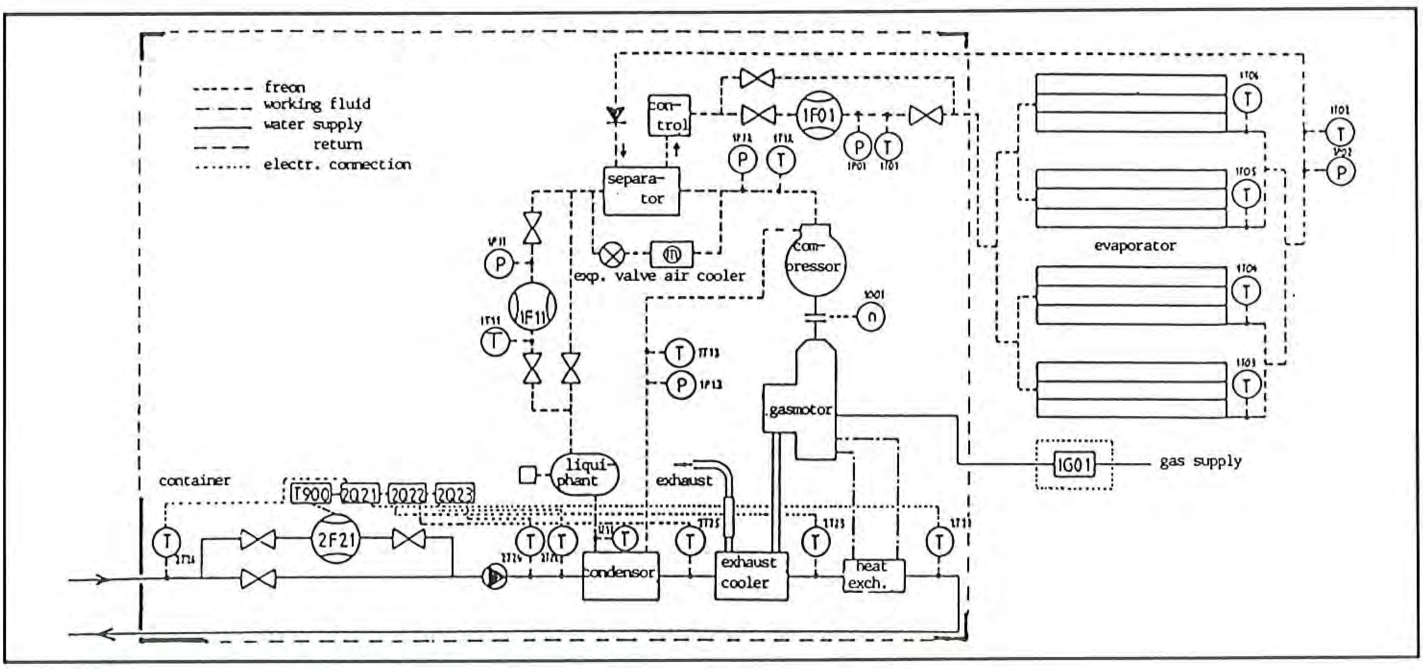

Figure 1. Schematic diagram of the compressor heat pump

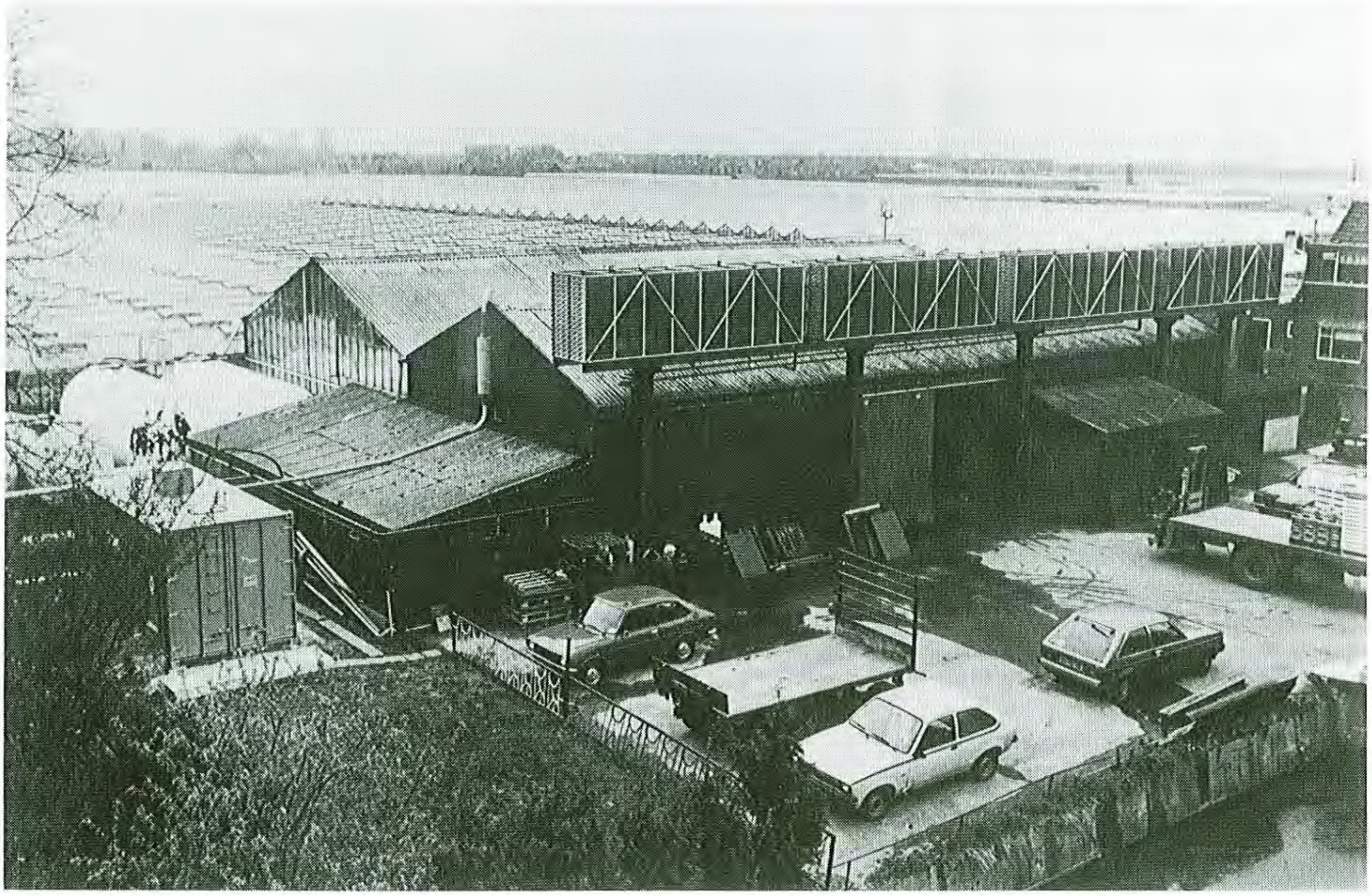

Figure 2. Overview of the greenhouse showing the evaporator

nology developed the model of the heat pump unit itself.

-- The developed computer models were validated and were used to evaluate the realized heat pump project.

-- The torsional vibration behavior of motor and compressor was studied.

-- Various noise level measurements were performed and the exhaust gasses were analyzed.

In the second phase of the project (now underway) the validated computer models will be used in a parameter analysis and system optimization study in a more general sense leading to suggestions for optimal design of heat pump systems for application in greenhouses.

The heat pump system

Due to the requirement of flexible implementation in greenhouse heating systems, an air-to-water compressor heat pump (CHP) was selected. The CHP, schematically shown in Figure 1, has a so called "silent evaporator" consisting of four blocks of plain piping. Figure 2 shows, among other things, the evaporator. An essential characteristic is that defrosting, as with forced ventilation evaporators, is not neces-

	Ambient		Boiler		CHP		Time
	T_o (°C)	v_w (m/s)	Q_{boil} (MJ/m^2)	Eff (-)	Q_{CHP} (MJ/m^2)	COP (-)	N (h)
Sep	10.1	2.1	96	.973	27.7	1.80	503
Oct	9.8	2.8	95	.968	22.5	1.84	501
Nov	7.4	5.8	0	-	0.1	0.00	0
Dec	4.8	5.2	185	.944	37.4	1.73	593
Jan	-0.4	4.1	374	.900	45.3	1.55	828
Feb	1.7	3.0	239	.921	36.3	1.64	647
Mar	0.5	3.9	278	.906	37.3	1.66	639
Apr	8.0	3.0	154	.890	42.4	1.80	690
May	8.1	2.9	144	.984	0.1	0.00	1
Jun	10.9	2.2	101	.980	21.4	1.84	428
Jul	15.4	2.0	80	.946	24.8	1.92	522
Aug	15.6	3.2	68	.985	21.4	1.86	486
Year	7.6	3.3	1826	.927	319.	1.73	5883

Legend:

T_o	Average ambient temperature
v_w	Average wind velocity
Q_{boil}	Total heat production by boiler
Eff	Average total boiler efficiency
Q_{CHP}	Heat production by CHP
COP	Average Coefficient of Performance
N	Operation time of the CHP

Table 1. Various monthly and yearly values measured in the heat pump project

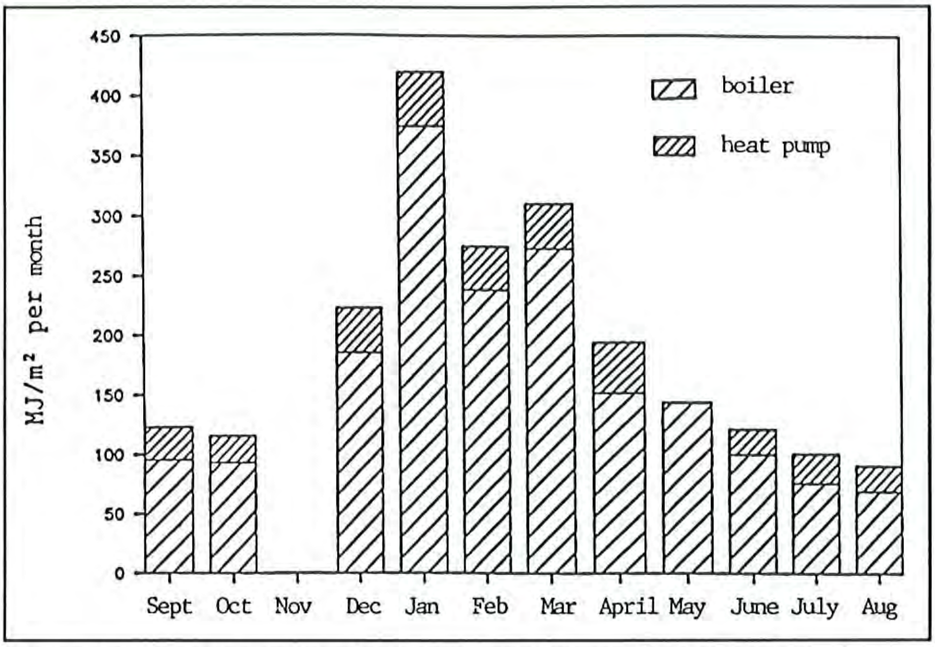

Figure 3. Monthly heat production of both boiler and CHP per m^2 greenhouse area during the period September 1986-August 1987

sary. The CHP also functions with satisfactory efficiency when a layer of ice is present on the evaporator tubing.

Besides the evaporator the CHP consists of a 6-cylinder gas motor, a 6-cylinder compressor and a shell-and-tube condenser. Two heat exchangers recuperate waste heat from flue gasses and cooling jacket of the motor. The working fluid is freon 22, which is pumped through the evaporator by a special ejector system causing "wet evaporation."

The control of the CHP is achieved in two ways. Going from high to low capacity in the first place, the number of revolutions is reduced from 1600 to 1000 rpm. Further reduction of capacity is achieved by disconnecting cylinders from the compressor from 6 to 4 or 2 cylinders. respectively. The CHP has a thermal capacity of 250kW. The measured NO_x emission was at the utmost 250 g/GJ.

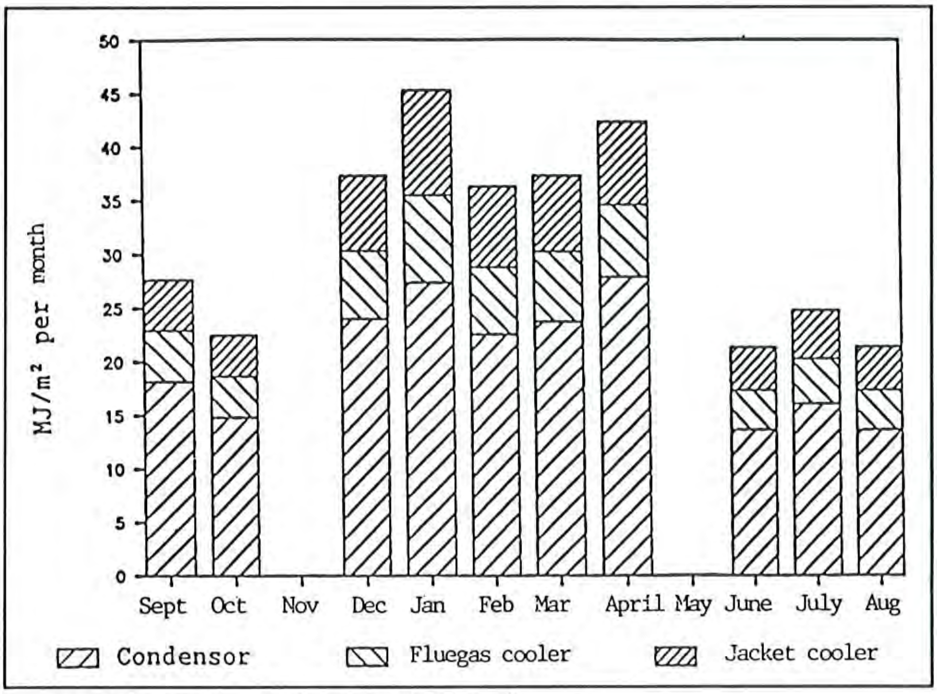

Figure 4. Monthly heat production by the CHP per m^2 greenhouse area

The greenhouse heating system

The greenhouse concerned covers an area of 13500m². Tomatoes are cultivated on a substrate basis. In November there is a cultivation break of a month. The original heating system consists of a conventional boiler (3500kW) equipped with two flue gas heat recuperators. Two heat distribution circuits are employed: a high temperature circuit (operation temperatures: 70-90°C) fed by the boiler and the first flue gas cooler, and a low temperature circuit (operation temperature: 40°C) fed by the second flue gas cooler.

Ambient temperature	9.5	(°C)
Average wind velocity	3.1	(m/s)
Average greenhouse temperature	18.56	(°C)
Heat loss by transmission	1751.9	(MJ/m²)
Heat loss by natural ventilation	259.5	(MJ/m²)
Heat loss through windows	494.7	(MJ/m²)
Heat loss through soil	-228.4	(MJ/m²)
Heat gain from solar radiation	454.5	(MJ/m²)
Heat supply high temp. circuit	1370.8	(MJ/m²)
Heat supply low temp. circuit	452.2	(MJ/m²)
Average CHP capacity	94.13	(%)
CHP heat production	19.98	(%)
Operation time CHP	6592.	(hours)
Average COP	1.69	(-)
Gas consumption boiler	44.81	(m³/m²)
Total gas consumption	50.96	(m³/m²)
Gas savings	5.19	(m³/m²)

Table 2. Reference calculation: various annual values for the greenhouse

Furthermore the installation has two heat storage vessels of 25m³ each. These are used when there is a CO_2-demand but no heat demand, in which case the superfluous boiler heat is stored. CO_2-fertilization is achieved by leading the flue gasses through the greenhouse.

The CHP is installed parallel to the second flue gas cooler feeding the low temperature circuit, which was substantially enlarged for this purpose.

The control of the greenhouse climate is originally managed by a computer.

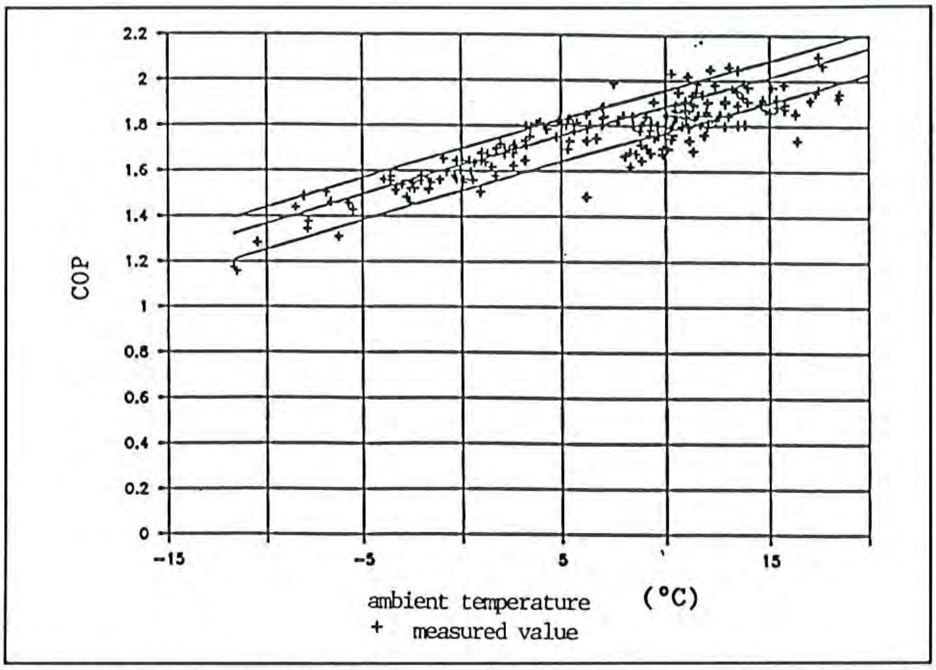

Figure 5. The COP calculated from the measured data (on a daily basis) and the regression lines for wind velocities of 0.3 and 6 m/s.

Software adjustments were made for controlling the heat pump so that the CHP has priority over the boiler. However, CO_2-fertilization, not possible with the flue gasses of the CHP motor, always has top priority when needed. This also counts for the "minimum pipe temperature" which is achieved only with the high temperature circuit, when necessary to enhance air movement stimulating plant transpiration. For maintaining the desired greenhouse climate the computer allows about 50 adjustable parameters.

The monitoring program

The monitoring program on one hand was set up to gain insight in the performance of the complete greenhouse heating system. On the other hand the heat pump parameters, e.g., various freon flows, pressures and temperatures, were measured in behalf of the validation of the heat pump model. For both aims the nature of the signals and the sampling rate were the same, thus permitting the use of one measuring system. Continuously eight heat flows were sampled and average hourly values for the greenhouse and ambient climate were measured. Various status signals and valve positions were registered to decide the behavior of the control of the system. A detailed summary of the various sensors and measuring procedures is given in reference 4.

Simulation models

Computer models were developed of the CHP and the greenhouse heating system. The heat pump model calculates the hourly heat production and the gas consumption depending on climate conditions, the temperature at the condenser side and the control signals. With this also the efficiency is decided. This rather extensive computer model was validated against the measured data after which a simplified heat pump model was derived. The simplified model, deduced from the extensive model by statistical techniques, is more suitable for the simulation studies in which the yearly values are calculated.

The greenhouse model simulates the greenhouse climate (temperature, humidity) and calculates the hourly heat

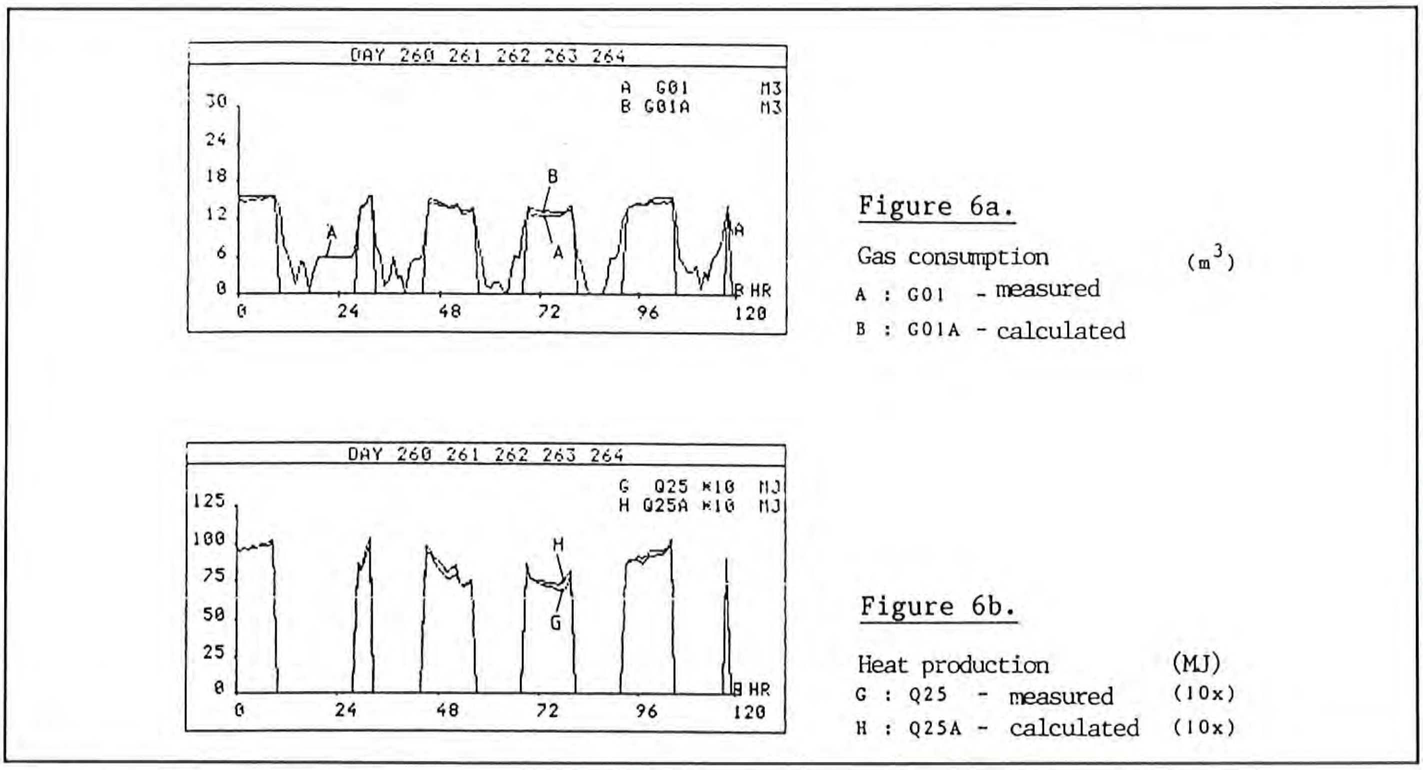

Figure 6. Calculation result of the CHP model and measured values for a number of consecutive days.

demand depending on outside climate conditions, greenhouse management and state of the vegetation. On the basis of this, the heat production is calculated considering the presence of a boiler as well as a heat pump.

Results - general

In general it can be concluded that the CHP system performed reliably. Average number of full-load operating hours was 4000 to 6000 a year, average yearly efficiency (COP) was 1.73, based on a combustion value of 35.1 MJ per m^3 of natural gas. Analysis of the torsional vibration behavior points to a long life of gas motor and compressor. After 10000 operation hours no major overhauls were necessary.

A point of concern was the noise level problem. An admissible sound level of 35 dBA at 1m from the neighboring property applied to this region, which was less than the measured background level. Despite this, the required level could be met.

An important aspect was the failure of the CHP flue gas cooler, which occurred twice in three years. This is ascribed to traces of freon in the combustion air and/or lubricating oil as a consequence of freon leakage. Freon leakage, in this project slightly enhanced by the installation of the freon flow meters, is an important problem, not in the least because of environmental arguments.

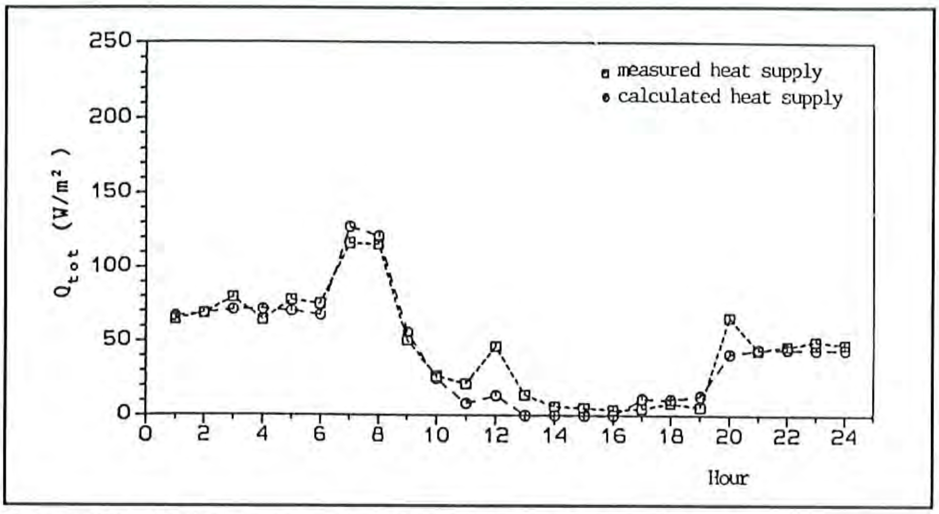

Figure 7. Calculated and measured hourly greenhouse heat demand for one day.

Results - monitoring program

Table 1 shows the performance of the heat pump in the monitoring period 1986/1987 on a monthly and a yearly basis. The efficiency values of the CHP and the boiler are based on the upper combustion value of natural gas (35.1 MJ/m^3). The COP is defined as the ratio of the totally produced heat on the condenser side to the combustion value of the consumed gas. Figure 3 shows the contribution of CHP and boiler to the total heat demand. In November the heat demand is zero due to the cultivation break, in May the CHP was out of operation due to the faulty flue gas cooler. The obtained average COP of 1.73 is certainly a good result, considering also the fact that due to extremely cold weather in January and March the high and the low temperature circuits were connected to the boiler leading to relatively low COP values. The number of operation hours (about 6000) is extremely high due to, among other things, the silent evaporator. Also the capacity of the heat pump is rather limited leading to frequent full-load operation.

From the heat production of the heat pump a savings of 62500m^3 natural gas can be calculated during the monitoring period, assuming an equivalent heat supply from the boiler with an efficiency of 0.9. The heat flows of the CHP are shown in Figure 4. Flue gas cooler and jacket cooler supply 36% of the total CHP production.

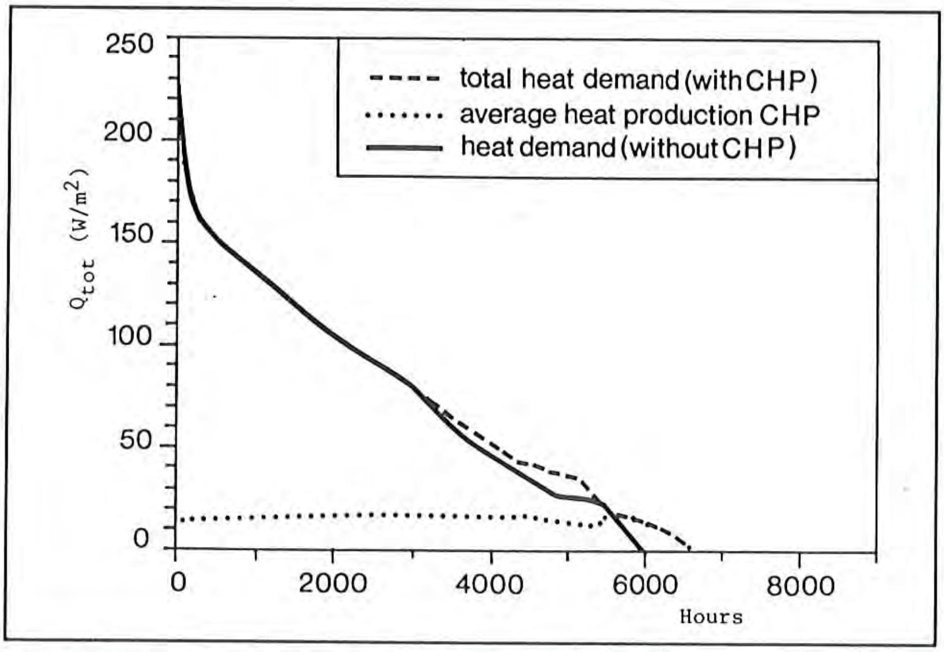

Figure 8. Annual heat load duration curves for the situation with and without compressor heat pump.

In the statistical analysis of the COP in first instance only full-load operation is considered on an hourly basis. The regression analysis showed that the wind direction has practically no influence on the COP although it should be mentioned that relatively little data were considered. The analysis also showed that the influence of the sun on the COP was negligible as was the influence of the frost layer thickness on the evaporator tubing.

The regression analysis shows a large influence of ambient temperature T_o (°C) and wind velocity v_w (m/s) on the hourly COP values and a negative influence of increasing water supply temperature T_w (°C).

On a daily basis, where also the part load behavior and start/stop losses are included, the statistical analysis led to the following equation:

$$COP_d = 0.0235*T_o + 0.0249*v_w + 1.527$$

showing the influence of water supply temperature is now missing. Figure 5 shows this regression.

Results - simulation study

Figure 6 shows some results from the validation exercises performed with the extensive CHP model. The agreement between measurements and calculations for various periods in the monitored year can be shown to be good.[5] This is also the case for the simplified heat pump model. Figure 7 shows some validation results for the greenhouse model. The hourly heat demand is well simulated as is the daily and weekly demand.

In the numerical evaluation of the project the hourly climatological data of the EC-Test Reference Year (TRY) for de Bilt-Netherlands was used.

Table 2 gives the results for the calculated reference situation, analogous to the situation in practice. The natural gas savings, as compared to the situation where all the heat is supplied by the boiler is about 5.2m^3 per m^2 greenhouse area.

The thermal capacity control of the heat pump, which guarantees priority for the CHP, was described before as was the fact that CO_2-fertilization and minimum pipe temperature always have priority when needed. As a check on the CHP-control a simulation was performed with and without the presence of a heat pump. Figure 8 gives the annual heat load duration curves for both situations, which shows that the CHP control strategy provides a certain amount of surplus heat. A simulation where the CHP control is achieved only by controlling the number of revolutions (thus without disconnecting cylinders) shows that although the savings are lower than the reference situation (namely 4.84 m^3/m^2), less surplus heat is produced.

Conclusions

- The realized heat pump has proven to be reliable and the efficiency is high. The applied concept is thus successful.
- Freon-leakage is a major problem.
- The heat pump supplies 15 to 20% of the total heat demand while the capacity is 7% of the total capacity.
- The COP of the CHP is hardly dependent on wind direction, frost layer thickness or solar radiation. The dependence on ambient temperature, wind velocity and water supply temperature is deduced from the regression analysis of the measured data.
- From validation calculations it can be concluded that the computer model of the CHP as well as the model of the greenhouse give a good representation of the practical situation.
- The applied capacity control of the heat pump appears to cause surplus energy production, besides priority.

References

1. Huijs, J.P.G. and P. Kaspers. "Heat/electricity-ratio in greenhouse horticulture." (in Dutch) Institute of Agricultural Engineering (IMAG), Report Nr. 211, Wageningen, June 1985.
2. Krekel, van der Woerd, Wouterse. "Market research on the applicability of small-scale cogeneration systems and heat pumps in non-industrial sectors." (in Dutch) Management consultants, Rotterdam, March 1984.
3. Nieuwkerk, J.P. van. "Energy in the Dutch greenhouse horticulture." (in Dutch) NOVEM, Utrecht, May 1988.
4. Dalhuijsen, A.J. and H.P. Oversloot. "Delfland greenhouse-heat pump project, phase 1." (in Dutch) TNO Institute of Applied Physics, Final report no. 614.006, Delft, April 1988.
5. Liem, S.H., et al. "Delfland heat pump project-phase 1." (in Dutch) Delft University of Technology, Final report no. K-130, June 1988.

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Bibliographic Review

Heat pumps in steam plant cycles. Parts 1 and 2. Robb, G.A., C.A. Robb, G. Arcand. Ottawa, Ont. Canada. May 1988.

This paper is a review of the experience with a heat recovery system at a Quebec paper mill, and a discussion of other applications, trends and developments in heat pump technology. After preliminary studies, CIP Inc. installed a $1 million heat recovery system at its plant in Gatineau, Quebec. Surplus white water from various locations in the mill is pumped in cascades to one central location from which it is pumped to sewer after passing through the heat recovery system. After a start-up and break-in period, and after solving corrosion problems, stable and reliable system operation was achieved. Net energy savings were 378 GJ/d (September 1986) and 566 GJ/d (December 1986). In the light of this plant's experience it was felt that in any waste heat recovery system, most of the heat should be extracted from the source with heat exchangers and a heat pump can be used mostly to facilitate the reinsertion of the energy in the process by increasing temperatures. Firms with a great deal of industrial process experience are offering industrial heat pumps. While there are heat pump applications throughout pulp and paper mills, this paper focuses on steam cycle applications: heat boiler feed-water make-up; staged feed-water heating, and combustion air preheating. Heat sources, location of heat pump in relation to heat exchangers, source temperature limitation, microprocessor control, and the cogeneration effect are discussed. It is seen that new developments such as steam generating R114 heat pumps and better centrifugal steam compressors will make heat pumping in the pulp and paper industry more attractive. Experience to date is with modified chillers. Units designed specifically for industrial applications should provide better performance.

Seven demonstration projects with heat pumps in industry. Bouma, J.W. Nederlandse Energie Ontwikkelings Maatschappij (NEOM), Sittard, Netherlands. Nov 1987, v. 42(11), p. 52-59.

With NEOM's support seven demonstration projects with heat pumps integrated in industrial processes are realized in the Netherlands. Applications are: a closed compression heat pump in a malt house, a heat transformer in a chemical factory, and mechanical vapor recompression installations in 5 different chemical and food factories. These projects are described and experience is discussed and evaluated. Energy prices are important for the estimations of profit and efficiency, but it is likely that, even with low energy prices, the use of heat pumps is often economically justified.

Mechanical vapor recompression in the food industry. Becker, F.E., A.E. Ruggles, D.S. Severson. Proceedings of the 1986 international gas research conference. Rockville, MD: Government Institutes Inc., 1986, pp. 949-964.

This paper discusses the application of mechanical vapor recompression as applied to evaporation processes in the food industry; in particular, in connection with the rendering of animal byproducts. The concept is based upon recycling normally exhausted contaminated steam evaporated from the raw feed and recompressing it for use as process steam to further evaporate additional water. In so doing, energy savings of up to 70 percent can be expected compared to conventional single-effect evaporators. To demonstrate this concept, a gas-powered reciprocating engine driving a modified positive displacement rotary screw compressor was integrated into a specially adapted cooker and operated in an industrial facility. Engine exhaust heat was also recovered to enhance the system performance. Testing of the system demonstrated energy savings of nearly 60 percent as well as increased product discharge temperature uniformity.

Experience with heat pump systems for energy saving in distillation columns. Meili, A. Distillation and absorption 1987. 2 Vols. Proceedings of a symposium held at Brighton (UK), 7-9 September 1987.

Conventional distillation columns are very large energy consumers. In spite of that fact, the breakthrough of energy saving operating methods, such as heat pumps, has only started to take place recently. C3-splitters, Ethylene Styrene Separation and EDC-Purification are only a few examples where direct vapor recompression is an extremely economical solution to save energy. This is especially true whenever the distillation columns are equipped with regular packings which have a low pressure drop.

The following papers appear in the proceedings of the **XVIIth International Congress of Refrigeration, Volume E, Air Conditioning and Energy Recovery**, Vienna 1987. Edited by Austrian Association of Refrigeration and Air Conditioning for the International Institute of Refrigeration 177, bd. Malesherbes, F-75015 Paris, France. All papers are in English unless otherwise noted.

Long-term aspects of heat pump research. Moser, F. Technical University of Graz, Austria, pp. 201-203.

The present situation of low energy prices is not favorable to any energy saving efforts including heat pump research. Nevertheless, economically justified targets for such research are developed and compared with such for other countries and their realization potential is discussed. The following questions are dealt with in this paper: (1) Can we justify research into heat pumps in times where energy prices are at the current low level? (2) What contribution could the large scale application of heat pumps bring for the environmental situation? (3) Can research to improve heat pumps be justified economically? To what extent can we expect to improve the economics of heat pumps by improving their technical performance? What kind of technological targets are economically justifiable? (4) What is the present state of the art and can we expect to achieve these technological targets?

Experiences with heat pump timber driers. Cox-Smith, I.R., P.G. Baines, and C.G. Carrington. Department of Physics, University of Otago, Dunedin, New Zealand, pp. 207-212.

A number of heat pump timber driers have been monitored while in normal commercial operation, some for up to 12 months, as part of an investigation funded by the New Zealand Forest Service and the New Zealand Ministry of Energy. The purpose of the study is twofold. First, specific assistance is given to operators to enable them to use the driers more effectively. Second, the data is required to validate simulation models of the plant, both static and dynamic. These in turn are being used in connection with control and design optimization problems. The model calculations, which are in substantial agreement with measurements, demonstrate how sensitive drier efficiency is to kiln conditions. Comparison of the operating conditions of the two driers illustrates the need for operators to be well informed of the importance of good drier control. The high fraction of the electrical power used by the fans in heat pump driers suggests that reductions may be possible in fan usage.

Energy saving in drying process with closed circulating heat pump. Wang Hong, Beijing Institute of Chemical Technology, and Liang Minghan, Tianjin University, pp. 213-218.

In chemical, light, and food industries, before products are tranported and stored, usually the moisture and other liquids must be removed from the solid materials. Drying is usually a requisite operation in those industrial processes.

A study on energy in drying operations is a very important subject because product drying is one of the most energy-intensive operations in industry. An analysis of a closed circulating heat pump drying unit using R142 as the working fluid is described in this paper. An exergy analysis is carried out to calculate the exergy efficiency. The coefficient of performance of the unit is also calculated. Through experiments in drying clay it was found that the energy consumption of the heat pump is 20-60% lower than direct electric drying.

A gas-driven heat pump applied to grain drying and chilling. Elefsen, F., Refrigeration Laboratory, Technical University of Denmark, pp. 220-226.

During the last 10-15 years the use of heat pumps in the process industry has increased radically, mainly because of the energy savings obtained. In the drying industry heat pumps have not been used widely yet because of the high drying temperatures. In grain drying, the drying temperatures are as low as 50-100°C on account of the product. In this field heat pumps are suitable, but with the short operating periods involved, the investment has until now been considered too high. Most grain is combined at a moisture content of 19-21%. Unless precautions are taken, mold growth will spoil the grain at these moistures. Normally, drying is used as preservation, but by means of the heat pump, chilling can be used as preservation. Today the most commonly used large grain drying plants are continuous flow dryers. Many of these plants use heavy fuel oil with a high content of sulphur. The sulphur is directly exhausted through the grain into the atmosphere. This pollution will be almost totally removed by using a natural gas powered heat pump, because of the low content of sulphur in the gas and the reduction of fuel consumption. By recycling a part of the exhaust air the emission of dust will also be decreased. A steady state computer model of a grain drying and chilling plant with a heat recovery system and heat pump was developed. Simulations were carried out under different operating conditions to determine the thermal performance factor as a function of capacity and energy savings ratio.

An industrial application of non-azeotropic mixture. Blaise, Jean-Claude, et al, pp. 256-261 (French).

Many controls, simulations and tests have been realized in laboratories or on test rigs on the subject of non-azeotropic mixtures for heat pumps. Consequently, Electricite de France (EDF), L'Institut Francais du Petrole (IFP), and Company QUIRI have decided to build a unit on an industrial site, using all their own top technologies to optimize the operation and the control: electronic expansion valves with memorization of saturation curves, binary and tertiary mixtures, coiled exchangers, and a control system named VIGILE. Such a test in an industrial environment is absolutely necessary to measure the reliability of this new technology in real usage rather than on test rigs. Even if similar units already exist, mainly in the field of domestic heat pumps, they are always small-size units built with hermetic compressors and very compact circuits. However, it is well known that the maintenance of larger units with "industrial" compressors is far more complex mainly in relation to refrigerant leakage. This paper describes how tests in real size have already shown that the problem of controlling the refrigerant fluid load with a mixture is not more difficult than with a standard pure fluid.

Development of compressors for mechanical vapor compression systems. Banquet, F., B. Deguerce, and C. Missirian, Electricite de France, pp. 297-302 (French).

The paper describes development of compressors for mechanical vapor compression made by Electricite de France in collaboration with different manufacturers. The aim of these developments is to promote a rational use of electricity in industry. Research and development work was undertaken to make available for steam compression volumetric compressors already used in industry for air compression. Centrifugal steam compressors were developed for higher rotational speed. New concepts are now developed to ensure a satisfactory behavior of material for corrosion, to replace gear boxes and to simplify sealings.

Experimental energy-saving heat pump installation in industry. Vighi, Francisco, et al, Department of Thermal Engineering, Madrid Polytechnic University College of Industrial Engineering, pp. 308-313 (French).

In 1985, IDAE-ADAE commissioned the Department of Thermal Engineering of the Madrid Polytechnic University College of Industrial Engineering to carry out studies on the use of heat pumps in energy-saving applications in industrial processes. Of the different cases in which the heat pump was applied in industrial processes, the results obtained in the aluminum anodizing company Metal-Air, S.L., located in Valladolid, are worthy of special mention. In this particular application a water/water heat pump was used as a replacement for the anodizing bath cooling-water refrigeration plant and the plant used for generating hot water for heating the de-greasing baths. Throughout 1985, and six months of 1986, working conditions have been controlled by means of an automatic data acquisition and control system. Study of the results obtained and comparison of these results with previous electrical energy and liquid fuel consumption levels has permitted the energy savings achieved and its impact on production costs to be determined. The details of the installation, measurements taken during the 18 months of the test and final results are the subject of this report.

Industrial heat pump manual. Technical and applications resource guide for electric utilities. Linnhoff March, Inc., Leesburg, Virginia, USA, October 1988.

When properly applied, industrial process heat pumps offer process owners the opportunity to achieve significant energy cost and efficiency improvements. This resource guide presents the basic technical features of heat pumps and describes the differences between industrial process (or more simply, industrial) heat pumps and commercial units. The current state of the industrial heat pump market is reviewed and analyzed. Industrial heat pumps are not the only energy saving technology available; there are many "competitors" including passive heat recovery, process modifications, and cogeneration. The way in which the industrial heat pump interacts and competes with these other alternatives is discussed in detail. Pinch Technology is introduced as the best method for finding and assessing all industrial heat and power options. A comprehensive industrial heat pump design methodology is presented. It can be used in industrial plants to identify heat pump opportunities and to ensure good design and operation. This methodology is based on practical experience and will help those interested in promoting and using industrial heat pumps benefit from the last decade of heat pump development. NOTE: To obtain a copy of this report, contact the Research Reports Center (RRC), Box 50490, Palo Alto, CA 94303, telephone 01-415-965-4081. There is no charge for reports requested by EPRI member utilities and affiliates, U.S. utility associations, U.S. government agencies (federal, state, and local), media, and international organizations with which EPRI has an information exchange agreement.

Heat pumps: industrial applications. April 1981-December 1987 (citations from the National Technical Information Service data base, USA).

This bibliography contains citations concerning design, development, and applications of heat pumps for industrial processes. Included are thermal energy exchanges based on air-to-air, ground-coupled, air-to-water, and water-to-water systems. Specific applications include industrial process heat, drying, district heating, and waste processing plants. Other published searches in this series cover heat-pump technology and economics, and heat pumps for residential and commercial applications.

Heat pumps: residential and commercial applications. April 1981-December 1987 (citations from the National Technical Information Service data base, USA).

This bibliography contains citations concerning design, development, and applications of heat pumps for residential houses and apartments, and commercial applications, such as office buildings and greenhouses. Heat-pump energy exchanges are based on air-to-air, ground-coupled, air-to-water, and water-to-water systems. The citations cover costs and reliability of the heat-pump systems, and studies of operations in differing climates and seasons.

Heat pumps: technology and economics. April 1981-December 1987 (citations from the National Technical Information Service data base, USA).

This bibliography contains citations concerning the technology and evaluation of heat pump systems and components. Evaluation includes coefficient of performance and economics of various types of heat pumps and energy-exchange media for varying climatic or seasonal conditions. Solid state heat pumps are part of the new technology referenced in the citations. Heat pump energy exchanges are based on air-to-air, ground-coupled, air-to-water, and water-to-water systems.

Energy answers '87: questions and answers on energy conservation and management in buildings and the built environment. Swedish Council for Building Research, Report G17:1988.

This report, published and sponsored in part by the Swedish Council for Building Research, provides answers to questions concerning energy conservation and energy management in the built environment. Emphasis is on the residential sector which in Sweden has received the most attention for energy conservation efforts. The authors have presented various important elements in the present state of knowledge. Some of the questions addressed include: How far can energy conservation be taken? Does energy taxation encourage conservation? How can energy be saved in offices, schools, hospitals, etc.? Heat pumps, do they pay? The answers to each question are short, concise, and backed up by actual experience and research work. References are given at the end of each article for further reading. For additional information, contact the Swedish Council for Building Research, St. Goeransgatan 66, S-11233 Stockholm, Sweden.

Proceedings of the 2nd DOE/ORNL heat pump conference: research and development on heat pumps for space conditioning applications. Prepared by the Oak Ridge National Laboratory, Oak Ridge, Tennessee, USA, August 1988.

This conference was planned to provide information on current R&D activities on residential and commercial heat pumps in the U.S. Department of Energy (DOE) and Oak Ridge National Laboratory (ORNL) Building Equipment Research (BER) Program. It was primarily for the benefit of HVAC equipment manufacturers who could use the technology in future products, but was also of value to a wider audience, including utilities, independent R&D organizations, universities, other government groups, and research funding and management organizations. Because this conference was intended to be a significant part of technology transfer efforts, the presentations emphasized energy performance objectives, technical results, development status, and potential HVAC equipment applications. In addition, the conference was organized to facilitate information exchange among attendees, speakers, and the DOE and ORNL Program staff, and to create an opportunity to receive constructive input from the private sector. The technical presentations were grouped into two principal subject areas: electric heat pump technology and thermally activated heat pump systems. Electric heat pump topics included capacity modulation experiments and system design analysis development, non-azeotropic refrigerant mixtures, and ground-coupled systems. In the area of thermally activated heat pumps, presentations centered on the development of absorption systems, engine-driven heat pumps, and Stirling engine-driven heat pumps. For a complete listing of the papers in these proceedings or for information about obtaining a copy of these proceedings, contact the Heat Pump Center (see address on back cover).

News Briefs

IEA Heat Pump Center Discusses the Future Role of Heat Pumps

On January 23 and 24, 1989, more than 40 representatives from industry, research institutions, and governments of twelve IEA/OECD countries discussed how to strengthen the IEA Heat Pump Center to better support heat pump applications. This international workshop was organized by the IEA Heat Pump Center and took place in Mainz, Federal Republic of Germany.

The heat pump, an end-use energy technology able to protect the environment and reduce primary fossil energy consumption, presently has a large market in the USA, Canada, and Japan. Due to several reasons, particularly economic ones, the heat pump boom of the late 1970s and early 1980s in many European countries declined to special applications only.

The workshop analyzed the different situations in individual countries and underlined the environmental advantages, in particular the inevitable role of heat pumps in reducing the CO_2 (greenhouse) effect. It expressed unanimous support for the development of new working fluids and technologies to cope with the CFC problem.

The workshop identified the major directions of the Heat Pump Center's work which are: to foster well-defined international cooperation on research, development, and demonstration; to coordinate promotional activities in the various countries; and to activate governmental and utility support for increased heat pump use. In all of these activities, industry should play an important role.

The new program of the Heat Pump Center and its close relation with experts representing various sectors of heat pump activities in member countries will create an international forum for directing the already available heat pump technologies and ongoing developments into broad environmentally sound and economically acceptable applications.

Modern Absorption Technology at the Technical University of Munich

Research and development of modern absorption technology for energy conservation and protection of the environment is the major goal of the Institute for Technische Physik (Director Prof. Dr. G. Alefeld) at the Technical University of Munich. On November 11, 1988, the status of work at the Institute was presented to the interested public. About 250 participants from industry, utilities, universities, and research institutions took part in this seminar, and about 150 were admitted for visiting the laboratories of the institute.

The goal of the one-day open house was to show the many possible applications of absorption technology which are being investigated. After a briefing on absorption technology and the actual research program by Professor Alefeld, the following experimental and demonstration plants were presented.

Heat transformer

In addition to an 8kW laboratory model, a 100kW multistage heat transformer is being investigated which can take 30% of a waste heat stream at 80°C and upgrade it to a temperature of 140-150°C. The remaining heat is rejected via a cooling tower at a temperature of 30°C. Industrial processes where the temperature of the waste heat stream is 30-60°C higher than ambient temperature are candidates for the application of heat transformers, such is often the case in paper or wood drying. Other possible application areas for heat transformers are chemical drying processes, alcohol distillation, production of condensed milk or juice, industrial waste water concentration, or other energy intensive processes such as the injection molding of plastics. Another promising application of heat transformers are district heating systems.

Double effect/double lift absorption heat pump

In comparison to other working fluid pairs, water-lithium bromide offers the advantage of higher COPs but the temperature lift possible using this working fluid pair is limited to approximately 30°C in single-stage units.

The development goal of the Institute is to improve the temperature lift or the COP by using a multiple-stage unit. An experimental unit was built to test these possibilities. In addition to the four main components that normally comprise a single-stage absorption heat pump, the high COP two-stage unit has an additional condenser and generator operating at a higher pressure level while the two-stage unit designed for high temperature lifting has an extra evaporator and absorber. The latter unit can supply hot water at temperatures up to 100°C. An optimal heat pump system has been designed which can be switched from high COP operation to high temperature lift operation. This type of heat pump is useful for cases where cooling and heating requirements must be met simultaneously, as, for example, in the air conditioning of buildings and in refrigeration combined with hot water production. The two-stage unit can be operated over the range from maximum refrigeration capacity to maximum heating capacity.

Heat pump transformer

As a result of detailed investigations concerning specific applications of absorption technology, for example, in breweries and in sea water desalination plants, it was determined that neither heat transformers nor absorption heat pumps separately presented an optimal solution.

A combination of an absorption heat pump with a heat transformer (a so-called "heat pump transformer") was developed and tested at the Institute. The heat pump transformer is driven with 6-8 bar steam and is cooled by 25-30°C chilled water. At the intermediate temperature level (100°C) the unit can deliver approximately 4.2 to 4.5 times as much heat as is input as drive energy. The unit also uses low temperature waste heat (80°C) as an additional drive source upgrading a portion of it for use at the higher temperature level. The temperature levels at which heat is delivered by the heat pump transformer are adjustable.

Hybrid compression absorption system

In many industrial plants, both heating and cooling is required. A compression absorption system was shown which could provide refrigeration at -20°C and hot water or even steam at the higher temperature end. This hybrid unit is basically a two-stage refrigeration system where the high temperature stage is replaced by an absorption cycle.

The main advantage of this hybrid system is that only one compressor is required but with a larger displacement. Yet a solution pump and an extra heat exchanger in the solution circuit are needed. The high temperature stage can be used for replacement of R114. The cycle can be operated using ammonia-water as the working pair and can achieve heat output at a temperature level of 100°C. An experimental unit (10kW) has been constructed at the Institute.

Compression absorption refrigerator using ammonia-water

A compression absorption refrigerator was developed. This unit combines a compression cycle with an absorption cycle. The purpose of the absorption cycle is to improve the performance of the compression cycle. This is achieved by utilizing the superheat from the compressor to drive the generator of the absorption cycle and by absorbing the refrigerant vapor at an intermediate pressure level after throttling. In this way the performance of a standard electric motor-driven compression cycle can be improved by 5-15%. In the case of an engine-driven compressor the performance improvement can be 25-50%.

Zeolith/water heat pump, heat storage and heat transformer

This system uses the sorption principle with zeolith as the sorbent and water as the working fluid. One of the main advantages of this design compared to conventional electric powered heat stores used in Germany is the system's heat pumping capability.

To unload the heat store, steam is added to the zeolith. This steam is generated at low pressures (low temperature) using heat available from the environment (solar, ambient air, ground). The sorption reaction which takes place between the zeolith and the water (steam) is exothermic. The heat generated is available at temperatures up to 90°C and can be used for space heating. The heat absorbed from the environment to generate the steam is in this way pumped to a higher temperature level.

To recharge the store, heat is added at a temperature of 250°C to drive out the water vapor from the zeolith. This process can be carried out using off-peak electricity, oil or gas burners, or even heat from a district heating system during low use periods. In this manner the energy can be stored for long periods of time with minimal losses.

This technology is presently being tested in industrial applications.

The 2nd International Workshop on Research Activities on Advanced Heat Pumps

Three institutes of the Graz University of Technology work together in the development of advanced heat pumps. The fields of their research work include the finding of new working media to replace CFCs, the improvement of mechanical equipment, and the integration and better use of heat pumps in industrial and residential/commercial installations.

At the beginning of the project in 1986, a first workshop on advanced heat pumps was held in Graz, Austria. It yielded fruitful ideas to the participants for further work. Therefore it was decided to organize a second workshop two years later.

From September 26 to 29, 1988, nearly 100 chemical, mechanical, and refrigeration engineers from 19 countries in Europe, America, Africa, and Asia met at the Institut fuer Verfahrenstechnik, Abteilung fuer Grundlagen der Verfahrenstechnik, Graz, to present the results and conclusions of their research programs. To give enough time for specific discussions and detailed information a special form of organization was chosen.

First, there were so-called "poster presentations." Every speaker prepared a poster and gave a 10-minute overview of his work. This gave the opportunity to get a comprehensive overview of the activities of the participating scientists. After these presentations there was enough time for each member of the audience to select points of special interest and discuss them with the authors.

Each day focused on the problems of a special topic: the first day was concerned with the problems of the absorption heat pump, the second with the compression heat pump, the third with process integration. On the last day, six specialists summarized the presentations and compared them with the world-wide standard. Additionally, two roundtable discussions were held on the issues of research needs for advanced heat pump systems and of the CFC replacement in heat pumps.

The following presents a short summary of the results of the workshop.

CFC problem

The impact of CFCs on the ozone layer calls for a reduction of the emission of CFCs to the atmosphere. There are three possible ways:

-- Careful handling and recycling of refrigerants, careful maintenance of units
-- New refrigerants to replace CFCs
-- New systems

The experts agreed that the emission of CFCs can be reduced by one third through more careful handling. But the prices of the recycled CFCs might be ten times higher than the actual prices at present.

New refrigerants with less harmful environmental impact are under development in several institutes and companies. One can also expect that water and ammonia will again be used in heat pump systems on a larger scale. The discussion of possible substitutes for CFCs led to the following table of possible media for the different working temperature ranges:

-100°C to -60°C	hydrocarbons
-60°C to +30°C	ammonia
+30°C to +120°C	new media (e.g., ethyl chloride)
+120°C to +180°C	water

In this context, the exact determination of thermophysical data is of growing importance.

Another alternative is to develop new systems. Absorption cycles offer many possibilities to circumvent the CFC problem.

Other points concerning compression heat pumps

Though the main research efforts aim at the replacement of the CFCs as working media, new working fluids are also sought for widening the application field of heat pumps in the high-temperature range.

Greater acceptance of the compression heat pump requires:

- Suitable application range
- High durability and lifetime
- High energy efficiency
- Low volume and small size
- Suitable capacity control

Much research work is being done to improve compressors. Progress is made in minimizing leakages and friction losses, working at higher speed and in speed capacity control. Non-azeotropic refrigerant mixtures will also be used for capacity control purposes and as replacements, especially for R12.

Small engine-driven heat pumps are under development.

Absorption heat pumps

In the field of systems with a heating or cooling capacity from 70 to 350 kW or more, the absorption heat pump will replace the compression heat pump. This is especially valid for the refrigeration and air conditioning sector.

It was reported that in Japan, 85% of the cooling systems over 350 kW are systems with an absorption heat pump. In Japan there is also great progress in directly fired absorption heat pumps.

Two main fields of research work on absorption heat pumps or heat transformers were clearly pointed out. First was the development of new working pairs for greater temperature boosts and a higher working temperature. For this aspect a short-term development is the improvement of existing working pairs, like water-LiBr. For the water-LiBr system corrosion inhibitors and viscosity-reducing agents are necessary. The second way of improving the performance of absorption heat pumps and absorption heat transformers is to introduce multistage cycles.

Modelling and simulation

About one third of the presentations were concerned with the problem of modelling and simulation. The explanation for this great number of theoretical studies is that this was a workshop on research activities. One can summarize these presentations with three slogans:

- From ideality to non-ideality
- From thermodynamics to kinetics
- From black boxes to phenomena

The advantage of computer models is a better knowledge of the process and therefore an optimized design.

Process integration

Another important aspect of the workshop was the process integration of heat pumps. For this topic, some principles have been elaborated very clearly:

- Heat pumps are only one technology to improve utilization of energy
- Heat pumps must be regarded as a part of a greater system
- Optimization of the heat pump does not necessarily lead to the optimization of the whole system

Heat pumps have to compete not only with oil or gas-burning systems, but also with other technologies for heat recovery. The best heat pump becomes uneconomical if integrated in the wrong way.

Therefore, there are two ways of process integration:

- Adapt the heat pumps to process needs, or

- Adapt processes to heat pump possibilities

Many papers treated this problem by showing different methods of calculating the most economical way of integrating heat pumps into industrial processes. Most of the papers concerning this topic dealt with computer-supported calculation methods for integrating heat pumps into existing or new plants.

Another result was that one should not regard heat pumps as heat recovery devices with a payback period but as a heat supply system like a heat or power station that has to provide heat as cheaply as possible.

The 2nd International Workshop on Advanced Heat Pumps held in Graz provided a platform where the participants could find a common language between chemical, mechanical, and refrigeration engineers.

A third workshop will be held in September 1990. For more information, please contact the Institut fuer Verfahrenstechnik, Abteilung fuer Grundlagen der Verfahrenstechnik (Prof. Dr. F. Moser) in Graz, Austria. The proceedings of the second workshop may be obtained there as well. They include all presentations, speeches, reports, and roundtable discussions.

J. Fresner and H. Schoeffmann, Institut fuer Verfahrenstechnik, Graz, University of Technology, Austria.

Schedule of Conferences

April 17-21, 1989

Hobart (Australia); **Federal Conference and Exhibition of the Australian Institute of Refrigeration, Air Conditioning and Heating (AIRAH).** Sponsored by AIRAH. Contact: Australian Institute of Refrig., Air Conditioning and Heating (Inc.), P.O. Box 1533R, G.P.O., Hobart, Tasmania 7001, Australia.

May 1989

London (UK); **Industrial Energy Management Conference.** Sponsored by the Institute of Energy, London. Contact: Institute of Energy, Conferences Dept., 18 Devonshire St., London W1N 2AU, UK.

May 1989

Prague (Czechoslovakia); **Symposium on the Optimum Use of Primary Energy Resources in Final Heat Consumption.** Sponsored by the United Nations, Geneva, Switzerland, and the Economic Commission for Europe. Contact: UN ECE Energy Division, Palais des Nations, Attn: F. Romig, CH-1212 Geneva 10, Switzerland.

May 28-29, 1989

Sweden; **2nd Working Meeting of IEA Annex XIV "Working Fluids and Transport Phenomena in Advanced Absorption Heat Pumps."** Contact: 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.

June 7-8, 1989

Essen (Fed. Rep. of Germany); **Tagung ueber Blockheizkraftwerke und Waermepumpen: Betriebserfahrungen unter aktuellen Umweltschutz- und Wirtschaftlichkeitsaspekten.** Sponsored by Verein Deutscher Ingenieure (VDI) - Gesellschaft Energietechnik, Duesseldorf (Germany, F.R.) Contact: Verein Deutscher Ingenieure, Abt. Tagungen, Graf-Recke-Str. 84, Postfach 1139, 4000 Duesseldorf 1, FR Germany.

June 24-28, 1989

Vancouver (Canada); **1989 ASHRAE Annual Meeting.** Sponsored by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE). Contact: ASHRAE International Headquarters, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA, telephone 01-404-636-8400, telex 705343.

July 5-7, 1989

Bristol (UK); **Meeting on the Impact of Electrotechnology on the Refrigeration/Heat Pump Industries (IMPEL).** Contact: University of Bristol, Faculty of Engineering, ATTN: Dr. R.T. Moses, Queen's Bldg., Clifton, Bristol BS8 1TR, UK.

August 6-9, 1989

Philadelphia, Pennsylvania (USA); **National Heat Transfer Conference and Exhibition.** Contact: American Society of Mechanical Engineers, United Engineering Center, 345 East 47th St., New York, NY 10017 USA.

August 27-September 1, 1989

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

September 18-21, 1989

Seoul (Republic of Korea); **International Conference and Exhibition on Energy Sources Management and Energy Saving Technology and Equipment (KORENERGY '89).** Contact: SHK International Services Ltd., 22/F, 151 Gloucester Rd., Hong Kong.

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 13-19, 1989

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

November 22-24, 1989

Hannover (FR Germany); **DKV - Kaelte-Klima - Tagung 1989.** Contact: Deutscher Kaelte- und Kilmatechnischer Verein (DKV), Pfaffenwaldring 10, D-7000 Stuttgart 80, Fed. Rep. of Germany.

March 12-15, 1990

Tokyo (Japan); **The 3rd International Energy Agency Heat Pump Conference.** Contact: 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.