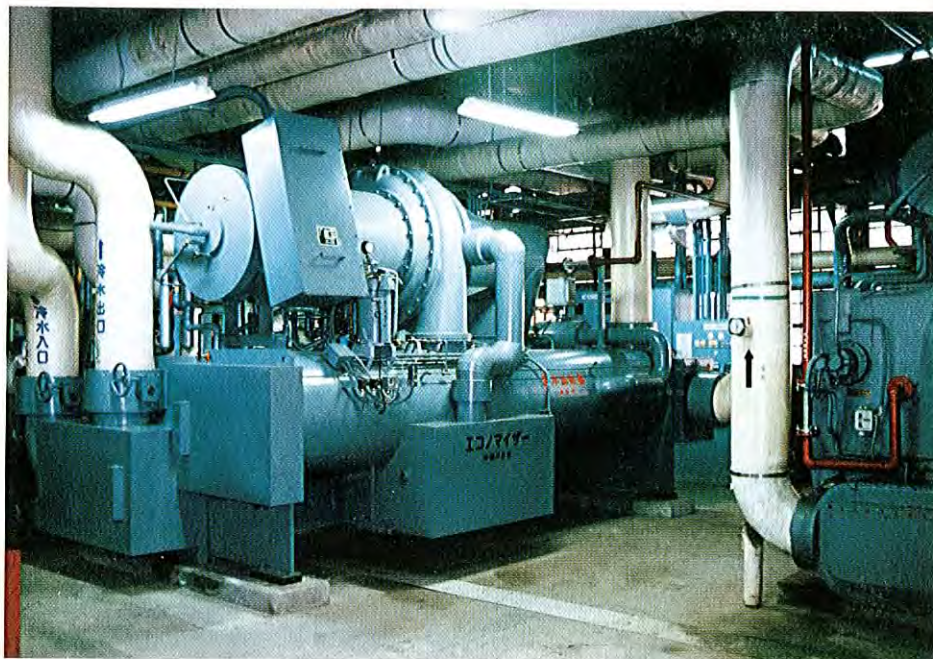


NEWS LETTER

PERIODICAL OF THE
IEA HEAT PUMP CENTER

IEA  heat pump
center

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Centrifugal heat pump (see page 8)

S. Richlen*

The U.S. Department of Energy's Industrial Heat Pump Program

The Industrial Heat Pump Program is one of a number of research programs being conducted by the U.S. Department of Energy's (DOE) Office of Industrial Programs in Washington, D.C. The Office of Industrial Programs has been assigned by the U.S. government to conduct R&D activities that are directed toward improving energy use efficiency in industry. These activities are to be complementary and non-redundant to private sector activities. Programmatic goals of the Office are to: (1) increase the end-use energy efficiency of industrial and agricultural operations; and (2) expand the energy options for manufacturing processes by providing technologies which utilize all types of fuels such as coal, renewables, oil, and natural gas. The emphasis of these activities is on long range, high risk research which will result in high energy savings.

A major program conducted by the Office of Industrial Programs is the Industrial Heat Pump Program. The goal of

the Program is to advance the technology available for designing, fabricating, and using heat pumps to increase the

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energy use efficiency in industry. This generally does not involve space heating, but rather using heat pumps in a role of both increasing the process efficiencies and recovering and reusing waste energy emitted by the process.

In the past, in pursuit of this goal, the Industrial Heat Pump Program has funded studies and research for the development of new hardware. In one case, the Brayton Cycle Solvent Recovery System, development has successfully proceeded through design, hardware development, and prototype testing at the laboratory (see IEA Heat Pump Center Newsletter Vol 4 No. 2, *Brayton Cycle Solvent Recovery Heat Pump*). The system is presently installed at 3M's Hutchinson, Minnesota plant where it is scheduled to go into service. It appears there will be further uses and installations of this type of equipment.

Presently, efforts in the Program are divided between identifying promising applications for the heat pump in industry and developing chemical heat pump technologies.

Recent review meetings and discussions with industry and utility representatives have revealed an interest in the industrial heat pump but also a concern that information about heat pumps, their performance under industrial conditions, and their effect on plant operation was not readily available. To address this need, DOE is gathering information on a world-wide basis of industrial heat pumps and their use. It is expected that DOE's participation in IEA Heat Pump Annex IX will contribute to this effort.

To support the distribution of information, the Industrial Heat Pump Program hosted a meeting at the Department of Commerce to discuss the development of an information network among electric utility companies for industrial heat pump technology transfer.

As a result of this meeting, the Industrial Heat Pump Advisory Group (IHPAG) was formed under the accommodation of Edison Electric Institute and the Electrification Council. Since that time, the

IHPAG has planned an effort of organization, training, and promotion to further the use of the heat pump in industry.

Chemical heat pump technology development has received a great deal of support from the Program. Recent efforts have included the design and fabrication of a high temperature absorption heat pump pilot plant using a ternary nitrate working pair (see IEA Heat Pump Center Newsletter Vol 4, No. 3, *260°C Aqueous Absorption Working Pair Under Development*). The pilot plant is expected to begin test runs during the fall of 1987. Continued testing and improvement of the ternary nitrate mixture is expected to provide information on behavior, operation, durability, performance, and problems.

A new research area expected to support chemical heat pump development is the measurement of thermochemical and thermophysical properties of advanced absorption fluids. Heat and mass transfer rates of absorption of re-

frigerant by sorbent will also be measured.

DOE's Industrial Heat Pump Program has developed an approach that successfully identifies the kinds of heat pump developments that are required for industrial applications ranging from process heating to process enhancement. Current activities are focusing upon identification of additional industrial applications that are appropriate for heat pumps and the subsequent development of advanced technologies responsive to these applications. Chemical heat pump research is developing the information needed to facilitate the development and implementation of a workable technology within industry. Finally, it appears that utilities will play a central role as facilitators of market penetration of both conventional and advanced industrial heat pump technology.

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N.V. Nguyen*

Steam Compression with Screw Compressors

In many factories, heat is lost in the form of waste steam. By compression of the waste steam, it is possible to produce process steam in an energy-saving way. Process steam is usually produced by boiler plants. The energy content of the boiler feed water is low. Thus, a great deal of heat is needed to produce process steam. Only a small part of this heat is used to raise the temperature. The biggest part is needed to evaporate the water. If that waste steam is available, the process steam can be produced by mechanical compression with only a small consumption of energy. The energy content of the waste steam is already very high, therefore only a small quantity of energy is needed to produce process steam.

Pilot plant

In 1983, a pilot plant with a screw compressor was put into operation in the chemical factory of Hoechst AG, Gersthofen, near Munich (Figure 1). The steam supply for this factory contains three networks with working pressures of 3.5, 9, and 20 bar. Recovery of waste

heat generates much waste steam during the production of the textile fiber. This waste steam is transferred into the 3.5 bar network exceeding the quantity needed.

To take advantage of the excess steam, it is compressed to 9 bar. The screw compressor used is a dry running proc-



Figure 1. Pilot plant for steam compression

ess gas compressor of Aerezner Maschin-fabrik with special seal construction. The prime mover is a speed-controlled electromotor with 650 kW driving power.

By varying the motor speed from 1500 to 3000 rpm, the output of steam can be adapted in the required range of 2 to 7.5 t/h. To avoid superheating, the temperature of the compressed steam is controlled at 190°C by water injection at the suction side. The pilot plant is equipped with an automatic control, which guarantees that the proper amount of excess steam needed is compressed as process steam of 9 bar.

The specific energy consumption is 75 to 85 kWh electrical energy per ton of process steam. In contrast, in a boiler plant, 700 to 750 kWh primary energy per ton is required. With 2000 to 4000 operation hours annually, the break-even point of this pilot plant is one to two years.

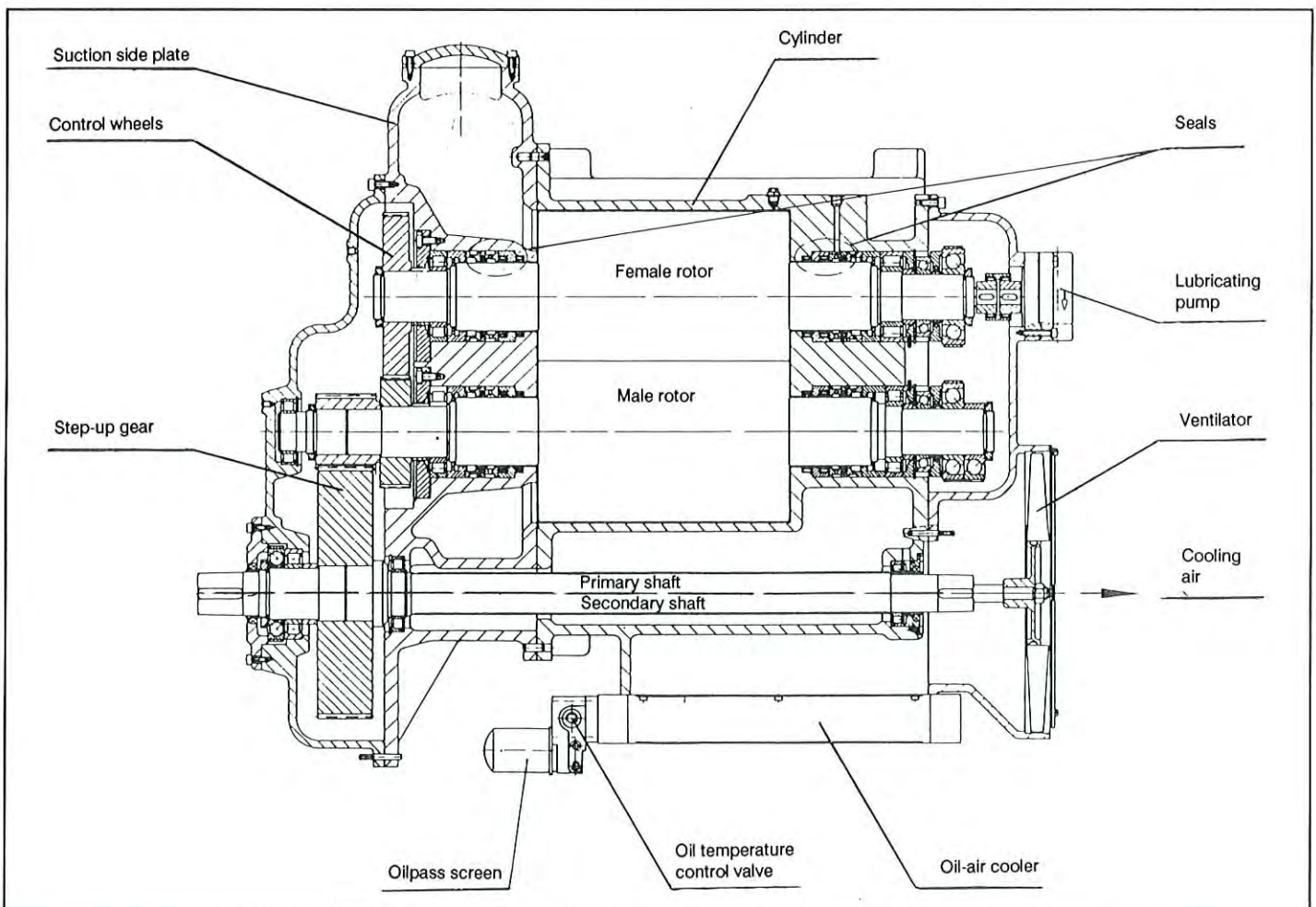


Figure 2. Construction of screw compressors of line VM

Test of low-priced compressors

In the above-mentioned case, waste steam has a pressure of 3.5 bar and a density of nearly 2 kg/m³. In most cases, waste steam of about 1 bar at a density lower than 0.6 kg/m³ is available. To produce the same quantity of steam, the volumetric flow must be more than three times greater. This means a bigger screw compressor with considerably higher costs of investment.

To save costs of investment, low-priced dry running screw compressors (type VM by Aerzener Maschinenfabrik, Figure 2) to generate oil-free compressed air up to 3.5 bar above atmospheric were modified for steam compression by integration of a water injection device and by variation of the labyrinth box.

A test stand for steam compression was installed at the SEP GmbH, Munich. The compressors are cooled by water injection. The water flow is controlled by the steam temperature after compression. The labyrinth box had to be modified in order to make sure that no condensed steam combined with the lubricating oil. To obtain higher pressure ratios for steam compression, two compressors were connected in series (this is possible without problems for the line VM by a common primary shaft). Pressure ratios between 2 and 10 with suction pressures ranging from 0.9 to 3.6 bar were investigated at various speeds in the range between 1500 to 3000 rpm.

Results and conclusions

The quantities of steam and water injection due to various pressure ratios are shown in Figure 3 for the screw compressor VM 137. In Figure 4 the volumetric efficiency and in Figure 5 the specific energy consumption are shown for various compression loads.

Keeping in mind that the price ratio between primary energy and electrical or mechanical energy is 2 to 3 and that approximately 750 kWh of primary energy are used to produce one ton of steam in a boiler plant, steam compres-

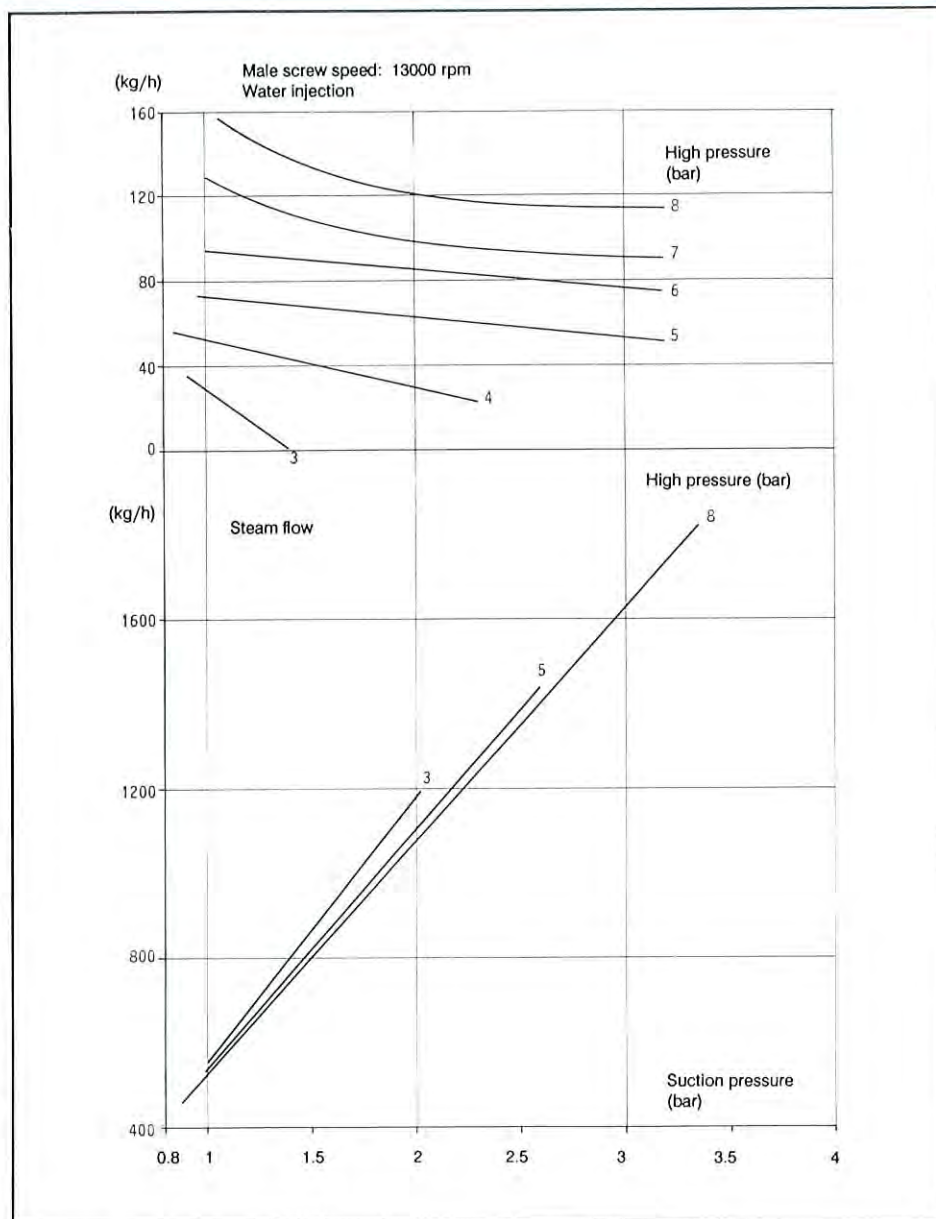


Figure 3. Steam flow and water injection of the screw compressor (VM 137)

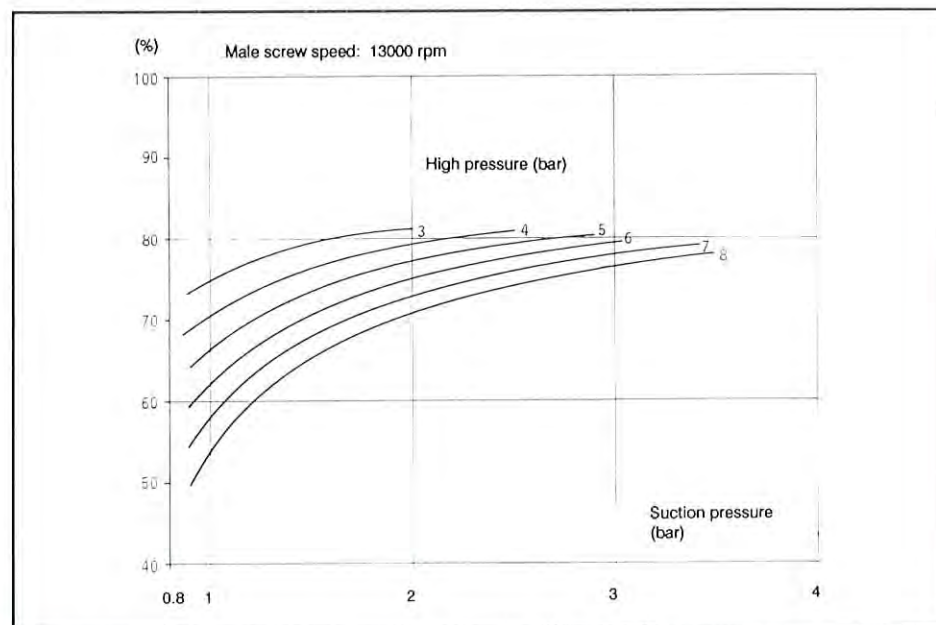


Figure 4. Volumetric efficiency of the screw compressor (VM 137)

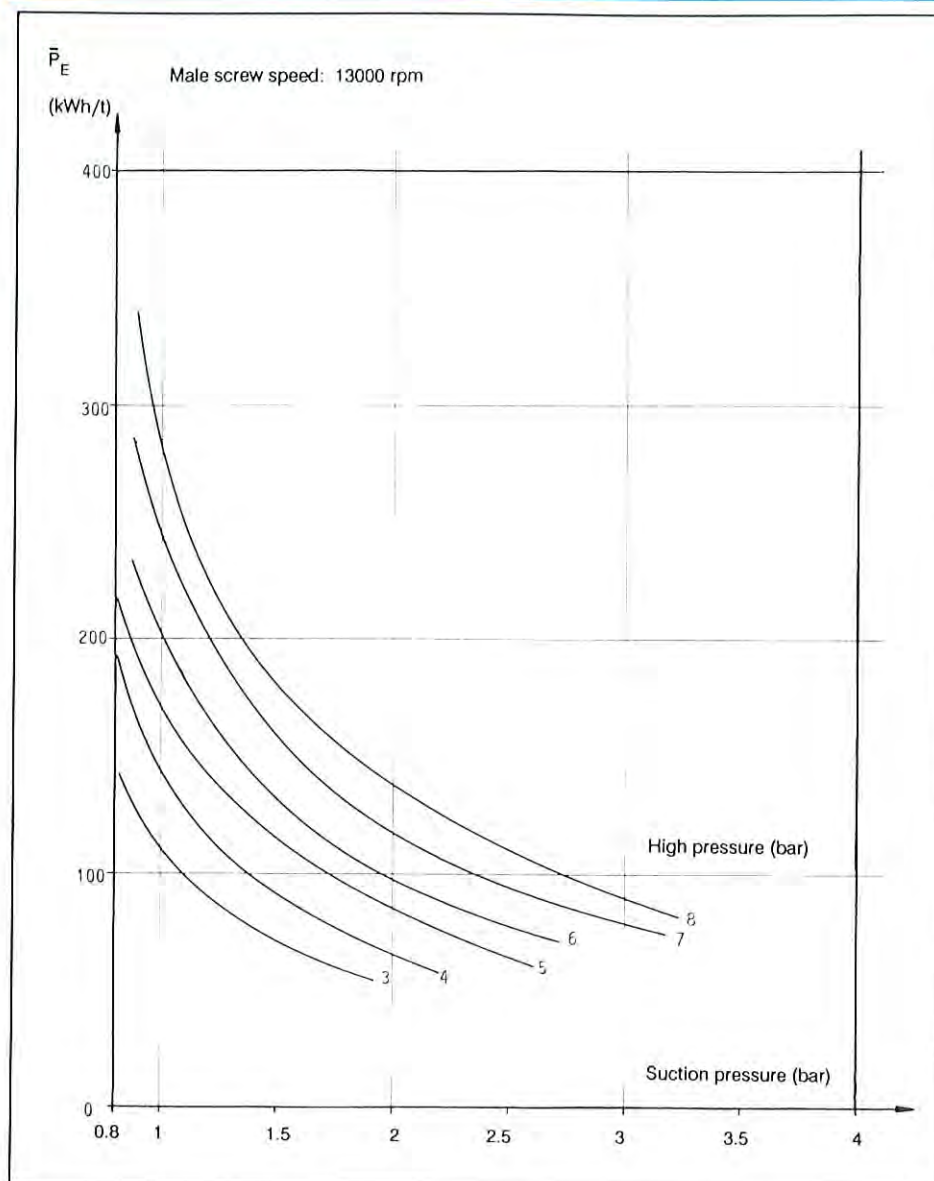


Figure 5. Specific energy consumption of the screw compressor (VM 137)

sion is no longer profitable when the specific energy consumption exceeds 250 to 350 kWh. Figure 5 shows, therefore, that the pressure ratio between high and low pressure should not exceed 7 to 8.

With respect to the costs of investment, in most cases the application pressure ratio shall be lower than 4 to 5. In many cases, as boiling or evaporation processes waste steam has a temperature of approximately 95°C and process steam is required at about 120°C, pressure ratio is about 2.5 and steam compression may be a profitable alternative.

Until now, turbo compressors and screw compressors were used for steam compression. Screw compressors may be used mainly at higher pressure ratios or at smaller flows typical for turbo compressors and/or if a speed-controlled flow variation is needed independently of the pressure ratio.

This project received financial support from the Ministry of Research and Technology of the Federal Republic of Germany.

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T. Hirabayashi*

An Example of Heat Pump Utilization at a Photo Film Manufacturing Plant

As a major plant of Konishiroku Photo Ind. Co., Ltd., Hino Plant is producing photosensitive materials, including photographic film. Yearly total energy (fuel and electric power) purchased for consumption at this plant amounts to 33,700m³ in terms of heavy oil. Since 1980, steps have been taken to reduce energy consumption. The first efforts achieved significant improvements in efficient energy utilization in production processes. In the second phase, attention was focused on the cost of supplied energy, and considerable energy cost reduction was achieved. Introduced below is an example of energy saving accomplished by installing an electric heat pump with a power input of 855 kW for simultaneous supply of process heat and cooling.

1. Energy consumption structure of the plant

At this photo film manufacturing plant, the majority of energy is required for the photosensitive material coating and drying process. This process is shown in Figure 1.

1.1 Energy flow (Figure 2)

The film is dried by hot, dry air; therefore, the three steps, cooling, dehumidifying, and reheating, are necessary to condition the circulating air.

Since the film drying speed is proportional to Δt (difference between dry-bulb and wet-bulb temperatures) of air for drying, rapid dehumidification (cool-

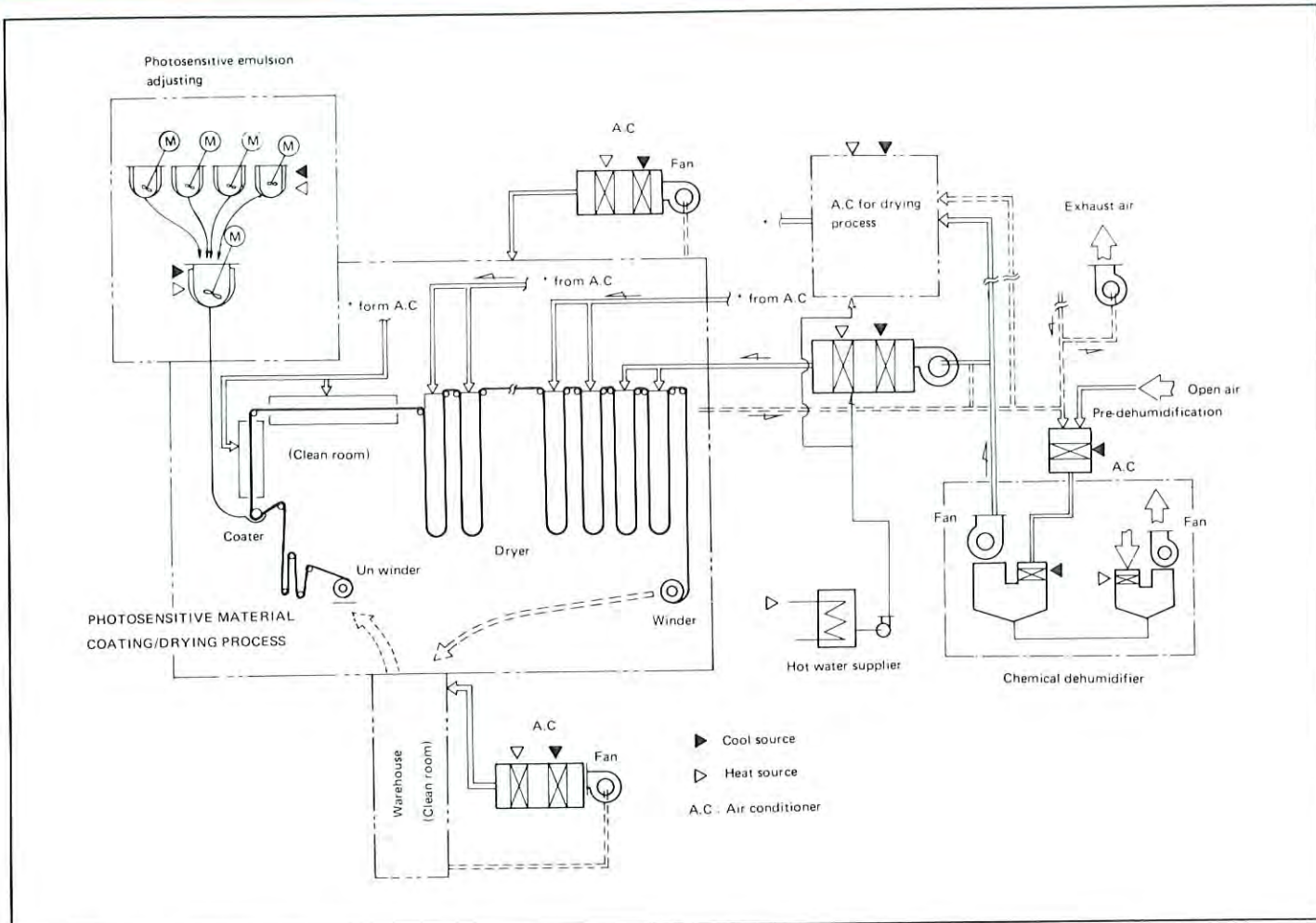


Figure 1. Outline of production facilities and energy uses

ing and chemical) and reheating are necessary. Thus, in the air drying process, large quantities of energy are con-

sumed for cooling and heating. The air used for drying contains moisture as a result of drying the film, and is dis-

charged from the production process. Depending upon temperature-humidity conditions of open air, it is either partially

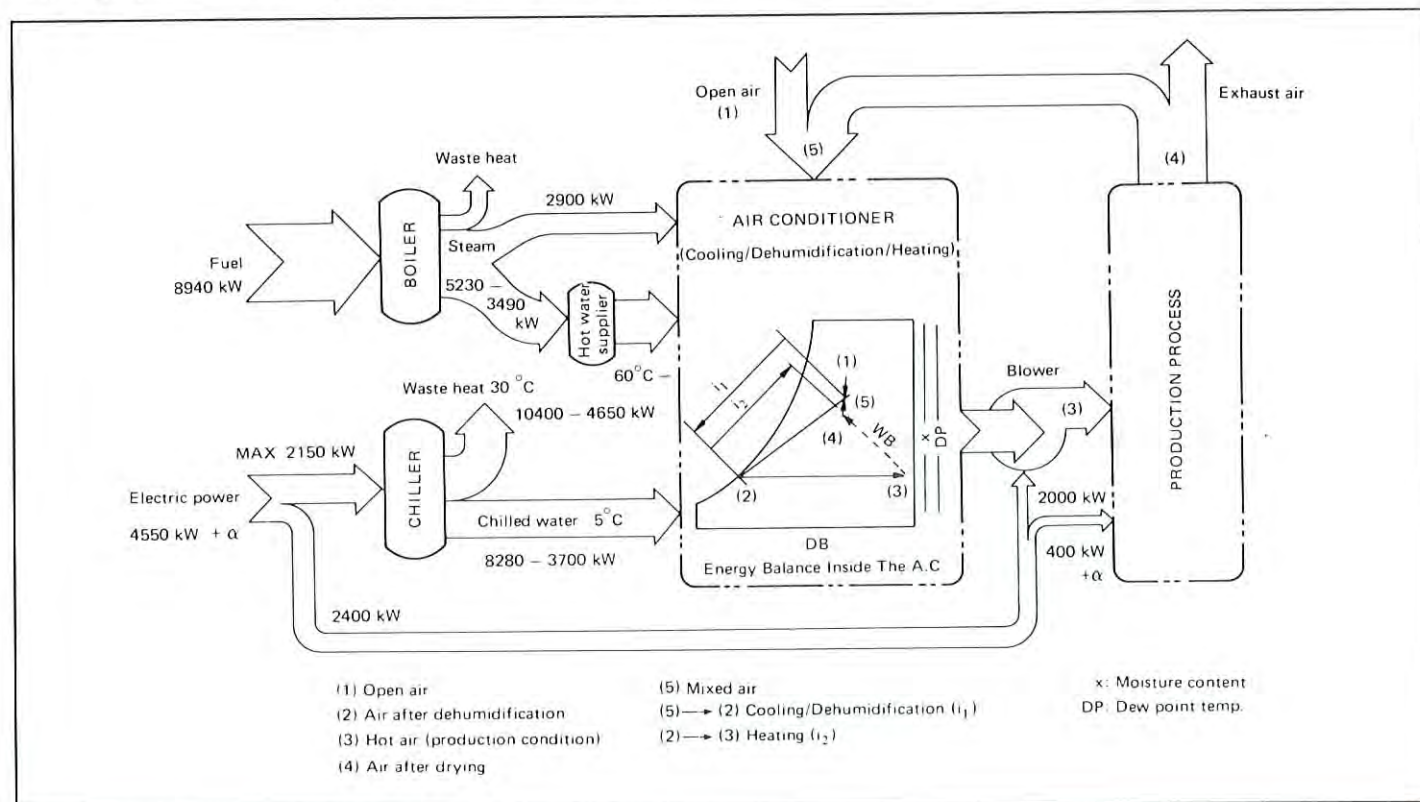


Figure 2. Energy flow for production process

recycled or discharged into the open air, whichever produces the greater energy savings. Meanwhile, to produce a cooling source (chilled water), large quantities of waste heat are released from the cooling tower of the chiller.

1.2 Energy saving steps (introduction of the heat pump)

As shown in Figure 2, waste heat from the condenser, generated as chilled water, is produced by the chiller, and is released into the open air from the cooling tower. Although the temperature level is low at 30-37°C, the waste heat has a very large quantity of heat. Figure 3 shows the relation between the quantity of heat available from hot water for the film drying process and waste heat available from the chiller.

Since the temperature of waste heat from the chiller is low compared to that of hot water required for the film manufacturing process, it cannot be used directly. If, however, a heat pump is introduced, it can effectively be utilized by collecting approximately 3,840 kW of

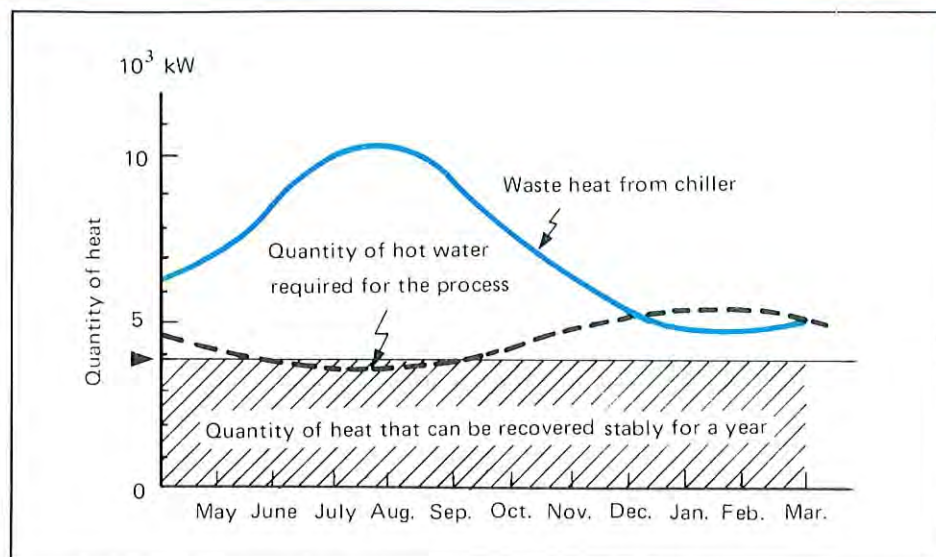


Figure 3. Relation between waste heat from the chiller and hot water required

heat (approx. 6,000 kg/h in terms of steam for producing hot water).

In selecting a heat pump system, the following criteria were used:

1. A high coefficient of performance (COP) in the chilled water production process was the main goal.

2. The energy cost of hot water should be less than one half of the conventional cost.

3. Hot water produced should be the same as that for the conventional film production process.

In light of the above criteria, a number of

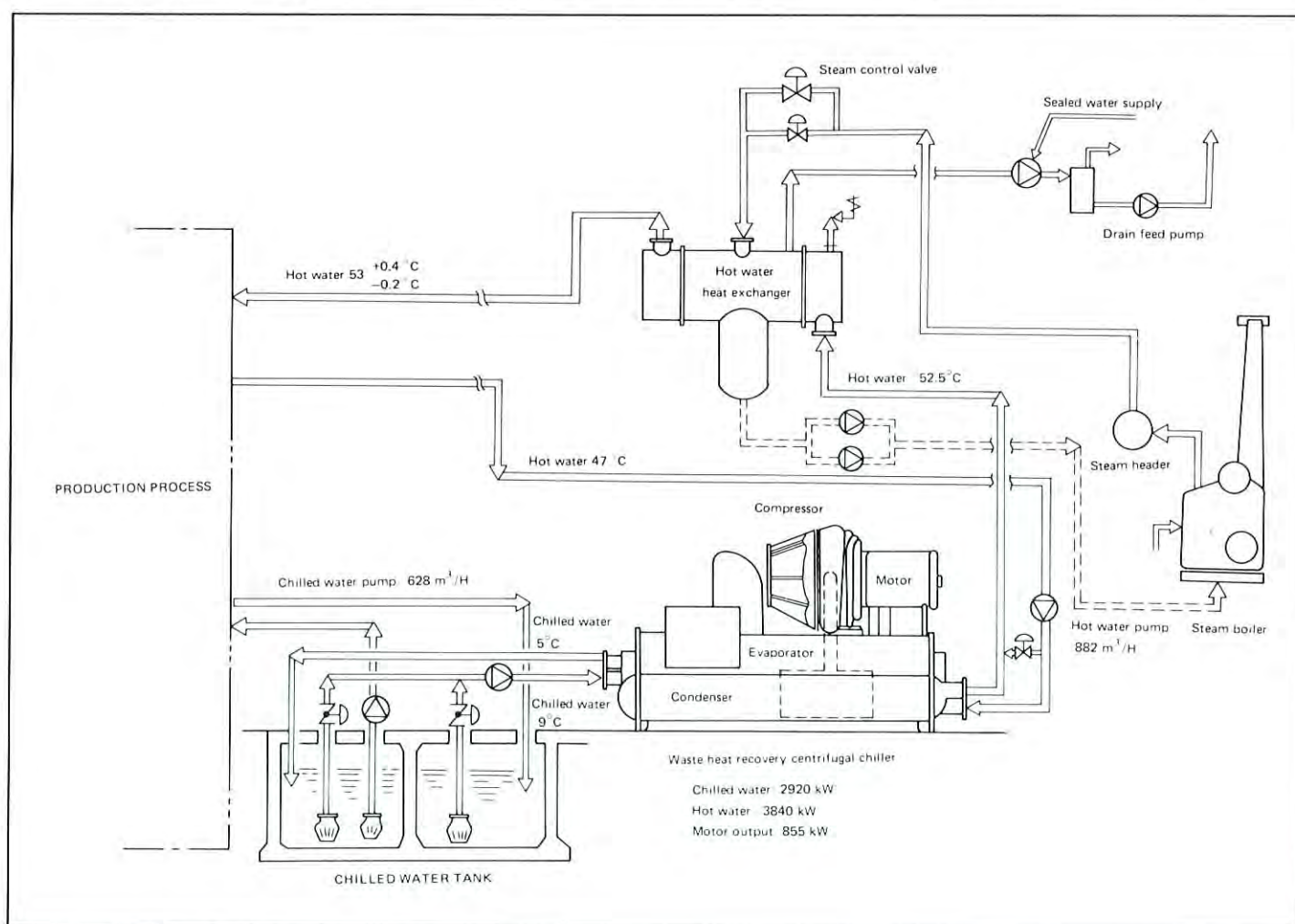


Figure 4. Outline of the system

systems were examined. The conditions of hot water (temperature level, temperature precision, pressure, etc.) used in the film production process were reviewed in order to minimize cost. As a result, a heat recovery system using a large-capacity heat pump was chosen.

2. Large-capacity heat recovery heat pump system

2.1 Outline of the system (Figure 4)

This system consists of a heat recovery type centrifugal heat pump (cover photo), designed to use waste heat from the condenser that occurs when chilled water is produced as its main heat source, and thereby deliver hot water for film production processes. It can stably supply energy against heat quantity fluctuations of chilled and hot water. The COP of the system is higher than 7.0.

2.2 Heat pump specifications

Type:	centrifugal
Heating capacity:	3,840 kW
Cooling capacity:	2,920 kW
Temperature of hot water:	47°C at inlet, 52.5°C at outlet
Temperature of chilled water:	9°C at inlet, 5°C at outlet
Motor output:	855 kW
Refrigerant:	R11

2.3 Beneficial features

2.3.1 Load fluctuation controllability

Hot water -- By setting the heat pump outlet temperature at 52.5°C and by raising it to the range of 52.5-53.0°C by means of quick-responding steam/hot water exchanger, hot water load fluctuations corresponding to 20-25% of heat pump capacity which occur at the film plant could be smoothed, at a very high precision of 53°C (+0.4/-0.2°).

Chilled water -- In order to smooth the hot water load fluctuations occurring at the film plant, the heat pump capacity is either increased or decreased. In this case, however, temperature of chilled water, which is delivered together with

hot water, also fluctuates. In this system, since temperature of chilled water is prevented, while preventing hot water load fluctuations, a fixed temperature of chilled water 5°C (+1/-0°) is delivered, without dropping the flow rate of chilled water (by preventing the chiller from freezing inside), by mixing chilled water to be used (5°C) and chilled water used (9°C) (the mixing ratio is automatically controlled by detecting the outlet temperature of chilled water = 5°C).

2.3.2 Improved responsibility of steam heat exchanger

A pump drain system was employed in which a drain sump is provided beneath the drum housing the heat exchanging tube, and a liquid level is maintained in order to prevent the condensate from reaching the tube. In this way, the heat transfer area of the heat exchanging section is always kept at maximum, thus capable of quickly responding to load fluctuations. Additionally, since the internal pressure is designed to be kept at a certain level below the atmospheric pressure, it has become possible to have stable control characteristics and to utilize the latent heat effectively.

2.3.3 Energy saving for auxiliary devices

Hot water supplied to the film plant is handled by a variable flow rate system which responds to hot water load. With this system, pump power can be reduced when the load is small, leading to power saving and also to stabilized water feeding pressure. Incidentally, a pump driving motor employs an inverter to control the feeding pressure by detecting it.

3. Results

The introduction of the heat pump produced several positive effects. The temperature precision and pressure stability of the hot water used for the film plant have been improved; the cost of energy used for hot water has been reduced by half compared to the costs for the conventional system (steam/hot water heat exchange); and yearly savings realized in terms of heavy oil have been 2,400 m³/year.

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Studies on Chemical Heat Pump Systems Using Ammonia in Japan

In recent years, various chemical reactions have been studied to apply them to heat storage, chemical heat pipes or chemical heat pump (CHP) systems. This research has been aimed at finding different uses of thermal energy and applying it in various industrial fields. This is mainly due to characteristics of reaction employed, such as: (1) the high energy density of heat storage while heat dissipation is low; and (2) the potential to reuse heat by, for example, improving its quality of heat source. Most proposed reactions may be classified according to working fluid involved, such as alcohol, ammonia, carbon dioxide, hydrogen, water, etc. This report describes the studies in Japan of the CHP using ammonia as a working fluid.

1. Ammonia Reaction Systems

Ammonia has been used as the working fluid of refrigerators. As shown in Table

1, its heat of evaporation compared with other media is very high, second only to that of water, and its evaporation and condensation can be controlled without

Working fluid	H ₂ O	NH ₃	H ₂	CH ₃ OH	CO ₂	SO ₂
Molecular weight	18.02	17.03	2.01	32.04	44.01	64.06
Heat of evaporation ¹ (kcal/kg)	539.1	327.7	108.5	130.3	83.1 ²	93.1
Specific heat of gas (cal/kg · °C)	444.8	498.2	3425.8	327.2	204.4	148.7
Boiling point (°C)	100	-33.4	-252.8	64.1	-78.5	-9.97
Melting point (°C)	0	-77.7	-259.2	-97.8	-56.5	-72.7
Critical temperature (°C)	374.2	132.5	-239.9	240.0	31.05	157.7
Critical pressure (kg/cm ²)	225.9	115.0	13.2	81.1	74.85	80.4

¹At boiling point²At triple point

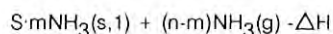
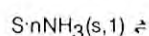
Table 1. Characteristics of typical working fluids

difficulty. Moreover, since there are many chemicals that react with ammonia over a wide temperature range, those reactions are readily usable for heat storage and CHP systems, and have high energy density.

On the other hand, since ammonia is combustible, corrosive and toxic, applications are limited. It is necessary to examine both the positive and negative aspects of ammonia before putting it to practical use.

Generally, the reactions in which ammonia takes part may be divided into the solid and liquid systems, whose reactions are expressed by the following formula:

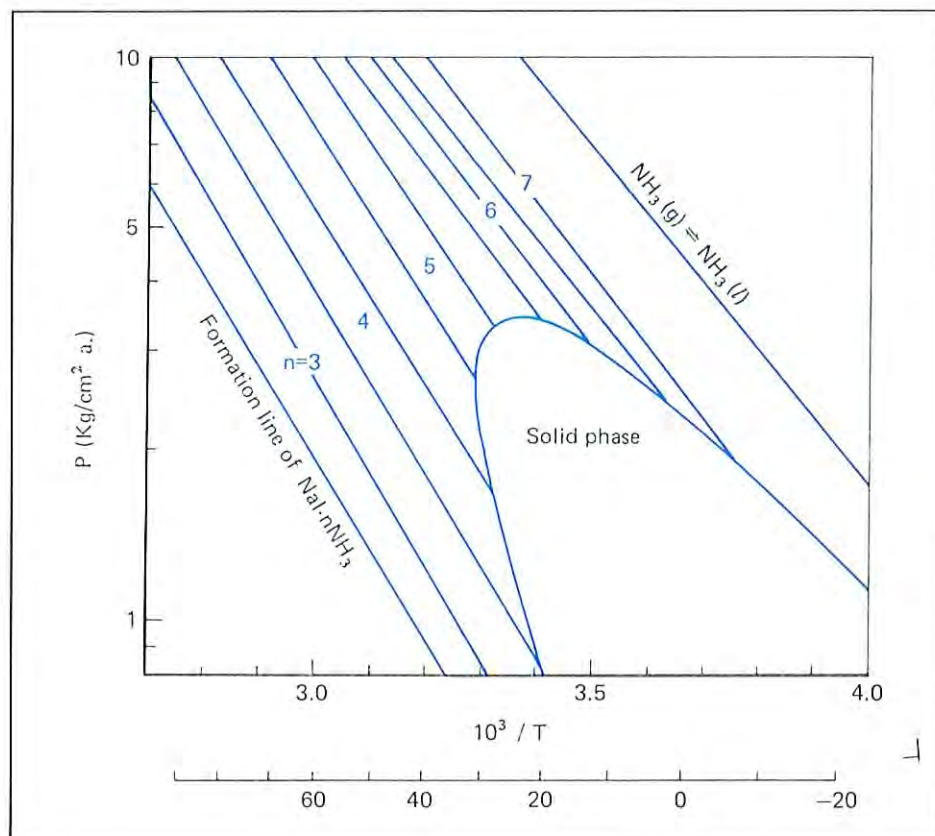
Equation 1:



(S = Salt)

$S \cdot n\text{NH}_3$ is called an ammine complex, ammoniate, or ammoniated salt, and it exists in the solid or liquid state. In the case of the solid system, the value of n , degree of ammoniation, is a certain integer peculiar to each salt. In the liquid system, n varies continuously depending on temperature and pressure conditions. Therefore, taking such features into account, each reaction may be applied to heat storage or CHP systems.

Examples of studies on liquid and solid systems in Japan are introduced below.

Figure 1. Equilibrium diagram of NaI-NH₃ system

2. Liquid System

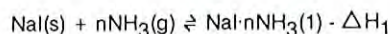
In this system, the ammoniates, $S \cdot n\text{NH}_3$, are in the liquid state, and are salts (S) that are dissolved in liquid ammonia. So, unlike the solid system, its ammoniation ratio (degree of ammoniation, n) varies continuously after it exceeds a certain value. Once two of the three variables (temperature, pressure and composition) are specified, the state of the system can be fixed (degree of freedom on phase rule is 2). Those salts which exhibit such properties consist of some salts of ammonium NH_4X ($\text{X} = \text{Cl}, \text{NO}_3, \text{I}, \text{Br}, \text{SCN}$) and sodium NaX ($\text{X} = \text{I}, \text{SCN}$). Since those reaction systems do not exist at a high temperature range compared with that of the solid ammoniate

systems, they may not be suitable for delivering high-temperature outputs. In the liquid state, however, those reactions can readily be accomplished. Furthermore, the system can be operated continuously, such as an absorption heat pump. Heat transportation (as a chemical heat pipe) will also be applicable.

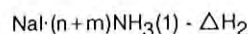
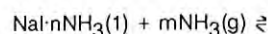
The authors have examined an ammoniate of sodium iodide (NaI)¹. This examination was made to achieve effective use of low-temperature waste heat from atmospheric temperature to 200°C. It was conducted in accordance with the national project "Research and Development of Technologies for Utilizing Waste Heat" as part of the Moonlight Project, Ministry of International Trade and Industry (MITI).

As shown in the formula given below, solid NaI becomes a liquid ammoniate, and by increasing the ammonia gas pressure or by lowering the system temperature, it changes into another ammoniate exothermically in which the ammonia ratio is increased.

Equation 2:



Equation 3:



These reactions are reversible and proceed rapidly. ΔH_1 is about 150 kcal/kg-NaI ($N=3$). Figure 1 shows the chemical equilibrium of this system. The inner part, surrounded by the formation line of $\text{NaI} \cdot n\text{NH}_3$ and pure ammonia's vapor pressure line and border line of solid phase (deposition of $\text{NaI} \cdot 4.5\text{NH}_3$), is liquid phase. The isocomposition lines in liquid phase can be approximated by straight lines and the ammoniation ratio, n , varies up to infinity. When the ratio is smaller than 7, the density of this system lies between 1.3 and 1.7 and its viscosity is only several centipoises. Taking into account the above features, it becomes possible to apply this reaction not only to a heat storage system using single reversible reaction, but also to CHP systems which can upgrade the heat source. The CHP system works on the same principle as the absorption heat pump ($\text{LiBr}-\text{H}_2\text{O}$ system) and can be accomplished by combining a pair of reactions, e.g., interchange of two different ammoniates and condensation/evaporation of pure ammonia, respectively.

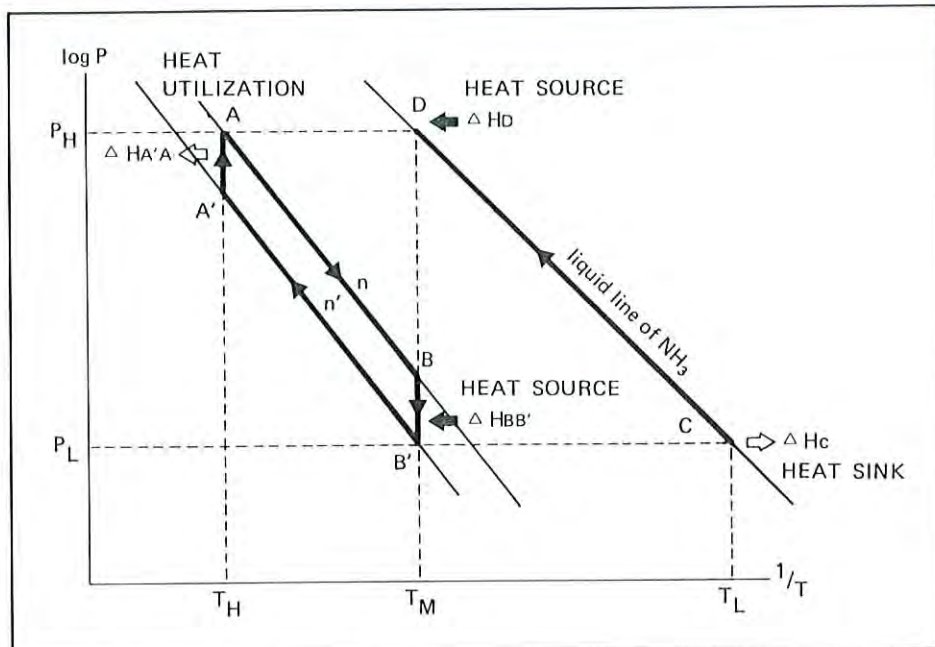


Figure 2. Working principle of CHP cycle (heat transformer mode)

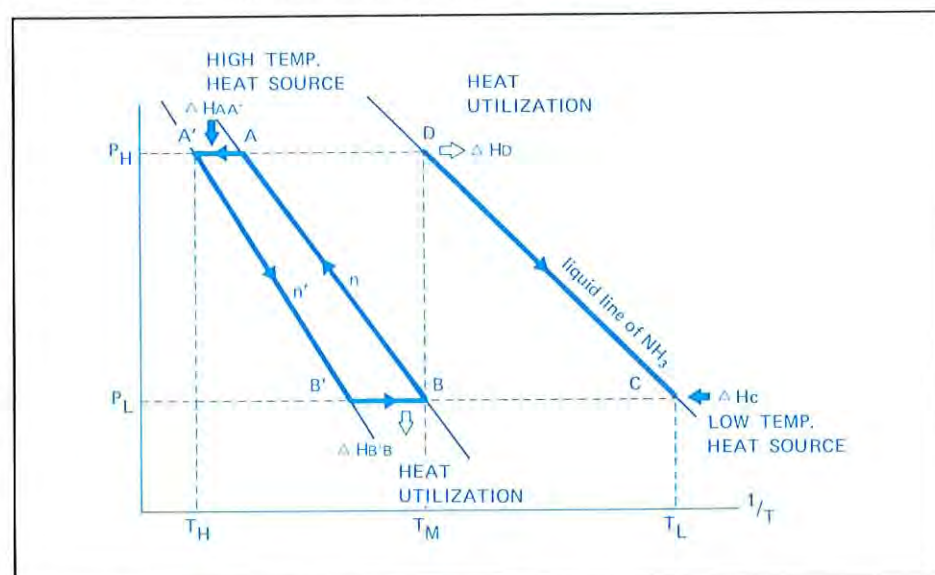
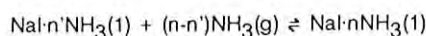


Figure 3. Working principle of CHP cycle (heat amplifier mode)

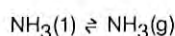
High-temperature side:

Equation 4:



Low-temperature side:

Equation 5:



Various thermal operations can be performed by selecting the temperature and pressure conditions, cycle directions of substances, etc. Some working modes have been examined by using the liquid phase obtained in Figure 1.

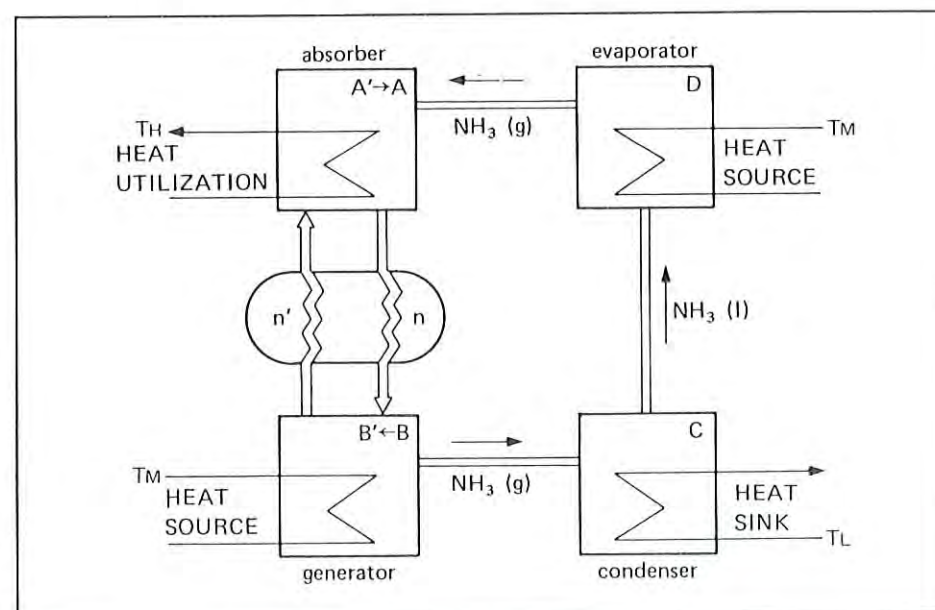


Figure 4. Schematic diagram of heat transformer mode

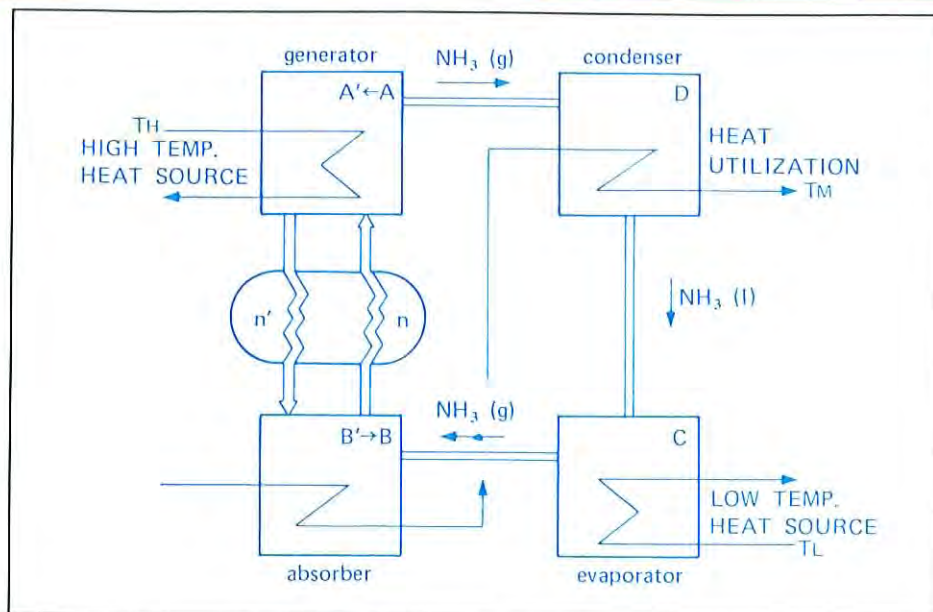


Figure 5. Schematic diagram of heat amplifier mode

Figure 2 shows the heat transformer mode, in which a higher temperature than the heat source is obtained. To achieve this, an ammoniate with composition n is endothermically decomposed at the heat source temperature T_M into a substance with lower composition n' (B-B' step). The ammonia released as a gas is then condensed, releasing heat to the heat sink (e.g., the atmosphere) at T_L . The liquefied ammonia is again heated up to T_M by the heat source (C-D) and evaporates so as to be absorbed by previously formed ammoniate of com-

position n' , where the heat of reaction at T_H is finally available (B'-A'-A).

The heat amplifier mode operation is shown in Figure 3. In this mode, the amount of heat available can be boosted, though the temperature, T_M , is lower than the heat source temperature, T_H . In contrast with the heat transformer mode, this one is driven in a reverse manner: an ammoniate with composition n is endothermically decomposed into an ammoniate with lower n' (A-A') at the heat source temperature, T_H . After

the heat of condensation is used at T_M , liquid ammonia is cooled to the low-temperature heat source (D-C, e.g., atmosphere). It evaporates there to be absorbed by the ammoniate with composition n' and the heat generated is utilized again at T_M (A'-B'-B). Incidentally, cooling is possible in the same cycle. Endothermic evaporation of ammonia is utilized at T_L .

Figures 4 and 5 show the conceptional flow for putting both modes to practical use. Reactions corresponding to the forward and backward directions in Equation 4 occur in the absorber and generator, respectively, and the phase change described in Equation 5 takes place in the evaporator and condenser.

Judging from Figure 1 and other properties of the NaI-NH₃ system, a suitable composition range for the heat storage system under 100°C, for example, is expected to be between $n=4$ and 5. Although the crystallization region should be avoided when utilizing the system as a continuous-type chemical heat pump, this liquid system has the advantage that ammoniates with different composition (n, n') can be freely chosen to match the necessary conditions. As a matter of fact, from the results of operations of test apparatus (Figure 6) for air conditioning (CHP) system, the authors have confirmed that a stable supply of heat is possible by controlling ammonia gas flow.

It is also possible to set up such a system that has a compressor as in Figure 7. In this case, however, it is desirable to avoid operation in the crystallization region to prevent pipe plugging. Regarding materials used for the apparatus, it was adequate to take care of those for ammonia gas.

On the other hand, the authors have also made studies on ammonium thiocyanate (NH₄SCN) to determine its chemical equilibrium, thermal properties, etc.² The crystallization region of this system does not exist over its usual temperature range so that one can expect it as a new liquid reaction system for the CHP system. It may be necessary, however, to consider its corrosion characteristic.

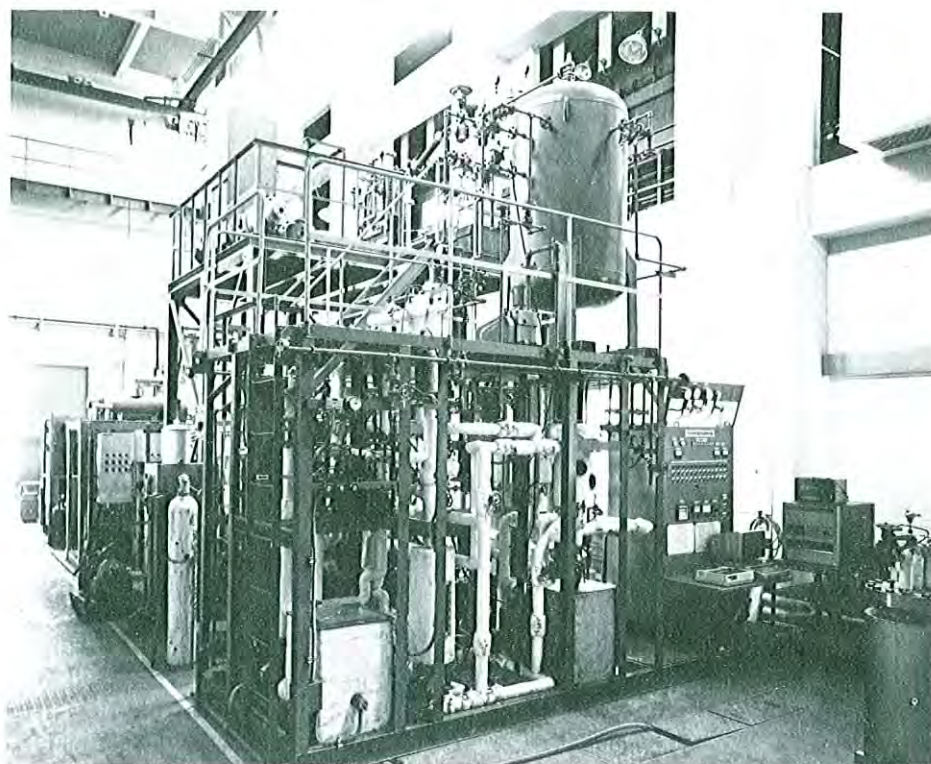


Figure 6. Test plant for thermal storage with liquid ammoniated salt

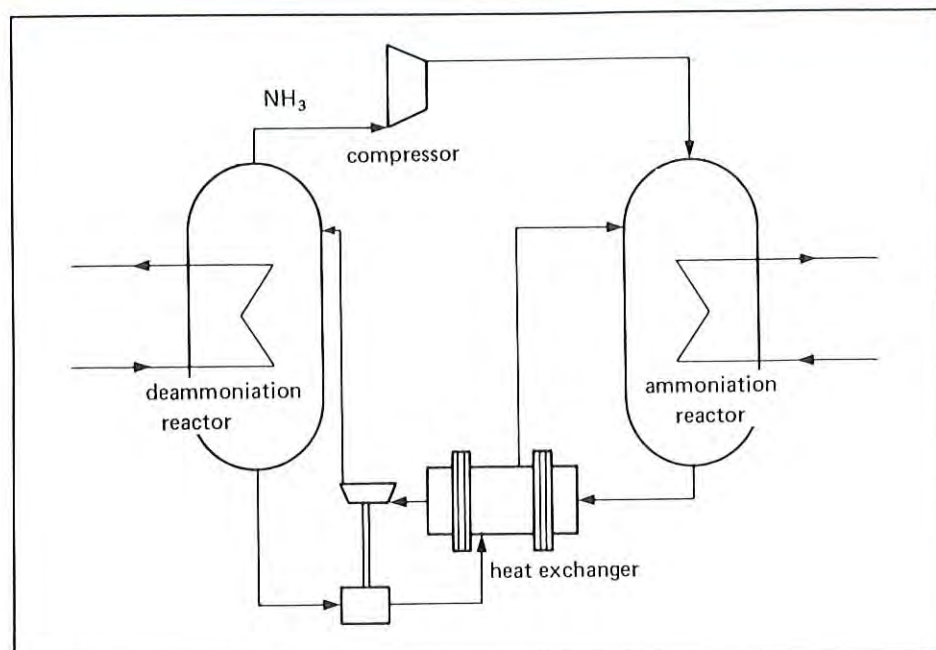


Figure 7. CHP system with compressor

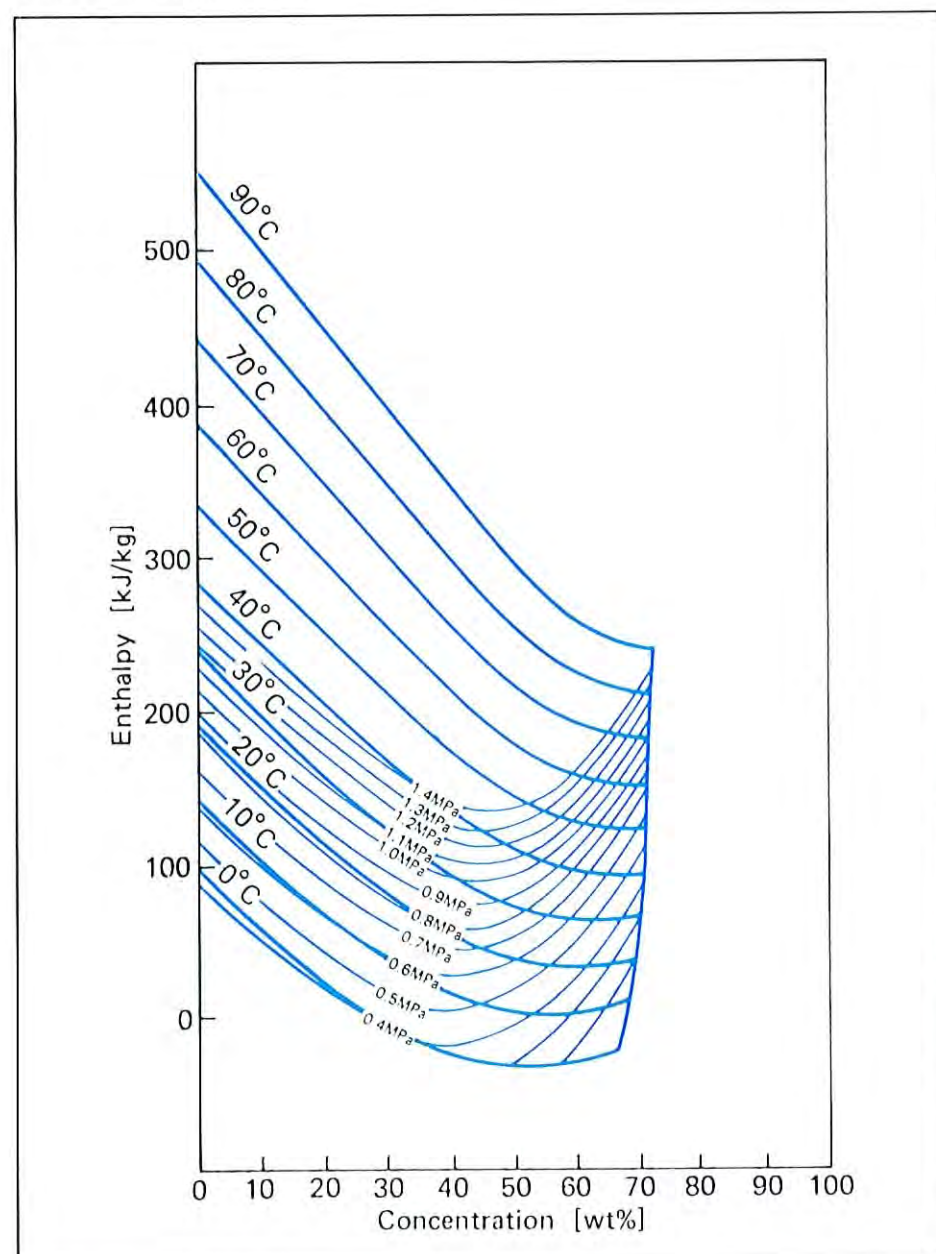


Figure 8. Enthalpy-concentration chart of $\text{NH}_4\text{Br}-\text{NH}_3$ system

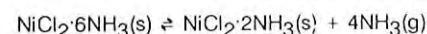
Yamamoto, et al, of Kansai University, have examined ammoniates, such as ammonium bromide (NH_4Br) and ammonium iodide (NH_4I).³ In order to apply these substances to solar-assisted air conditioning systems, they made enthalpy concentration diagrams (Figures 8 and 9) and estimated the performance of those systems.

3. Solid System

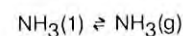
Many halides of alkali metals and alkali earth metals have a tendency to react with ammonia gas to produce solid ammoniates. The reaction scheme is similar to that of hydration/dehydration reaction. As those ammoniation reaction systems lie over a wide temperature range, they can be used for thermal operation at higher temperatures compared with the liquid systems.

Yoneda, et al, of Tokyo Science University, have been studying a CHP system which combines the ammoniation/deammoniation reaction of nickel chloride and the evaporation/condensation of ammonia.⁴

Equation 6:



Equation 7:



The main purpose of this study is to store solar energy. The ammonia gas generated in the heat storage process is liquefied by a compressor to condense it. The reason for using a compressor is that when ammoniation of $\text{CaCl}_2 \cdot 4\text{NH}_3$ is taken as an ammonia gas received, this reaction rate becomes so small that the system efficiency is decreased. This method, which uses a reaction vessel as shown in Figure 10, could convert thermal energy at 230°C into the form of liquid ammonia at 0°C and 8 atm. In a reverse reaction as a heat recovery process, the efficiency of the energy storage was reported to be as high as 60%, with an average of 14%. It was pointed out, however, that the low heat conductivity of solid material would be the largest problem.

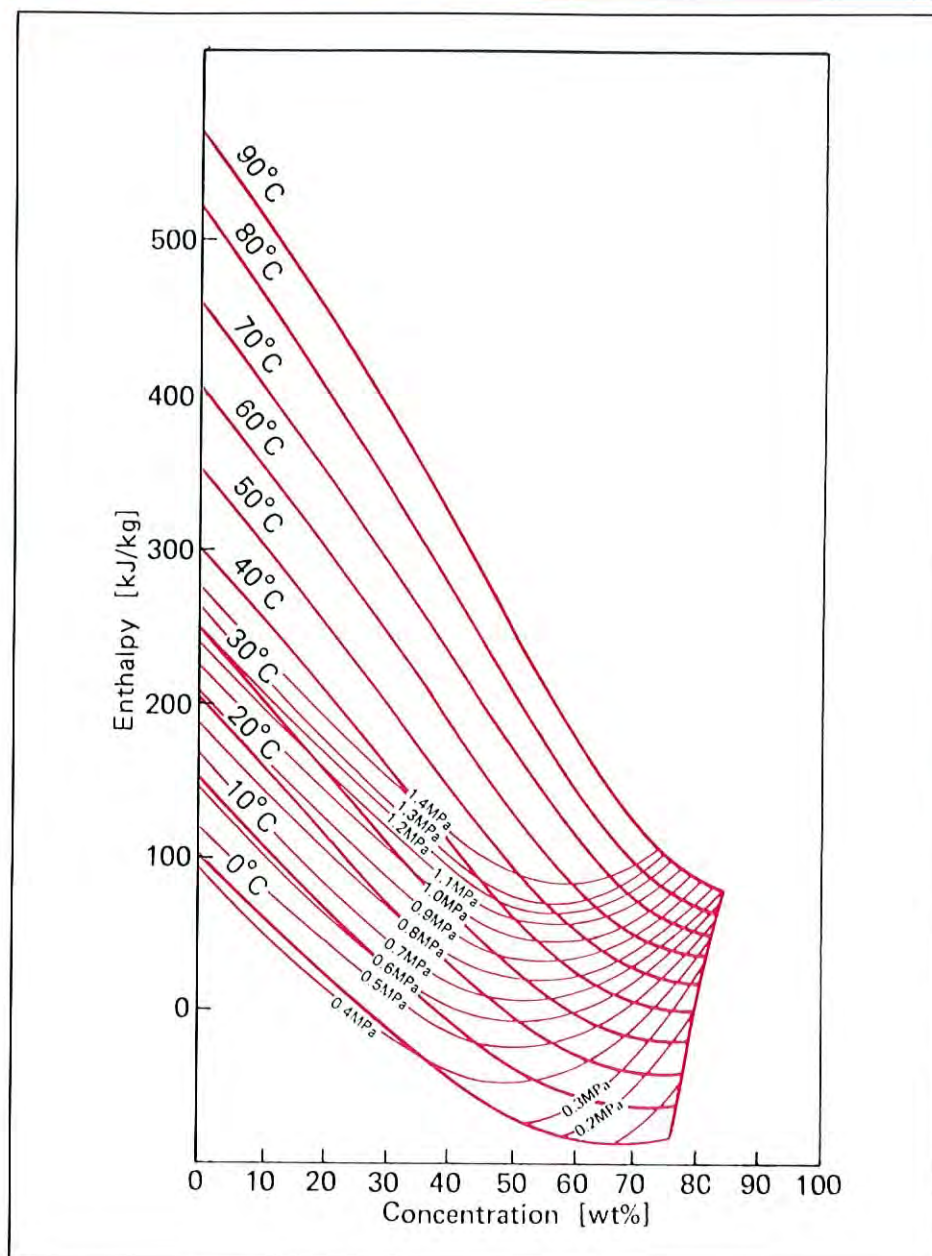


Figure 9. Enthalpy-concentration chart of $\text{NH}_4\text{I}-\text{NH}_3$ system

NEDO (New Energy Development Organization), which is in charge of the national project, Super Heat Pump Energy Accumulation System of the Moonlight Project, has commissioned

IHI (Ishidawajima Harima Heavy Industries Co.) with the task of studying the CHP system using the pair of reactions given below. They are a combination of solid and liquid reactions which use

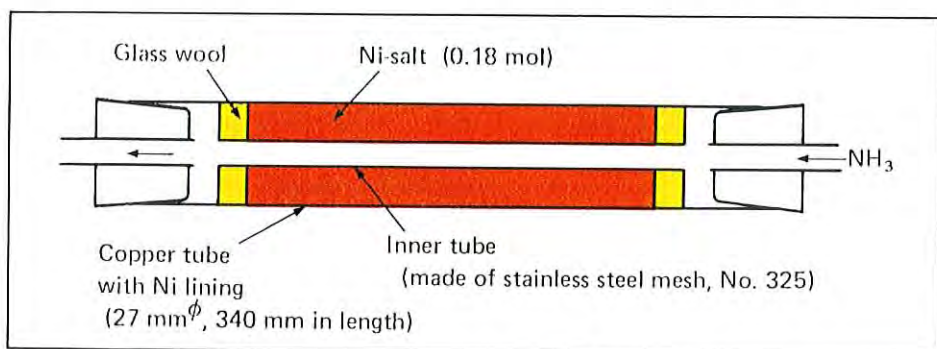
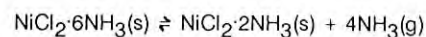


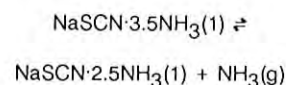
Figure 10. Reaction tube of $\text{NiCl}_2-\text{NH}_3$ system

ammoniates of nickel chloride and sodium thiocyanate, respectively:

Equation 8:



Equation 9:



Thermal energy storage and CHP systems in Japan, which use ammonia as the working fluid, have been outlined. In spite of its excellent features as a working fluid, ammonia seems not to be in practical use because of its shortcomings.

This tendency may be more apparent in Japan than in Europe and America. However, there is another fact to consider, that is, that freons, frequently used as working fluids or refrigerants these days, are causing destruction of the ozone layer in the atmosphere. This is becoming a well-known social issue. It seems that now is the right time to reconsider the use of ammonia.

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R. Monahan*

The Development of a Natural Gas-Fired Stirling Engine-Driven Heat Pump

The Gas Research Institute (GRI), USA, has funded the development of numerous natural gas-fueled appliances. In one current program, a 33 kW (10 ton) light commercial heat pump package has been built around the V160 Stirling engine.

While the Stirling cycle is one of the oldest thermodynamic cycles in use today, it is also one of the least utilized, having been pushed aside by Otto and Diesel cycle internal combustion engines. However, inherent characteristics such as low maintenance and long life give the Stirling engine the potential to meet the requirements of a durable, economical heat pump system. A Stirling engine differs from a conventional powerplant in that combustion takes place continuously outside the cylinder and heat is transferred through a wall to a captive working fluid, usually helium or hydrogen.

The Stirling cycle closely resembles the Carnot cycle, as shown in the P-v and T-s diagrams in Figure 1. Heat is transferred away from the working fluid (typically helium or hydrogen) during the compression phase (1-2) on the "cold" side of the engine, thus ideally maintaining a constant temperature. The working fluid is then displaced (2-3), passing through the regenerator and the primary heat exchanger, picking up thermal energy while ideally maintaining a constant volume. Isothermal expansion (3-4) occurs on the hot side of the engine in the expansion cylinder and provides the major force on the piston which is trans-

lated into rotary motion through the crankshaft. The working fluid is then moved back to the "cold" side of the engine (4-1), storing thermal energy in the regenerator while maintaining a constant volume before entering the gas cooler.

Shown in Figure 2, the V160 engine is a two-cylinder, 90-degree "V" kinematic, single-acting Stirling engine of 160cc swept volume. In contrast to the free piston variety, which transmits its linear power through a diaphragm, or a double-acting unit which typically requires four cylinders to achieve the proper phasing, the primary heat exchanger, regenerator and gas cooler of the V160 are located between the separate expansion and compression cylinders.

The V160 engine has a peak power output of 20kW at 3000 rpm and peak torque of 80N-m at 1000 rpm with an efficiency exceeding 30% (HHV). As shown in the energy balance in Figure 3, the coolant system typically receives 50% of the input energy and can be used for domestic hot water and/or supplemental heating in HVAC and cogeneration packages. A heat exchanger between

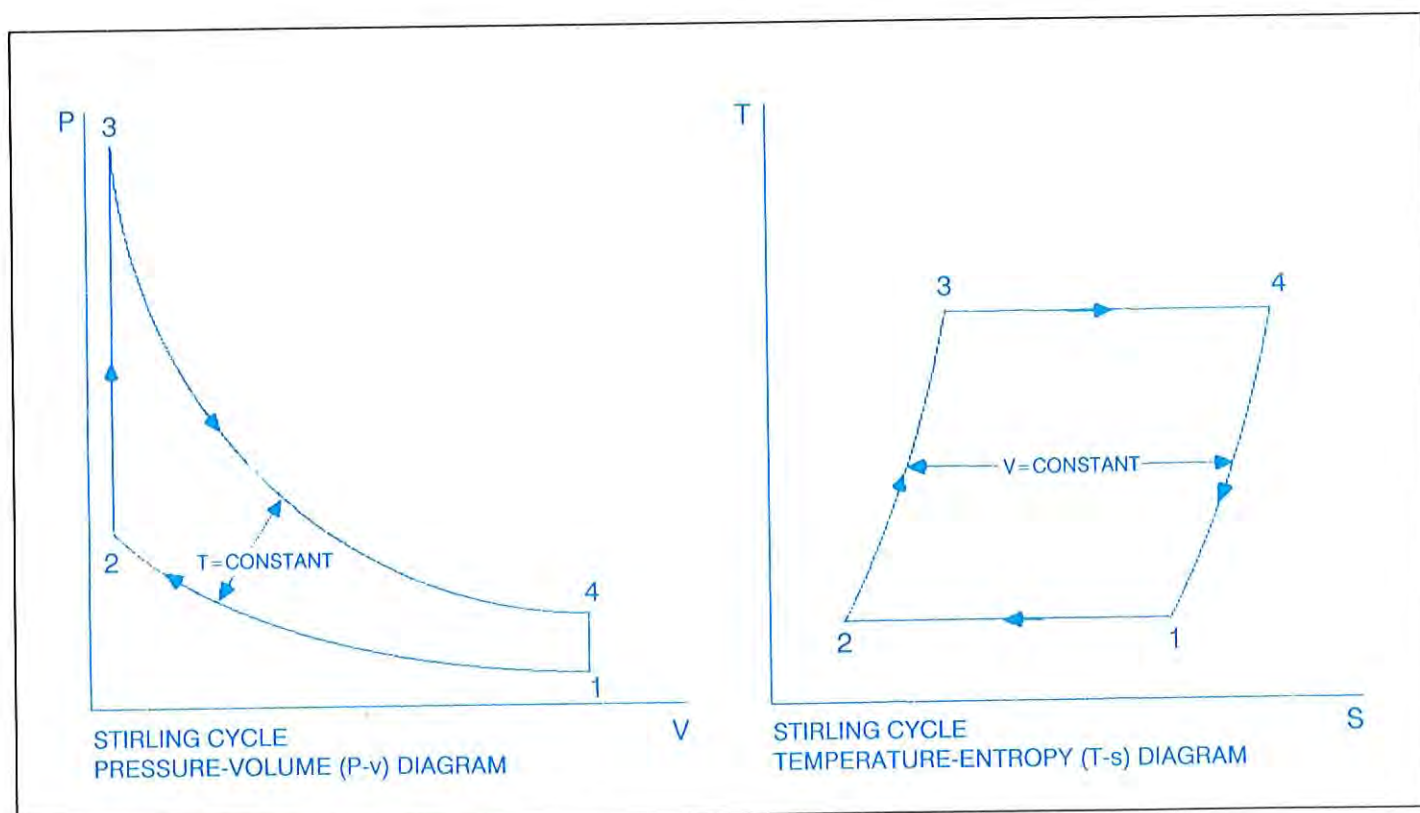


Figure 1. Stirling cycle -- P-v and T-s diagrams

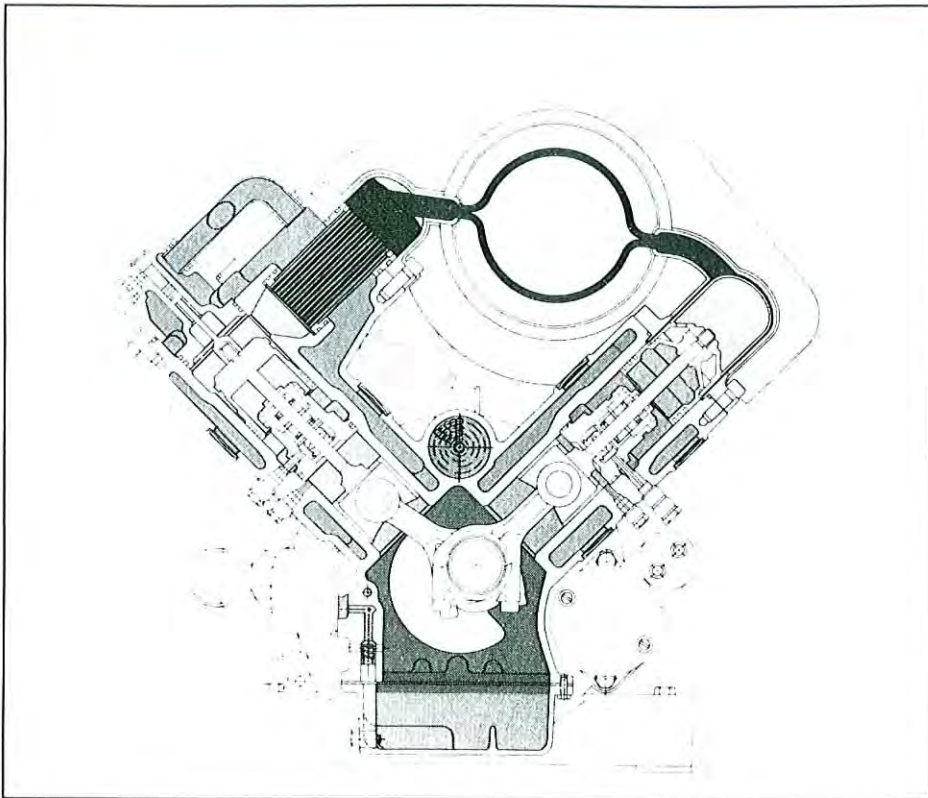


Figure 2. Cross-section of V160 Stirling engine

the incoming combustion air and the exiting products of combustion reduces exhaust losses and adds to the efficiency of the packages.

Two major control loops are monitored by the engine's microprocessor control system. Engine speed and load are modulated through a series of solenoid valves which regulate the flow of the working gas into and out of the engine. Working gas temperature is measured with a thermocouple and maintained at a preset level by adjusting the flow of air and fuel into the combustion chamber.

Each cylinder is equipped with a rod seal between the crosshead and the piston, isolating the high pressure working gas from the crankcase. This also prevents the flow of lubricating fluid into the cycle space, and therefore the piston rings must run dry against the cylinder walls. Helium storage reservoir refilling intervals depend on the tank size, load and speed cycle and usage factors but is targeted at once every two years for the heat pump application.

For ease of testing and fabrication, a split-system York Champion heat pump package was chosen as the basis for the first working prototype, built at the Borg-

Warner Research Center. The indoor and outdoor modules were modified to include heat exchangers through which a three-way valve controlled the flow of engine coolant. The engine-compressor assembly was attached to the outdoor module.

Preliminary cooling mode testing was performed using a closed loop rig coupled to the indoor module and allowed the verification of the computer model. Full-scale environmental evaluation was later carried out at York International Corporation's laboratories. The results were used as the basis for a seasonal performance analysis which allowed direct comparison with anticipated outputs from other heating and cooling packages.

Data from the heating and cooling mode testing at the York facility appears in Table 1. While cooling mode performance and capacity fell short of the goals,

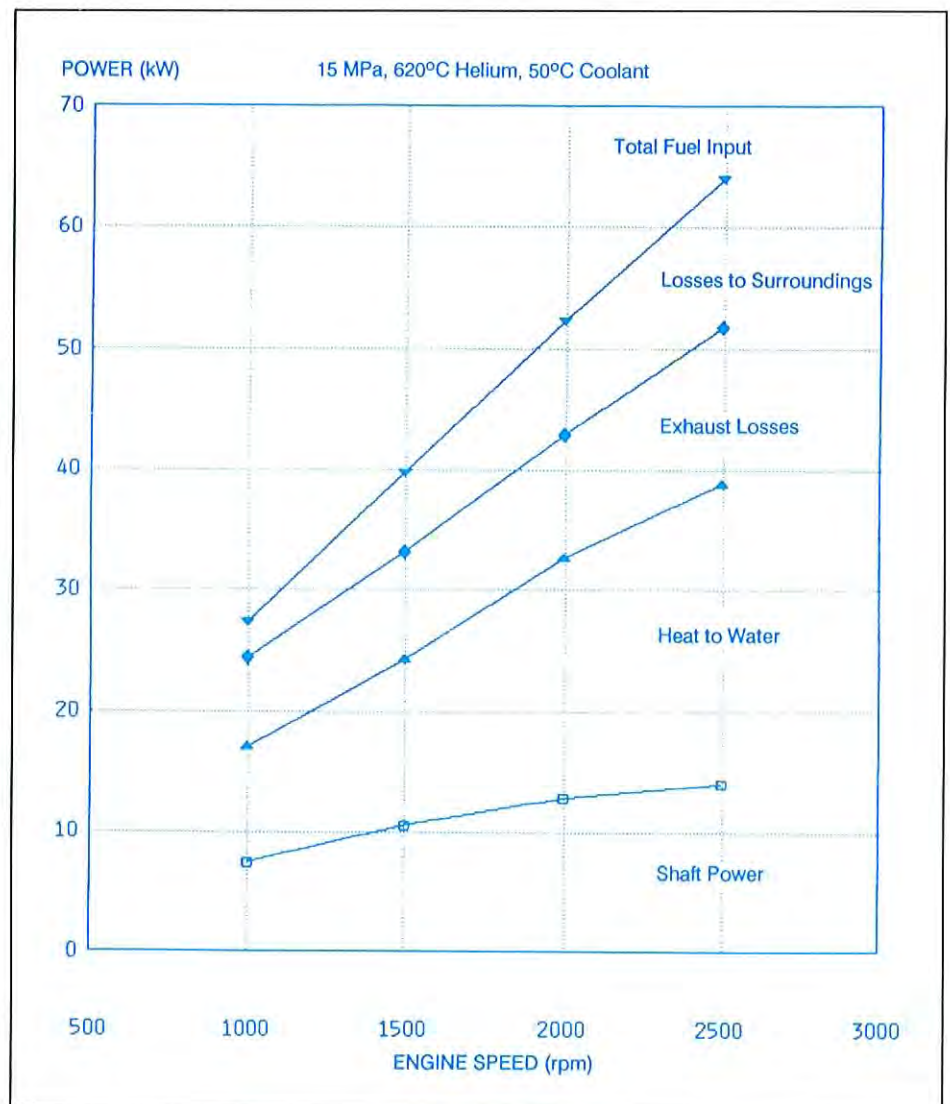


Figure 3. Heat balance of V160 Stirling engine

Run No.	DB/WB Temps		Engine Data				System Data				
	ID°C	OD°C	Speed (rpm)	Torque (N-m)	Power (kW)	Eff. (%)	No. Cyl.	Capacity (kW)	Q. to Rad. (kW)	% Total by Rad.	COP's Thermal Cost
Cooling mode											
1	27/20	35	1700	66.7	11.86	25.8	4	32.8			.713 .541
2	27/20	28	2000	64.3	13.50	25.7	4	37.0			.706 .550
3	27/20	35	2700	39.5	11.19	20.5	2	27.3			.500 .393
4	27/20	28	2900	38.5	11.71	20.8	2	29.8			.530 .417
5	27/20	35	1000	60.1	6.29	26.6	4	23.6			.996 .629
6	27/20	28	1000	55.2	5.78	26.5	4	25.7			1.17 .723
7	27/20	20	1000	47.5	4.97	26.8	4	26.8			1.44 .825
8	27/20	28	1000	23.9	2.50	16.5	2	15.3			1.01 .532
9	27/20	20	1000	16.5	1.73	13.7	2	15.3			1.22 .578
Heating mode											
10	21	8/16	2050	58.2	12.46	24.1	4	65.1	26.5	40.8	1.26 .979
11	21	-8/-9	2400	46.9	11.79	25.5	4	48.9	23.6	48.3	1.08 .798
12	21	-21/-21	3000	41.8	13.13	22.2	4	50.3	32.4	64.5	.850 .677
13	21	-22/-22	2400	42.4	10.67	28.3	4	35.1	18.7	53.3	.934 .674
14	21	8/16	2700	33.6	9.47	27.4	2	56.0	24.4	43.6	1.16 .889
15	21	-8/-9	3000	26.6	8.36	19.7	2	39.1	23.6	60.4	.939 .682
16	21	-22/-22	3000	41.8	13.13	22.2	2	50.3	32.4	64.5	.850 .677
17	21	8/16	1000	47.6	4.98	25.7	4	33.4	9.0	26.9	1.72 .991
18	21	-8/-9	1000	40.9	4.51	25.1	4	23.9	9.4	39.3	1.33 .744
19	21	8/16	1000	21.0	2.20	17.3	2	20.7	6.2	30.1	1.63 .778
20	21	-8/-9	1050	21.3	2.34	17.5	2	15.1	5.8	38.4	1.13 .547

Table 1. Results of environmental laboratory testing of V160-driven system

output in the heating mode was sufficient to obviate any requirement for supplemental resistance heating with this package. The seasonal performance analysis used a bin method approach, using gas and electric prices and climatic data for Chicago and Dallas. This allowed the calculation of sea-

sonal operating costs based on 24 hour/day operation, a cyclic degradation coefficient of .25 for all cases and did not take into account any maintenance costs. Only one cooling load line was considered, although both maximum and minimum design heating requirements (Max and Min DHR) were in-

cluded in the analysis. Table 2 shows the computed operating costs for the standard electric heat pump, a natural gas furnace (efficiency rating of .8) and the breadboard V160-driven system. Substantial savings are shown for the V160 system in both the cooling and heating modes in Chicago and Dallas.

Operating Mode	Cost COP's								Annual Operating Costs																											
	V160		Std.		Furnace		V160/Std.		Std.		V160		Furnace		V160/Std. Savings																					
	Chic	Dall	Chic	Dall	Chic	Dall	Chic	Dall	Chic	Dall	Chic	Dall	Chic	Dall	Chic	Dall																				
Heating - Min DHR	.633	.663	.411	.458	.616	.610	1.540	1.448	3525	1188	2288	820	2353	892	1237	368																				
Heating - Max DHR	.638	.727	.357	.469	.696	.692	1.787	1.550	8115	2321	4545	1498	4164	1572	3570	823																				
Cooling	.497	.509	.370	.390	N/A	N/A	1.343	1.305	1160	2975	807	2127	N/A	N/A	353	848																				
Assumes Cyclic Degradation Factor $C_D = .25$ For All Cases																																				
Prices																																				
<table><tr><td></td><td colspan="2">Chicago</td><td colspan="2">Dallas</td></tr><tr><td></td><td>Winter</td><td>Summer</td><td>Winter</td><td>Summer</td></tr><tr><td>Natural Gas (\$/10⁶ Btu)</td><td>5.49</td><td>5.14</td><td>5.25</td><td>4.79</td></tr><tr><td>Electric (\$/kW-hr)</td><td>.1010</td><td>.1124</td><td>.0955</td><td>.0990</td></tr></table>																		Chicago		Dallas			Winter	Summer	Winter	Summer	Natural Gas (\$/10 ⁶ Btu)	5.49	5.14	5.25	4.79	Electric (\$/kW-hr)	.1010	.1124	.0955	.0990
	Chicago		Dallas																																	
	Winter	Summer	Winter	Summer																																
Natural Gas (\$/10 ⁶ Btu)	5.49	5.14	5.25	4.79																																
Electric (\$/kW-hr)	.1010	.1124	.0955	.0990																																
Average charge for commercial equipment including demand, usage and connect charges (Reference: GRI Report 85/201)																																				

Table 2. Operating cost comparison - V160-driven heat pump vs. standard electrical vs. gas furnace

Although the thermal COP of any engine-driven heat pump system will naturally be considerably lower than its electric counterpart, the difference in natural gas and electricity prices in most of the United States still makes gas-fired heat pumps economically attractive. For purposes of comparison between all-electric and hybrid (natural gas and electric) heat pump systems and other space conditioning equipment such as furnaces and air conditioners, the "Cost COP" and "electric-based COP" were developed.

The Cost COP is defined as:

$$\text{COP}_{\text{cost}} = \frac{\text{COP}_{\text{thermal}}}{1 + \frac{E}{G_{\text{cost}}} \times \frac{(3413 \times P_{\text{aux}})}{Q_{\text{gas}}}}$$

And the electric-based COP is defined as:

$$\text{COP}_{\text{electric-based}} = \frac{\text{Useful Heating and Cooling}}{(G/E) \times Q_{\text{gas}} + Q_{\text{elec}}}$$

where:

P_{aux} = auxiliary parasitic electric power (kW-hr)

E/G = electric-to-gas price ratio

G/E = gas-to-electric price ratio

Q_{gas} = natural gas energy used

Q_{elec} = electric energy used

While both parameters depend on the local utility rates, they offer a purely economic method of performance measurement, and one that is most meaningful to the consumer. Cost COP's are shown in Table 2.

The usefulness of the Cost COP is evident in the comparison of the performance of the V160-driven system and the hypothetical case with a 40% improvement in engine efficiency. Figure 4 details the energy balance for the case of low ambient temperature (17°F/-22°C) where an increase in engine efficiency actually results in increased operating costs, as indicated by the reduced Cost COP. This seemingly paradoxical result is due to the fact that the heat recoverable by the indoor unit's radiator is reduced and the zone load requirements must be met with relatively expensive electric resistance heat.

The environmental laboratory test results showed several weaknesses in the

breadboard system, indicating the need for a second generation design which could implement the appropriate improvements. Specifically, line losses and instrumentation caused high pressure drops in the refrigeration system, poor distribution of the refrigerant and air were discovered to be occurring through the indoor coils, high electrical parasitic losses of the indoor and outdoor fans had to be addressed and package aesthetics needed to be improved to resemble a commercial product.

SPS constructed a "black-box" engine/compressor module including controls and auxiliaries, a package design which could be incorporated into a number of HVAC/R applications from water chillers to air conditioning systems. Figure 5 shows the engine/compressor package assembled with the York outdoor module, undergoing testing at York.

By controlling separately the two fans in the outdoor unit, and through the use of a continuously variable, inverter-driven indoor blower, the cost savings shown in Table 3 are predicted. Further reductions in electric usage are anticipated through the use of a belt-driven engine coolant circulation pump.

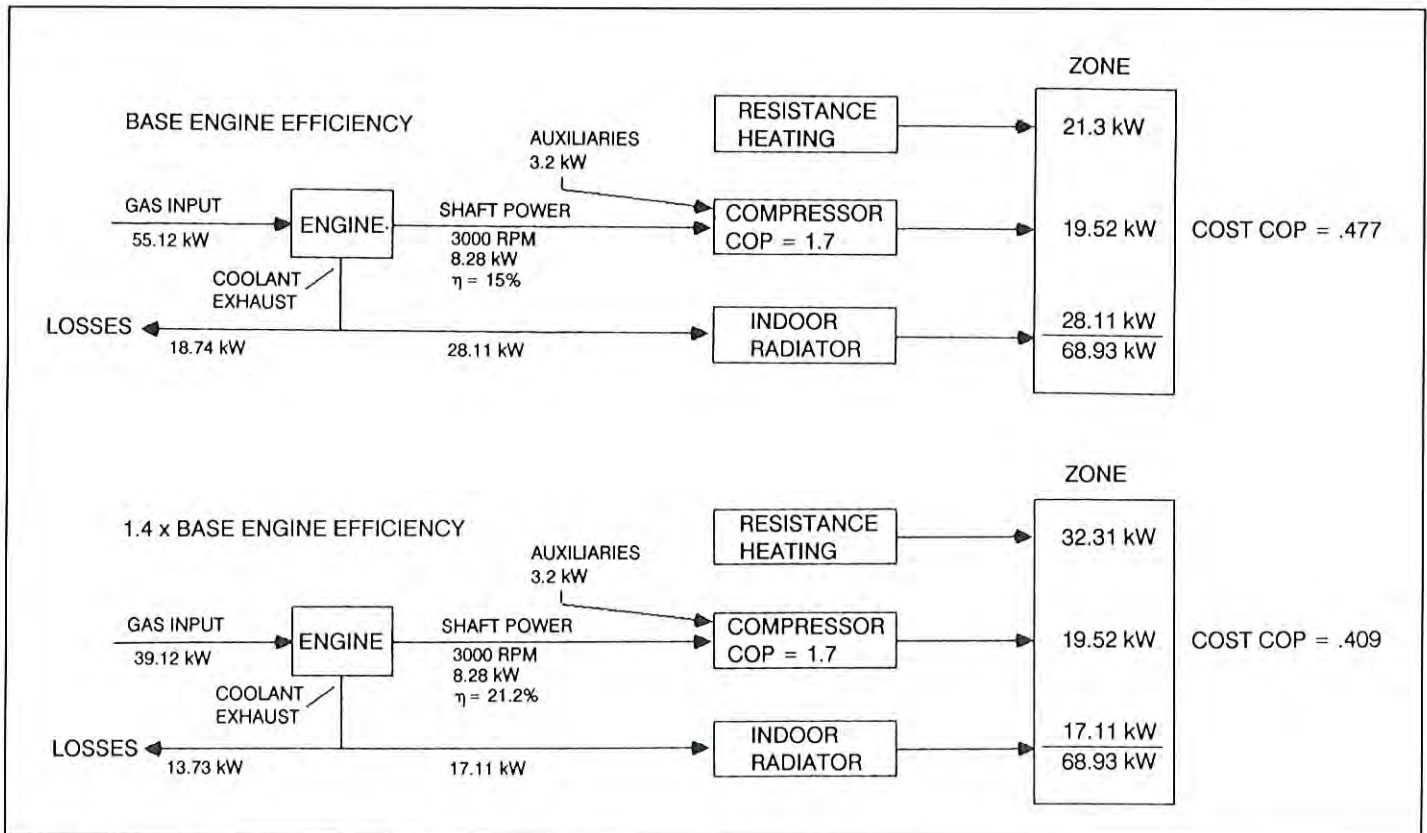


Figure 4. Effect of engine efficiency - heating mode (17°F/-22°C outdoor, maximum DHR)

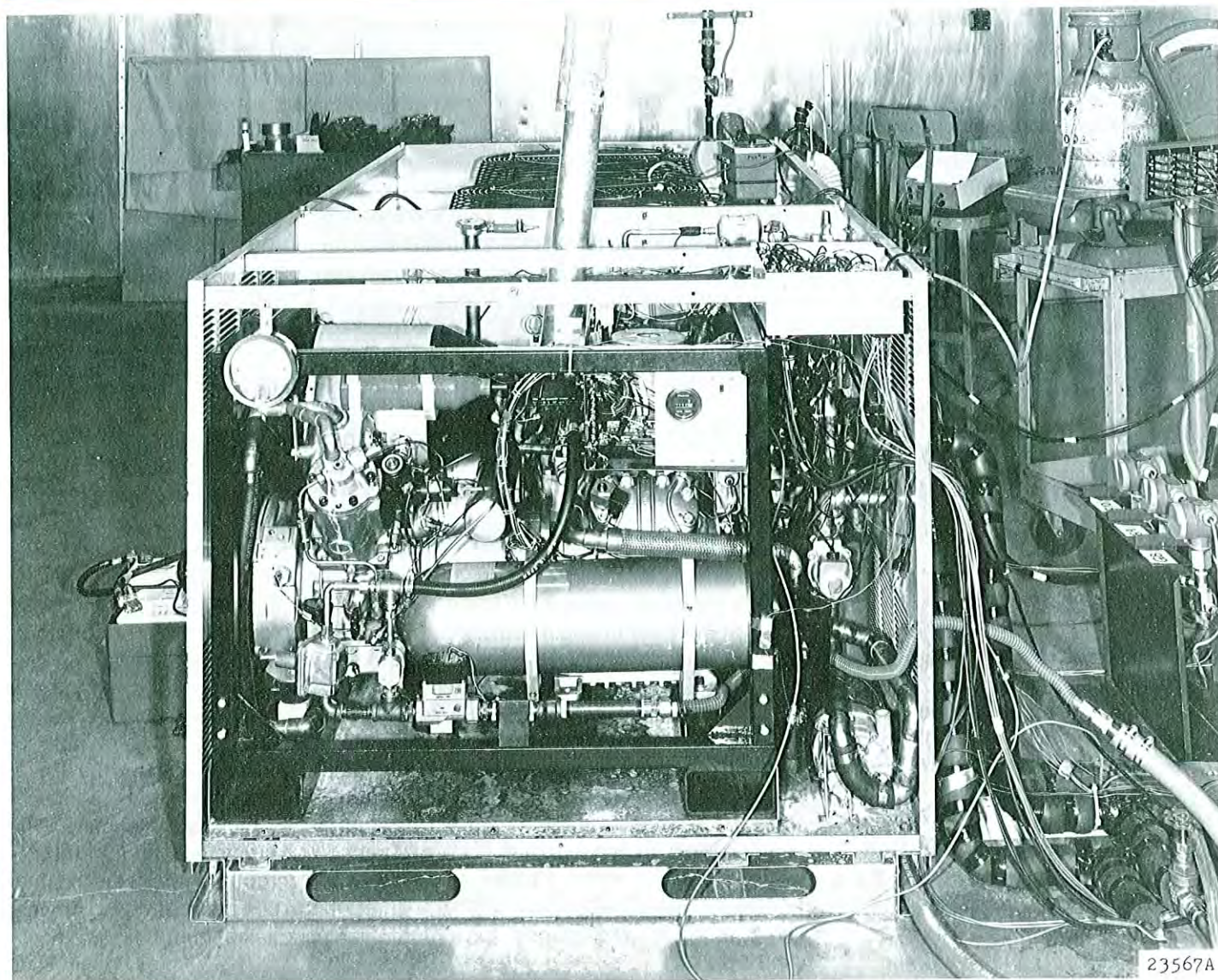


Figure 5. Second generation prototype heat pump, outdoor unit

Operating mode		Cooling		Heating		Annual	
		Oper. Costs	Savings	Oper. Costs	Savings	Oper. Costs	Savings
Standard elec. heat pump	Chicago	\$1948	--	\$3951	--	\$5899	--
	Dallas	3937	--	934	--	4871	--
Breadboard engine-driven system (ID + OD airflow fixed)	Chicago	1277	\$ 671	2192	\$1759	3469	\$2430
	Dallas	2675	1262	599	335	3274	1597
Fixed ID, optim. 2-step OD fan	Chicago	1245	703	2047	1904	3292	2607
	Dallas	2638	1299	557	377	3195	1676
optim. variable ID blower, fixed OD	Chicago	1143	805	2102	1849	3245	2654
	Dallas	2452	1485	579	355	3031	1840
Second generation prototype system (optim. variable ID blower, 2-step OD fan)	Chicago	1088	860	1935	2016	3023	2876
	Dallas	2381	1556	527	407	2908	1963

Table 3. Revised seasonal analysis projections and effects of airflow control changes

Full environmental laboratory testing of the second generation system is expected to be completed in late April 1987. Initial results show significant performance improvements, particularly through the reduction in electric consumption.

These test results will be used to identify further changes to be made in the design of the engineering field experimental units to be constructed later this year. In these modules, SPS will utilize the new "F"-series engine, incorporating a number of improvements for reliability, performance, cost, weight, and size, all resulting from over 122,000 hours of durability testing of the V160 engine.

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The Economic Margin for Alternatives in New Energy Technology

The following report extract was prepared by the Heat Pump Center. This report is the result of work carried out by a Systems Group to prepare material needed to assist the Swedish Council for Building Research in drawing up its Program Plan for Energy Technology for the three-year period 1987-1990. The Systems Group was formed within the Department of Building Services and Engineering at Chalmers University of Technology, and consists of the following persons: Enno Abel, Stefan Aronsson, Jan-Olof Dalenbäck and Per-Erik Nilsson.

Structure of the report

In order to assist the reader in locating those parts of the report which may be of particular interest, below is a brief description of how the report has been arranged, and a general indication of the contents of each chapter.

Introduction

The first chapter, Introduction, presents an overall review of the philosophy of research and development in the area of new energy technologies. An objective for any and all research and development work must be to arrive at design principles and actual plant design that genuinely assist the energy supply situation. Another equally important objective of research and development is that it should result in an addition to the general level of knowledge. In this report, only matters relating to economic evaluation are considered. This has been done primarily with the objective of identifying those technical solutions or lines of development that have little or no viability outside the research and development field. It also provides a means of ranking systems that are economically competitive.

Method of Assessment

The second chapter, Method of Assessment, describes how the economic margin for various alternative energy technologies is considered, and also how a number of basic concepts are

defined. Considering this purely in terms of the principles involved, two fundamentally different steps in the assessment process become apparent:

1. Weeding out of systems which have little prospect of becoming economically viable.
2. Assigning priority to, and ranking, systems which do have economic potential.

As far as the first of these steps is concerned, i.e., ascertaining whether a system is economically realistic or not, it is important that, if anything, assessment should err on the side of favoring the system. Weighing-up work in this preliminary "weeding-out" stage, must therefore be done by checking whether the limiting cost for the system, i.e., the lowest possible cost that can be attained in the future, lies below an upper cost limit which has been set on the high side. To determine this upper cost limit, for example, the cost for heat produced from expensive gas oil is used.

As far as the second step is concerned, i.e., that of relative ranking of different systems, a realistic cost for each system is considered. This is then compared to a reduced upper cost limit. The lower the upper cost limit that the system can beat, the higher it can be ranked.

All assessments are made with the basic assumption that each part of the system is either essential for the working of the system in its entirety, or that it can pay for itself by resulting in greater heat output.

Thus, for example, in the case of heat stores, the cost of heat storage must be recovered by the increase in heat production which the presence of the store allows. This marginal cost for heat must lie below the upper cost limit.

The value of any heat producing installation is determined by the necessary resources in the form of investment, operation, and maintenance that are needed to produce the heat. Some form of evaluation yardstick which objectively quantifies this is needed to evaluate any installation. The units or principle of the evaluation yardstick must be easily comprehensible, and there must not be any hidden meanings or effects which are not directly indicated by the name or type of units of the evaluation yardstick. This is a basic prerequisite if this means of quantifying the problem is not to result in misunderstandings or incorrect conclusions. Further, the evaluation yardstick should be some criterion which is natural to the reader, i.e., it must not give the impression of being artificial or unusual. In this report, the criteria of energy cost has been consistently employed, expressed in SEK/kWh, as the evaluation yardstick. Limiting cost, upper cost limit, marginal cost, economic margin, etc., are therefore always expressed in SEK/kWh.

The reasoning applied and the evaluations made must be generally valid in that they may not be linked to, or dependent on, one or a few specific designs only. Further, the reasoning must be sufficiently simple and easily understood to allow it to be easily checked. More advanced computer models have therefore been avoided, as the marginal increase in accuracy provided by such models does not offset the greater difficulty of the reader in checking the validity of the results.

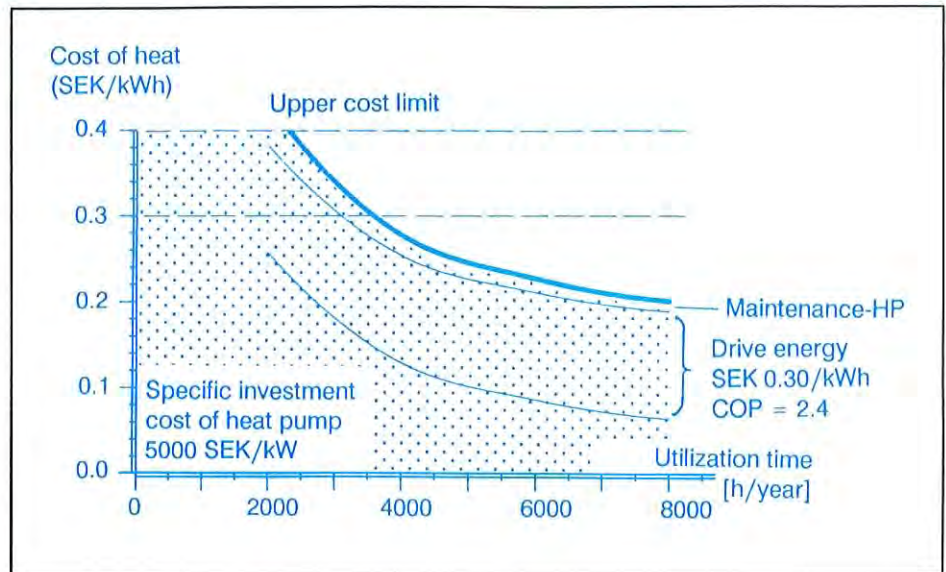
Alternatives Investigated

The third chapter, Alternatives Investigated, deals with costs and cost margins, mainly for variations/combinations of systems within the heat pump and solar heating sectors. The cost of heat from these systems with various specific capital costs is presented as a function of some parameter which is of importance for the system.

For heat pumps, one arrives at the relationship between the total cost of heat and utilization time. This is illustrated in Figure 1.

From such diagrams, it is possible to see how the cost of heat from heat pump systems with the same specific investment cost (SEK/kW) is affected by the annual utilization time and the cost of drive energy for the heat pump.

Over and above this, the diagrams can also be used to illustrate how, for example, the total cost of heat will differ from the design heat cost if the actual utilization time differs from the design utilization time.



Sample calculation for Figure 1

What annual utilization time is needed for a heat pump having a specific investment cost of SEK 5000/kW, if the cost of heat from it is not to exceed 0.30 SEK/kWh? The average cost of heat of electricity is 0.30 SEK/kWh, including the power charge element. It can be seen from Figure 1 that the utilization time must be at least 3900 h/year.

The heat cost graph obtained by adding the specific operating costs to the specific investment costs can also be used to reveal the economic margin for more expensive system alternatives. This is done simply by drawing in the upper cost limit in the heat cost/utilization time diagram. The economic margin for a more advanced system alternative then lies within the area between the upper cost limit and the heat cost curve.

As an example of this, try to find the economic margin for a heat engine as the prime mover of a heat pump.

This is illustrated in Figure 2.

It can be seen immediately from Figure 2 how extremely important a high utilization time is if a heat engine prime mover is even to be considered for the heat pump.

Figure 1. Total cost of heat as a function of utilization time

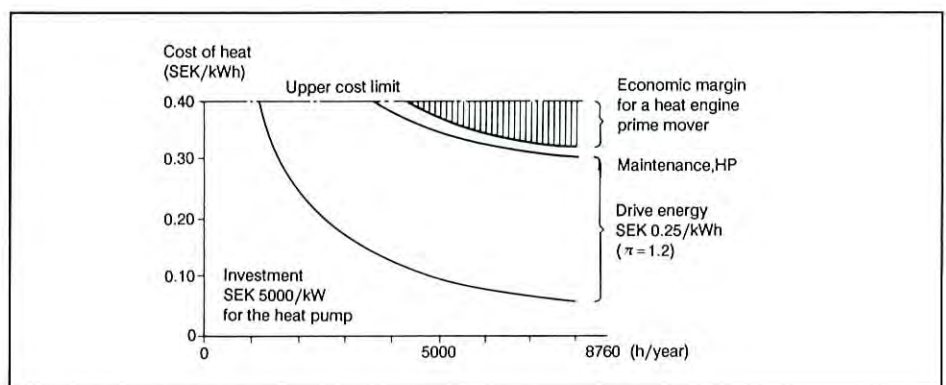


Figure 2. Economic margin for a heat engine prime mover for driving a heat pump

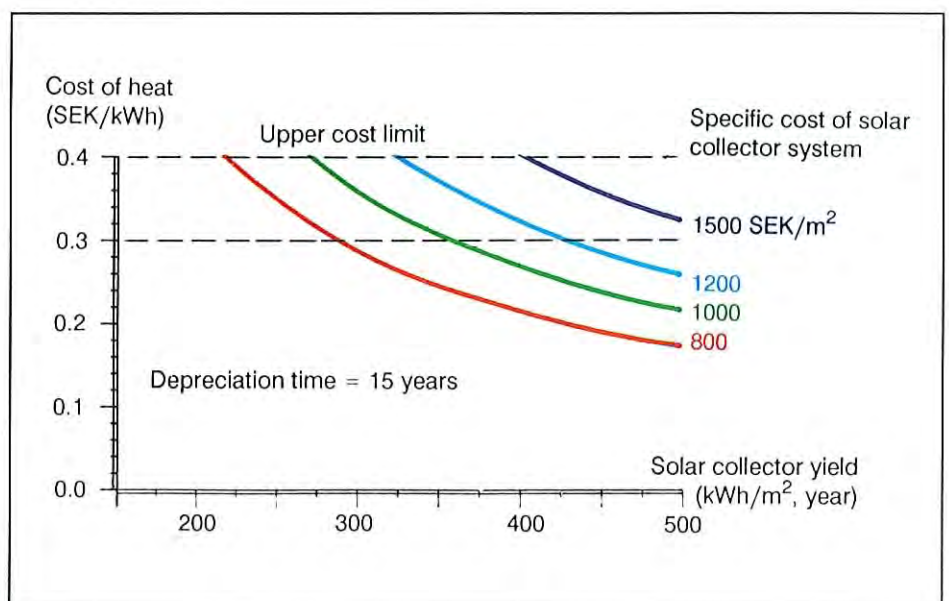


Figure 3. Cost of heat as a function of energy yield for a solar energy installation (6% real rate of interest)

For solar energy installations, one arrives at the relationship between the total cost of heat and solar collector yield. This is illustrated in Figure 3.

Sample calculation for Figure 3

What is the maximum specific cost of a solar heating system without seasonal heat store if it is connected to a large district heating system? Given:

Depreciation time	15 years
Upper cost limit	0.20 SEK/kWh
Solar collector yield	400 kWh/m ² , year

It can be seen from Figure 3 that the maximum permissible specific solar heating system cost is 800 SEK/m².

Economic Potential for Seasonal Heat Stores

In the fourth chapter, Economic Potential for Seasonal Heat Stores, the principles and methods described are illustrated by examples. Each example is intended to show the amount of economic margin available for heat stores in various systems.

The method of working as such can also be applied to parts of heat systems other than heat stores; it can be used, for example, to investigate whether a longer run of underground piping is viable in order to utilize some particularly good heat source. It can also be used to check whether extra piping can be justified in order to gain operational benefits. However, such points are not considered here.

A great number of systems have been considered, containing low-, medium-, and high-temperature heat stores, with and without heat pumps using different heat sources.

The following sample calculation (Figure 4) is given for the "storage strategy I" system using outdoor air. Similar calculations are performed for the remaining systems, mentioned above, in the complete report.

Assumptions

- Low-temperature heat store, charged with heat directly from outdoor air.
- Store working temperature: +5°C to +15°C.

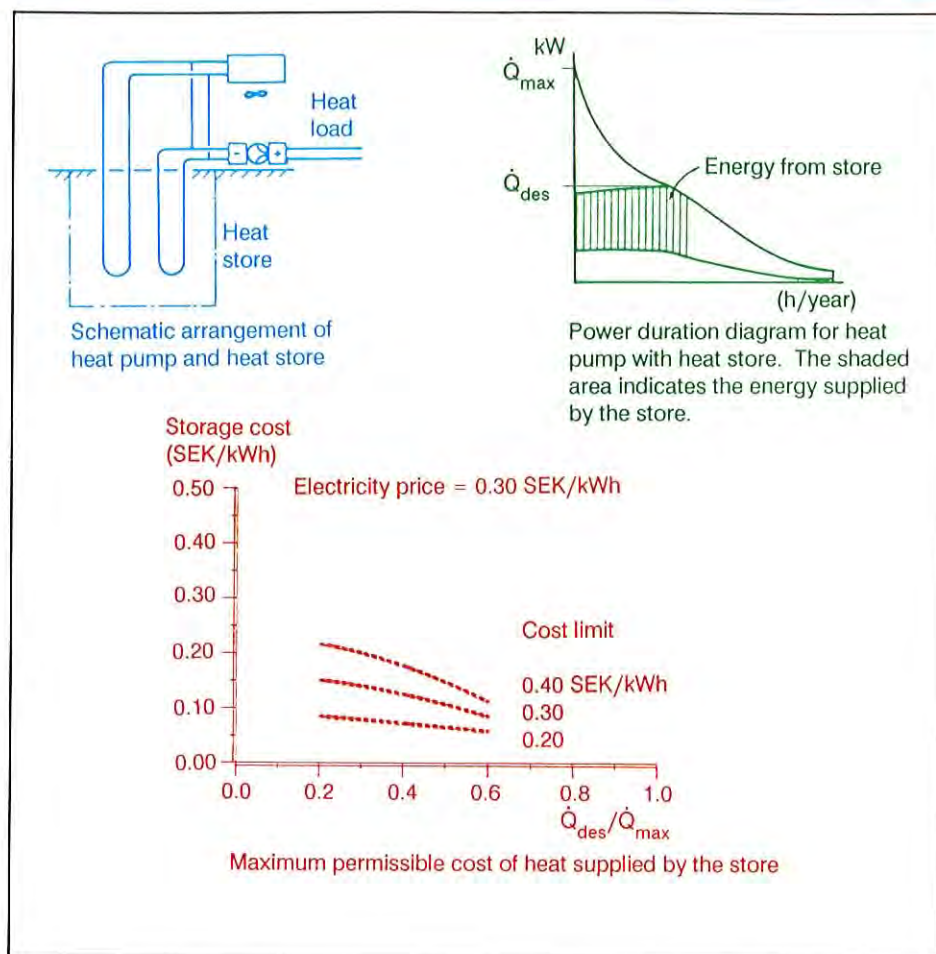


Figure 4. Heat from outdoor air (store operating strategy I)

- The heat pump extracts heat from the store only when the store temperature exceeds outdoor air temperature.
- The reference installation is an outdoor air heat pump.

In Figure 4, this cost has been calculated on the assumption that the marginal cost of the additional heat supplied by virtue of the presence of the store does not exceed the specific cost limit, which is the highest marginal cost that can be accepted without choosing to provide the heat in some other manner. The reference heat source/system chosen here is gas oil; three alternative oil prices have been used:

0.40 SEK/kWh	(= SEK 3,000/m ³)
0.30 SEK/kWh	(= SEK 2,300/m ³)
0.20 SEK/kWh	(= SEK 1,500/m ³)

The maximum permissible cost includes not only the cost of the store itself but also all costs for necessary connections on the supply and discharge side, as well

as additional costs involved for the heat absorber and other system parts as a result of incorporating the storage function.

A prerequisite for the diagram is that the cost of the total heat energy supplied by the system is less than the cost limit.

Appendix 1 describes in detail how the calculation of maximum permissible cost of a heat store is determined.

Reference

The Economic Margin for Alternatives in New Energy Technology, Swedish Council for Building Research, Report No. D4:1987, Stockholm, Sweden. This report is available in English.

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J. Bouma*

Mechanical Vapor Recompression in a Separation Process

Mechanical vapor recompression (MVR) is considered a reliable technique and is accepted by industry. The majority of the installations realized operate with water vapor as working medium. MVR is often applied in evaporation processes in the food industry, for instance, in the dairy, starch and beer industries in The Netherlands. Installations operating with a medium other than water vapor are not yet widely used. However, for separation and recovery of solvents, MVR also offers interesting prospects. In the following, an MVR installation for the separation of fatty acids in the chemical industry is described and some results of measurements are presented. This MVR project is part of the Netherlands Energy Development Company (NEOM) demonstration scheme. The installation was put into operation in 1985 at the Unichema Chemie plant in Gouda, The Netherlands.

System description

In producing fatty acids from tallow, a solution of unsaturated fatty acid (olein), saturated fatty acid (stearin), and water is obtained. Separation of the two acids results from their different solubility in methanol at low temperature. From the two fractions obtained, stearin and olein both in a methanol-water solution, methanol is recovered by distillation. In the previous installation, the distillation process was heated by steam. Both acids are distilled in two steps, i.e., in a pre-evaporator and in a column. In the pre-evaporator, preheated acid methanol water solution is heated and part of the methanol and water is evaporated. Final separation of the methanol occurs in the columns. Methanol and water vapor is then condensed and re-used. In the previous installation, 90% of the energy in the fresh steam was lost during condensation of the recovered methanol-water solution by cooling water.

The MVR technique was introduced in order to save on energy costs. Methanol-water vapors from the two pre-evaporators are compressed by one electrically driven centrifugal compressor. After compression, the vapor is condensed in the two pre-evaporators, where heat of condensation is transferred to the solutions. General data on the MVR installation are summarized in Table 1.

Heating capacity	3 MW	(design)
COP	15	(design)
Working medium	Methanol/water	
Annual operating time	7630 hours	(design)
Pressure ratio	2	(design)
Suction pressure/temperature	Atmospheric/72°C	(design)
Type of compressor	Centrifugal	(Alstom)
Type of drive	Electric motor	
Type of pre-evaporator	Falling film	(Wiegand)

Table 1. General data

Performance

From a first monitoring session, data have been gathered that enable quantifying the performance of the heat pump after approximately six months of operation. Performance data are summarized in Table 2. The COP of the heat pump is 19. When electric auxiliaries (pumps) are included, the COP is 12.4. Compared to expected performance data based on design values, the measured values are higher. The system is operated under steady conditions without significant load variations.

Heating capacity	3.17 MW
Driving energy	0.166 MW
Elec. consumption auxiliaries	0.088 MW
COP	19
COP, incl. auxiliaries	12.4

Table 2. Monitored performance

Operating experience by the user is positive. Besides energy cost savings, the MVR system also offers process

technical advantages. Capacity control is easy and the system offers good flexibility. Control of the total process is computerized. Moreover, quality of separation with the MVR system is as expected and constant.

There have been no major failures with MVR equipment in the first year of operation. The only incident that occurred was a damaged inlet guide vane in the control section. As far as scaling is concerned, no problems have occurred. A second monitoring session is scheduled two years after the first session, i.e., November 1987, in order to incorporate optimization and other effects.

Economical aspects

The estimated annual saving is 3 million m³ natural gas equivalent. Total capital investment equals 3 million Dutch guilders and includes compressor, two

falling film evaporators, two separators, heat exchangers, tanks, pumps, piping, instrumentation, spare parts, erection, insulation and civil works. Parts of the previous installation could be re-used like heat exchangers, pumps, reboilers, columns and tanks. The decision of investment was made on an estimated three year payback time. Since heat pump performance is better than expected and no severe breakdowns have occurred, the real payback time might be even shorter than three years. The real energy saving capability and return on investment will be available after the second monitoring session.

The results thus far have shown that MVR can be well applied for separation and evaporation processes in chemical industry for solvents other than water vapor.

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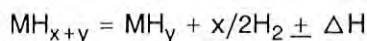
Recent Developments of Metal Hydride Heat Pumps in Japan

A large amount of thermal energy is involved in the dehydriding and hydriding reactions of hydrogen with hydride-forming materials. Heat involved in each reaction can be used to extract and supply heat for thermal energy storage and conversion, air conditioning, heating, drying, and humidity control devices. Such hydride-forming material is called a "metal hydride" and has been considered as the material extensively applicable to energy storage and conversion. A heat pump is a typical application for the dehydriding and hydriding characteristics of metal hydrides, particularly as a cooling and temperature upgrading device. This is a summary report on the recent developments of metal hydride heat pumps in Japan. Details of R&D activities in Japan and technological topics are referenced.^{1,2}

Principles of metal hydride heat pumps

The reaction between hydrogen and hydride-forming materials is reversible. It is characterized by rapid kinetics and large amounts of heat in an endothermic desorption reaction ($\rightarrow$ in Equation 1) and exothermic absorption reaction ($\leftarrow$ in Equation 1).

Equation 1:



The amount of heat ($\pm \Delta H$) in each reaction has an average value of 6.4 to 9.2 kcal per mole of H_2 or 160 to 230 kJ per kg of alloy. In the case of Mg-based alloys, much higher values are obtained. In the former case, approximately 1000 Kg of the alloy are necessary for a heat pump with a capacity of 100 kW.

To execute a heat pump cycle, a set of two paired metal hydride heat exchangers are necessary. With this system, a continuous countercurrent flow of hydrogen can be maintained even though the hydriding and dehydriding reactions are executed in a batch-wise manner. Figure 1 shows a metal hydride heat pump cycle providing a low temperature (T_l) sink which can be used for air conditioning and refrigeration purposes. The points 1-4 indicate the final states that the metal hydride and metal achieve during execution of the cycle.

Lines labeled MH_l and MH_h are lines of constant concentration for the low vapor pressure (low concentration of H_2) and high vapor pressure (high concentration of H_2) metal hydrides. These equilibrium temperature and pressure relations are given as van't Hoff plots.

The cycle is executed as follows (for clarity, a single pair of high and low pressure hydride heat exchangers is used to describe the cycle). Let metal M_h be at temperature T_l and pressure P_3 , and let metal hydride MH_l be at temperature T_m and pressure P_4 . With these conditions, metal hydride MH_l is heated at point 1 to raise its temperature from T_m to T_h . At the same time, it is used to supply hydrogen to metal M_h . Metal M_h , reacting with hydrogen from MH_l , releases energy causing its temperature to rise from T_l to T_m . At point 2, metal M_h has absorbed all the hydrogen to be-

come MH_h at T_m and P_2 . Metal M_l is now cooled to T_m by rejecting heat to the atmosphere. MH_h with its vapor pressure $P_2 > P_4$ is then used to supply H_2 to metal M_l . Since the release of H_2 is an endothermic reaction, the temperature of MH_h drops. Once it reaches temperature T_l , heat is added to the metal hydride, MH_h , to maintain this temperature until all the hydrogen has been driven off. The hydrogen is absorbed by the metal M_l until the metal hydride MH_l is formed. The energy released during this exothermic absorption of H_2 is rejected at point 4, thereby maintaining the low pressure metal hydride at T_{m-m} and P_4 . The cycle can now be repeated. To summarize, in Figure 1, heat is rejected to the atmosphere (T_m) at points 2 and 4. At point 1, a high temperature source (T_h) adds heat to the cycle; and at point 3, heat is absorbed from a low temperature source (T_l).

Figure 2 shows a metal hydride heat pump cycle useful for temperature upgrading and heating. Heat is supplied at temperature T_m at both hydrogen absorbing points (1 and 3). At points 2 and 4, heat is added from a high temperature source (T_h) and a low temperature source (T_l), respectively.

Several double-stage types of heat pumps have been proposed as shown in Figures 3 and 4. For these cycles, six heat exchangers composed of three different types of metal hydrides are used. The hydrogen is circulated in a stepwise manner through the heat exchanger system. In Figure 3, one part of thermal input at point 1 generates two parts of low temperature source (points 3 and 5) and three parts of heating source (points 2, 4, and 6). In a counter-clockwise cycle (Figure 4), the temperature can be upgraded to a much higher level than is possible with a conventional single-stage heat pump. In this case, points 1, 5, and 3 supply useful energy at temperature T_m .

Developments of metal hydride heat pumps in Japan

The world's first attempt to develop a heat pump for residential use had been initiated by Sekisui Chemical Industry in 1980. The three-year project was supported by the Japan Research & Development Corporation (JRDC).³ Chudenki Kogyo constructed a demonstra-

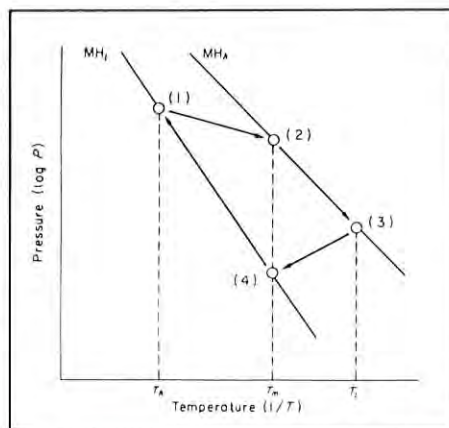


Figure 1.

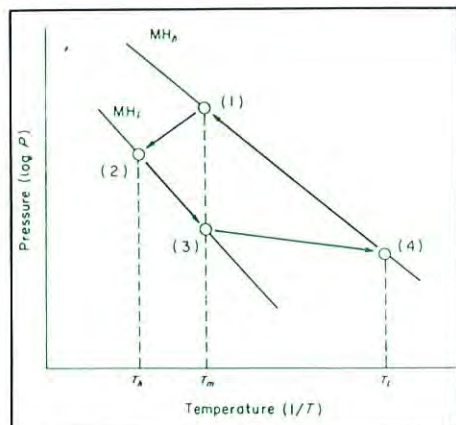


Figure 2.

tion plant (Figure 5) with a capacity of 1.75 kW to produce chilling water in its Kashima farm in 1982.⁴ In 1982, Japan Metals and Chemicals also developed a test unit with a 1.75 kW cooling capacity.^{5,6}

These three developments have attracted strong attention from other Japanese industries because of the

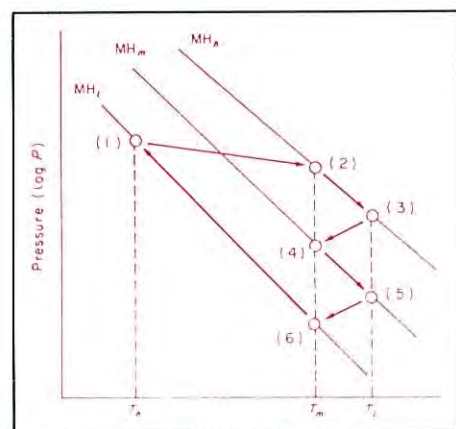


Figure 3.

potential for the recovery of industrial waste heat and the utilization of natural energy resources such as solar and geothermal energy.

Governmental projects have been playing an important role in the area of metal hydride R&D in Japan. Several national projects operated by the Sunshine Project Promotion Headquarters, Agency of Industrial Science & Technology (AIST), and Ministry of International Trade and Industry (MITI), are being carried out at the National Chemical Laboratory for Industry (NCLI) and at the Osaka Government Industrial Research Institute (GIRI). A testing unit for thermal energy storage was developed in NCLI in 1978 which used Mg-based hydrides⁷. These

research results have been used for further development of energy conversion devices. GIRI has engaged in the development of a heat pump (Figure 6). Special attention is being given to the development of heat exchanger modules composed of Cu-encapsuled metal hydride particles which are mechanically compressed under pressure to form a rigid heat transfer layer.^{8,9} An alternative governmental project was carried out by AIST in collaboration with

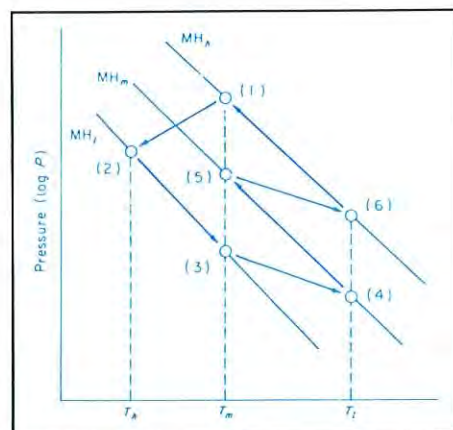


Figure 4.

National Research Institute for Metals, National Aerospace Technology Research Institute, Tokai University, and Kawasaki Heavy Industry in 1980 and 1985.^{10,11} Wind energy is collected to generate thermal energy by adiabatic compression of air. The thermal energy is then used to release hydrogen from a metal hydride reservoir which is stored in a gas holder. The hydrogen is in turn transferred back to the metal hydride

reservoir where the heat of absorption is released and available for heating.

Table 1 illustrates the current status of heat pump developments in this country. Excluded are unpublished works such as those performed by Matsushita Electric (double-stage heat pump) and Sumitomo Metals (heat pump for temperature upgrading above 300°C).

Feasibility studies on the applications of metal hydride heat pumps in industrial sectors have been reviewed in order to develop a computer simulation procedure for plant design and technical evaluation purposes.^{12,13} A computer has also been used to control the complex hydrogen flow in a six-reactor system.

An off-peak power storage unit with a capacity of 420 kWh has been successfully operated from February of 1987 to date for air-conditioning and heating of a research building in Kogakuin University (Hachioji Campus) (Figure 7). Approximately 10 tons of hydriding alloys are used to generate chilling water and hot water under temperature levels of 0-5°C and 45-65°C, respectively.

Technological problems

Hydriding materials

Economical feasibility of metal hydride heat pumps is strongly dependent on the properties of the hydriding material; particularly important are the equilib-



Figure 5.



Figure 6.

rium pressure- temperature- composition relations (P-T-C relations) and the amounts of hydrogen transferable between a paired metal hydride system. The latter is defined as the available hydrogen contents (AHC). It should be noted that this value is not necessarily proportional to the absolute amount of hydrogen to be contained in a given metal hydride, but is affected markedly by such equilibrium properties as isothermal pressure hysteresis and the slopes of the plateau region. Therefore, alloy cost should be evaluated not by weight base but by the value of AHC per weight of hydriding alloy.² It is obvious that rare-earth based materials are less expensive than Fe-based materials from such a viewpoint. Among the many candidates for the constituents of a heat pump, La-Ni based and Mm*-Ni based multi-component hydrides are presently the most practical and realistic materials. Unfortunately, the AHC value of any hydride forming material is too small to make the metal hydride heat pump competitive against conventional compression and absorption heat pumps. For this reason, materials research should strongly be pursued with the purpose of discovering hydriding materials which have at least 1.5 to 2.0 wt. % of AHC. International collaboration in this area is strongly recommended.

(*Mm [mixed metal] is a mixture of rare-earth elements and is thought to be a less expensive source for forming hydriding materials than refined Lanthanum.)



Figure 7.

Heat transfer properties

The average particle size of well-activated metal hydrides, after repeating many cycles of the dehydriding/hydriding reaction terminates in the range of a few tenths of a micrometer. The size distribution causes poor thermal conductivity in the metal hydride bed, and as a result, the heat of reaction transferred through the reactor wall is reduced considerably. Improvement of the thermal conductivity of a metal hydride bed is a major problem in this field.^{14,15} Two noteworthy approaches to this problem are the work done by GIRI (heat exchanger module of Cu-encapsulated hydride)⁹ and NCLI (low melting point hydride with excellent heat transfer properties)⁹. The heat and hydrogen transmission char-

acteristics of heat pumps have been studied extensively at the Kogakuin University Chemical Energy Laboratory (KUCEL).¹⁶

Conclusions

Metal hydride heat pumps have been actively developed in Japan by more than twelve different institutes, universities, and private firms. The present status of such developments were briefly reviewed here. The economical feasibility of the industrial and commercial application of this technology is largely dependent upon the metal hydride properties, particularly the AHC-

value. Continuous efforts in the area of material science must be devoted to the discovery of hydride-forming materials with a high hydrogen capacity.

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Developed By	Types & Uses	Thermal Output (kcal/h)	Temp. Cond.	Hydriding Alloys Used	Total Alloy Amounts (kg)
Sekisui Chemical Ind.	Heating & air-cond. (+ hot water supply)	9,660 15,000	$T_h = 92-11^\circ\text{C}$ $T_m = 40^\circ\text{C}$ $T_l = 10^\circ\text{C}$	LaNi- LaNi-Al-	--
Chuo-denki Kogyo	Chilling water	1,980	$T_h = 164 \pm 4^\circ\text{C}$ $T_m = 19.5 \pm 1^\circ\text{C}$ $T_l = 13.5 \pm .5^\circ\text{C}$	MmNi- LaNi-	40
Japan Metals & Chemicals	Air-cooling	1,500	$T_h = 140^\circ\text{C}$ $T_m = 45^\circ\text{C}$ $T_l = 10^\circ\text{C}$	MmNi- LaNi-	19 19
	Heating & air-cond. (high overall coeff.)	4,000	$T_h = 150^\circ\text{C}$ $T_m = 50^\circ\text{C}$ $T_l = 5^\circ\text{C}$	La-Ni ₅ LaNi _{4.7} Mn _{0.14} Al _{0.15} LaNi _{4.42} Mn _{0.48} Al _{0.05}	48
Toyobo and Kubota & Works	Temp. upgrading	3,000	$T_h = 90^\circ\text{C}$ $T_m = 70^\circ\text{C}$	LaNi _{4.7} Al _{0.3} LaNi ₅	46
			$T_l = 15^\circ\text{C}$	LaNi _{4.7} Al _{0.8} MmNi _{4.5} Al _{0.5}	49
Matsushita Elec.	Cooling	1,600		Ti _{0.9} Zr _{0.1} Mn _{1.8} Cr _{0.2} V _{0.2} LaNi ₅	17 18
			$T_h = 150^\circ\text{C}$ $T_l = 15^\circ\text{C}$	TiMn-	4
	Heating & cooling (double-stage)	6,000	$T_h = 280^\circ\text{C}$ $T_m = 144^\circ\text{C}$ $T_l = 5^\circ\text{C}$	TiMn-	180
Nippon Steel Works	Heating & air-cond.	13,000	$T_h = 72^\circ\text{C}$ $T_m = 50^\circ\text{C}$ $T_l = 2.5^\circ\text{C}$	Ca _(9.1wt%) - Ni _{85.1} Mm ₅ Al _{0.79}	140
				Ca _(3.2wt%) - Ni _{67.8} Mm _{25.9} - Al _{3.11}	140
Japan Chem. Eng., Chuo-denki Kogyo and Sekisui Chem.	Temp. upgrading/ steam generation (double-stage)	66,000	$T_h = 150^\circ\text{C}$ $T_m = 80^\circ\text{C}$	3-different rare-earth nickel hydrides	2,400
Kurimoto & Works	Temp. upgrading	1,000	$T_h = 100^\circ\text{C}$ $T_m = 80^\circ\text{C}$ $T_l = 20^\circ\text{C}$	LaNi ₅ LaNi _{4.7} Al _{0.3} *	--
Materials & Energy Research Inst.	Heating & air-cond. (off-peak power storage)	362,000	$T_h = 160^\circ\text{C}$ $T_m = 45 \quad 65^\circ\text{C}$ $T_l = 5 \quad 10^\circ\text{C}$	LaNi- LmNi- MmNi-	10,000
T_h = Highest temperature T_l = Lowest temperature T_m = Intermediate temperature L_m = La-rich mixed metals					

Table 1. Developments of metal hydride heat pumps in Japan

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Heat Pumps in Evaporation

Many unit operations in the process industries are designed in isolation without adequately considering the potential for energy integration with the surrounding process. This practice is evident in the design of evaporators for the process industries. Evaporation system design is often completed and only partially optimized within the constraints of the evaporation requirement itself. However, the real question remains unanswered: Has the correct evaporation system been specified, especially when considering integration potential with the surrounding process? This paper presents the results of a Process Synthesis Study by Union Carbide Corporation for the Electric Power Research Institute (EPRI) of Palo Alto, California, USA. The study objective was to evaluate the potential for utilizing electric heat pumps in evaporation processes within the process industries. Using process synthesis techniques, a study was completed on ten evaporation intensive processes from within five process industries. The results for each process are summarized, especially the feasibility of utilizing electric heat pumps in the evaporation system. The effect of considering the surrounding process in the design of the correct evaporation system is also discussed for both retrofits and new designs. Conclusions are presented to facilitate generalized guidelines for proper evaporation system design. Process synthesis techniques that are relevant for proper data extraction and evaporation system design are discussed to appropriately demonstrate correct application of the technology. Identification of various types of background processes and how they interact with the evaporation system are also discussed.

1. Introduction

The evaluation techniques utilize recent and powerful developments in the field of Process Synthesis, such as the process pinch, process performance targets, and appropriate placement of heat pumps with regard to the pinch. These concepts have been incorporated into a computerized engineering workstation for process synthesis and optimization known as The ADVENT™ System, which was used for this particular study.

1.1 Study Objectives

The major objectives of this study were:

- To evaluate the potential for electric heat pumps in evaporation processes within the process industries using process synthesis technology.

-- To draw specific conclusions regarding this potential for the individual processes and industries selected for the study.

-- To draw general conclusions regarding this potential for the entire process industries.

-- To develop "Applications Guidelines" to assist potential users of electric heat pumps to identify valid applications.

-- To identify specific problems that stand in the way of the broader use of electric heat pumps and suggest research programs that might overcome these problems.

1.2 Study Assumptions

In this particular examination of evaporator systems, ten generic processes were

evaluated for potential electric heat pump applications. Each generic process was subjected to a process synthesis evaluation considering both the existing process design (i.e., retrofit) and a new grassroots design. The criteria for acceptable designs included both industry-accepted economic criteria and technical feasibility. However, the following limitations were placed on the study:

- Only open cycle heat pumps were considered for this study. All ten generic processes evaporated water as the solvent and the assumption was made that open cycle steam elevation compression systems should be used. These are generally considered to be best in such service.
- Electric driven compressors were considered to be single stage with compression ratios of less than 2:1.
- The required utilities specified at each process location were accepted for each generic process as all that were available to the process.
- Representative prices were assumed for various utilities.

2. Evaporation industry survey

2.1 Visits with evaporator system vendors

Most evaporator systems are purchased as a package from vendors. Consequently, many of the innovations in evaporator design originate with evaporator system suppliers.

A large, representative cross-section of evaporation system vendors in the United States were visited to obtain information relevant to the following areas:

- Industrial application of evaporation processes, potential market for heat pump technology, and industrial processes for further study.
- Recommendations for industry process consultants.
- Competing technologies.
- Application and acceptance of heat pump technology by evaporation system users.

- Restrictions, both real and perceived, which might limit the application of heat pump technology.

2.2 Industrial segment processes

This study groups evaporation processes into five industrial segments:

- Chemicals and petroleum
- Dairy
- Food
- Pulp and paper
- Textiles

Approximately 40 processes that use evaporation were identified. From these, ten processes were elected for more detailed study based on several criteria. At least one process from each industrial segment was selected. "One-of-a-kind" processes were avoided. Preference was also given to well-known processes

on which industry consultants could supply generic process data. Results are shown in Table 1.

2.3 Limitations to application of heat pumps

Several factors are real or often perceived to be limitations to the wider application of heat pump technology in evaporation processes. Perhaps the major limitation is the users' lack of understanding or familiarity regarding heat pump evaporator systems. Better education and publicity is required to counter this. There have also been a few misapplications of heat pumps and a few poorly conceived and designed systems which resulted in problems during operation. Some of these misapplications occurred in the Pulp and Paper Industry and the Food Industry. Sometimes the equipment was incorrectly specified or

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 - Dyeing - Finishing - Caustic recovery
	* Indicates process selected for additional study

Table 1. Industry segment processes

simply not suitable for the application. Most of these occurrences happened when heat pumps were just beginning to be seriously considered for evaporation systems. These few cases have received wide publicity, and the many successful installations may be ignored by someone not familiar with heat pump applications.

There are several factors that must be adequately considered when evaluating heat pump technology:

- Capital costs
- Electrical cost and availability
- Compressor maintenance
- Equipment limitations
- System start-up
- System turndown

2.4 Advantages of heat pumps

The most important advantage offered with proper application of heat pump technology is reduced energy requirements. Several other advantages may also be afforded by heat pumps.

- Isolate process fluids
- Low temperature processing
- Space requirements
- Cooling water
- Falling film evaporators

3. Process synthesis technology

The identification of the optimal evaporation system is complex. However, there are several tools available to the process engineer which will make the selection of the appropriate system much easier. A discussion of several of the most valuable tools and their implications is the primary focus of this section.

3.1 Composite curves - energy targeting^{1,2,3,4}

The temperature-energy diagram, often referred to as Composite curves, is a convenient way to view the heating and cooling going on in a process. Consider two hot streams being cooled. Their cooling curves on a T-Q diagram are shown in Figure 1. The same can be done for all hot streams being cooled in a process. A similar procedure can be followed by all cold streams being heated in a process.

The hot composite is the sum of all hot streams being cooled, and the cold

composite is the sum of all cold streams being heated (see Figure 2).

The composite curves can be drawn at any location along a Q axis and still convey the same information (i.e., same heat duty and T) as long as the temperature intervals and slopes are maintained; however, the position of the curves rela-

tive to one another does have meaning. As shown in Figure 3, the region of overlap represents the opportunity to recover energy from the hot streams by transferring it to the cold streams. This is possible as long as the minimum approach temperature, ΔT_{min} , is greater than zero. The remaining heating of the cold streams and cooling of the hot

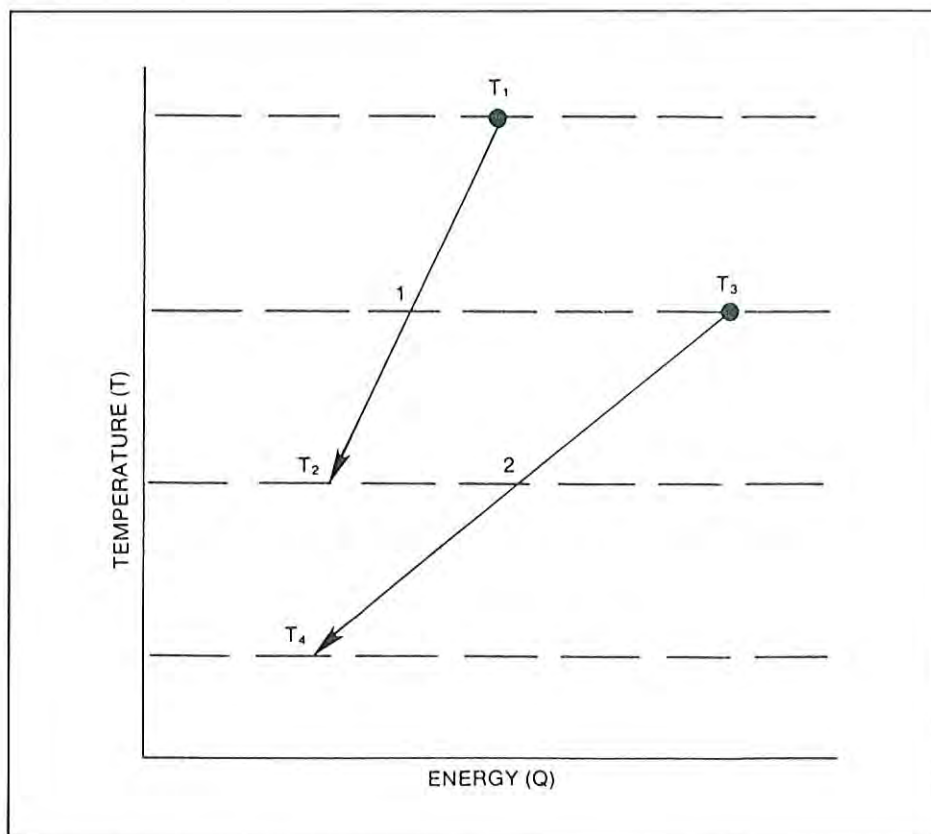


Figure 1. Cooling curves for two hot streams

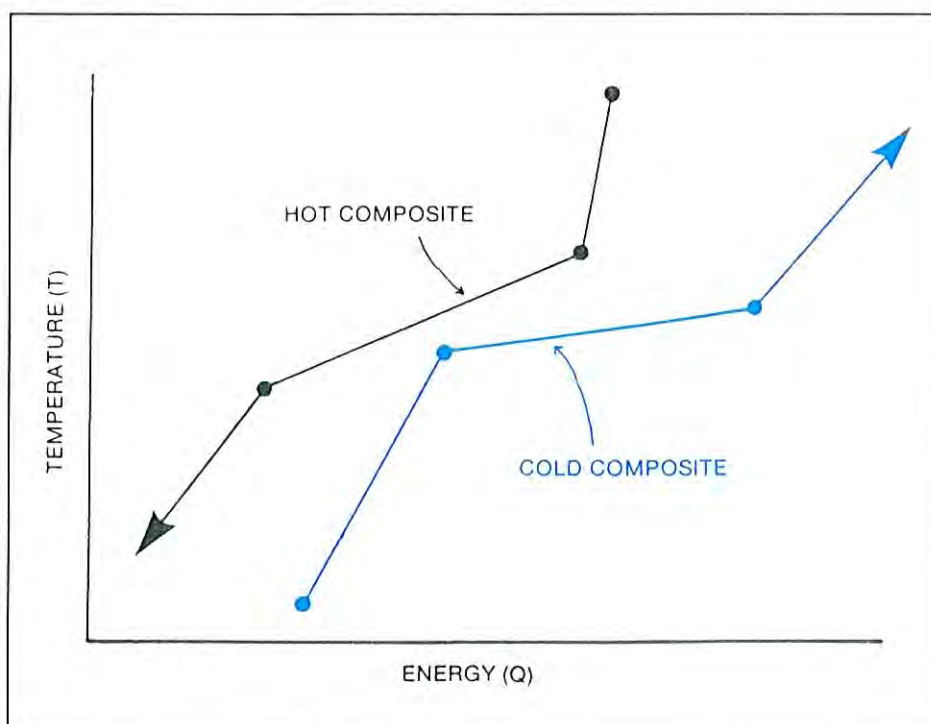


Figure 2. Hot and cold composite curves

streams must be accomplished with hot and cold utilities, typically steam and cooling water. The required amount of each utility is uniquely determined by $\Delta T_{\min}$.

As $\Delta T_{\min}$ increases, energy recovery decreases and utility usage increases. The heat exchange surface to recover the energy, however, decreases since driving forces are larger.^{1,4,5,6} For any $\Delta T_{\min}$, the composite curves define the minimum utility requirements for the process and, thus, establish the energy targets for the process. Note that this is done before the design of the heat exchange network is known.

The composite curves (shown in Figure 4) provide a new and useful view of the process. In general, these curves have a point of closest approach called the "process pinch."³ The pinch divides the process into two parts:

- Above the pinch there is insufficient energy in the hot composite curve to provide the energy needed to heat the cold composite. The deficiency is made up by a hot utility. The amount is the "hot utility target." The process above the pinch is deficient in energy and can be thought of as a heat sink.
- Below the pinch there is more energy available in the hot composite than is needed by the cold composite. This excess must be discharged into a cold utility. The amount is the "cold utility target." The process below the pinch has excess energy and can be thought of as a heat source.

When a process has been characterized as shown, some simple yet powerful rules emerge that have a significant impact on the process and minimum targeted utilities.⁴ These rules are as follows:

- Don't transfer heat across the pinch.
- Don't cool with utilities above the pinch.
- Don't heat with utilities below the pinch.

The significance of these three rules is that a violation of any of these rules results in heat transfer across the pinch and an increase in minimum hot and cold utilities.

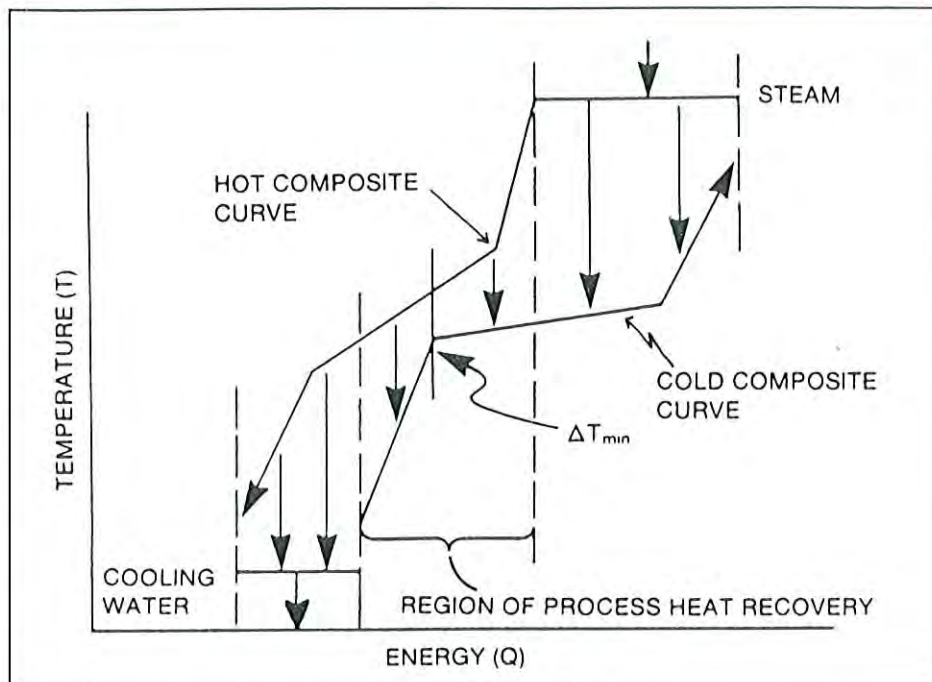


Figure 3. Process composite curves

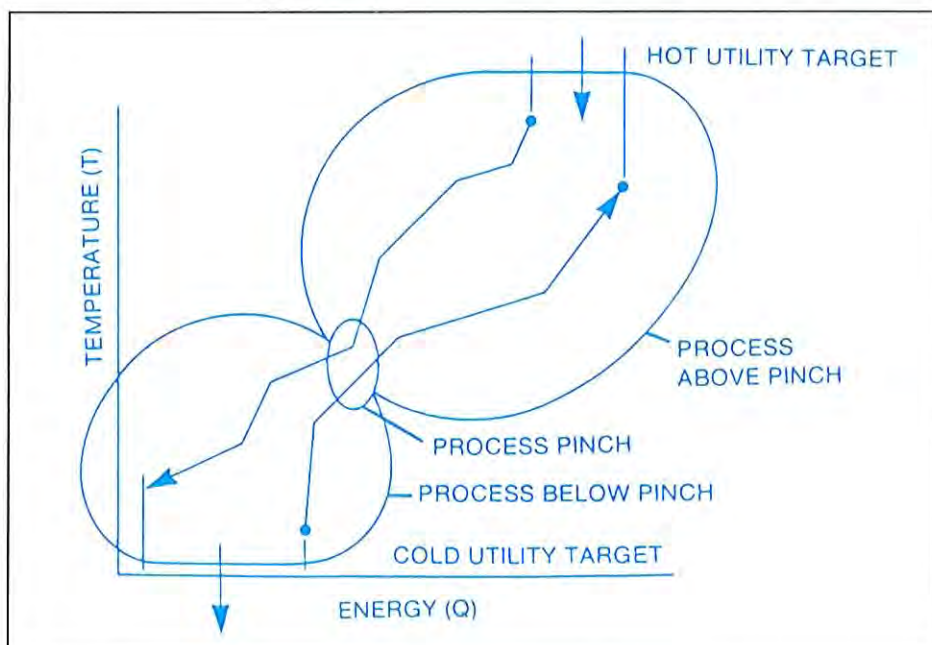


Figure 4. Process divided at the pinch

3.2 Grand composite curve⁷

The Grand Composite curve is another useful tool in evaluating the proper design of minimum energy processes. This type of curve (as shown in Figure 5) is derived from the Composite curves previously described. The Grand Composite combines both the hot and cold Composite curves together in a single curve which represents the utility targets, heat exchange driving forces and energy to be transferred.

The Grand Composite curve, just like its Composite curve counterpart, is represented on a temperature-enthalpy plot. The curve is constructed by shifting the hot composite curve down (normally one-half $\Delta T_{\min}$) and the cold composite curve up (normally one-half $\Delta T_{\min}$) until the two curves "touch" at the pinch. The Grand Composite curve is a plot of the horizontal enthalpy difference between the shifted hot and cold Composite curves versus the temperature at that particular enthalpy difference.

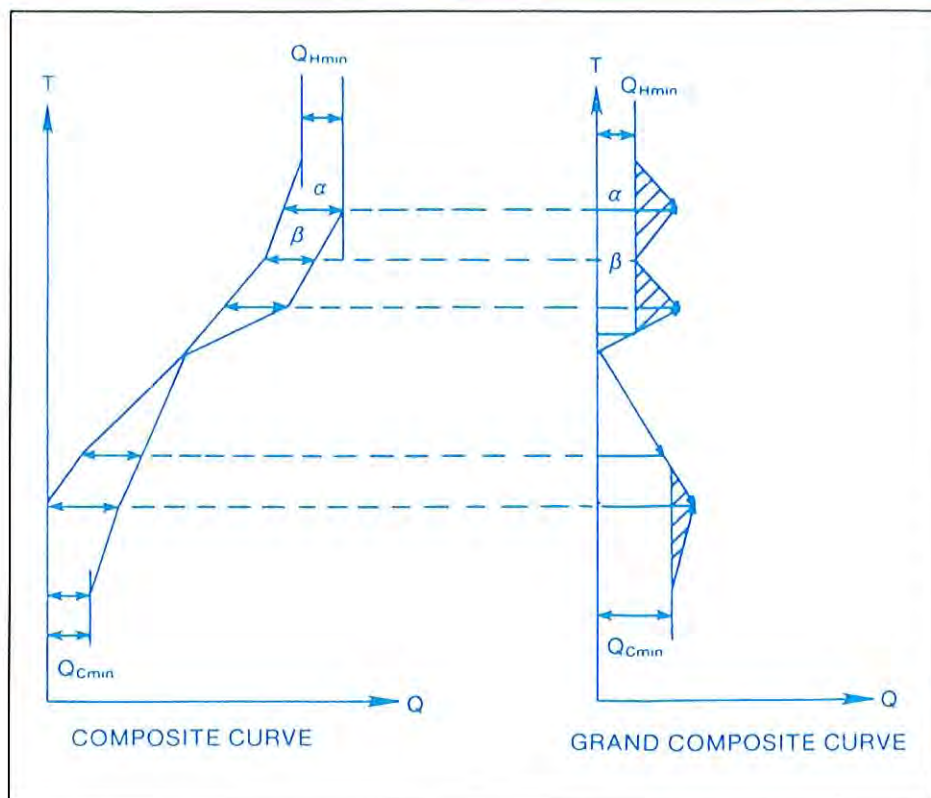


Figure 5. Construction of grand composite curve from composite curves

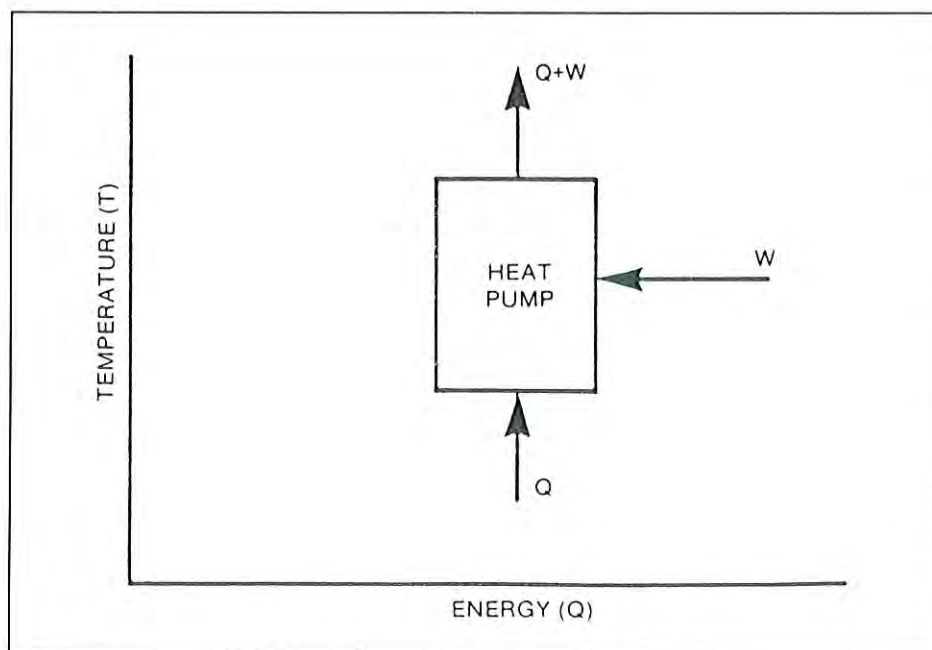


Figure 6. Thermodynamic model of a heat pump

3.3 Appropriate placement of unit operations⁴

The proper integration of various unit operations into a process can be determined by developing a thermal model (i.e., temperature vs. heat relationship) of the unit operation and then considering how this should be incorporated into a background process.

A heat pump may be thought of as a heat engine running in reverse.⁸ This is shown in Figure 6 as taking an amount of heat (Q) at a low temperature, adding work to it in the form of shaft power, and rejecting the heat ($Q + W$) at a higher temperature.

Consider two alternatives for integrating the heat pumps into the process. For the first case shown in Figure 7, note the following points:

- There is no impact on the process above the pinch, so the energy required remains A ; and
- Below the pinch, there is a net addition of W units of energy, which increased the cooling load by W .

The overall effect is that work, e.g., electricity, is cascaded into the cold utility.

The next configuration in Figure 8 shows the only economical way a heat pump has a chance to work. Energy is removed from below the pinch as the process desires and deposited above the pinch, again in agreement with process needs. Now, both heating and cooling are reduced. The heat pump is justified if the value of the utilities avoided, i.e., $Q + W$ hot utility and Q cold utility, exceeds the cost of capital to install the heat pump and the cost of the shaft work, W , required for it to operate.

4. Conclusions for applications of electric heat pump evaporators

This section discusses conclusions from the electric heat pump study. The discussion is divided into three parts: specific conclusions for each candidate process; general conclusions for all processes; and general rules for application of electric heat pump evaporators.

4.1 Specific conclusions for each candidate process

The ten candidate processes, representing five industries, are extremely different. As shown in Table 2, the ten candidate processes vary considerably in terms of energy consumption and current evaporation system design. In some cases, the target energy reduction might show considerable potential for improvement. In other cases, the target seems to indicate little potential for improvement. Sometimes, improvement potential was realized with process improvements.

In spite of the broad diversity among the ten processes, some generic conclusions can be made. These conclusions are summarized as follows:

- Most evaporation systems can be improved, especially when considering integration potential with the background process.

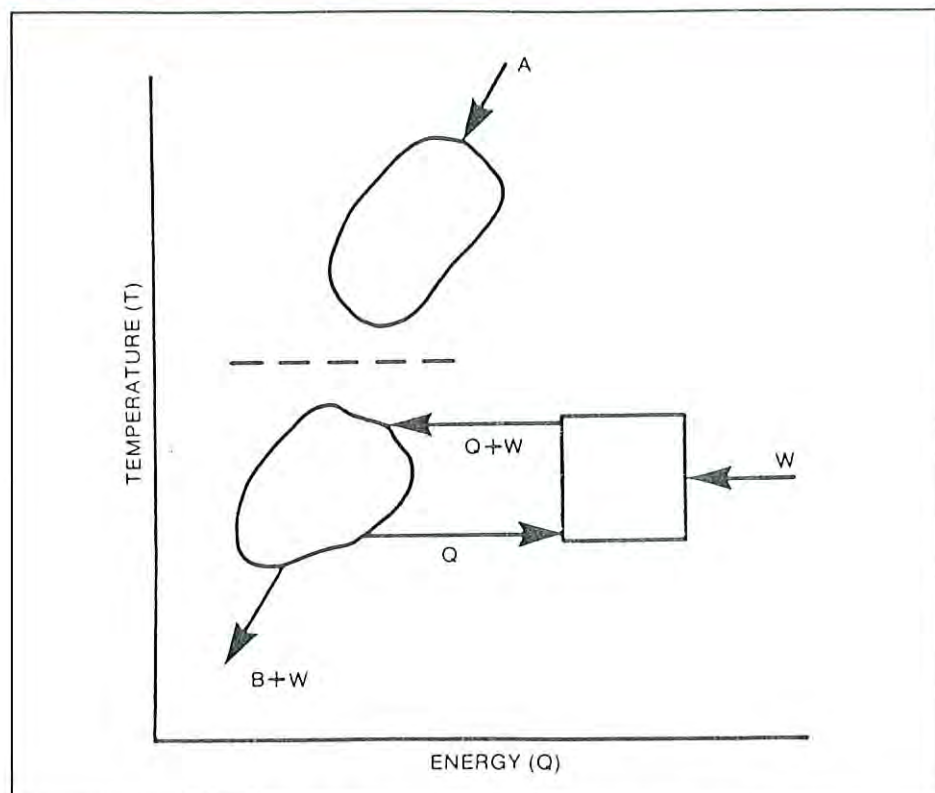


Figure 7. Improper integration of a heat pump into the process below the pinch

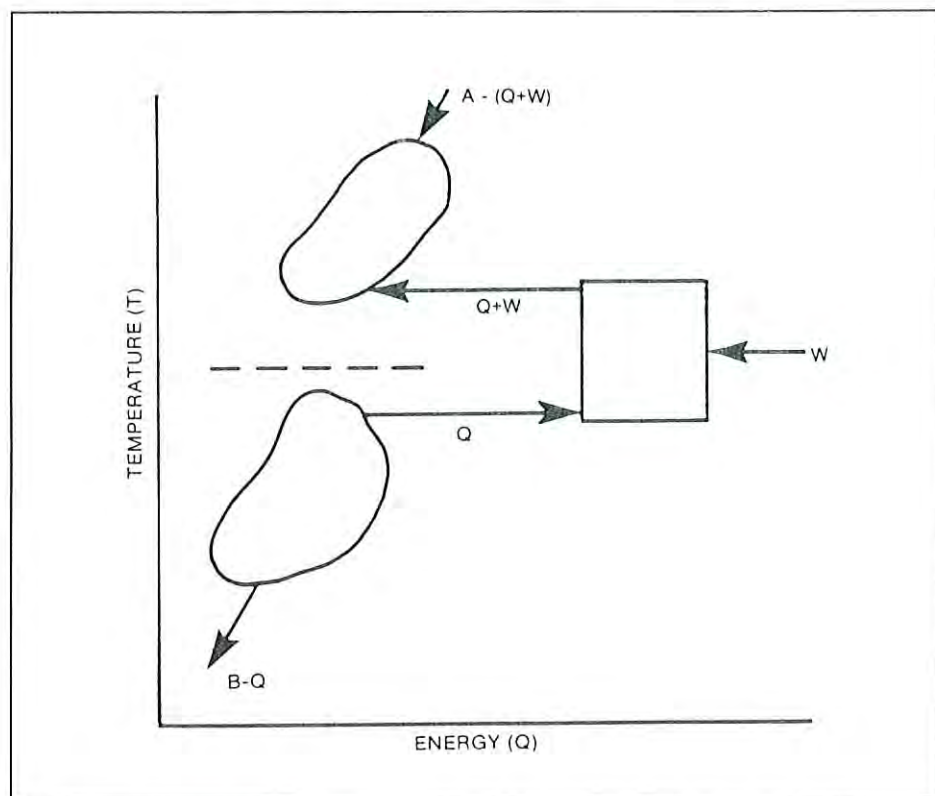


Figure 8. Proper integration of a heat pump into the process across the pinch

was confirmed as near optimal with the existing MVR/TVR evaporation system. The citrus juice concentration process could be improved but project economics were extremely poor.

Electrical MVR evaporation systems (or some combination or revision) were recommended for six of the ten candidate processes. The associated economics for these ideas generally averaged approximately a two year payback with variations between one and five years.

4.2 Generalized conclusions - all processes

The generalized conclusions are summarized below.

- The evaporation industry normally designs evaporation systems in isolation without consideration of the surrounding process.
- Evaporation systems are generally over-designed with missed integration opportunities from the surrounding process for integration improvements.
- Significant opportunities exist for improving evaporation system performance for both retrofit designs and new designs.
- These significant opportunities generally consider integration of the evaporation system with the surrounding process.
- In general, additional potential exists to utilize electrical MVR heat pumps for evaporation.
- Project economics appear to be attractive with simple economic paybacks averaging approximately two years.

An important point concerning evaporation system design needs to be emphasized. If the evaporation requirement is isolated from the remainder of the process and designed accordingly, the resulting design is probably not optimal. Evaporation equipment manufacturers attempt to design the best possible evaporation system (i.e., capital and operating cost) with the information that is given to them. However, failure to

-- Electrical MVR evaporation systems could be recommended for approximately one-half of the processes.

-- Where MVR is appropriate, project economics generally appear to be attractive for both retrofits and new designs.

Of the ten candidate processes, improvements were proposed for eight processes. Of the remaining two, one currently utilized an MVR/TVR evaporation system (cheese-whey) and the other currently utilized a multiple effect evaporation system (citrus juice concentration). The cheese-whey powder process

consider integration potential with the surrounding process often means missed opportunities for additional savings. Even though the resulting design may be more complex and require additional controls (the resulting system might even consider unequal heat duties and unequal areas), customers should know this information to pick the best possible design for their requirement.

Based on this premise, one of the most important conclusions can be made:

Do not design in isolation

To fulfill this requirement, three scenarios can be proposed:

- Train evaporation equipment manufacturers to practice process synthesis technology. The customer would then provide a more complete process package including flowsheets and heat and material balances. Through interaction with the customer, evaporation equipment manufacturers would design evaporation systems potentially integrated with the process.
- Train the customer's engineers, who might specify or operate the evaporation system, to practice process synthesis technology. The idea is for the engineer to conceptually design the evaporation system first. The conceptual design would consider integration opportunities with the surrounding process. Once completed, the conceptual design would be supplied to equipment manufacturers for more detailed design. Obviously, close interaction between the engineer and the equipment manufacturer would be necessary to complete the final design.
- Utilize a consultant who is knowledgeable with process synthesis technology. The customer and/or equipment manufacturer would utilize the consultant to conduct the process integration study and propose the best design. Both the customer and equipment manufacturer must be an integral part of the design process to make sure they understand what is being proposed, and more importantly, why it is proposed.

Process	Actual Hot Utility Consumption (MM Btu/Hr)	Target Hot Utility Consumption No Process Modifications (MM Btu/Hr)	Target Hot Utility Consumption After Process Modifications (MM Btu/Hr)	Current Evaporation Design
1. Fermentation alcohol	145	87	87	MVR
2. Chlor-alkali				
-diaphragm	151	150	100	4 Multiple effect
-membrane	43	41	41	3 Multiple effect
3. Potash	49	49	38	5 Multiple effect
4. Nylon 6-6	14	7	7	1 Single effect
5. Citrus juice	12	11	11	5 Multiple effect
6. Corn milling/HFCS	73	58	58	2 Effect TVR
7. Cheese whey	6	4	4	MVR/1 Effect TVR
8. Skim milk powder	14	4	4	5 Effect TVR
9. CTMP	27	16	16	MVR
10. Kraft black liquor	181	108	108	6 Multiple effect

Table 2. Candidate process summary

One final general conclusion should be mentioned. When process synthesis technology has been properly utilized in the design of evaporation systems, a truly optimal system can be developed, given the current state of the process technology. This optimal design should help to move the total process closer to minimum energy targets with little additional potential for improvements. Also, no surprise in the design will arise since the system is already near optimum.

4.3 General rules for application of electric heat pump evaporators

There are four principles which should be considered when evaluating an evaporation system for possible MVR application. The four important criteria are: heat loads, boiling point rise with concentration, the relations of the evaporation to the process pinch, and the service requirements of the unit.

Heat load and savings. The heat load of the required evaporation plays an important role in the design and economic advantage of an electric heat pump system. One consideration, generally, is that the operating cost of the MVR system is smaller than the operating cost of the multi-effect system; however, most calculations show that for a given effective steam cost, the operating cost advantage is diminished for lower total evaporation loads even to the point of favoring multi-effect evaporation for a very low rates (i.e., less than 20,000 lb/hr evaporation duty).

This point is explained by the fact that the cost of energy for low evaporation loads

(i.e., less than 20,000 lb/hr) is a minor portion of the total cost of the system (operating cost plus capital investment) and the capital investment is a much more significant factor. Therefore, economic paybacks (i.e., time to pay back the capital investment) may tend to increase to the point of becoming unacceptable.

The conclusion is that for very small evaporative loads, there may be no total cost advantage for using the MVR system or that the payback on the MVR unit may be very long due to the low operating cost advantage inherent in the small evaporative load systems.

Very large evaporative loads, i.e., those above 1,000,000 lbs/hr, cannot be accomplished in a single radial flow compressor and would require either an axial flow compressor or multiple radial flow compressors. Both of these alternatives increase the cost of the compression units and thus result in longer paybacks for the units.

Boiling point rise. The boiling point elevation with increasing concentration is important due to the mechanical limitations of single-stage compression. Single-stage compressors may provide up to a 2.0:1 compression ratio for low throughput systems. In general, practical single-stage compression ratios are in the 1.5 to 1.7 compression ratio range. The dew point of the condensing vapor must be raised above the boiling point elevation and the minimum approach temperature for heat transfer. The minimum approach temperature will usually increase for systems with low heat trans-

fer coefficients but for illustrative purposes, assume a 15°F minimum approach temperature. The compression ratios of 2.0:1, 1.7:1, and 1.5:1 result in an increase in the dew point of 37°F, 28°F, and 21°F, respectively. Thus, the maximum boiling point elevation for each respective compression ratio is 22°F, 13°F and 6°F, in order of decreasing compression ratio. Therefore, a good approximate rule might be to consider mechanical vapor recompression when the proposed concentration results in a boiling point rise of approximately 20°F for small loads and 10°F for larger loads with accompanying good heat transfer coefficients.

Pinch relation. The design of the evaporation system should consider the overall plant energy balances. It is possible for many evaporator designs to be integrated with the existing process and consume essentially no additional utility beyond the background process. An examination of the Process Grand Composite curve may demonstrate that MVR should not be used even when physically feasible. In general, the amount of the evaporative load supplied via vapor recompression should be less than the total load due to process integration. Only that portion of the evaporative load which extends beyond the pinch of the background process should be provided by a heat pump. Multiple effect evaporation that straddles or even causes the pinch may have reduced possibilities for economical heat pump use due to physical limitations (i.e., boiling point rise, etc.). However, even these heat duties can be reduced by considering combinations. One such combination may include using an MVR to pre-concentrate the feed to the existing evaporation system.

The tradeoff between capital investment, utility cost and operability must be examined for each identified design. The selected designs may all be identified on the Grand Composite curve in a systematic manner and their energy consumption noted. The final step in the design process must be to evaluate the effect on plant operability of each of the designs, ranking them in order of operability.

Service. The service of the unit is also an important factor. Highly corrosive materials, toxic chemicals and explosive

materials may require special handling techniques not normally applied to compressors, which may result in a higher capital cost or total infeasibility. Concentration of fluids which have extremely poor heat transfer characteristics and require higher minimum approach temperatures may also cause problems for heat pump systems.

Compressors will also require more general maintenance for fluid changes, oil leakage, or seal replacements. The plant scheduled downtimes should be examined in order to determine the effect of the compressor maintenance requirements.

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*W.D. Witherell, Linnhoff March, Leesburg, Virginia, and F.A. Connaway, Union Carbide Corporation, South Charleston, West Virginia, USA

LITERATURE REVIEW: INDUSTRIAL HEAT PUMPS

Macriss, R.A., J.M. Gutraj, and T.S. Zawacki. **Absorption Fluids Data Survey: Final Report on Foreign Data.** Institute of Gas Technology, Chicago, Illinois, USA, 1987. (English)

In a 12-month evaluation, 300 different foreign publications containing data that relate to properties of binary, ternary, and multicomponent absorption fluids were identified. From these publications, 127 were selected as sources of relevant foreign data. Fluids covered include combinations of each of 20 different refrigerant compounds, with one or more of 41 single, 13 binary, and six ternary absorbent compounds. Coarse screening and evaluations were performed. Data gaps for key fluids are summarized and unresolved conflicts in data are noted. Results show that with very few exceptions, all candidate fluids for devel-

opment activities dealing with advanced absorption heat pump concepts have data gaps that must be addressed in the near future.

Anonymous. **Optimum Heat Pump Placement in Industrial Processes: Final Report.** TENSA Services, Houston, Texas, USA, 1987. (English)

A study was carried out on the optimum placement of heat pumps in industrial processes. Ten processes from a variety of industrial sectors were selected from a total of 26 candidates and were analyzed for optimal heat pump placement using the methods of pinch technology. Numerous potential applications were identified for current heat pump systems and considerable scope was also found for possible advanced heat pump tech-

nologies. The scope for replication of the heat pump options identified in this study was estimated. This evaluation indicated that the total potential for energy savings nationally in the ten processes studied is about 32.9×10^{12} Btu/year or about 10% of the current energy usage.

Williamson, R.C., W.R. Williamson, and C.H. Roy. **Energy Effective Contaminant Removal from Closed Loop Recovery Systems for Cyanide Based Electroplating.** LI-CON, Inc., Pensacola, Florida, USA, 1987. (English)

An ultra-low-temperature heat pump evaporator was developed to provide a feasible alternative to conventional chemical treatment and sludge disposal of heat sensitive cyanide based-electroplating wastes. The evaporator system, trademarked *Fridgevap*, was successfully tested on a cadmium cyanide plating line, proving the feasibility of recovering and returning the plating solution to the bath. The purpose of the project reported here was to develop and test an auxiliary crystallizer to continuously precipitate sodium carbonate decahydrate (wash-

ing soda) contaminants, thereby enhancing the effectiveness of materials recovery. The combined evaporator and crystallizer system underwent limited testing on both a cadmium and a zinc cyanide plating line. This report describes and discusses the development and testing of the prototype closed loop recovery system and the patented heat exchanger which made the system possible. Included is a comparison with conventional chemical precipitation and sludge disposal.

NEWS BRIEFS

ASHRAE Standard Reaffirmed

ASHRAE has announced the publication of ASHRAE Standard 17-1986, *Method of Testing for Capacity Rating of Thermostatic Refrigerant Expansion Valves*. This standard is a reaffirmation with minor editorial revisions and updated references of the standard originally published in 1982. This standard prescribes a method of testing for capacity rating of thermostatic expansion valves which have nominal capacities up to and including 135,000 Btu/h (39.6 kW) of refrigeration effect at 40°F (4.4°C) evaporator temperature. This standard does not include permanent bleed-type valves, nor does it specify values for the test conditions to which the expansion valve shall be subjected. If standard test condition valves are needed, they may be found in related industry standards dealing with equipment performance. The price of ASHRAE Standard 17-1986 is \$18.00 (ASHRAE member price is \$12.00). To order, send payment in U.S. funds to ASHRAE Publication Sales, 1791 Tullie Circle, NE, Atlanta, Georgia 30329, USA. Credit card orders (VISA, MasterCard, and American Express) are accepted by telephone (404-636-8400) or by mail (include full credit card number and expiration date).

ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers), founded in 1894, is an international organization of 50,000 persons. Its sole objective is to advance through research, standards writing, and con-

tinuing education the arts and sciences of heating, ventilation, air conditioning, and refrigeration for the public's benefit.

Call for Papers

Research Activities on Advanced Heat Pumps

The Organization Committee of the Workshop on Research Activities on Advanced Heat Pumps is seeking papers for the Second Workshop on Research Activities on Advanced Heat Pumps to be held in Graz, Austria, on September 26-29, 1988. After the success of the first workshop in October 1986, it was decided to organize this event bi-annually. The primary interest of the Organization Committee for the 1988 workshop is to provide a forum for discussion between specialists on the forefront of heat pump research. Topics to be addressed include: heat pump related fundamental work, construction materials, corrosion problems, new developments for compression heat pump cycles, new development for sorption heat pumps and heat transformers, new working media and CFC replacements, new applications for advanced heat pump cycles, advanced equipment, advanced controlling, simulation techniques, and heat pump related ecological problems. Abstracts should be sent to Doz. Dr. H. Schnitzer, Organization Committee, Workshop on Research Activities on Advanced Heat Pumps, Institute fuer Verfahrenstechnik, Inffeldgasse 25, A-8010 Graz, Austria.

The Third Workshop on Solar-Assisted Ground-Coupled Heat Pumps

The Third Workshop on Solar-Assisted Ground-Coupled Heat Pumps will be held in Sweden in January 1989. The aim of the workshop is to exchange information and to agree on common algorithms and rules of thumb for an expert system model developed for this workshop. This activity dates back to the first workshop held in Ispra, Italy, in 1982, where the state of the art was established. The second workshop in Vienna concentrated on system performance. This third workshop will concentrate on the performance and costs of the systems in various climates.

All papers presenting a system with at least a ground storage and a solar collector system or a heat pump are welcome at this workshop. As in the case of the two previous workshops, participants will be asked to present their reports in accordance with a reporting format. This will facilitate comparison of the projects and is prerequisite for the analyses with the expert model. After the workshop, each participant should have a good idea of the compatibility of his system both in his country and in other climatic regions.

If you wish to receive more information, please write to Jan Olof Dalenbaeck, CTH/Installationsteknik, S-412 96 Gothenburg, Sweden, or Egil Ofverholm, Swedish Council for Building Research, St. Goransgatan 66, S-112 33 Stockholm, Sweden.



Selected Book and Report Reviews

Review of the most recent publications on heat pumps

Lambrecht, Juergen. **Potential for the Application of High Temperature (Greater than 100°C) Thermal Energy Storage Systems in German Industry.** (Einsatzpotential von Waermespeichern fuer Abwaermetemperaturen von mehr als 100°C in der deutschen Industrie) Bundesministerium fuer Forschung und Technologie Forschungsbericht (BMFT-FB-T 86-219), 1986. (German)

The applicability of High Temperature Thermal Energy Storage systems (HTES) to various industrial processes was investigated. Processes from the iron and steel, aluminum, ceramic, and textile industries were considered. A detailed analysis of five of the most promising processes is performed. Cost benefit analyses are done for realistic applications. Results indicate that the use of HTES in the iron and steel, aluminum, and ceramic industry are technically and economically feasible. Proposals are also presented for development of HTES modules having broad applications.

Coutelle, Rainer, Walter Danninger, Peter Gaiser, Nhu-Vinh Nguyen, and Klaus Thielecke. **Development and Testing of a Process Heat Generator System: Phase 1.** (Entwicklung und Erprobung eines Prozesswaermegenerator-Systems: Phase 1) Bundesministerium fuer Forschung und Technologie Forschungsbericht (BMFT-FB-T 86-163), 1986. (German)

The objective of this project was to develop and test a mechanical steam compression system using a screw compressor. The system was installed at the Hoechst AG factory in Gersthofen. It can transfer up to 7.5 t/h of excess steam from a low pressure (4 bar) to a high pressure (9 bar) steam network. Compression is done using a modified dry running compressor from the Aerezener

machine factory GmbH. Components modified were seals and bearings. The steam feed rate is adjusted by compressor speed control. The compressor is cooled by water injection based on the compression end temperature. To determine the technical and economic limits of applicability the system was operated over the following ranges: speed -- 1250 to 3000 RPM; suction pressure -- 2 to 3.5 bar; final pressure -- 4 to 9 bar.

Hane, G.J., and R.A. Hutchinson. **Stirling Engine Research at National and University Laboratories in Japan.** Pacific Northwest Laboratory, Richland, Washington, USA, 1987. (English)

Pacific Northwest Laboratory (PNL) reviewed projects that are related to the development of Stirling engines and that are underway at Japanese national laboratories and universities. It was found that research and development in Japan is focused on component rather than on whole engine development. Information for the investigation was obtained from literature reviews and interviews conducted at laboratories and universities. Contact with university professors involved in Stirling engine work and selective examination of their projects is suggested as a good way of keeping abreast of Japanese Stirling developments.

Heat Pump Technology Center of Japan. **Heat Pumps in Japan: Product and Technical Trends in 1986.** Heat Pump Technology Center of Japan, Tokyo, Japan, 1987. (English)

An overview of technical developments and new products, with information about the general situation and the trends of Japanese government and professional associations. Also discussed are the trends of market and

technical developments and the trends of application systems in Japan. The intention of the Heat Pump Technology Center of Japan is to compile and publish this information on a regular basis, possibly annually.

Grove, Lorraine, ed. **Proceedings of the 3rd International Symposium on the Large Scale Applications of Heat Pumps.** BHRA, The Fluid Engineering Centre, Bedford, England, 1987. (English)

Proceedings of the 3rd International Symposium on the Large Scale Applications of Heat Pumps which was held in Oxford, England, March 25-27, 1987. A brief report of this symposium was published in the Heat Pump Center Newsletter Vol. 5, No. 2, June 1987.

Zimmerman, Kay H. and Raleigh H. Powell, Jr., eds. **Heat Pumps: Prospects in Heat Pump Technology and Marketing.** Lewis Publishers, Inc., Chelsea, Michigan, USA, 1987. (English)

Proceedings of the 1987 International Energy Agency Heat Pump Conference, Prospects in Heat Pump Technology and Marketing, held in Orlando, Florida, USA on April 28-30, 1987. The conference was previewed in the Heat Pump Center Newsletter Vol. 5, No. 1, March 1987. This book contains six papers on Operating Experience with Heat Pumps for Residential and Commercial Space-Conditioning Applications; seven papers on Advances in Electric Heat Pumps; nine on Heat Pumps for Industrial Applications; six on Advances in Thermally Activated Heat Pumps; three on Heat Pumps for District Heating Applications; six on Economics, Marketing, and Promotion; three papers on Government's Role in Energy Conservation, and additionally, one paper on Estimating International Heat Pump Use.

Schedule of Conferences and Trade Fairs

October 1-2, 1987

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

January 26, 1988

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

January 30-February 3, 1988

Dallas, Texas (USA); **1988 ASHRAE Winter Meeting.** Sponsored by the American Society of Heating, Refrigerating and Air-Conditioning Engineers. Contact: Judy Marshall or Judith Breese, ASHRAE International Headquarters, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA, telephone (404) 636-8400, telex 705343 ASHRAE.

February 1-3, 1988

Dallas, Texas (USA); **Southwestern Air-Conditioning, Heating, Refrigerating Exposition.** Sponsored by ASHRAE and the Air-Conditioning and Refrigeration Institute (ARI). Contact: Judy Marshall or Judith Breese, ASHRAE International Headquarters, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA, telephone (404) 636-8400, telex 705343 ASHRAE.

March 9-10, 1988

Tokyo (Japan); **1988 JAR International Symposium on Recent Developments in Heat Pump Technology.** Sponsored by the Japanese Association of Refrigeration (JAR). Contact: JAR, San-ei Bldg., 8 San-ei-cho, Shinjuku-ku, Tokyo 160, Japan, telephone 03-359-5231, cable REITOJAR, TOKYO, telefax 03-359-5233.

June 27-29, 1988

Ottawa (Canada); **1988 ASHRAE Annual Meeting.** Sponsored by the American Society of Heating, Refrigerating and Air-Conditioning Engineers. Contact: Judy Marshall or Judith Breese, ASHRAE International Headquarters, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329, USA, telephone (404) 636-8400, telex 705343 ASHRAE.

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District Heating

The December 1987 issue of the Heat Pump Center Newsletter will focus on district heating. The following contributions are planned:

- Measures to minimize CFC leakage
- Systematic research on condensation of binary mixture of vapors for Rankine or heat pump cycle
- Sewage water-source heat pump
- Outline of the second district heating and cooling plant for Sibaura 4-Chome area
- District heating and cooling of urban areas

Our regular features (book reviews, schedule of conferences and trade fairs, news updates) will be included. If you would like to contribute an article on the subject of district heating, please send it to the Heat Pump Center.

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