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INTERNATIONAL ENERGY AGENCY

IMPLEMENTING AGREEMENT ADVANCED HEAT PUMPS

ANNEX IX - HIGH TEMPERATURE INDUSTRIAL HEAT PUMPS

HPC-copy

FINAL REPORT

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SUMMARY

In this report the state of the art of high temperature industrial heat pumps is described. Overviews are given of the different types of heat pumps on the market including where possible listings of manufacturers. A detailed description is also given of the R & D efforts going on in the field.

It is found from these overviews that open cycle compression heat pumps are able to reach temperature levels of up to 200 °C. The closed cycle compression systems in actual use are limited to temperatures of some 130 °C. For sorption machines in actual use only the heat transformer is able to reach temperatures significantly higher than 100 °C. However here also the temperature limit presently is located at some 135 °C.

The economics of high temperature heat pumps is investigated. It is found that under the present conditions of fuel oil costs heat pumps can be applied in only a few countries where the cost of electricity to drive the heat pumps is extremely low (e.g. Norway). The run time of heat pumps must be as high as possible and must typically be of the order of at least 6.000 hours per annum in order to give rise to acceptable payback periods.

The report concludes with detailed case studies in which the energetic and economic performance of existing heat pump systems are analysed. In general it is found that the systems perform according to expectations except for the economics. Several systems built a few years ago would be deemed uneconomical under the present cost conditions.

PART II : ECONOMICS

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PART I

INTRODUCTION

This report contains the results of a study executed by a team of researchers of the Katholieke Universiteit of Leuven, Belgium, concerning high temperature industrial heat pumps. This study constitutes the main activity undertaken under Annex IX : High Temperature Industrial Heat Pumps of the Implementing Agreement on Advanced Heat Pump Systems of the International Energy Agency.

The countries that participated in this Annex are : Belgium (operating agent), Japan, United States, Switzerland, Finland, Netherlands, Federal Republic of Germany and Sweden. The study was initiated in 1986 and terminated in 1989.

The present report consists of a report produced by the operating agent describing the state of the art, R and D and the economics of high temperature industrial heat pumps. In addition it contains a number of case studies produced by the participants in Annex IX which provide insights in the actual use of these heat pumps. It is hoped that the report provides a better insight into the problems which high temperature energy conservation systems face, where one stands at this moment and where future R & D efforts should be directed to.

1. OPEN CYCLE COMPRESSION HEAT PUMPS

1.1. OVERVIEW OF MANUFACTURERS

The state of the art of vapor recompression systems in Europe was investigated by J.F.Reynaud (2). He made a comparison between mechanical compressors and steam ejectors.
The most important disadvantages of the steam ejector are :
- consumption of a significant amount of expensive energy
- high noise level
- limited pressure range of induced vapor. Mechanical Vapour Recompression

Therefore the ejector can be used in units equipped with M.V.R. as stand-by device or to reduce make-up steam consumption. The most used mechanical compressors are centrifugal compressors while the largest plants are equipped with axial compressors. A few plants are equipped with volumetric compressors (piston non-lubricated compressors, liquid ring compressors). There should be roughly 1500 to 2000 MVR plants throughout the world and most existing applications can be found in :
- concentration and crystallization by evaporation
- distillation
- drying
- thermal effluents valorization

Recently, an investigation of MVR plants was undertaken by the International Union for Electroheat (5) from which one can conclude that :
- two thirds of the plants were installed after 1979
- the food industry and the chemical industry consume 40 % of the total compressor power
- the number of plants can be classified as follows :
    - 80 % for concentration by evaporation
    - < 2 % for crystallization by evaporation
    - 5 % for distillation
    - < 1 % for drying
    - 8 to 10 % for thermal effluents valorization
- the average consumed power is 610 kW

- the distribution of compressor drives is as follows :
  - 89 % electric motors
  - 8 % steam turbines
  - 3 % combustion engines
- centrifugal compressors are used in 80-85 % of the plants.

Concerning the trends of development in Europe, the number of installed plants is decreasing compared with the increasing trend during 1980-1985. This is due to the lower fuel costs. Still there are some branches of industry where MVR applications look very promising. Continued development can be found in the milk, chemical and to a lesser extent in the pump and paper and sugar industries. According to J.F. Reynaud one can expect 40 to 60 new MVR plants in Europe per annum. This number is dependent on fuel oil prices. According to reference (1) a reduction in investment costs could increase this number to a considerably higher level.

The second part of the report of M. Wakabayashi (1) concerns a common study on the present conditions and future prospects for industrial heat pumps in Japan. Based on declining oil prices, positive appreciation of the yen and a cheaper electricity price, one can calculate that the economic marginal COP has to be at least 5, 12 in 1986 for an electromotor driven heat pump. The COP of the installed system must be larger than 5,12 before being economically acceptable. From this point of view it is necessary to reduce the initial costs and to raise the COP of the heat pump installation at least to 5,5. It is clear from this that the open cycle compression heat pump is superior to the closed cycle type.

In ref. 3 the future prospects for different types of industrial heat pumps in Europe are investigated by T. Berntsson.
He considers Europe as one market because here the technical conditions for industrial heat vary little. The outlook of each industrial branch is almost the same from~~e~~ country to country.
Berntsson concludes that vapor recompression and open-cycle heat pumps have the advantage to achieve a high C.O.P. for a relatively low investment cost. Some problems due to air leakage can occur.

The practical possible evaporation temperature has a lower limit of 80-90 °C for turbo compressors and 90-100 °C for screw compressors. This fact decreases the potentional of steam compressors.

A similar investigation was carried out by Laborelec (5) in Belgium. A statistical review was made concerning the practical application of MVR in industry. One concludes that 70 % of the total installed MVR systems can be found in distillation processes and 30 % in concentration processes. The most used compressor brands are Sulzer and Allis Chalmers which account for 11 and 17 % of the total number of installed compressors. Both brands represent 30 % and 47 % respectively of the total installed power. The main objective of this report was on the one hand to study the energy consumption of distillation, concentration and drying and on the other hand to identify the MVR applications in different industrial branches.

Ref. 6 presents the methodology and results of an investigation of the range of economic applicability of heat pumps to distillation processes. The investigation was performed by Radian Corporation under contract to the Electric Power Research Institute. The first step in the study was to investigate the activities of the most important manufacturers in the U.S.A.
Westinghouse, General Electric, York, Vilter and Sullair are the only companies which supply packaged heat pumps of industrial size. This means that the delivered thermal capacity reaches 300 - 3000 kW. None of their packaged systems had ever been applied to distillation columns. Concerning distillation heat pump systems, York has several taylor made designs in propane/propylene and butane/isobutane splitters in gas plants.
Manufacturers of heat pump components were also contacted.
The Linde Division of Union Carbide supplies an extended surface heat exchanger. One surface of the tube has a porous metallic coating, the other side is fluted. The heat transfer coefficients are four times larger than in a plain surface tube. It is clear that this results in a smaller temperature approach and a reduction of the

compressor power. This special heat exchanger is applied in 7 propane/propylene splitters and 3 butane/isobutane splitters.
Another application, although not used commercially, of an extended surface heat exchanger is developed by NRC, Inc. The tubes have an array of helical wire loops bonded to the inner and outer tube surfaces in order to increase the fluid turbulence.
The most important component of a heat pump is the compressor. Ingersoll-Rand has supplied many centrifugal compressors in different distillation plants where the working fluid is the column overhead vapor. Sulzer Brothers Limited of Switzerland has a lot of experience in the design of heat pumps for vacuum distillation. Their knowledge has been licensed by Koch Engineering in the U.S.A.
Siemens Corporation is under contract to EDF to develop a vapor recompression system. Similar developments have been noted in Japan, West-Germany, Canada and Australia.
Based on available information of existing plants in Canada and the U.S.A., it is estimated that 75 to 200 distillation heat pumps are currently in operation worldwide. Most of them are vapor recompression systems and about a third have extended surface tubing, the rest plain surface tubing.
Two case histories are presented in ref. 6 : a heat pump assisted isobutane/normal butane splitter (ISO-FRAC INC. at Hutchinson, Kansas), and a heat pumped propane/propylen and isobutane/normal butane splitter (ENTERPRICE PRODUCTS at Mon Belvieu, Texas).
The second part of the study (6) concerned the development of an economic evaluation program for heat pump retrofits. This consisted of two independent programs, the PROCESS simulation package for conceptual design and the COST program for cost estimation and life-cycle economic analysis. The evaluation programs were applied to five candidate distillation processes. In this part of the study a parametric representation of the five processes was derived. In this way it was possible to make a sensitivity study of the design factors (reboiler/condensor; plain vs. extended surface tube; centrifugal vs. reciprocating compressor, ...) and economic factors (electricity cost, equipment cost, ...).

Another computer program was developed by Mészaros (7). A design strategy was proposed for selecting the best heat pump assisted distillation column. Three possible configurations were compared in order to find the most economic solution. These three systems were : closed cycle processes, vapor recompression with reboiler and a bottom flashing system. The design strategy was based on energy cost and payback time.
Similar investigations are presented in ref. 8. A design model for heat pump assisted distillation systems is developed here. Analysis of ethanol/water mixtures (9) and a comparative analysis on three alcohol mixtures (10) are presented by other researchers.

In Spain (Dep. de Quimica Tècnica, Univ. of Barcelona) a comparison between a vapor recompression system and a conventional distillation column (propane/propylene splitter) was made, based on a sensitivity study using a computer program. The vapor leaving the column is preheated before compression. This preheating is made possible by heat exchange with the reflux. This reflux (high pressure) is then expanded and sent back to the distillation column. In the conventional system the reflux is cooled by water. After optimisation, results show a decrease in cooling water consumption by 68 % and the total consumption is reduced by 66,5 %. The payback time is 2,1 years.
Another application of direct compression of vapors is also presented in reference 14.

Ref. 11 presents the results of a process synthesis study of heat pumps in evaporation processes by Union Carbide Corporation for the Electric Power Research Institute (EPRI), California. The study of evaporation processes is concentrated on five industrial sectors (chemical and petroleum, dairy, food, pulp and paper, and textiles) and uses process synthesis techniques such as the process pinch. From these five sectors 40 processes (table 1) were selected which are potential candidates for assisted compression heat pumps. Out of this list 10 representative processes are selected for further investigation (marked with "*"). The list of 40 possible application areas for heat pumps was obtained through company visits (table 2).
A review of consultants who supplied flow sheets and data for the evaluation of the application of heat pump technology to candidate processes is listed in table 3.

Table 1

INDUSTRY SEGMENT PROCESSES (6)

Industry	Process
Chemicals and Petroleum	*Fuel Alcohol by Fermentation
	*Potash
	Salt
	Sugar
	Sulfuric Acid and Sulfates
	*Caustic Soda
	Enhanced Oil Well Water Recovery
	Water Desalination
	Waste Water Concentration
	Solvent Recovery for Paints
	Cationic Resin Manufacture
	Heavy Metals Processes
	Ceramics
	Specialty Chemicals
	Phosphoric Acid
	Urea Process
	Ammonium Nitrate Process
Dairy	*Cheese-Whey Powder Process
	*Skim Milk Powder Process
Food	
	*Fruit Juice Concentration
	*Corn Milling/Corn Syrups
	Beer By-product Recovery
	Coffee/Tea Concentration
	Tomato Paste
	Candy Manufacture
	Gelatin and Similar Dry Powders
Pulp and Paper	*Kraft Black Liquor Pulping
	Thermo-mechanical Pulping
	*Chemi-thermo-mechanical Pulping
	Magnified Pulping
	Soda Pulping
	Sulfite Pulping
Textiles	*Nylon Process
	Polyester Process
	Solvent Recovery from Latex
	Dying
	Finishing
	Caustic Recovery

* Indicates process selected for additional study.

Table 2

EVAPORATOR MANUFACTURERS VISITED (6)

Manufacturer	Industry[1] Chemicals and Petroleum	Dairy	Food	Pulp and Paper	Textiles
APV Crepaco, Inc.	Y	Y	Y		
Aqua-Chem, Inc.	Y		Y	Y	
Artisan Industries, Inc.	P		Y		Y
Blaw-Knox Food and Chemical Co.	Y	Y	P	Y	Y
Dedert Corporation	Y		P		
GEA Weigand		P			
Goslin-Birmingham, Inc.	Y			P	
HPD Incorporated	Y			P	
LUWA Corporation	P		Y		Y
Niro Atomizer		P			
Resources Conservation Co.	Y			Y	
Rosenblad Corporation	Y		Y	P	
SACDA[2]			Y	P	
Signal Swenson Division	P	Y	Y		
Unitech Division of the Graver Co.		Y		Y	

Legend: Y indicates Yes, previous experience
P indicates primary industry experience.

Notes: [1]Manufacturers indicated their previous experience in the five industrial segments shown. They may have experience in other industries as well.

[2]Not an evaporator manufacturer. Provides simulation services for evaporation systems.

Table 3

PROCESS INDUSTRY CONSULTANTS (6)

Process	Consultant
Fuel Alcohol by Fermentation	R. A. McKenney Union Carbide Corporation P. O. Box 8361 South Charleston, WV 25303
Chlor-Alkali	T. H. Yohe Signal Swenson Division 15700 Lathrop Avenue Harvey, IL 60426
Potash	S. M. Glasgow Union Carbide Corporation P. O. Box 8361 South Charleston, WV 25303
Nylon-6,6	George C. Colbert and Associates, Ltd. 2315 Jamaica Drive Wilmington, DL 19810
Citrus Juice Concentration	Kenneth L. Holladay Orange Juice Consultant 4762 N. W. 1st Court Plantation, FL 33317
Wet Corn Milling High Fructose Corn Sweeteners	Warner Weiss Equipment Sales Company 10129 Powers Avenue Englewood, CO 80111 Friedrich H. Jahn Starcosa GmbH Am Alten Bahnhof 5 D-3300 Braunschweig Federal Republic of Germany
Cheese-Whey Powder	Frank Wilderspin Food & Dairy Process Consultant 901 South Washington St. Park Ridge, IL 60068
Skim-Milk Powder	Frank Wilderspin See above address
Chemi-Thermo Mechanical Pulp	Eva Sebbas Ekono, Inc. 410 Bellevue Way, S.E. Bellevue, WA 98004
Kraft Black Liquor Pulping Cycle Fine Paper Mill	Eva Sebbas See above address

From the study one may conclude that in general most of the evaporation designs are isolated from the surrounding process and generally overdesigned. The evaporation systems can be improved by integration with the surrounding processes and by utilizing electrical MVR heat pumps. Payback periods of 2 years can be achieved in many cases. Many specific conclusions are drawn from each particular process.

A similar study on evaporation plants was worked out in Sweden (13). Due to moderate and low electricity prices MVR is an interesting alternative to multi-effect evaporation especially for pre-concentration applications in this country.

The economic and practical problems of mechanical steam compressors in the Danish process industry are studied in ref.(21). A comparison between the advantages and disadvantages between the different compressor types is made. It seems that centrifugal compressors are most frequently used for large systems and the roots blowers for smaller installations. The cycloid, the piston and the screw type compressors might be useful in special applications. Traditional markets for steam compressors are the sugar, milk, and salt industry. In these industrial sectors, there is competition from efficient multistage evaporators. Therefore the steam compressor is a better retrofit system in less traditional markets (brewing, distillation, district heating).

An analogous study (22) was made in the U.S.A. where the largest potential applications for vapor compression heat pumps can be found in the food, textile, pulp and paper, and petrochemical industries.

A sensitivity study concerning energy savings in distillation processes, working at amospheric pressure and equipped with open cycle compression heat pumps was made by S. Weiss (14). In this study the energy savings per Kmol overhead vapor are calculated. These savings are based on the reduction of cooling water and steam consumption taking into account the electricity consumed to drive the compressor.
Figures 1a and 1b represent the yearly savings as a function of the pressure ratio (P_2/P_1) with the vapor flux (D^*) and the column diameter as parameters. Figures 2a and 2b represent the payback time as a function of the above mentioned parameters. When one assumes a yearly cost saving of at least 0,6 x.10^6 DM/year, the following maximum values of the pressure ratio are derived :

1.	Steam pressure : 0,35 MPa		P_2/P_1
	column diameter :	1,6 m	< 2,5
		2,4 m	< 4,5
		3,6 m	< 5,5
2.	Steam pressure : 1,7 MPa		
	column diameter :	1,2 m and	
		1,6 m	< 5
		> 2 m	> 6

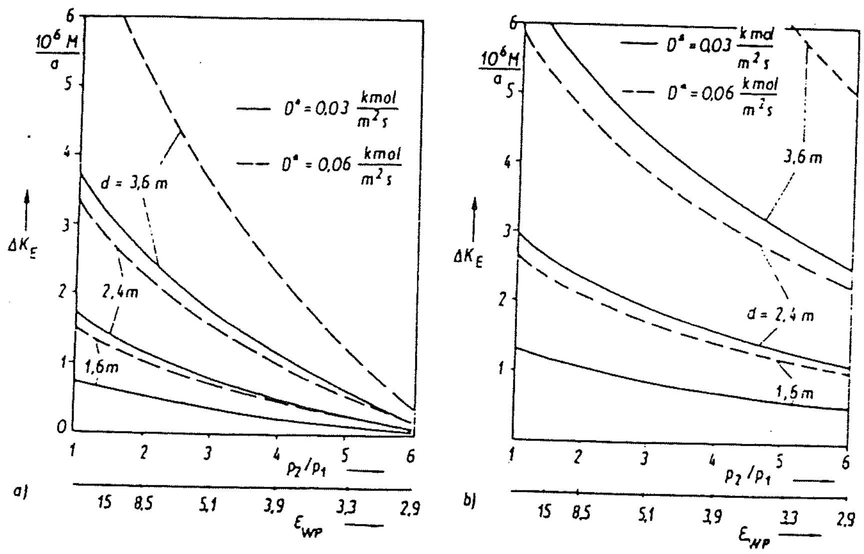

Fig. 1a, 1b : Cost savings per year as a function of the pressure ratio and column diameter (0,35 MPa; 1,7 MPa steam pressure) (14)

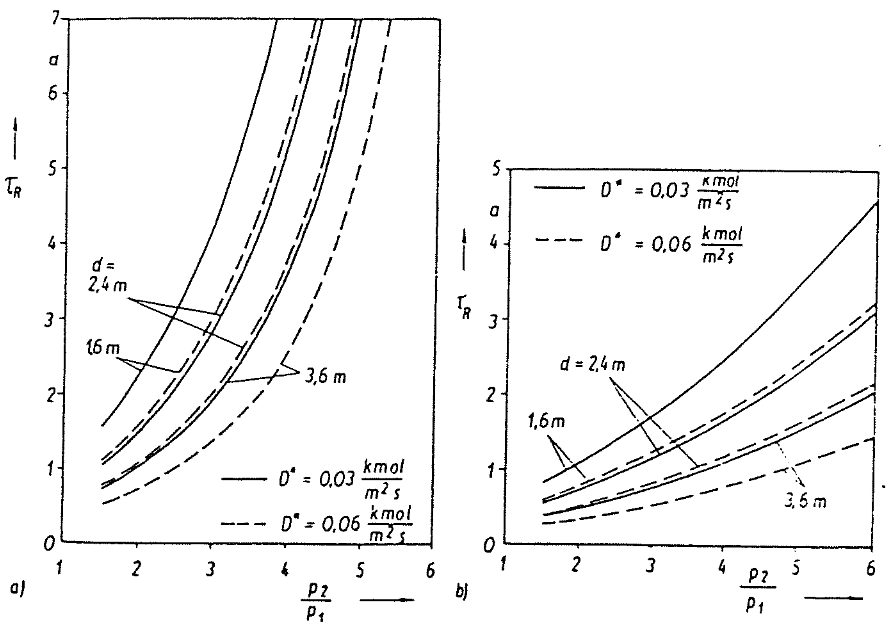

Fig. 2a, 2b : Payback time as a function of the pressure ratio and column diameter (14)

1.2. APPLICATIONS

The first part of the report of M. Wakabayashi and S. Sakashita (1) contains some successful applications of open and closed cycle compression heat pumps developed over the past few years in Japan. Actual installation numbers in different industrial branches, such as distillation, brewing and drying, are given. The described case studies show the benefits achieved by correct placement of a heat pump with respect to the process affected.

The first application described concerns ethanol vapor produced (96 vol %; 78 °C; 8800 kg/h) in a distillation column and then led to the shell side of a falling film type heat exchanger. In the tubes water is evaporated (steam 0,37 kg/cm^2, 74 °C; 3200 kg/h) and led to a screw compressor. The steam is then raised in temperature and pressure (1,28 kg/cm^2, 114 °C superheat, 3600 kg/h) to provide the necessary heat for the distillation column. The annual operating time is 8000 hours and the achieved COP is 5,73. This process started operation May 1986.

400 wat inje ↓ niet 96. (*)

Another example concerns a beer brewing process. This is a batch process (2000 batcher/year and 1,5 hr/batch). Vapor (1,033 kg/cm^{2}a; 100 °C; 10.000 kg/h) leaves the wort kettle and is compressed to 3,28 kg/cm^2 (136 °C; 10793 kg/h). The superheated vapor is condensed in an external heater of the wort kettle. The compressor is driven by a diesel engine (830 kW) and the obtained COP is 10,1.

Ref. (16) describes a screw compressor developed to provide excess steam (7,5 t/h) from a 2 to 3,5 bar network to a 4 to 9 bar network at the Hoechst AG factory, located at Gersthofen (W. Germany). The screw compressor is a dry machine with water injection, supplied by the Aezener Machinenfabrik GmbH. Speed regulation between 1250 to 3000 rpm was necessary because the steam quantities of the higher and lower pressure networks varied in time.

A screw compressor for high temperature application was tested out in Japan (17) at the Mayekawa Moriya Factory in 1983. The steam flow rate of the steam compressor varied between 1 ton/h to 15 ton/h. The temperature level of the heat source was between 60-120 °C and the heat sink level between 100-180 °C with a maximum difference between condensation temperature and evaporation temperature of 60 °C.

The construction of a practical plant in the Suntory Tonegawa Brewery was started in March 1984. Presently this plant is in operation. A similar project as in (16) is described in ref. (18). A prototype screw compressor recompresses 10.000 lb/h of waste steam while using only 50 % of the fuel that would be required to produce the steam in a conventional boiler. The screw compressor is driven by a 500 hp natural gas engine and was tested during a period of one year.

For the European Communities (19) E.D.F. (France) worked out some interesting projects.
One of these is the heat recovery from a cyclohexane distillation column with direct recompression. The vapor leaving the column is recompressed and condensed in a reboiler at the bottom of the column. The annual saving is 5300 t.o.e. and 1 kWh replaces 4,8 Mcal. A second project is the steam recompression from a titaniumoxide concentrator, replacing a single-acting evaporator (3000 t.o.e annual savings). Some problems occurred due to corrosion and fouling. A last application is found in an alcohol stripping column. Water of 35°C is evaporated to 90°C (0,7 bar) and compressed to 1,9 bar (550 kW compressor output power). The steam is then condensed at 120°C in a reboiler at the bottom of the alcohol stripper. 1 kWh of the compressor replaces 8,72 Mcal of steam and the overall saving for the plant is 1910 kt.o.e.

R. Wimmerstedt (26) in his paper presents a review of vapor recompression heat pumps used for the rectification of propylene (C_3 splitter). The overhead vapor is recompressed and afterwards used to provide the necessary heat at the bottom of the distillation column. The following table lists references concerning the above process.

Table 4.

plant location	Installation year	compressor type	number of stages	number of compressor	suction flow rate (m^3/h)	pressure ratio	suction pressure (bar)
Jela, Italy	1970	radial	3	1	1560	1,51	13,8
Sarralbe, Fr.	1975	"	3	1	3600	2	5,2
Antwerp, Bel.	1976	"	3	1	5640	1,76	11
Felluy, Bel.	1977	"	2	1	13500	1,76	10,1
La Porte, USA	1977	"	2	1	5940	1,66	12
Cuyo, AR	1987	"	1	1	7407		14,8

Other examples of industrial heat pump demonstration projects, supported by the European Community are listed below (24).

- Waste steam compression by a gas-engine-driven screw compressor in a brewery, Germany.

  The waste steam (5650 kg/h; 1,1 bar; 99 °C) normally exhausted to the air is now compressed by a screw compressor to 1,6 bar and 114 °C and re-used in the wort copper.
  Hot condensate (100 °C), extracted from the kettle, together with waste heat from the gasmotor are used to heat up water which is used in the brewery.
  Technical data : primary gas energy input : 493 kW
  produced useful energy : 4269 kW

- Heat pumps using steam generated by a thermomechanical pulp process in France.

  Steam generated in thermomechanical pulp production is fed into a cyclone washer to remove the impurities and then led into the evaporators of the heat pumps.
  Fresh steam is generated coming as waste steam from a pressurized preheater. This steam is sent to two compressors (132 kWe; 1500-2500 kg/h and 55kWe; 700-1300 kg/h) and is used in a steam network.
  This project was quite successfull and a payback time of 2,5 years was achieved (1982).

- Heat recovery from sugar production in France.

  Exhaust steam from a crystallization and a syrup condensation unit is recompressed and pumped back into the heating systems.
  The operation and measurement is expected to start in September 1987 and continues through January 1988. The results will be analyzed by EDF and the Sugar Industry Research Institute of France (IRIS).

The following tables review different industrial branches where most open cycle compression heat pumps are installed on a country by country base for IEA member countries.

1. Norway

Table 5.

Industrial branch	Installation year	Capacity MW	Plant address
fish processing (evaporators)	1984-1985	3 x MW-range	
pulp and paper	1977	20	Hunsfoss Fabrik N-47 00 Vennesla
pulp and paper	1978	60	Borregaard Fabrik N-1700 Sargsborg
pulp and paper	1982	15	Borregaard Fabrik N-1700 Sargsborg

2. Japan

Most heat pumps installed in Japan are closed cycle heat pumps and absorption heat pumps.

MVR applications are found in the diary, brewing and chemical industries, and total number of the applications be estimated at approximately one hundred.

3. SWEDEN

In Sweden 13 evaporation plants with MVR have been installed since 1980. One of those is in the pulp industry, three in the chemical industry, six in dairies and three in other food applications. Furthermore three MVR plants for distillation have been built, two plants for compression of vapour from thermo mechanical pulping and one in the ammonia production.

Table 6 : Review of MVR heat pumps in the Netherlands

Industrial branch	Year of instal-lation	Process or application	Consumed power (kW) Capacity (kW)	Working fluid	Manufacturer	Company/plant address
chemical	1985	evaporation of methanol and water from fatty acid	200/300 (centrifugal compressor) electr.driven cogeneration	methanol	Wiegand	Unichema Chemie Gouda
chemical	1967		220/3500 (centrifugal compressor) steamturbine driven	water		NSM/Sluiskil
chemical	1983	concentra-tion of glucose solution	500/ - (centrifugal compressor) electr. driven	water	Borsig	Cargille Bergen op Zoom
chemical						DSM, Geleen
dairy	1983	concentra-tion of whey	250/500 (centrifugal compressor) electr. driven	water	Wiegand	Coop. Zuivelfabriek De Toekomst St. Nicolassga
dairy	1982	concentra-tion of whey	900/9500 (two stage centrigugal compressor) electr. driven	water	Linde	DMW Campina - Veghe
dairy	1982	concentra-tion of whey	750/ - centrifugal compressor electr. driven	water	Linde	DMW Campina - Veghe
dairy	1985	concentra-tion of whey	220/ - three stage centrifugal compressor, electr. driven, cogeneration	water		Coop. Fabrieken van melkprodukten
food	1981	concentra-tion of starch	632/14200 steamturbine driven centrifugal compressor	water		CPC Nederland Sas van Gent

Table 6 (continued)

food	1986-1987	Concentration of glucose solution	800/ - steam turbine driven centrifugal compressor	water	Borsig	CPS Nederland Sas van Gent
food	1986	concentration of starch		water		Latenstijn
slaughtery		concentration of meat	centrifugal compressor	water		vleesafvalverwerking Geko Rotterdam
slaughtery	1984	concentration of meat	centrifugal compressor	water		vleesafvalverwerking Son, Breugel (NCB)
chemical	1986	concentration	20/285 roots blower, electrically driven	aceton	Demag	Duphar, Neememba
food	1986	concentration of starch	1300/≈ 28000 two stage-centrifugal compressor, electr. driven in combination with a steam ejector	water	John Brown Dedert.Corp. Borsig	CPS, Nederland Sas van Gent
brewery	1986	concentration of wort	100/2000 roots blower, electr. driven	water	Klein	Heineken
chemical	1986	concentration of a gelatin solution	centrifugal compressor electr. driven	water	Wiegand supported by EG	Trobas, Dongen

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4. Netherlands (23)

Table 6 reviews practical data subdivided into the types of industry, installation year, process specification, consumed power and capacity, working fluid, manufacturer and plant address for open cycle heat pumps in the Netherlands. Most MVR systems use water vapor which is widely accepted in industry. At this moment the use of other vapors is investigated by means of pilot plants

5. Austria

Four open cycle heat pumps are installed in Austria. All of them are thermocompression plants. Table 7 lists some information concerning those plants :

Industry or process	Year of installation	Heating capacity (MW)	Plant address
salt production	1979	2 x 33	österreichische Salinen AG A-4802 Ebensee, Austria
dairy	1986	14	Alpi Milchindustrie schillerstrasse 35 A-5021 Salzburg
fruit	1987	15	Grünewald Obstverwertung Grazerstrasse 6 A-8510 Stainz
dairy	1987	2,5	Molke Verwertung A-6933 Doren 72

Table 7 : Review of heat pump plants in Austria

6. USA

Table 8 lists single stage radial turbo compressors used in different industrial branches. All of them are installed by Borsig.

7. France

Table 9 lists also single stage radial turbo compressors installed by Borsig in France. All of them are installed in 1980.

Table 8 :

Single stage radial turbo compressors for vapor (25)

Customer	Order placed	Number	Suction volume m^3/h	Suction pressure bar	Discharge pressure bar	Speed rpm	Input kW	Drive	Applicati
oster Wheeler SA	1962	1	56.900	0,8	1,08	4.730	520	steam turbine	process pressure increase
qua-Chem-Rhinelander Paper Co. SA	1963	1	28.270	1,08	1,47	7.560	435	steam turbine	black lye distillat
osenblad Corp. SA	1964	1	38.400	1,03	1,41	6.700	490	E-motor gear drive	black lye distillat
osenblad Corp. SA	1965	1	54.000	1,03	1,41	6.150	686	steam turbine	black lye distillat
osenblad Corp. SA	1965	1	73.900	1,03	1,41	5.000	970	steam turbine	black lye distillat
entennial Mills Div. SA	1971	1	39.900	1,014	1,43	8.340	597	E-motor gear drive	waste heat recovery
tandard Brands, Inc. SA	1971	1	100.000	1,03	1,34	4.510	1.037	steam turbine	vapor compressi
Sandwell Int'l. USA	1975	1	152.900	1,034	1,49	4.452	2.275	E-motor gear drive	vapor compres
Resources Conservation Co.+ USA	1975	2	114.900	0,847	1,068	3.550	875	E-motor gear drive	vapor compres (saline
ITT Rayonier, Inc. USA	1975	1	163.400	1,051	1,50	4.415	2.386	E-motor gear drive	vapor compres.
Weyerhäuser Co. USA	1975	1	241.600	1,034	1,34	3.655	3.789	E-motor gear drive	vapor compres.
AB Rosenblad Patenter Norwegen	1975	1	73.100	1,207	1,775	6.614	1.785	E-motor gear drive	vapor compres:
Midwestern Food USA	1975	4	113.700	0,706	1,083	5.690	1.265	motor gear drive	vapor compres:
Aqua-Chem. Inc. Weston Paper Co. USA	1975	1	53.200	1,055	1,503	7.326	794	E-motor gear drive	vapor compres.

Single stage radial turbo compressors for vapor

Customer	Order placed	Number	Suction volume m^3/h	Suction pressure bar	Discharge pressure bar	Speed rpm	Input kW	Drive	Application
tandard Brands, Inc. SA	1971	1	32.100	1,014	1,42	7.730	455	E-motor gear drive	vapor compressi
tandard Brands, Inc. SA	1972	1	27.700	1,014	1,42	8.300	399	E-motor gear drive	vapor compressi
rain Processing Co. SA	1973	1	86.400	1,034	1,41	5.445	1.063	steam turbine	vapor compressi
.E. Seagrams & Sons SA	1974	1	52.600	1,034	1,413	6.890	653	steam turbine	vapor compressi
J.M. Huber Company USA	1974	1	54.600	1,013	1,427	7.233	745	E-motor gear drive	Vapor compres
Resources Conservation Co. USA	1976	3	102.300	0,809	1,04	3.550	820	E-motor gear drive	vapor compres. (saline

Table 9 :

Single stage radial turbo compressors for vapor

Customer	Order placed	Number	Suction volume m^3/h	Suction pressure bar	Discharge pressure bar	Speed rpm	Input kW	Drive	Applicati
Ste. Rocuette Frères Frankreich	1960	1	28.210	0,748	1,0825	8.615	312	Motor	vapor compression
Kestner Frankreich	1980	2	29.340	1,013	1,61	10.000	567	Motor	Vapor Compression
Lurgi Frankreich	1980	1	89.510	1,013	1,46	6.840	1.220	Motor	Vapor Compression

8 . Other countries

Table 10 lists the single stage radial turbo compressor, installed by Borsig in Canada, Finland, Belgium and Germany.

Table 10 :

ngle stage radial turbo compressors for vapor (25)

stomer	Order placed	Number	Suction volume m^3/h	Suction pressure bar	Discharge pressure bar	Speed rpm	Input kW	Drive	Applicatior
Rosenolad Corp. Kanaca	1974	1	241.600	1,034	1,434	3.655	3.190	E-motor gear drive	vacor compres.
[illegible] Ltd. anada	1971	1	29.200	1,014	1,43	7.700	412	steam turbine	vacor compressi.
W. Rosenlew A.B. nniand	1974	1	153.100	1,013	1,434	4.370	2.844	E-motor gear drive	vapor compressi
hiting Fermont eicien	1975	1	19.100	0,9973	1,525	9.927	341	E-motor gear drive	vapor compres
i cfurt	1980	1	62.840	1,47	2,05	7.850	1.121	Motor	Vapor Compression

1.3. COMPRESSORS

The main compressor types used for industrial MVR systems are : positive displacement compressors (subdivided in reciprocating and rotary compressors) and turbocompressors (radial and axial).

Concerning the high temperature positive displacement compressors it is very important that there is no oil contact with the compressed steam. Making use of contactless moving parts or lubricating by the compressed fluid itself, can be a solution.

(*) Another important problem is the cooling of the compressor. Mostly water injection or water jacket cooling is applied. Water injection can have a cleaning effect. Another advantage can be a decreasing internal leakage rate due to the water film.

The choice of the compressor used in a mechanical vapor recompression system depends on the flow rate, pressure rise, cost and a lot of general process conditions, (for example the nature of the vapor : steam, alcohol,...) erosion, corrosion (4), fouling, pressure levels, etc...

Especially turbo compressors are very sensitive to erosion and fouling. Screw compressors are very robust. Wet or polluted vapors can be compressed without any problem. Problems may occur when the suction temperature is lower than 90 - 100°C. In this case air can penetrate and the overall heat transfer coefficient will decrease. The same problem occurs with turbo compressors when the suction temperature is lower than 80-90 °C.

Table 11(4) reviews some properties of different compressor types. From this table one can conclude that the screw compressor can deliver a high pressure ratio. This can be a reason why screw compressors are not so much used compared to turbo compressors. The investment cost in fact is smaller and the energy recuperation is higher when the pressure ratio is smaller. In processes where the operating conditions are constant in time, turbo compressors are advantageous.

On the other hand, in practice, there are industrial processes of which the working conditions vary in time, for example the concentration of liquor. In such cases the boiling point increases with increasing concentration. Therefore a higher pressure ratio is necessary and the screw compressor may then be a solution (16).

Other volumetric compressors are the liquid ring and the lobe compressor. The first one is not often used in MVR systems because of its relatively low flow rate (8000 m^3/h) and low isentropic efficiency (0,2 - 0,5). The last one is used more often because this compressor runs without contact between the rotating parts. Further advantages are : stability, simplicity, attractive price (standardisation), good resistance against fouling (water injection). Disadvantages are : a high noise level, maximum pressure difference of 1 bar (due to internal leakage) (4).

Reciprocating compressors are not often used in MVR systems because of lubrication problems, sensitivity to liquid slugs and high costs. On the other hand turbo compressors are often used in different MVR plants. Recent developments of the centrifugal compressor (4) are impellers in composite material with carbon fibre and casings in glass fibre. This is necessary to resist the induced strains resulting from centrifugal forces and temperature effects. The most important reason why axial compressors are not frequently used in MVR is because of their high cost in spite of their compactness and good efficiency. Special care is needed to remove all the water from the incoming vapor because of erosion problems at the leading edge of the blades.
Table 12 reviews some advantages and disadvantages of different steam compressor types used in MVR (15).
Table 13 points out the existing and potential applications of heat pumps and MVR systems (15).
Table 14 gives measured performance values for steam compressors (16).

The following lists some manufactures of compressors used in mechanical vapor recompression (15).

HIBON : lobe type booster compressor
ATLAS COPCO : centrifugal and twin screw compressor
AIR INDUSTRIE NEYRTEC : liquid ring compressor
ALSTHOM-ATLANTIQUE RATEAU : centrifugal compressor
BORSIG : centrifugal compressor
NEU : centrifugal compressor
SULZER : centrifugal compressor
ALLIS-CHALMERS
CREUSOT-LOIRE
DRESSER
G.H.D.
GUIDAS
HOWDEN
TECHNOFAN

Figure 3 shows a curve which represents the increasing development of steam recompression in industry for different companies for the early eighties.

Category	Type	Volumetric flow rate	Pressure rise rate per stage
Turbo-compressors	— centrifugal — axial	7,000 to 500,000 m^3/h	1.8 to 2 1.2 to 1.8
Volumetric compressors	— reciprocating (piston) — rotary • lobe • screw • liquid ring	< 2,000 m^3/h 100 — 15,000 m^3/h 1,000 — 25,000 m^3/h < 8,000 m^3/h	4 to 6 $\Delta p \cong 1$ bar 3 to 6 1.4 to 1.5

Table 11: Compressor characteristics (4)

'able 12:

vantages, Disadvantages and Possible Developments of the Different Steam Compressor Types Used in MVR (15)

he different steam ompressor types used n MVR		ADVANTAGES General	ADVANTAGES Special	DISADVANTAGES	Possible Developments
Compressors	Non-lubricated pistons	- No surge region - Possibility of desuperheating liquid injection at the inlet (except for pistons)	Highest compression ratios	- Highly sensitive to liquid incursions - Complicated mechanics	Economic compressors derived from heat motors
	Non-lubricated screws		Compression ratio of 4	No industrial production	Economic compressors derived from the air compressor series
	Lobes		Industrial production	Pressure rise limited to approx. 1 bar	Improvement of the desuperheating liquid injection
	Liquid ring		- Simplest construction - accepting heavily contaminated vapours	Efficiencies not very well known in certain conditions of application	Increasing the efficiency
Compressors	Centrifugal	- Good efficiencies - Very low internal leakages	Industrial production	- Low pressure ratio per stage - Surge region Complexity	Compressors for the lowest capacities
	Axial		Best efficiencies		Compressors for the lowest capacities in order to compete with centrifugals

Table 13 .:

Existing and Potential (pot.) Applications of the Heat Pumps and Mechanical Vapour Recompression; Examples of Utilization.

(15)

	Heating of Liquids	Air Condition ., Space Heating	Concentration, Crystallization	Drying	Distillation, Rectification	Heat Recovery
Agricult./Food industry: - liquid products	pasteurization, washing water, wine	pot.	milk and derivates, juice, sugar, vinasse	milk and derivates	distilleries	hop vessels
- vegetable products	bleaching, washing water, mashing	canneries, ripening houses	alfalfa juice	malt houses, corn		condensers in refrigeration units
- meat products	scalding, washing water		pot. (effluents of fatty matter, blood)	meat processing, salting units		condensers in refrigeration units
Chemical industry: - heavy chem. eng.	pot. (heating of cuvettes, reactors ....)		chlorides, carbonates nitrates	pot. (fertilizers, powders, granulates ....)	misc. mixtures, eg ethane purification	exothermic reactions: sulphuric acid ...
- light chem. eng., pharmaceutical		dehydration (films)	adhesives, gelatines, synthesis vitamins..		perfumes	exothermic reactions, caoutchouc
Petrochemical industry	pot.		pot. (effluents)		methanol, C 5 fractions, propane-butane, dichlorethane	cooling water
Paper industry	pot.		liquors	pot. (pulp, paper)		cooling water
Metallurgical industries and mechanical engineering	treatment baths, leaching		effluents	misc. parts,coatings, pot. (ores)		hot effluents
Textile industry	dyeing	spinning and weaving mills	effluents	cloth, yarn bobbins		hot effluents
Misc. industries	swimming pools	clean rooms	sea water desalienation, effluents	wood, leather, gypsum		geothermic water, skating rinks
HEAT PUMP	X	X		X	X	X
M V R	pot.		X	pot.	X	X

Table 14 - MEASURED VALUES FOR STEAM COMPRESSORS (16)

COMPRESSOR TYPE	SPEED RPM	POWER kW	EVAPORATING TEMPERATURE °C	TEMPERATURE DIFFERENCE °C	VOLUMETRIC EFFICIENCY	PRACTICAL COP	PRACT. COP / CARNOT COP	SOURCE
Liquid ring	740		100 120 140	10 10 10		7.8 12.8 14.8	0,20 0,30 0,44	Textiles Research Center Mulhouse (5)
Lobe	2 500 2 760 2 000	6 6 6	60 81 81	26 11 17	0,63 0,78 0,67	5.8 18.6 9	0,42 0,56 0,42	EDF (3)
Monoscrew	3 000 3 000 3 750	20 11 28	110 86 110	32 36 38	0,61 0,57 0,68	5.52 4.79 4.92	0,42 0,40 0,47	EDF (4)
Twin screw	8 730 8 730	104 96	103 106	30 25	0,82 0,78	8.49 9.79	0,61 0,60	EDF
Centrifugal	37 900	73	100	17		14,7	0,62	EDF (5)

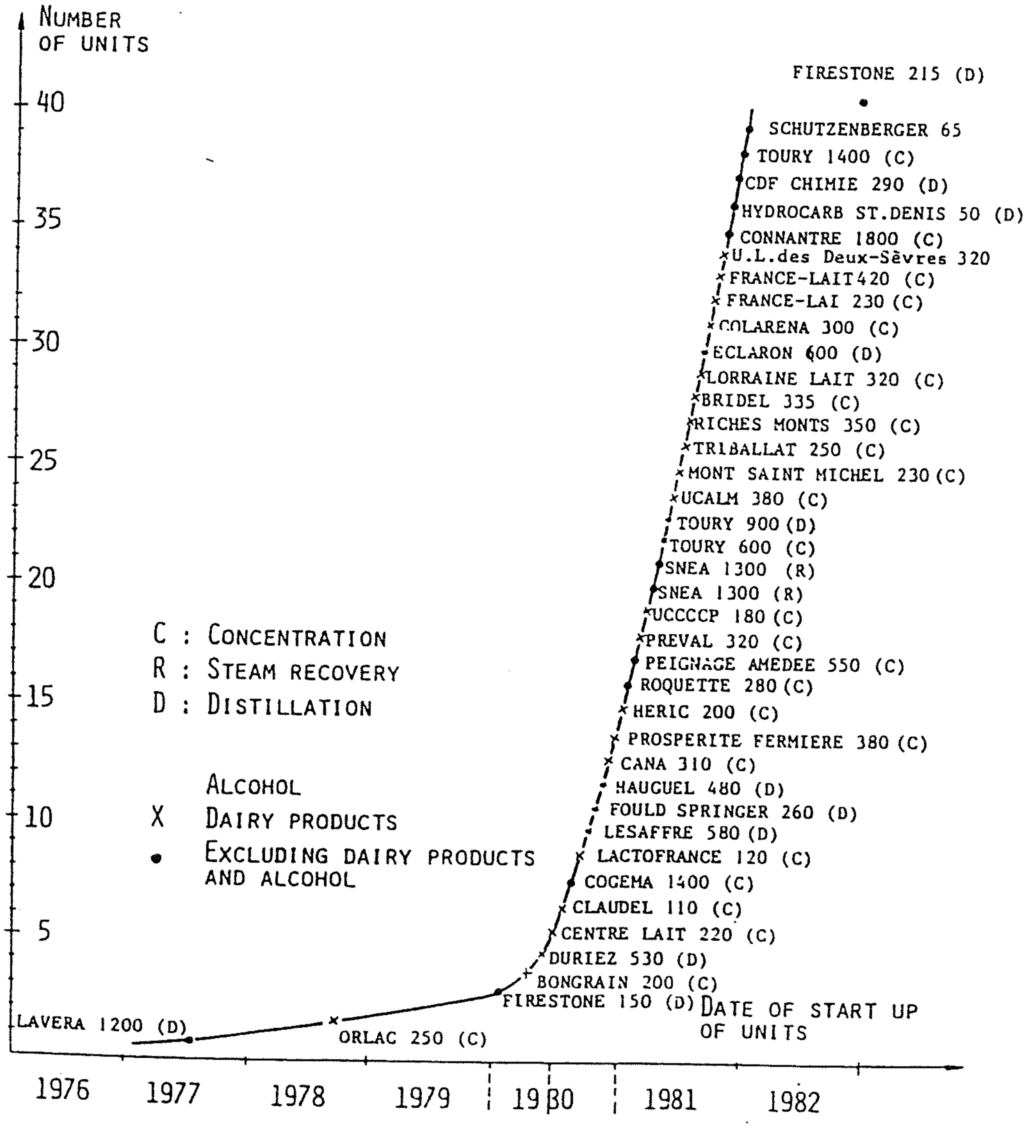

FIGURE 3 :DEVELOPMENT OF STEAM RECOMPRESSION IN INDUSTRY

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2. CLOSED CYCLE COMPRESSION HEAT PUMPS

2.1. APPLICATIONS

2.1.1. Introduction

Heat pumps have to be compared with other technologies in terms of energy and economy. Competing energy recovery systems are : heat exchangers, cogeneration, vapour recompression (open-cycle heat pumps), organic Rankine cycles (electricity from waste heat) and heat transformers.

For several industrial sectors and at different energy prices a comparison of the technologies mentioned above has been worked out (31). Heat exchangers turn out to be the most economic solution. From the economical point of view open-cycle recompression heat pumps are next. Heat transformers, high-temperature heat pumps and ORC's are economically in the same range. Further developments of the technologies can bring one or the other system ahead. However the final decision will always be based on the relation of electricity price to oil price.

Generally, an MVR system is more efficient and economic than a closed cycle HTHP if the heat source is a condensable vapour. Nevertheless the heat pump will be superior under certain conditions for example :

- vapour contains inert gases
- vapour conditions are not good for recompression (large superheating, low COP, near critical point)
- vapour is wet (compressor destruction by droplets)
- no suitable lubricant available
- compression conditions are not suitable (pressure ratio...)

Less economic than compression heat pumps are the heat transformers. Their greatest advantage is the small amount of mechanical energy needed however they need big amounts of cooling water.

The Organic Rankine Cycle is interesting compared to heat pumps in countries where the price of heat is low compared to the price of electricity. Depending on COP, temperature levels, running time, etc., the break even point seems to be at a cost ratio of 2,5 to 3 for 1 kWh electricity to 1 kWh of steam.

2.1.2. Applications

In Canada a newsprint mill in northern Ontaria installed two HTHP's. The heat pumps use R114 and are combined with mechanical recompression of low pressure steam. Steam was generated at 140 °C at a quantity respectively of 366,08 GJ/year and 347,00 GJ/year. This means a value of 1,832650 \$/year and 1737740 \$/year (Canadian dollars) (9).

In England V.A. Eustace tested an IRD gas engine driven heat pump in order to obtain data on the heat pump performance at off-design and part load conditions. The heat pump extracts the heat from an effluent stream at a nominal 80 °C. The gas engine recuperates its waste heat (exhaust gases) and combines this with steam produced by heat delivered by the heat pump condenser at 100 °C.
At design point the system consumes 242 kW of natural gas and produces 256 kW of useful heat; the COP is 3,19 and the heat recovered from engine and exhaust is 117 kW. R114 was chosen as the most suitable working fluid. Table 1 and figure 2 show the technical data and the scheme of the installation. The type of compressor and engine choses is dictated by the output of the unit. A screw compressor or a centrifugal compressor should be utilized in larger systems. A number of potential applications were investigated with success (25) (26). These involved recovery of heat :

1) by condensing vapours generated in a chemical reaction vessel at 80 °C. Steam produced by the heat pump system is used to heat the same reaction vessel;
2) from a process effluent stream used to cool batches of maize in a whisky distillery;
3) from whisky still condenser cooling water in a distillery. Low pressure steam generated by the heat pump is compressed in a thermocompressor to 2,4 to 2,7 bar and supplied to the still heating oils.

	Actual	Design
Gas power to engine	221 kW	242 kW
Steam power from system	333 kW	356 kW
Steam flowrate	0.141 kg/s	0.151 kg/s
Primary energy ratio (PER)	1.50	1.47
Engine RPM	1080 r/min	1100 r/min
Shaft power from engine	68 kW	75 kW
Heat pump coefficient of performance (COP)	3.12	3.19
Heat recovery from engine and exhaust	121 kW	117 kW
Heat lost from engine and exhaust	32 kW	50 kW
Percentage energy recovery from engine & exhaust	86 %	79 %
Refrigerant flowrate (R114)	2.54 kg/s	3.01 kg/s
Condensing temperature	120 °C	120 °C
Compressor discharge temperature	126 °C	125 °C
Evaporating temperature	60 °C	60 °C
Steam outlet temperature	110 °C	110 °C
Effluent inlet temperature	80 °C	80 °C
Effluent outlet temperature	65 °C	65 °C
Effluent flowrate	2.3 kg/s	2.6 kg/s
Heat removed from effluent	145 kW	164 kW
Compressor oil differential pressure	1.5 Bar	1 to 2 Bar

Table 1. Performance of IRD High Temperature Heat Pump (24) (3)

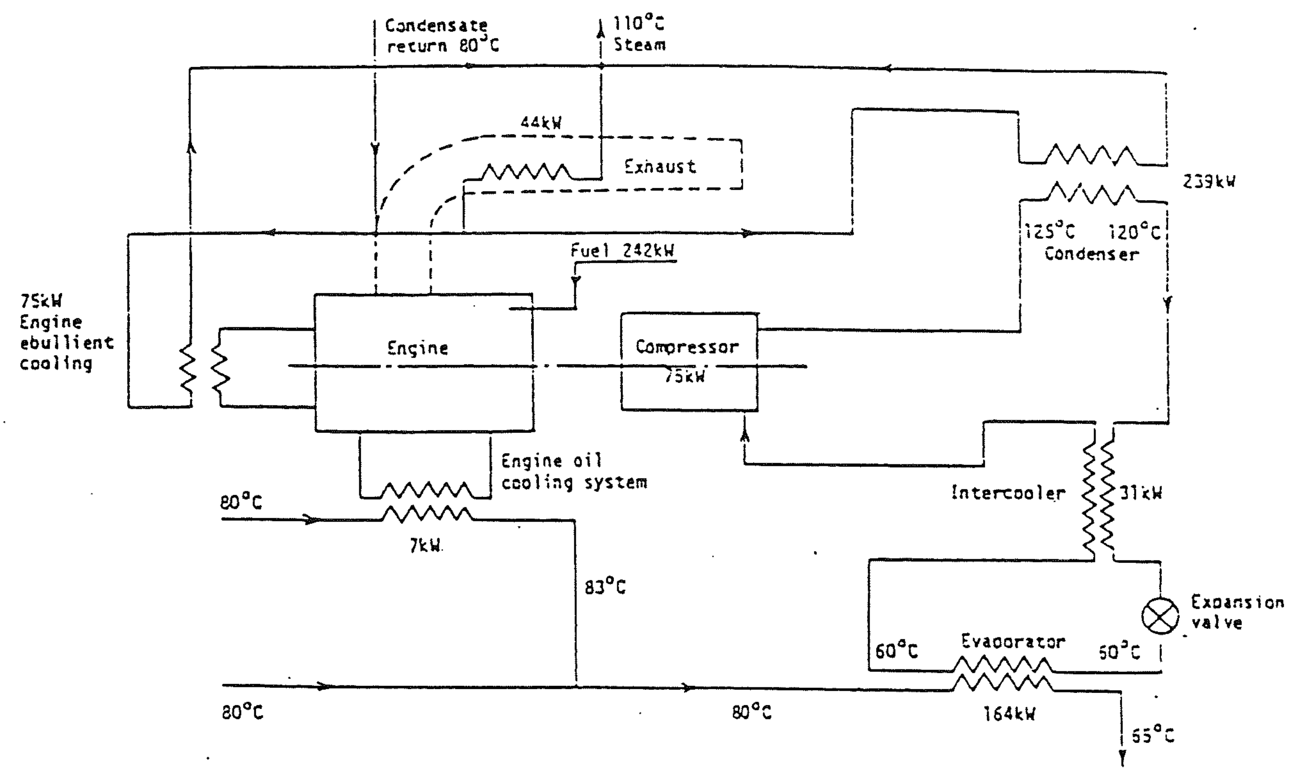


Figure 2 : Schematic of the IRD gas engine driven high temperature heat pump (24) (3)

Table 2 lists the main manufacturers of electric heat pumps in Japan (36).

Table 3 shows the technical data of two high temperature heat pumps of the Sulzer company (Switzerland). Figure 3 shows the limits of different working fluids for use in a high temperature compression heat pumps (37). Table 4 and figure 4 show the characteristics of a compression HTHP. The range of heating capacity of the compressors used in the Sulzer HTHP is indicated in figure 5. An estimation of the purchase cost for a HT gas and electric driven heat pump as a function of the temperature lift and the exploitation cost as a function of the utilisation temperature are given by figures 6 and 7 (37).

ELECTRIC HEAT PUMP MANUFACTURERS

Maker \ Product	Air-source heat pumps: Packaged air conditioners	Air-source heat pumps: Wall-through air conditioners	Air-source heat pumps: Chilling units	Air-source heat pumps: Heat recovery chilling units	Air-source heat pumps: Screw heat pumps	Air-source heat pumps: Heat recovery screw heat pumps	Air-source heat pumps: Hot water supply heat pumps	Air-source heat pumps: Cooling / heating & hot water supply heat pumps	Water-source heat pumps: Packaged air conditioners	Water-source heat pumps: Heat recovery small heat pump units	Water-source heat pumps: Chilling units	Water-source heat pumps: Heat recovery chilling units	Water-source heat pumps: Heat recovery screw heat pumps	Water-source heat pumps: Heat recovery centrifugal heat pumps	Water-source heat pumps: Hot water supply heat pumps
Ishikawajima Heavy Industries Co., Ltd.					*										
Ebara Corporation					*	*							*	*	
Kobe Steel, Ltd.					*	*							*		
Sanyo Electric Co., Ltd.	*		*												*
Sharp Corporation	*								*						
General Aircon, Ltd	*	*				*									
Daikin Industries Ltd.	*		*	*	*	*	*	*	*	*	*	*	*	*	*
Toshiba Corporation	*	*	*					*	*	*				*	*
Toyo Carrier Engineering Co., Ltd.	*		*												
Nippon PMAC Co., Ltd.		*							*	*					
Hitachi, Ltd.	*	*	*	*	*	*	*	*	*	*	*	*	*	*	*
Fujitsu General Ltd.		*													
Fuji Denki Sosetsu Co., Ltd.	*		*												
Mayekawa Mfg. Co., Ltd.					*	*							*		
Matsushita Seiko Co., Ltd.		*	*						*		*				*
Matsushita Electric Industrial Co., Ltd.	*	*					*		*						
Mitsubishi Heavy Industries, Ltd.	*	*	*	*	*	*	*	*	*	*	*	*	*	*	*
Mitsubishi Electric Corporation	*		*	*			*	*	*	*	*	*		*	*

Table 2. Manufacturers of electric heat pumps (not only HTHP) in Japan (36).

Kompressoren Compressors Compresseurs		Kältemittel Refrigerant Fluide frigorigène	Wärmequelle Heat source Source de chaleur		Wärmeverwendung Heat utilisation Utilisation de la chaleur			Anlage Plant Installation	Land Country Pays	Bemerkungen Notes Remarques
Anzahl Number Nombre	Typ Type Type		Art Type Type	Temp. Temp. Temp. °C	Art Type Type	Temp. Temp. Temp. °C	Heizleistung Heating capacity Capacité de chauffage kW			
1×2	22	R 114	4	+ 59	i	125	2450	Himont S.p.A., Ferrara	IT	

Table 3 : technical data of a HTHP installation constructed by Sulzer.

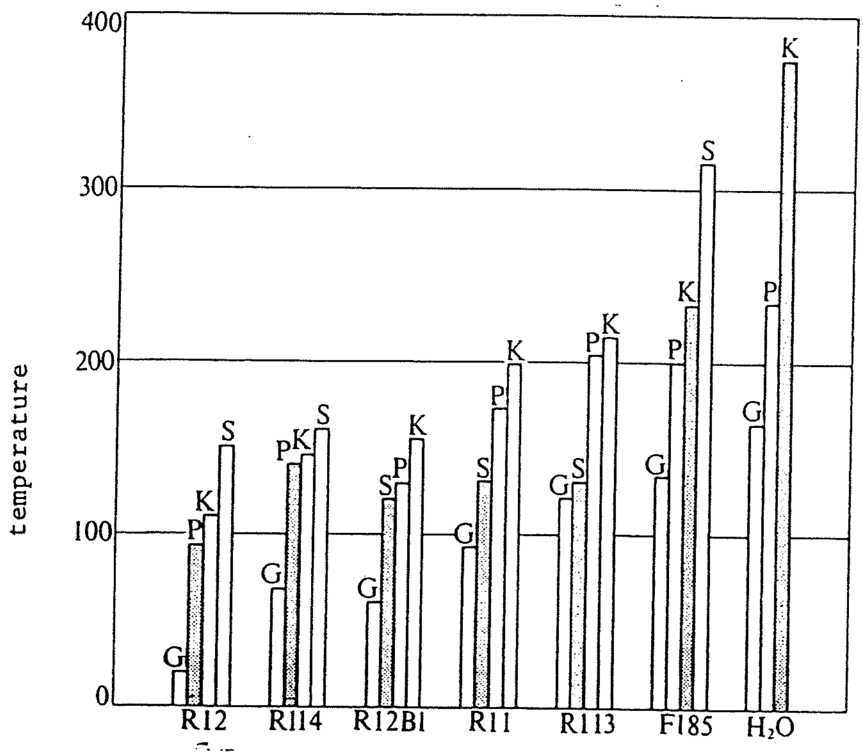


Figure 3 : active working fluid.

K = critical temperature

S = limits of stability

P = saturation pression (30 bar)

G = saturation pression (7 bar)

application zone

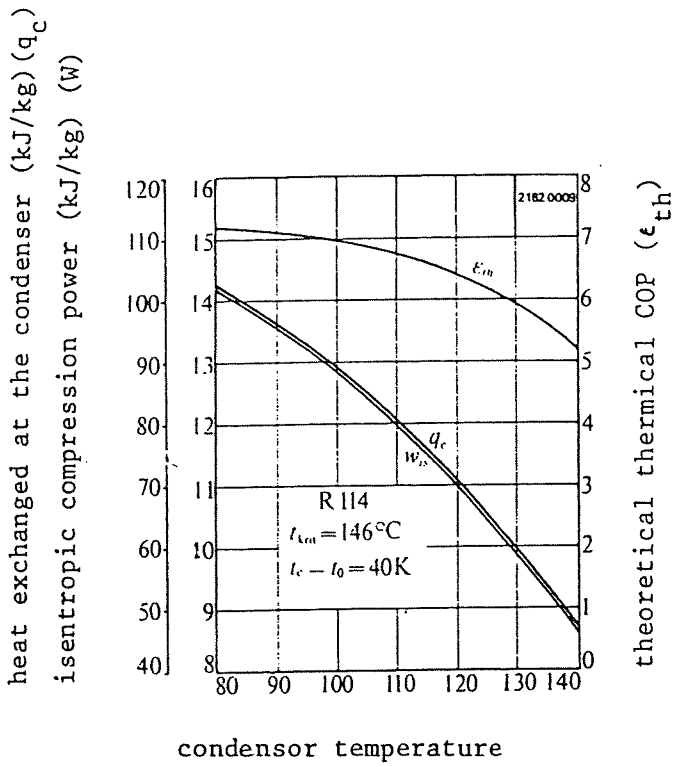
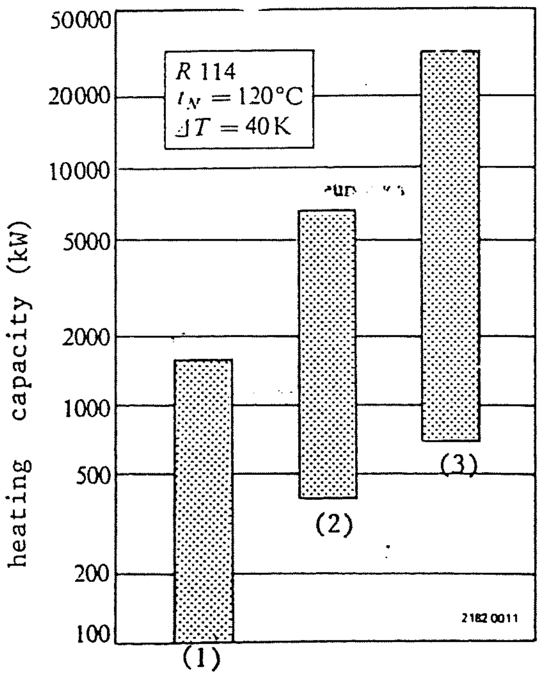


Figure 4 : Compression heat pump performance characteristics (37)

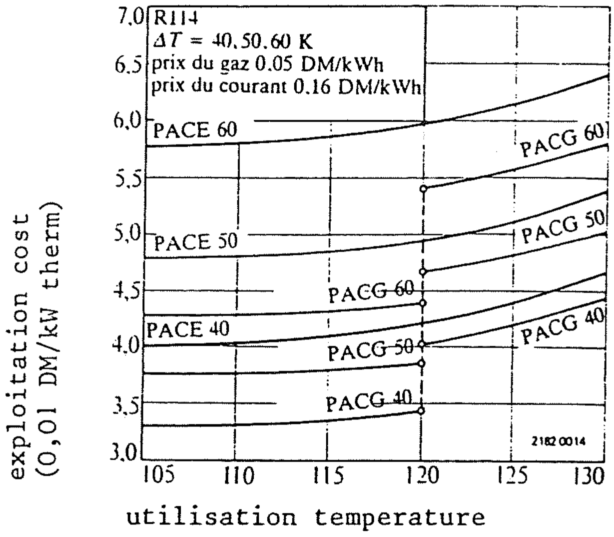
utilisation temperature	100 °C up to 130 °C
temperature lift	40 to 60 K
heat source and heat sink fluid	water
working fluid	R114
calorific capacity	700 kW therm.
compressor type	piston
compressor drive	gaz or electrical engine

Table 4. Characteristic data of a compression heat pump (37).



(1) piston compressors
(2) screw compressors
(3) turbocompressors

Figure 5 : Heating capacity versus compressor type (37)



PACE : electrical HP

PACG : gas driven HP

Figure 6 : Exploitation cost as a function of the utilisation temperature (37).

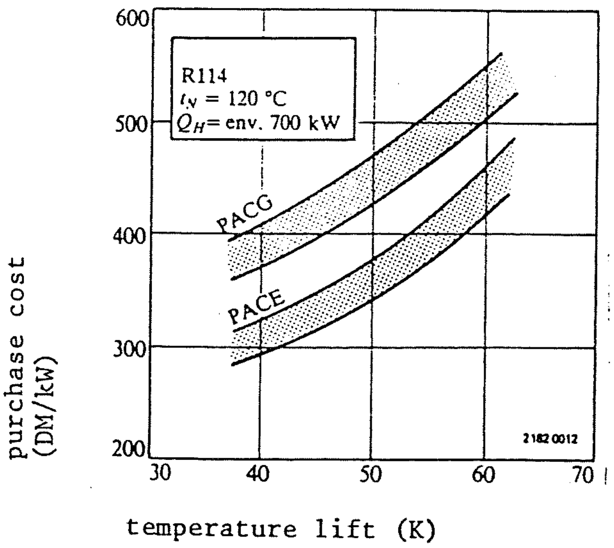


Figure 7 : Purchase cost versus temperature lift for electrical and gas driven heat pump (37).

In Japan the Japan Refrigeration and Air-conditioning Industry Association (JRAIA) investigated the application of heat pumps to industrial processes (1). In Japan progress in industrial energy conservation during the past 13 years is remarkable. However, low temperature waste heat is still being rejected, for the most as hot water.
The industrial energy consumption of the total energy consumption structure in major countries is as follows : Japan : 64,3 %; Germany : 36,9 %; France : 40,8 %; U.K. : 38,2 % and USA : 32,1 %. The temperature level of energy consumption in Japan is : (1)

< 100 °C	30 %
100 °C - 315 °C	31 %
315 °C - 600°C	9 %
600 °C -	22 %

Table 5

The heat consumed between 100 and 150 °C in a number of industrial sectors is given in table 6.

Industry	Temperature between 100° - 150°
food and tobacco	62,3 %
textile	50,3 %
lumber and wood	9,3 %
pulp and paper	85,9 %
chemicals	26,9 %
rubber	26,3 %
leather	100 %
stone, clay and glass	85,6 %

Table 6

To expand the use of the HTHP, it is necessary to reach output temperatures up to about 200 °C and to get a larger temperature lift with higher efficiency. It is also important to integrate the heat pump in the total system to suit the process best. Therefore in 1984 the "Super Heat Pump Energy Accumulation Project" was started by the Agency of Industrial Science and Technology (AIST) (1). This project is aimed at developing advanced heat pumps for large commercial and industrial applications :

1) advanced vapor compression heat pumps; high performance and high output temperature;
2) chemical heat storage and heat pump system.

In Japan Yoshiaki Kubo and Seiichi Sakuma (5) made a market investigation for high temperature heat pumps. The temperature level of useful and waste heat were considered.
The following levels of waste heat temperature and available heat temperature are reported.

industrial sector	waste heat temp. °C	useful heat temp. °C
petrochemicals	40° - 70°	50° - 500°
polymer chemistry	60° - 100°	100° - 150°
inorganic chemistry	40 - 80	60 - 100
chemical fortilizers	40 - 70	50 - 100
dyeing and finishing	20 - 45	50 - 70
sugar manufacturing	50 - 115	60 - 140
dairy	50 - 60	130 - 140
livestock processing	20 - 30	60 - 70
marine products processing	35 - 120	90 - 170
noodle manufacturing	70 - 100	105 - 130
frozen foods	20 - 30	60 - 70
seasoning	25 - 90	80 - 120
cooling leverages	25 - 90	60 - 140
pulp and paper	50 - 120	80 - 170
steel manufacturing	30 - 1000	50 - 1500

Table 7

In Japan several heat pumps are already installed in dyeing, finishing, sugar manufacturing, livestock processing and frozen goods processing. The temperature level of the heat produced is from 50° to 70 °C. In most cases a period of 2 years pay-back is normally considered necessary.

2.2. RESEARCH AND DEVELOPMENT

2.2.1. Working Fluids

2.2.1.1. Pure Fluids

A lot of heat pump research is related to new working fluids or mixtures with special emphasis on non-azeotropic mixtures.

From a thermodynamic point of view many of the known refrigerants can be used in high temperature heat pumps. The refrigerants R11, R12B1, R21, R113, R114, R114a, R114B2, R133a, R216, trifluoroethanol-water mixtures, fluorinol FL50 and 85 are acceptable when the restriction of a maximum pressure of 20 bar at condensing temperature of 100 °C is considered. For higher temperatures fluorocarbons of the perfluoro-type may be advantageous as well as of course water (28).

2.2.1.1.1. Refrigerants : R114 - R12 - R142b - R133a (1) (2) (3) (4)

Today's heat sink temperatures with freon R114 and R12 are respectively in the range of 110 °C and 70 °C. Halogenated hydro-carbons such as R114 are suitable for operating temperature up to about 120 °C and could allow the production of low pressure steam. However, steam generated with these heat pumps will not be sufficiently high temperature and therefore may not be suitable as process steam.

Asea-Stal in Sweden (3) proposed to the IEA to test a steam generating R114 heat pump to recover heat from humid air.
Use is made of a two-stage turbo-compressor with evaporating temperature of 31 °C and condensing temperature of 110 °C. The condenser is fed with water from a mill system and steam is boiled in the condenser at

a temperature of 105 °C or about 1,2 bar. 11,5 MW is produced with this low pressure

A two-stage screw compressor heat is being provided with screw expanders reaching a temperature level of 150 °C is being developed in Japan (2). It uses R12 as working fluid and can achieve temperature lifts of 100 °C with a COP of 3. The search for new high temperature refrigerants up to 200 °C, like fluorocarbons such as fluoro-alcohols or perfluoro-alcohols, is still continuing (2).

R142b can be used as an alternative refrigerant to R114 because R144 needs superheating before compression to avoid slugging and requires large size compressors because of its low density. However this fluid is only compatible with the production of hot water and up to 95 °C - 100 °C. J.C. Blaise and T. Dutto found that the stability of R142b in presence of oil and metals is the same as that of R114. However the heat production is highly superior to the one of R114 and this allows an important reduction in compressor size. In addition R142b does not need superheat before compression as the isentropic curves remain in the vapor domain during compression. The experimental COP of the tested installation was close to 65 % of the Carnot COP. This means COP = 4,03 for a T_c of 100 °C and a temperature rise of 60 °C.

As stated in ref. (23) R133a is considered as an alternative working fluid for R114 because of the superheat requirement of the latter. R133a has a higher density but due to its toxicity it cannot be applied in heat pumps.

A lot of work to enhance the output temperature and to reach a considerable temperature lift is done. Takeshi Yoshii of the Mitsubishi Heavy Industries, Tokyo-Japan developed an advanced compression heat pump. The aim is to double the COP and to attain higher output temperatures of 150 °C to 300 °C. Therefore working fluids as well as mixtures are part of the project (1).

2.2.1.1.2. Pentane

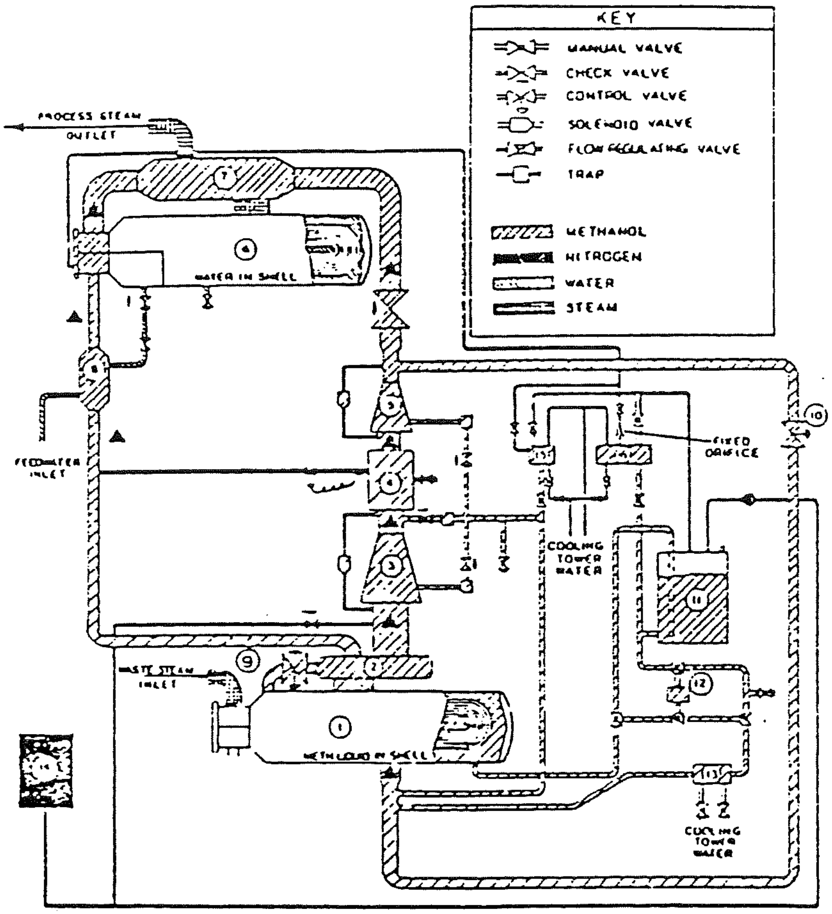
In Japan Kubo and Sakuma attempted to increase the utilisation temperature of an electrically driven screw compressor heat pump with a condensing temperature of 135 °C. After a study of different types of freon and hydrocarbon refrigerants which included : thermal stability, costs, durability tests (scaled tube test), safety, COP, pressure required compressor capacity, normal pentane was selected as the best fluid (5).

Normal pentane and R114 were first selected. Sealed tube tests at 170 °C during 17 days with various combinations of refrigerants and lubricating oils led to pentane as the most suitable fluid. However pentane is flammable so most applications will be **in** the chemical industry where flammable materials are acceptable materials.

2.2.1.1.3. Methanol (6) (3) (28)

During the development and design of an electric heat pump for the generation of process steam (5,4 bar, 11 °C superheat) Moreland and Wolfe selected methanol as working fluid. Figure 8 shows a schematic of the methanol heat pump system. Eight criteria were used to evaluate potential working fluids : thermodynamic performance - chemical stability - corrosiveness - cost and availability - volume flow rate - heat requirements for compression - hazard, explosive and flammable limits - toxicity - carcinogenicity - availability of thermodynamic properties. The working fluid also had to have a critical temperature greater than 205 °C (at least 30 °C above the maximum delivery temperature) and a boiling point below 105 °C.

Only 100 of 1000 fluids considered passed the last two requirements (T_c > 205 °C, T_b < 105 °C). Out of these 100 only four passed the 8 required demands : methanol, thiophene, n-hexane and fluorinol 85. Methanol was chosen on the basis of associated equipment cost, assuming that chemical stability and compatibility with materials used in the heat pump are acceptable. Analytical chemistry techniques were used to quantitatively measure the thermal stability of methanol. Minor amounts of dimethylether and water were detected but were not expected to give system problems. This high temperature heat pump has a pay-back period of 1,3 years and a net output of 30 10^6 Btr/hr (= 31,65 GJ/hr). The source leaving temperature must be at least 70 °C in order to avoid vacuum.



PARTS LIST

1. METHANOL EVAPORATOR
2. METHANOL SUPERHEATER
3. 1st COMPRESSOR STAGE
4. INTERCOOLER
5. 2nd COMPRESSOR STAGE
6. METHANOL CONDENSER
7. METHANOL DESUPERHEATER
8. METHANOL SUBCOOLER
9. EXPANSION VALVE
10. HOT GAS BYPASS VALVE
11. METHANOL STORAGE TANK
12. METHANOL PUMP
13. RAPID COOLDOWN HEAT EXC
14. NITROGEN STORAGE TANK
15. AUTOMATIC PURGE DEVICE
16. PURGE CONDENSER

Figure 8 : Schematic of methanol heat pump system (6)

2.2.1.1.4. H_2O

At the Nagasaki Institute of Applied Science, Japan, the development of a reciprocating steam heat pump with turbo supercharger delivering an output up to 300 °C is undertaken. The employed refrigerant is H_2O (R718) and the expected value of the COP is 3 with a temperature rise of 150 °C (2).

Reference (8) describes a second project to develop a high temperature (about 150 °C) h.p. Water is used as working fluid and various operational parameters that influence the performance of the heat pump are investigated.

The Ministry of Trade and Industry of Japan sponsors studies of the critical pressure and temperature of working fluids. Physical and thermodynamic properties undergo dramatic changes and in addition there is chemical instability in this high temperature region.

A comparison of the COP of heat pumps working with water and fluorinated hydrocarbons is made by Tolle (28). Tolle proposes to compress moist steam to avoid high temperatures at the end of compression.

2.2.1.2. <u>Non-Azeotropic mixtures</u> (8)

It is known that non-azeotropic fluid mixtures enhance the COP and/or thermal capacity in compression hp's. However little data is available concerning non-azeotropic mixtures of refrigerants in high temperature heat pumps.

The main parameters that may be influenced (in addition to the COP) by the use of mixtures are : heating capacity, compressor discharge temperature, higher temperature gradients in the heat sink and the heat source, pressure levels in the condenser and evaporator, heating capacity control (32) (33) (34) (35).

The introduction of a proper second fluid results in a decrease in condensing pressure and/or compressor outlet temperature to such a level that the condensing temperature is increased compared with the pure working fluid. Mixtures capable to increase both the heating capacity and the COP will in the future probably be competitive with the pure working fluids used today (33) (32).

The condenser pressure can be decreased allowing higher condensing temperatures. The evaporator pressure can be adjusted to be slightly above 1 bar. Compressor life will be extended due to the decreased condenser pressure and consequently decreased bearing loads in the compressor.

The proper choice of a mixture allows the decrease for a given condensing temperature of the compressor discharge temperature.

With the aid of a distillation column, cfr. figure 9, varying the non-azeotropic mixture composition, a continuous control of the heat output of the heat pump is possible (34) (35).

Without decreasing the COP, higher temperature gradients in the heat sink and heat source are allowed. This partly results in smaller costs of the heat exchangers.

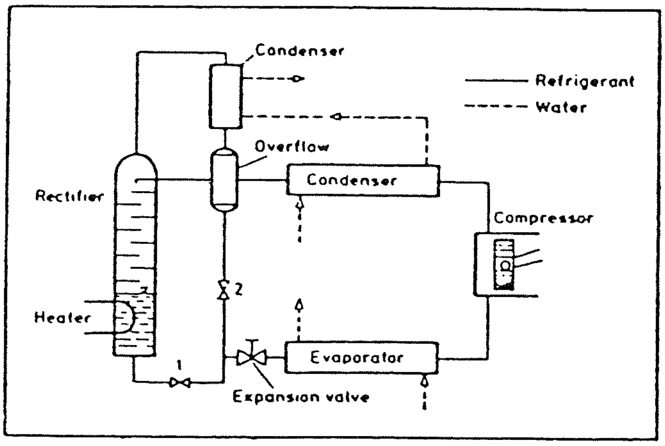


Figure 9 : Modification of a heat pump plant with capacity control by a rectifier for changing the mixture composition/B60/.

Blaise and Dutto try to enhance the COP as much as possible, to decrease the condensing pressure as much as possible and to provide capacity control by means of these fluids. They investigated the influence of the mixture composition on thermal and mechanical capacities of the heat pump and the influence of leakages. To obtain condensing temperatures between 60 °C and 120 °C they used R12-R114 mixtures. It was possible to vary the thermal and mechanical powers by varying the composition of the mixture. With the same h.p working at the same temperature conditions (80 °C) the thermal output can vary from 1 to 2 with the same value of COP (3,5). Blaise and Dutto found that leakages had no influence on the composition of the non-azeotropic mixtures (10).

Struck of MAN AG in Munich examined low-cost screw compressors for refrigerant R12 and R114 and their mixtures. He considered condensing temperatures up to 90 °C (11). Kruse tried to increase the application field of diesel-engine driven heat pumps for condensing temperatures up to 90 °C through the use of a screw compressor and the refrigerant mixture R12/R114 (12).

Radermacher measured the specific heat of 4 refrigerant mixtures up to the critical temperature of the mixture, for five compositions and pure components in the liquid and vapor phases (13).

The Commission of the European Community has taken into its four year Non-nuclear Energy R & D Programme the development of an industrial h.p, which can produce heat up to 300 °C. This includes investigations of fluid mixtures for high temperatures. Up to now no data is available concerning these R & D projects.

In the field of non-azeotropic mixtures the following large projects are going on :

- at the Technical University of Denmark measurements of cycle performance are carried out in a laboratory test plant;
- at the University of Graz, prof. Moser (Austria) searches methodically for more optimal mixtures;
- at the University of Hannover, prof. Kruse tries to solve problems related to lubricants in contact with mixture;
- at Chalmers University of Technology, professors Berntsson & Gron perform evaporator heat transfer measurements and collect PVT-data for mixtures (32);

- the National Bureau of Standards (USA) is working on expansion valves and heat transfer problems on the evaporator side;
- at the University of Illinois, prof. Stoecker measures heat transfer on the condenser side.

2.2.1.3. General information

For temperatures between 120 °C and 180 °C different working fluids can be used. For instance hydrocarbons and trifluoroethanols/fluorinol up to 160 °C. However the thermodynamic fluids and mixtures are still incomplete. Therefore projects to collect thermodynamic properties of prospective substances to support and conduct experimental studies are needed (16). Kanagawa Institute of Technology of Japan started to do this type of work in 1984 (14).

At TU Graz Schnitzer collected data on high temperatures working fluids and non-azeotropic mixtures for compressor driven heat pumps (15).

2.2.1.4. Data banks

To be able to do engineering calculations, thermal and thermodynamic data is required. This data can be obtained from data banks. Pure fluid properties are stored here either as point values or fitted to equations so wide temperature and pressure ranges can be covered. The most important properties of the three data banks PPDS, Keydata and Detherm-SDS are the following : vapour density - liquid viscosity - vapour pressure - surface tension - latent heat - critical temperature, pressure, compressibility, molar volume, density-formula weight - melting, boiling point, etc. They concern most of the interesting working fluids such as R12, R12B1, R21, R113, R114, R114B2, R133a, R718, R112, R216.

2.2.1.5. Conclusions

Delivery fluid temperatures up to 130 °C are possible, depending on source temperatures. Up to 130 °C the working fluid R114 is used. Trifluorethanol (fluorinol) looks interesting for higher temperatures (up to 160 °C). At present trifluorethanol is only used in expansion machines. The volumetric efficiency is only about 10 % of R114 so large volumes have to be handled. As it has no problems of stability and decomposition water is a promising fluid above 200 °C. To avoid high temperatures at the end of compression of the water, Tolle (28) proposes to compress wet steam. A COP close to the Carnot COP (for an ideal loss-free compressor) can be obtained, as since the enthalpy and the entropy lines in the wet steam area do not diverge considerably. Water gives the best CC compared with R11, R113, R114, R12B1, R114B2, R216. The main disadvantage is the large suction volume.

2.2.2. Working Fluid/Lubricant Combinations

One of the major problems is to find a suitable working fluid/lubricant combination for use in heat pumps at high temperatures. Two primary requirements of the lubricant are that at the appropriate pressure and temperatures, chemical and thermal stability and suitable viscosity should be achieved.

2.2.2.1. Thermal and chemical stability

In the U.K. Bertinat, Drakesmith and Taylor investigated alternatives for both the standard working fluids and the mineral oils (17). At high temperatures not only the working fluid has to be thermally stable but also the fluid/lubricant combination must be chemically unreactive in the presence of metals and impurities.
They found that R114 was the most stable of the fluids investigated whatever the type of lubricant involved. Refrigerant 11 and R113 were unstable even with the most stable lubricant. Refrigerants 12B1 and R21 were intermediate. For R114 and R21 it was found that the COP's of the two fluids are not much different but that R21 was more chemically active and gave considerably more problems because it attacked the materials used in seals and gas hots. A refined mineral oil with R114 is sufficiently stable to be used. If necessary a more expensive synthetic hydrocarbon oil is recommended or if all else fails a synthetic fluorinated lubricant provides ultimate stability.

Drakesmith reports on the chemical performance of the following systems :
- R12B1/Nujol (mineral oil) : poor
- R114/fluorosilicone oil (Dow Corning) : good
- R114/Foublin (polyperfluoroether) : very good

at temperatures of 180 °C and at 200 °C.
While no decomposition occurred in the system R114/Foublin the system R12B1/Nujol showed bromide overstabilisation and copper and bromide plating.

At the university of Salford (U.K.) Watson, Abbas, Srinivasan and Devotta found that R11 is stable up to 220 °C, however in the absence of lubricants. It is less stable in the presence of alkylbenzone lubricants and still less stable in the presence of napthenic lubricants. R11 is more stable than R12B1 in all combinations of lubricants and metals (copper, steel). Metal, particularly copper, is less subject to attack by R11 than by R12B1. However lubricant effectiveness of n-oils is less impaired by R12B1 than by R11 (19).

Giolito and de Moncuit of Rhone-Poulenc investigated a synthetic lubricant for high temperature heat pumps (20). They found that stability of R114 and R133a at 120 °C is quite good in the presence of lubricant oil 801E23 and metals Fe, Cu, and Al. At a condensing temperature of 120 °C and after a 500 h run in a real system the lubricant remained clear. They reported that no important problems of corrosion have been found with the R114/801E23 system at 120 °C. Generalised corrosion of steel was very low.

In ref. (5) selection of lubricating oil was made by the following requirements :
- good thermal and oxidation stability
- proper viscosity in the working temperature range (134 °C) and a small solubility for the refrigerant.

A viscosity of at least 30 sCt (1 stokes 10^{-4} m^2/s) was required.
A polyglycol based synthetic oil was selected. This oil has a high stability against thermal cracking, low oxidation rate and has small solubility in refrigerants. During 1500 hours of operation the lubricant was stable.

Not only refrigerant/lubricant combinations are studied.
Moreland and Wolf made clear that contact between lubricant and methanol is considered to be unacceptable from a chemical compatibility viewpoint. Therefore a reciprocating and a screw compressor could not be used and a centrifugal unit was made with a very high impeller tip speed (6).

2.2.2.2. Thermophysical properties

High solubility of refrigerants in oils influences to such a high degree the thermophysical properties of the refrigerant that the machines don't work at the design point. Therefore the real dynamic viscosity, density, surface tension, heat transfer coefficient and thermal conductivity are to be taken in consideration by investigators. In the high temperature range only little data about kinematic, dynamic viscosity and solubility is available.

Daniel, Anderson, Schmid and Tokumitsu report on the kinematic viscosity of a number of lubricants. Viscosity data for the mixtures R22/ISO 50Noil; R22/SMC22b and R12/glycoyle 80 is available but only up to 100 °C (21) (18).

On weight basis refrigerants are better soluble in n-oils than in p-oils. In the case of complete miscibility, solubility of refrigerant in oil is mostly expressed in terms of pressure-temperature concentration diagrams. In case of a mixture gap, critical solution temperatures are given (CST). High aromatic contents in oils lower the critical solution temperature (CST). High viscosity oils have a higher CST than low viscosity oils. p-oils have a higher CST than n-oils. In the high temperature range ref. (21) gives p-T-c data up to 100 °C.

2.2.2.3. System aspects

In general the oil acts not only as a lubricant but also as a coolant and sealant in the compressor. However adequate precautions are required to ensure the return to the compressor. Oils also inhibit heat transfer particularly in the evaporator and can result in blocking of the expansion device at low temperatures. To avoid such problems, oil free systems can be considered. They use aerodynamic compressors (centrifugral or axial) only applicable for large systems, or oil free reciprocating or screw compressors. These however are more expensive and have lower efficiencies than lubricated compressors (22).

2.2.2.3.1. Heat transfer

Experimental studies domenstrate that an oil-refrigerant mixture has a higher boiling temperature than that of the pure refrigerant taken at the same pressure (29) (30). Viscosity and surface tension are increased due to the solubility of the oil. This results in different boiling heat transfer characteristics of the refrigerant-oil mixture (29, 30). The refrigerants completely miscible in oil, such as R12 and R11 generally have a drecreasing heat transfer performance with increasing percentages of oil in the mixture. However the heat transfer coefficient can be increased due to the foaming action of the oil for concentrations going from 0 % to 3 %. For R114-oil mixtures with up to 3 or 4 % oil in the refrigerant, an increase in the heat transfer coefficient is experienced. At higher concentrations a decrease is observed.

For oil below 3 % by weight the increase in heat transfer coefficient is small so that design or performance calculations based on oil-free heat transfer data can be made without serious errors (29, 30).

2.2.2.3.2. Lubrication and compressor performance

The type of compressor influence strongly the choice of oil because the lubrication requirements of the different compressor types vary considerably (11) (20).
For reciprocating compressors some particular aspects have to be considered. Foaming of the oil in the crankcase on starting, pumpdown of the compressor when stopping and crankcase heating during shut down should be mentioned here (ref. 17, 21, 23).

In ref. (23) (24) (17) some indications of the precautions to be taken with respect to system control are given. They ensure safe and reliable operation of the systems with condensing temperatures up to 120 °C. Special attention is given to the transient conditions during stopping and starting.

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3. ABSORPTION HEAT PUMPS

3.1. OVERVIEW OF MANUFACTURERS

The best known working fluids for AHPs are $NH_3/H_2O/LiBr$.
The maximum generator temperature for the latter pair is approximately 150 °C. This means that the heat sink temperature has an upper limit slightly above 100 °C. Such heat pumps cannot operate at high temperature conditions and therefore have a limited potential in industry (2).
Aside from this limitation there are other disadvantages such as :
- limitation in temperature lift (40 - 45 °C)
- high investment cost.

To avoid these disadvantages special efforts are made in the way of finding new working pairs for a higher temperature range and improved heat exchangers (cheaper and more compact). An investigation in the Netherlands (3) does not expect very much practical results in the next 10 years concerning new working pairs, from present research efforts. It is suggested to pay more attention to improving the components of the absorption heat pump. This way the AHP will be more competitive with closed cycle electrical or gasmotor driven heat pumps.
The limitation of the sink temperature at approximately 100 °C is the reason why in this study the lower temperature range also is taken into account.
Table 1 gives a listing of presently existing absorption heat pump manufacturers and their available products. The table also lists the number and the application field for absorption heat pumps and heat transformers.

Country/ Manufacturer	AHP (Type I) Q<100 kW	AHP (Type I) Q>100 kW	Heat transformer (Type II)	Installation figures Type I	Installation figures Type II	Applic. field	Remarks
Japan				107*	5*		* 1980–83 [3]
Ebara Co	x	x		6		I, C	
Hitachi Ltd.	x	x	x	3		I	
Hitachi Zosen Co.		x	x		1*	I, C, R	* Export NL in 1985
Kawasaki Heavy Ind.			x		1*	I	* Cooperation with GEA
Mitsubisni - York		x	x			I, C, R	
Tokyo Sanyo Electric	x	x	x	22*	4*	I, C, R	* 1980-84 (installation list)
Yazaki	x						
F.R. Germany							
AWT	x			>200		I, C, R	
Borsig		x		1		I	
GEA			x		2	I	
Linde		x		6		I, C	
Schiedel	x*						* Zeolite/Water
USA							
American Yazaki	x						
Arkla	x	x		2*		I	* Export NL in 1981
Carrier		x		1*		I	* Export NL in 1982
Trane		x		2*		I, C	* Export NL in 1982/83
France							
Collard & Trolart	x*						* available 1987
GDR							
Maschinenfabrik Halle		x		1		I	
Romania		x		3		R	22 units in planning stage

13

Abbreviation: R: Residential C: Commercial I: Industrial

oud A

Table 1. Survey of manufacturers of AHP, products and installation figures (4)

HPC

3.2. APPLICATIONS

The following gives an overview of absorption heat pump applications.

3.2.1. Federal Republic of Germany

Borsig GmbH (Berlin)

- 1 two-stage AHP installed as a demonstration project for district heating (Fernwärmeschiene Saar) (NH_3/H_2O)
- operation start : 1980
- operational data : see table 2
- heat output : ± 2 MW; 15.000 MWh; max. 105 °C

Linde AG (Cologne)

- 6 installed single stage plants (300 kW - 20 MW) in industry and the commercial sector (NH_3/H_2O)
- know-how to produce multistage AHP
- Taylor made designs
- a survey about the most important data up to now is given in table 3 (NH_3/H_2O).

3.2.2. Japan

The heat output range of the AHP and heat transformers for some manufacturers is listed in table 4. More detailed information can be found in table 5.

Production up to 1984 : 107 units were delivered

- one exported to Costa Rica by Ebara Co
- one exported to Sweden (Tokyo Sanyo) for district heating (heat output 7,2 MW)

1985 : by compiling information from 6 manufacturers, 19 installations are applied in the following sectors : (steam driven) spinning mill, paper and pulp industry, dye works, photofilm plant, steel mill, chemical plants. The heat output ranges between 1,74 and 3,62 MW.

Heat source: district heating return flow			
$V = 124\ m^3/h$			
$t_E = 40{,}6\ ^oC$		$\dot{Q}_o$	$= 1240$ MW
$t_A = 32{,}0\ ^oC$			
Drive:	steam $p_D = 12{,}2$ bar		
	$t_D = 188\ ^oC$		
	power input	$\dot{Q}_H$	$= 2{,}825$ MW
	steam output	$\dot{Q}_D$	$= 2{,}315$ MW
	boiler efficiency	n_K	$= 0{,}819$
Use: district heating system			
$V = 80\ m^3/h$			
$t_R = 52{,}2\ ^oC$		$\dot{Q}_C$	$= 1{,}433$ MW (Condenser)
$t_C = 67{,}6\ ^oC$		$\dot{Q}_{A1}$	$= 0.828$ MW (Absorber 1)
$t_{A2} = 76{,}5\ ^oC$		$\dot{Q}_{A2}$	$= 1{,}135$ MW (Absorber 2)
$t_{A1} = 88{,}7\ ^oC$		$\dot{Q}_{LK}$	$= 0{,}158$ MW (Solution cooler)
$t_V = 90{,}4\ ^oC$		$\dot{Q}_N$	$= 3{,}554$ MW
Process parameters:			
heat ratio		$\dot{Q}_O/\dot{Q}_D$	$= 0{,}536$
COP		$\dot{Q}_N/\dot{Q}_H$	$= 1{,}258$

Table 2. Typical operation parameters of the Borsig demonstration plant (4)

Location	Application	Driving medium	Capacity heating/ cooling (kW)	Source	Sink (°C)
Garmisch-Partenkirchen	swimming pool ice rink	n-Butan	7200/1950	ice generation	heating and hot tap water 55/42
Meckenheim	office building space heating air conditioning	natural gas	1750	soil absorber exhaust air waste heat	heating water 57/46
Bad Kreuznach	school space heating	natural gas	389	soil absorber	heating water 58/46
Waiblingen	district heating	sewage gas natural gas	2500	treated sewage water	heating water 65/40
Stuttgart	office building workshop space heating	natural gas	310	ambient air	heating water 50/41
Suhr (CH)	factory building cold-storage depot	hot water	2100/676	waste heat	heating water 40/30

Table 3. AHP plants installed by Linde AG (4)

Manufacturers	AHP (Type I)	AHP (Type II)
Ebara	260–3000	not produced
Hitachi	200–7500	200–4500
Hitachi Zosen	500–7000	500–2500
Kawasaki	not produced	500–3000
Mitsubishi-York	300–9000	150–4500
Tokyo Sanyo	70–8000	300–2100
Japan	70–9000	150–4500

Table 4. Products capacity in manufacturing (heating capacity Mcal/h) (4)

Manufac-turers	Model name	Working pair	Applica-tion	Drive heat	STD or order
Ebara	16JH 16JS	H_2O + LiBr	I	ST,HW	Standard
	16JR		C	TG,LNG	Order made
Hitachi	HAP		I	ST,HW	Standard
	HAP-FS		I	HO.	Order made
Hitachi Zosen	HHP-1W		I	ST,HW	Standard
Mitsubishi-York	ESH-F		R, C, I	ST,HW, WG	Standard
	ESG		R, C, I	TG	Standard
Tokyo Sanyo	AH GH-V		C, I	SW,HW	Standard
	TSA-SUH-B		R, C, I	TG,KE	Standard

Abbreviation:

Application
R: Residential
C: Commercial
I: Industrial

Drive heat
ST: Steam
HW: Hot water
TG: Town gas
LNG: Liquid natural gas
KE: Kerosene
HO: Heavy oil
WG: Waste gas
WST: Waste steam
WHW: Waste hot water

Table 5. Absorption heat pumps (Type I) (4)

3.2.3. USA

7

From ref. (4) one can conclude that there is practically no market for AHP's in the U.S.A.
Only five U.S. units were imported to the Netherlands during 1981-1983 :
- Arkla and Carrier : 3 units (H_2O/LiBr) in cogeneration plants
- Trane : 1 unit (driven by heat from a district heating system)

3.2.4. France

Referring to ref.4 there is no information available about large AHP systems in Fance, but in ref.8 a description about the performance of an AHP is made and various examples of industrial applications are presented and the economics are discussed.

3.2.5. German Democratic Republic

One large steam driven AHP (NH_3/H_2O) in a sewage treatment plant (3 MW output) is installed by VEB Maschinenfabrik Halle.

3.2.6. Rumania

At the present there are three existing plants using the working pairs H_2O/LiBr and applied in district heating using waste water from industry :
- 1983 : a 5,8 MW AHP is installed at Jassy
- 1984 : water/water compression absorption heat pump is installed in Bucarest (8,7 MW heat output) (hot water preparation for district heating, payback period : 5 years)
- 1985 : two 5,8 MW are installed at Pitesti
- 10 units of 2,9 MW and 12 units of 5,8 MW are prepared

3.2.7. Canada

Studies carried out by the National Research Council identified potential industrial sectors where AHP can be successfully applied.

A possible application was identified in the paper drying industry. A design was proposed by Acres International Ltd of Toronto. The system uses hot dry air to dry the wet paper. Moist air is leaving the drying system and fed into the absorber where heat and mass transfer takes place with NaOH. This diluted NaOH is recuperated in a generator which produces low pressure steam and concentrated NaOH which is fed back to the absorber. The water vapor is afterwards condensed and drained. The production of this plant is 400 ton/day. This concept is developed with the aim to reduce the amount of steam used for the extraction of moisture from the wet paper. A payback period of 2,65 years could be achieved. Still at the moment there is no practical installation of the above described principle. Instead of a practical plant, a laboratory model will be developed and investigated.

A second possible application of an AHP was investigated by Stone & Webster Canada Ltd. of Toronto in collaboration with Energy Concepts Company of Annapolis, Maryland. A butanol dewatering tower seems to give the best results when provided with an inverse working absorption heat pump using H_2O/LiBr as working fluid. The generator uses the overhead vapors as heat source and the heat rejected in the absorber is used as bottom heat (120°C) for the column. A potential payback of 2 years was estimated. Subsequent column operation data obtained from plant sources later showed the following errors in the design data :

- only 65 % of the heat of the overhead vapor was available
- the average condensing temperature was 10°C lower than first accepted
- the distillation column vapours contain organic compounds which were incompatible with copper.

The reduction by 65 % of the heat available for the generator of the AHP results in a steam saving in the reboiler from 2500 kg/h to 1600 kg/h. In spite of this the price for the new plant design was three times higher than the previous price. (The only bid was received from Hitachi Zosen Corporation of Japan.) The reason for this higher price can be attributed to the use of titanium tubing in the absorber and generator instead of a copper/nickel alloy and to a larger heat exchanger due to the closer approach temperatures used.

The result of this new investigation shows that the project was uneconomic. Nevertheless the whole study provides some useful guidelines to evaluate other suitable candidate systems. One of the main conclusions was that AHP systems are better integrated in new plants rather than in existing plants.

3.2.8. Netherlands

In the Netherlands an H_2O - LiBr absorption heat pump (type II) was installed in the chemical industry in 1985 to produce steam from a waste steam. This installation consumes 13.800 KW of waste heat and produces 6.400 KW usefull heat (see case studies)

3.2.9. Sweden

At present there are three large absorption heat pumps in operation in district heating systems in Sweden. Two of these have a heat output of 7 MW and one 50 MW.

3.3. RESEARCH AND DEMONSTRATION PROJECTS

The following lists some recent investigation, that are carried out in Europe and in the U.S.A.

3.3.1. Germany (1)

- University of Essen
  - Investigation of promising fluid pairs. Special attention is payed to heat and mass transfer problems, compatibility with materials and stability (especially decomposition of the fluid pairs in the generator).
  - Studies on different combinations of heat transformers and absorption heat pumps. The most promising system will be built (100 kW).
- University of Munchen
  - Studies on different combinations of heat transformers (low temperature range) and absorption heat pumps (high temperature range, using LiBr/H_2O).
  - A study on a discontinuous heat pump for heating and cooling at the same time, using water/zeolites. A heat input higher than 200 °C is possible and a temperature rise of about 100 °C is possible in

one stage. A heat pump with a 10 kWh heat storage is tested during two years.

- Battelle (11)

  -Two stage LiBr/H_2O absorption heat pump producing 100 kW of heat at 130 °C from waste heat at 105 °C (1).

  -A two stage AHP was developed after selection between different heating systems. The following gives a review of the technical data : working pair : H_2O/LiBr
  desorber temperature : 155 °C
  evaporator temperature : 50 °C
  useful achieved temperature level : 110 - 115 °C
  COP : 1,35
  heating capacity : 10 MW
  investment cost : DM 2 million
  operating hours : 8.000 h/year

The effect of different energy prices on the payback period for different countries is investigated. A payback time of less than 3 years is possible in Germany. The assembly of an installation in the paper industry should be started in february 1984.

3.3.2. Netherlands (1)

- Duintjer

  -Development of a cheap plate fin heat exchanger with improved performance.

3.3.3. Italy and France (1)

- CNR (Italy) and CNRS; UTC; CETIAT; BLM (France)

  -Research on solid gas combinations used in periodical operating AHP up to a temperature level of 250 °C (zeolite/water and active coal/methanol).

  -Development of suitable heat exchangers and reactors.

3.3.4. Denmark (1)

- Technical University of Denmark

  -Investigation on NH_3 (refrigerant) in combination with different metal halides (absorbants).

  -Design of a quasi continuous AHP to circumvert the disadvantages of periodic operation of a solid AHP.

  prototypes : -air/air heat pump for energy recovery in a drying process
  -a 2,3 kW two-stage heat transformer which produces heat at 300 °C.

3.3.5. France (1)

- ING

  -Investigation on NH_3 (refrigerants) in combination with graphite containing metallic salts which may have a better performance than metal halides and NH_3.

- Gaz de France and Creusot-Loire (7)

  -Investigation on a 100 kW prototype industrial high temperature absorption heat pump.
  Tests were carried out at evaporating temperatures of 55-65 °C and useful temperatures of 120-130 °C with an obtained COP between 1,20 and 1,56. This experimental plant is a representative design of the real absorption heat pump which would be installed in a paper mill drier.
  The evaporator takes the necessary heat from the hot and humid air coming out of the drier extraction hoods. The generator is heated by the combustion of gas and the useful heat is obtained on the condensor and absorber side to produce steam from water which would be sent inside the drying cylinders (figs. 2, 3). The fluid mixture used is water-lithium-bromide pair together with an efficient corrosion inhibitor type A42 which was chosen to be the best economic solution. Tests carried out at the CREUSOT-LOIRE corrosion testing laboratory have shown that even with very low oxygen contents

austenitic stainless steels type 304 L or 316 L suffer, in the presence of a LiBr solution, from intergranular pitting and stress corrosion.

3.3.6. Belgium (10)

A single stage absorption heat pump is developed which is able to upgrade industrial waste heat above 120 °C. From literature reviews tiffluorethanol and chinolin is chosen. A computer model is developed incorporating the latest heat transfer augmentation techniques. Based on heat and mass transfer models applied to the different components the results show a reduction of the heat exchanger areas of 30 %.
From the practical side TFE and chinolin have very good properties compared to H_2O/LiBr (high COP and no cristallization problems). A component size and cost analysis is available since 1984.

3.3.7. USA

- An absorption heat pump design has been made for upgrading industrial waste heat from 60 °C to 120 °C (9). The first step was the construction of a laboratory scale system. A reduced power prototype was built and succesfully operated to demonstrate the possibility of using relatively low waste heat temperatures to produce industrial process steam. The prototype single-stage system, using H_2O/LiBr as working pair, was succesfully operated and demonstrated close agreement with the computer model

- The objective of the project carried out by Energy Concept Co was to locate a retrofit site and bring one of the first absorption heat pumps in use for directly recycling distillation reject heat (6). The process uses overhead vapor heat upgraded by an absorption heat pump.
  The first step was to find a suitable application for the AHP augmented distillation column prototype. A preliminary design and a cost estimation is prepared.
  The second step was the assembly of the prototype. Its performance was monitored for 6 months. Afterwards several candidate sites were identified and preliminary designs and cost estimates have been prepared.

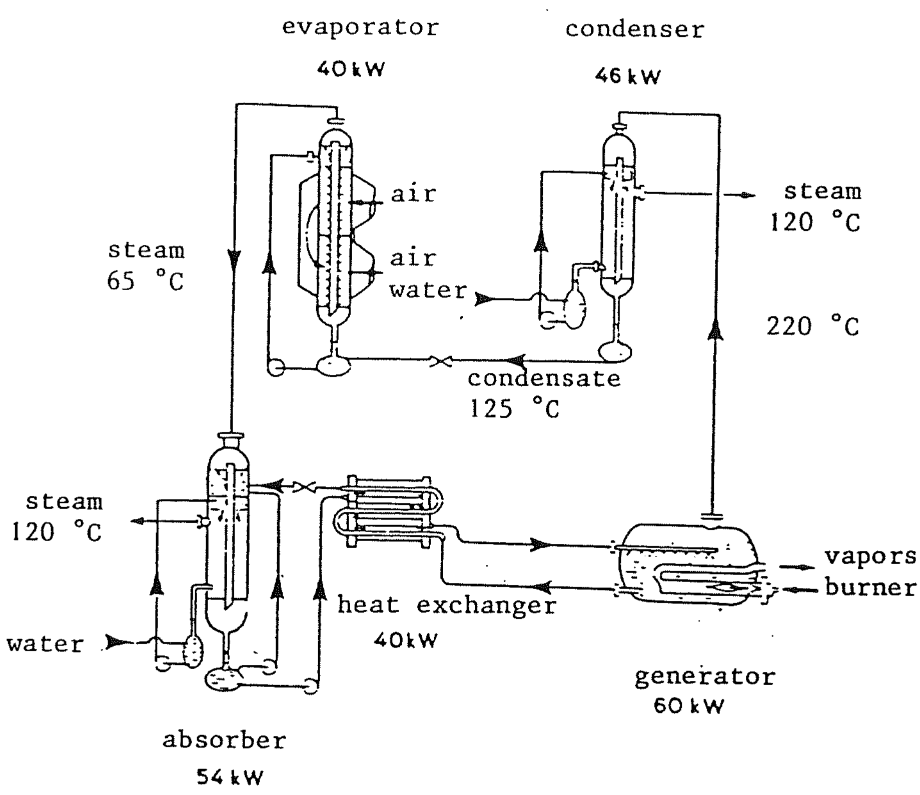

Figure 2 : Components of the 100 kW prototype absorption heat pump

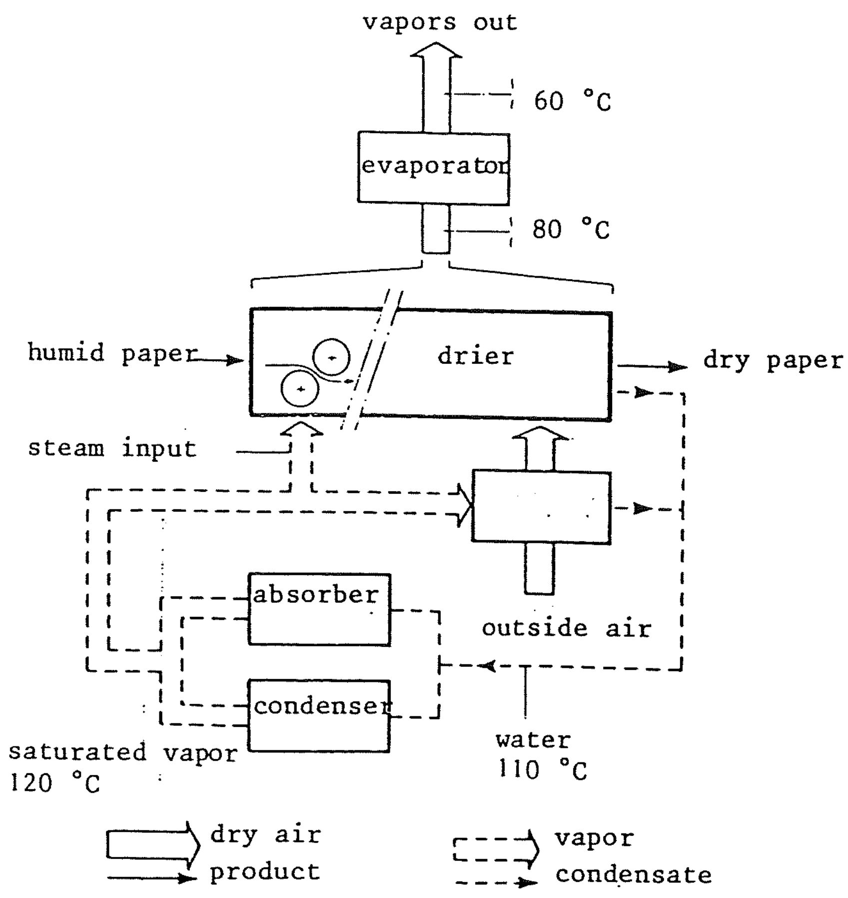

Figure 3 : Adaption of the absorption heat pump to the paper drier

3.3.8. Working Pairs

The choice of solute solvent working pairs to be used in the heat pump is crucial to the energetic performance of the heat pump, the size of its components and thus its economic success. For these reasons many extensive studies on suitable working pairs have been made, especially to develop a heat pump for industrial application, operating at high temperatures. Thermodynamic as well as thermal properties of solute and solvent and their solution have to be available in order to calculate the performance of the heat pump and the size of the components. Therefore it is necessary to perform experiments in order to gather sufficient properties to make heat pump design possible. Ref. 10 aims to develop an industrial heat pump which can recuperate waste heat at temperature levels from 90 °C to 120 °C and which can pump this heat up over a 30 °C temperature increase. Obviously the temperature of the driving heat should be in the neighbourhood of 200 °C. The chosen working pair in this study was TFE/quinoline.

Higher condenser and absorber temperatures can also be achieved by a two-stage heat pump cycle (ref. 12). Two-stage heat pumps are more complicated to realize and to design. They are definitely more expensive and would be difficult to justify economically. Ref. 13 investigates the use of different high temperature working pairs in a two-stage absorption heat pump. In the first stage NH_3/H_2O was used. The second stage working pairs were considered in detail. The result was that the best working pair is hexafluor-isopropanol in quinoline.

> Energy concepts C_0 was contracted by Oak Ridge National Laboratory (14) to develop a working pair capable to work at higher operating temperatures and higher temperature lifts than LiBr. As a result they found a ternary absorption working pair of $LiNO_3$, K NO_3 and $NaNO_3$ in H_2O, stable above 260 °C. Corrosion tests have been carried out between 120 °C and 315 °C and show a slight corrosion in the case of mild steel exposed to vapor above 220 °C. Mildly alloyed steel (2,5 / Cr) will be corrosion resistant in the vapor space up to extreme temperatures. The maximum temperature lift that can be achieved in the temperature range of 50 °C to 230 °C is 58 °C in the case of a heat transformer and 90 °C in the case of a temperature amplifier.

The high temperature absorbent has a 2,2 to 3,3 % better COP than LiBr. One may conclude that this new working pair will not replace LiBr in all the industrial installation due to the good and well known properties of LiBr. In those cases where high temperature lifts and high temperature working conditions are required this ternary absorption working pair can be used.

For binary systems, the two most used working pairs are $LiBr/H_2O$ and H_2O/NH_3. Both have their limits. The corrosivity of $LiBr/H_2O$ in connection with O_2 (subatmospheric working) limits its application up to 180 °C. NH_3 is poisonous and generates high pressures even with low temperatures. The high pressure limits the output temperature to 80 °C. The conclusion is that for high temperature applications only $LiBr/H_2O$ can be used, but there are a lot of disadvantages. Looking for better stability, pressure and corrosion properties, the working pair H_2O/zeolite is stable up to 180 °C (15), allows a great temperature difference, high COP, high solubility, high temperature by normal pressure (< 10 bar) and is not corrosive (15). This working pair can only be used in discontinuous absorption heat pumps. E181/Trifluorethanol is also stable up to 230 °C. The disadvantages are high viscosity and a low evaporating heat.
With the working pair $NaOH/H_2O$ temperatures above 250 °C can be realised but NaOH is very corrosive in this temperature range (16). This working fluid is under investigation in Japan and is not applicable for temperatures below 80 °C. Extremely low pressures occur in this temperature range.

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4. HEAT TRANSFORMERS

The application of heat transformers already having been covered in section 3.2, attention will be given here only to R & D activities.

4.1. RESEARCH ACTIVITIES

Most part of the research related to heat transformers concerns :
- expanding the operating temperature range (high output temperature and high temperature lift);
- improving the energy efficiency;
- matching source and sink for total system optimization.

In Germany, GEA has started a development project featuring the development of advanced heat transformers utilizing new working pairs and multi-stage cycles. Figures 1, 2, 3, 4 and 5 make clear the different possibilities worked out or being developed by GEA. Figures 2 and 3 show the design data of a TFE-DTG and a TFE-pyrrodine heat transformer. The results obtained at GEA are very promising.
Figure 5 shows the combination of a heat transformer and a high temperature heat pump using new working fluids. The problem is that high-grade heat must be supplied at a temperature level of about 150 to 200 °C in such a case.
Investigations will be made under the ECC R & D program and in cooperation with French companies under the EUREKA-Project (3).
Table 1 gives an overview of the ongoing collaboration.

A research team of Essen University undertook the task of finding new working fluids for absorption plants. Although the research program concerns new working pairs for absorption heat pumps mainly, some of the working fluids are also suitable for use in heat transformers. Under the project some 150 different working fluids were investigated with respect to vapor pressure, density, viscosity, solubility, thermal stability, phase transitions, enthalpy of mixture and specific thermal capacity. The thermodynamic properties of the working pairs PFPA-DTG and HFIP-NMP make them seem especially suitable for being used in heat transformers. Table 2 and 3 list solubility data, general information and a comparison at different working conditions (5).

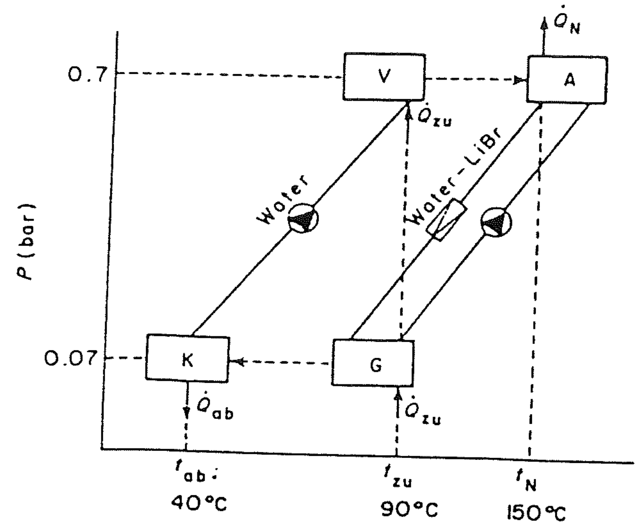

Figure 1 Design characteristics of a water–LiBr heat transformer. V, Evaporator; K, condenser; A, absorber; G, generator; $\dot{Q}_N$, usable heat flow at t_N; $\dot{Q}_{zu}$, heat input at t_{zu}; $\dot{Q}_{ab}$, waste heat output at t_{ab} [3]

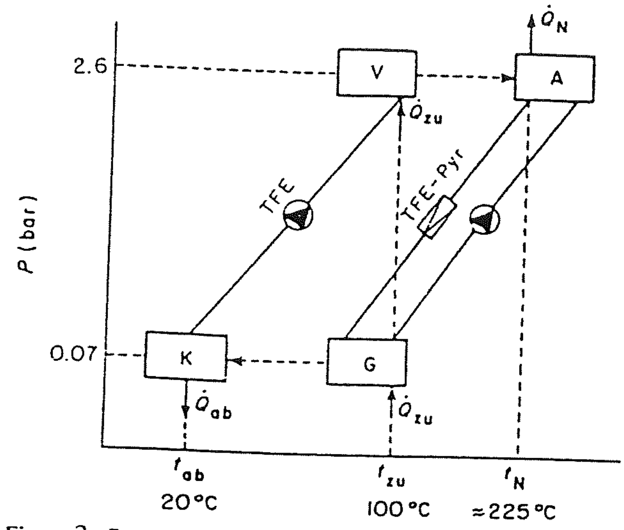

Figure 2 Design characteristics of a TFE–Pyr heat transformer. [3]

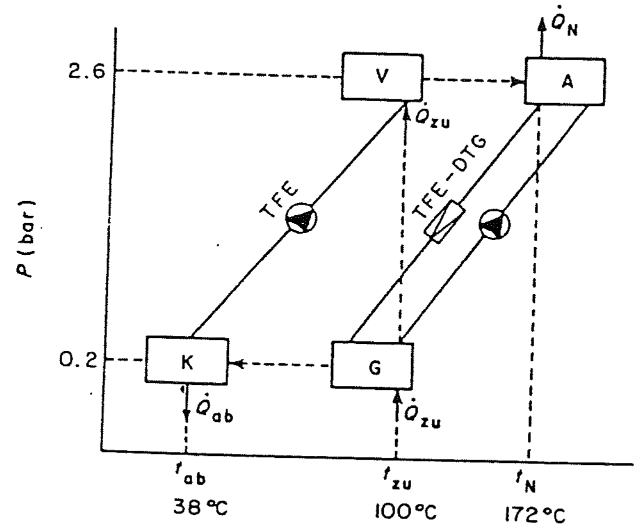

Figure 3 Design characteristics of a TFE-DTG heat transformer [3]

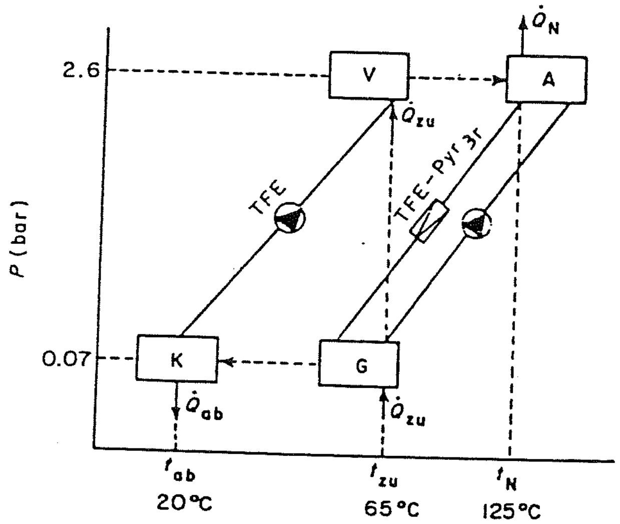

Figure 4 Single stage TFE–Pyr heat transformer for a temperature lift from 70 to 110°C. [3]

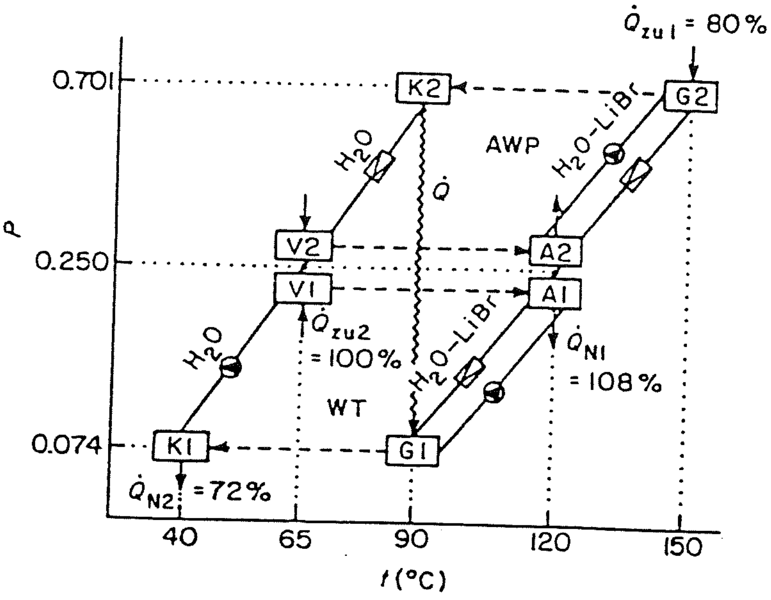

Figure 5 Combination of a H_2O–LiBr heat transformer with a H_2O–LiBr heat pump for the temperature lift from 70 to 110°C. AWP, Absorption heat pump; WT, heat transformer. [3]

THE HEAT TRANSFORMER

Working Fluid Water - LiBr	Working Fluid Water - LiBr	Working Fluid Organic Substances	Working Fluid Organic Substances
Single - Stage Cycle	Multi - Stage Cycle	Single - Stage Cycle	Combination of a Heat Transformer and an Absorption Heat Pump
Heat Input Temperature 70 ... 120°C	Heat Input Temperature 50 .... 120°C	Heat Input Temperature 50 ... 150°C	Heat Input Temperature e.g. 100°C
Temperature Raise 30 ... 50K	Temperature Raise > 50K	Temperature Raise > 50K	Temperature Raise e.g. 50K
Max Heat Output Temperature 150°C	Max. Heat Output Temperature 150°C	Max Heat Output Temperature > 200K	Heat Output Temperature 1. 90°C 2. 150°C
ON SALE	RESEARCH AND DEVELOPMENT STAGE		
EEC - Demonstration Project "Heat Transformer"	Eureka Project "PACA"	EEC - Research and Development Project "Advanced Heat Transformer"	

Table 1: State-of-the-art of heat transformer

System	Evaporation temperature (°C)	Refrigerant content in the solution at various absorption temperatures															
		170	160	150	140	130	120	110	100	90	80	70	60	50	40	30	20°C
TFE-Chi	0										0.003	0.054	0.107	0.197	0.287	0.392	0.494
	50			0.045	0.070	0.108	0.152	0.214	0.284	0.365	0.450	0.542					
TFE-DTG	0										0.029	0.052	0.091	0.153	0.247	0.369	0.507
	50			0.037	0.060	0.094	0.132	0.187	0.262	0.355	0.457	0.574					
TFE-EP	0										0.117	0.194	0.301	0.385	0.461	0.546	
	50			0.132	0.211	0.285	0.352	0.414	0.480	0.540	0.586	0.667					
TFE-ICh	0										0.005	0.061	0.109	0.198	0.295	0.396	
	50			0.062	0.090	0.126	0.175	0.239	0.312	0.390	0.463	0.549					
TFE-MP	0										0.093	0.144	0.216	0.298	0.390	0.485	0.588
	50			0.153	0.195	0.254	0.309	0.366	0.434	0.505	0.585	0.674					
TFE-NMP	0											0.220	0.328	0.423	0.500	0.572	0.649
	50			0.140	0.255	0.298	0.366	0.423	0.515	0.566	0.634	0.700					
TFE-Pyr	0										0.092	0.151	0.236	0.323	0.419	0.520	0.627
	50			0.153	0.200	0.257	0.320	0.379	0.449	0.534	0.617	0.703					
TFE-TEG	0										0.025	0.043	0.073	0.122	0.203	0.322	0.468
	50			0.052	0.066	0.090	0.122	0.166	0.230	0.315	0.410	0.527					
HFIP-NMP	10									0.629	0.664	0.700	0.732	0.766	0.802	0.839	
	40				0.605	0.637	0.666	0.697	0.728	0.750	0.781	0.811	0.846				
PFPA-DTG	0										0.519	0.630	0.676	0.720	0.755	0.794	
	50	0.140	0.258	0.464	0.525	0.601	0.649	0.691	0.727	0.762	0.795	0.833					

Table 2 : Solubility data [7]

	Working Fluid	General									
		r kJ/kg	t_{kr} °C	p_{20} bar	p_{100} bar	Price DM/kg sol	Tox	Korr.	Δt_m	Rect.	Data
1.	TFE-DTG	449	250	0,07	2,3	28	yes	no	201	no	--
2.	TFE-NMP	449	250	0,07	2,3	27	yes	no	129	(yes)	-
3.	TFE-Pyr	449	250	0,07	2,3	25	yes	no	171	(no)	--
4.	HFIP-NMP	429	195	0,16	5,5	(2500)	less(asspt.)	(no)	145	(yes)	-
5.	PFPA-DTG	(255)	?	0,03	ca.1	295	less(as spt.)	yes Inhibitor	179	(no)	--
6.	NH_3-H_2O	1260	132	8,6	63	2	yes	partly	133	yes	+
7.	H_2O-LiBr	2500	374	0,02	1	25	no	yes Inhibitor	-	no	+

Comparison at different Working Conditions

	$t_E = 100$ °C $t_k = 20$ °C				$t_E = 100$ °C $t_k = 50$ °C				$t_E = 50$ °C $t_k = 20$ °C			
	ξ_a	t_a	Δt_a	f	ξ_a	t_a	Δt_a	f	ξ_a	t_a	Δt_a	f
1.	0,05	185	20	19	0,18	155	8	16	0,05	128	15	19
2.	0,20	ca. 210	ca. 12	16	0,52	160	10	10	0,20	1,38	5	16
3.	0,15	ca. 225	ca. 20	17	0,45	165	15	11	0,15	140	10	17
4.	(0,65)	(220)	(10)	(7)	0,82	160	8	4	(0,65)	130	12	7
5.	(0.57)	(185)	(10)	(8,5)	(0.72)	(150)	(10)	(6)	(0.57)	(125)	(10)	(8,5)
6.	-	-	-	-	-	-	-	-	0,28	125	10	14
7.	-	-	-	-	0,38	146	14	12	-	-	-	-

Table 3 : Working Fluids for Heat Transformers [3]

Another pair suitable for high temperatures is $NaOH/H_2O$. Temperatures of 250°C can be realised. Research on this topic is done by a Japanese university. It is not suitable for temperatures lower then 80°C due to the low pressures (3).

At Chalmers University of Technology in Sweden, research is done on a laboratory-sized regenerator. This regenerator works according to the falling film principle. The advantages with this type are :

- high heat and mass transfer coefficients;
- possibilities to have countercurrent flow in the regenerator which is important if low temperature waste heat is to be used (2, 3, 4).

At Lund University in Sweden a concept with self-circulation is being investigated (11).

An experimental study is made at the chemical laboratories of the University of Nancy on the enhancement of heat and mass transfer in falling film evaporators and absorbers by turbulence promotors (4).

The performance of a composite absorber is studied at E.N.S.I.G.C., Toulouse, France. The absorber consists of a spraying zone and a falling film zone developed on industrial curregated tubes (3, 4).

At the University of Munich a combination of a heat transformer for the low part and an absorption heat pump for the high temperature part is investigated. As a working pair $LiBr/H_2O$ is used. This combination is interesting as it combines the advantages of providing sufficient heat at high temperature with the ability to also use all waste heat from the process at lower temperatures (1, 3, 4).
A two-stage heat transformer is constructed by Battelle, Germany. The working pair is $LiBr/H_2O$ and 100 kW of heat at 130 °C will be produced from waste heat at 105 °C (3).

4.2. INDUSTRIAL CONSIDERATIONS

In 1980 GEA started with research activities concerning heat transformers. In 1984 the first heat transformer was installed in Dörnten (Germany). Later on three more transformers were installed. Table 4 lists the four installations already in operation or being installed (8).

Location	Dörnten (FRG)	Stuttgart (FRG)	Yugoslavia	Wesseling (FRG)
first year of operation	'84	'86	'86	'88
capacity (MW)	1	1,4	1,1	2
temperature of waste heat	100°	100°	100°	97°
temperature of the up-graded fluid	145°	136°	144°	133°
industry	slaughtery	brewery	slaughtery	chemical

Table 4 :HTHT in operation or being installed

The heat transformers installed by GEA nowadays achieve a maximum output temperature of 150 °C, a temperature lift of 50 K and a COP of 0,48 (8). In the future working pairs PFPA-DTG (pentafluoropropionic acid - tetraethylene glycol dimethylether) and HFIP-NMP (hexafluoroisopropanol-N-methylpyrrolidone) may be used in heat transformers. However the cost at the moment ('87-'88) is too high to ensure a profitable use. The highest temperature for single-stage heat transformers is in the range of 150 °C. It should be possible to develop a "topping" MVR system recompressing the steam produced by the transformer. This system would make it possible to reach higher temperatures than 150 °C.

The double stage heat transformer is 10 % to 20 % more expensive than the single stage version. The double-stage heat pump/heat transformer is 30 % more expensive than the single stage system (8). The prices nowadays are about 650.000 DM for a 1 MW output single stage heat transformer (8).

> The characteristics of the heat transformer installation at the Degussa plant are interesting..

- 2 MW-5 MW output of 3,3 t/hour of steam at a pressure of 3 bar
- purchase cost : 1,2 million DM
- installation cost : 0,5 to 0,8 million DM
- running time : 7500 hours/year
- electricity price : 0,13 DM/kWh
- insurance control system : 50.000 per year
- cost of reparations : 3 % of the investment
- waste heat : 80 - 100 °C
- heat demand : 110° - 140°C
- temperature of the upgraded fluid : 133 °C

Using this data and taking into account that at Degussa a 10 % to 20 % net return on investment is required as a minimum for energy conservation investments the steam price must be 25 to 30 DM/ton steam. The steam cost nowadays is in the range of 18 DM/ton (9).

Table 5 presents a review made by Batelle (reparation of heat transformers) concerning the activities in the field of heat transformers.

In October '85 Hitachi installed a high temperature heat transformer at the Akzo Zoutchemie plant in Delfzijl in the Netherlands (10). Figure 6 shows a simplified flow scheme of the heat transformer. The design values and the measured values (table 6) at full load conditions during a 48 hour test are reported. The aim is to obtain low pressure steam at 150 °C, a pressure of 4,76 bar and a flow rate of 11 ton/hour.

In October (85 Hitachi installed a high temperature heat transformer at the Akzo Zoutchemie plant in Delfzijl in the Netherlands (10). This installation is described further in part III of this report.

Table 5 : A review made by Batelle concerning the activities in the field of heat transformers (9).

Company	working fluid	capacity	year of installation	sector of installation
Hitachi Zosen	LiBr-H_2O	5x60 kW - 3,3 kW	'79 - '83	chemical ind. (Japan)
		6,4 MW	'86	Akzo Zoutchemie - (the Netherlands) - chemicals
Sanyo	LiBr-H_2O	2,3 MW	'81	Japan plactics
GEA	LiBr-H_2O	1 MW°, 1,4 MW°, 1,1 MW 2 MW	'84 - '86 '86 '87	slaughtery brewery Degussa (FRG) (chemicals)
KRUPP	NH_3 - H_2O TFE-E 181	stopped the development		
AEG	H_2SO_4-H_2O	250 kW	'87	paper indust. (FRG)
Thermo consulting Heidelberg	NH_3 - H_2O	based on resorption principle		
Batelle	LiBr-H_2O	50 kW	'86	plastics industry (FRG)

Other companies working in the field of heat transformers are : Seebeck, Techno Product, Voest, Borsig, BBC Mitsubishi. However no further detailed information is available on their activities.

5. OZONE AND GREEN HOUSE EFFECTS

The depletion of the ozone layer and the green house effect present a challenge to high temperature heat pumps. In particular the closed cycle compression systems using R-114 are threatened as their working fluid is one of those that have to be fased out in the near future. Existing absorption systems are not threatened as their working pairs do not have an impact on the ozone and green house effects.

Obviously these two environmental phenomena have to be taken into consideration in the search for new working fluids.

R-114 has an ozone depletion factor of 0,6 and a greenhouse effect impact factor of up to 1,5, which are rather high values (R-12 has 1,0 for each case). The market phaze of R114 however is 20 times smaller than that of R12 in Europe. In the U.S. it is fifty times lower.

R143 (CHF_2CH_2F) is being considered as a replacement of R114 together with R142b. The latter however turns out to be flammable.

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