10. DEVELOPMENT PROJECT OF NEXT-GENERATION GAS-FIRED ABSORPTION WATER CHILLER-HEATERS

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Since they were first developed, gas absorption chiller-heaters have grown more efficient in performance, more compact in size and more sophisticated in function over the years. Machines of this type have increasingly found their way into not only large buildings but also medium and small ones. Recently, the trend has been for people to seek air-cooled heat sources, and more energy- and labor-saving features. At the same time, a growing number of the so-called co-generation systems have been introduced to meet varieties of applications.

Faced with these moves by developers of competitive equipment, we in the gas business decided to launch a project of developing our own next-generation machines. The project, started in October 1984, was intended to develop innovative absorption machines that would far surpass conventional absorption types in performance. The primary objective was to develop air-cooled, ultra efficiency machines that had been dismissed as impossible.

In launching the project, we held a number of hearings in which candidate manufacturers were screened for their development setups, technological levels, and willingness and motivation to take on the project. As a result, several manufacturers were asked to take part in the project.

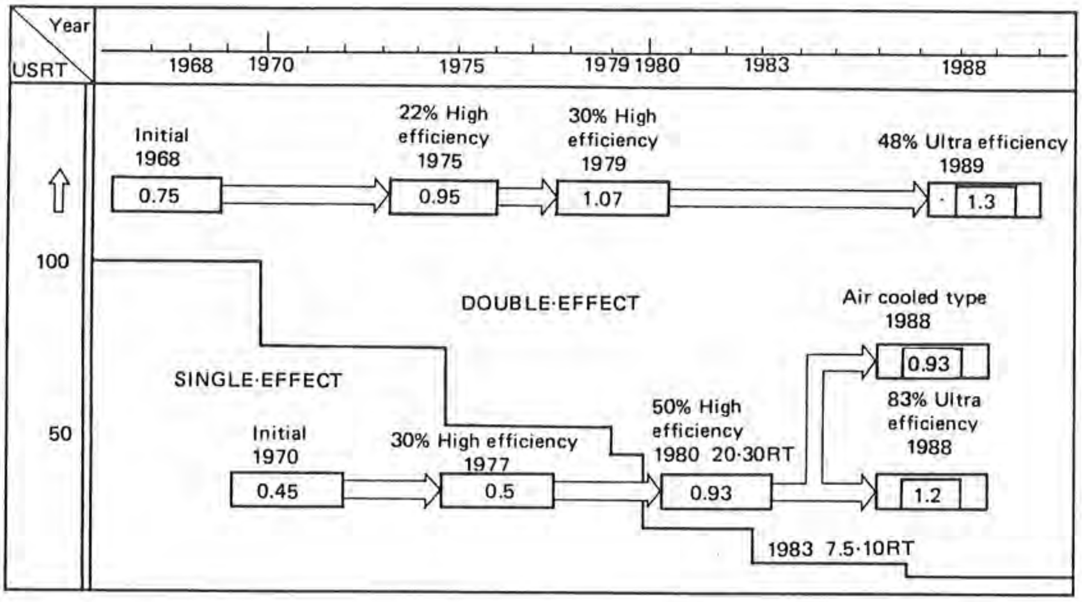
We, at TOKYO GAS CO., LTD. promoted the new project, and three gas companies and the three manufacturers entered into joint development agreements.

The air conditioning COP of gas absorption chiller-heaters was 0.75 when they were first developed. In 1975, the efficiency was improved by 20 to 22 percent. In 1979, the COP exceeded the "1" benchmark for the first time, reaching 1.07. The breakthrough came as a result of the use of an exhaust gas heat recovery device with heat pipes. Another contributing factor was an added process of heat recovery from internal cycles. Development of these techniques is a subject that needs to be pursued even at a time, like these days, when energy supply tends to surpass demand.

A recently recognized need for saving water is being addressed by an effort to develop air-cooled machines. Implementing these machines poses a stiff technological challenge. But once developed, the air-cooled type saves up considerably on annual running costs by eliminating the cost of cooling water.

The project, when launched, also addressed various targets in terms of installation space, ease of installation, maintainability and durability in addition to implementation of an air-cooled type and ultra efficiency type unit. All the targets were set for the first time in the world, and they were at the world's highest technological levels.

Transition in Development of Gas Absorption Chiller-heater



Next-generation Machine Development Project

	Air-cooled small- to medium-size gas absorption chiller-heaters	Ultra efficiency medium-size gas absorption chiller-heaters	Ultra efficiency large-size gas absorption chiller-heaters
Features	Epoch-making, air-cooled machines operating on H_2O-LiBr working fluid combination are to be developed. Developing these machines was once thought impossible.	Highly efficient, energy-saving machines are to be developed as an evolution from conventional medium-size machines.	Large-size energy-saving machines are to be developed as a significantly improved version of today's high-efficiency type (COP: 1.07).
Major targets	Air-cooled structure	Cooling COP: 1.2 Heating efficiency: 0.88	Cooling COP: 1.3 Heating efficiency: 0.98
Schedule	1984 - 1987	1984 - 1987	1984 - 1988
Participating developer	*TOKYO GAS CO., LTD. OSAKA GAS CO., LTD. TOHO GAS CO., LTD. HITACHI, LTD.	TOKYO GAS CO., LTD. OSAKA GAS CO., LTD. *TOHO GAS CO., LTD. YAZAKI CORPORATION	TOKYO GAS CO., LTD. *OSAKA GAS CO., LTD. TOHO GAS CO., LTD. KAWASAKI HEAVY INDUSTRIES LTD.

* A secretary for development

The development targets fell in three categories of machines, as listed below. These targets were pursued under a three to four-year mid-term development plan each.

Concrete results emerged from the project. In April 1988, marketing of a ultra high-efficiency medium-size gas absorption chiller-heater called GASMAX began. An air-cooled gas absorption chiller-heater and a ultra high-efficiency large-size gas absorption water chiller-heater are expected to be marketed beginning in spring 1989.

Air-cooled absorption chiller-heater is discussed in this paper.

11. DEVELOPMENT OF ULTRA EFFICIENCY MEDIUM-SIZE GAS-FIRED ABSORPTION WATER CHILLER-HEATER

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1. Overview

Since they were first developed, gas absorption chiller-heaters have been getting smaller in size, lighter in weight, and more efficient in energy consumption. This progress has been made possible by diverse technical innovations such as use of the double effect cycle. We set out to make small- and medium-size gas absorption chiller-heaters more efficient by modifying the various components of the machines. Our experiments led to successful development of commercial type machines whose cooling COP is 1.2, close enough to the theoretically optimum efficiency.

2. Outline of Components for Ultra Higher Efficiency

Not only large-size but also medium- and small-size absorption chiller-heaters are getting more efficient (Cooling COP: 0.95). To boost the efficiency of the medium- and small-size machines requires putting together improvements of various machine components.

Table 1 lists major items to be improved for higher efficiency of the absorption water chiller-heater system, along with some specific measures to achieve what was envisaged.

We studied the following four major items for development:

(1) Solution circulation cycle
(2) High-performance heat transfer tube
(3) Heat exchangers for heat recovery (exhaust gas, condensed refrigerator)
(4) Heating efficiency

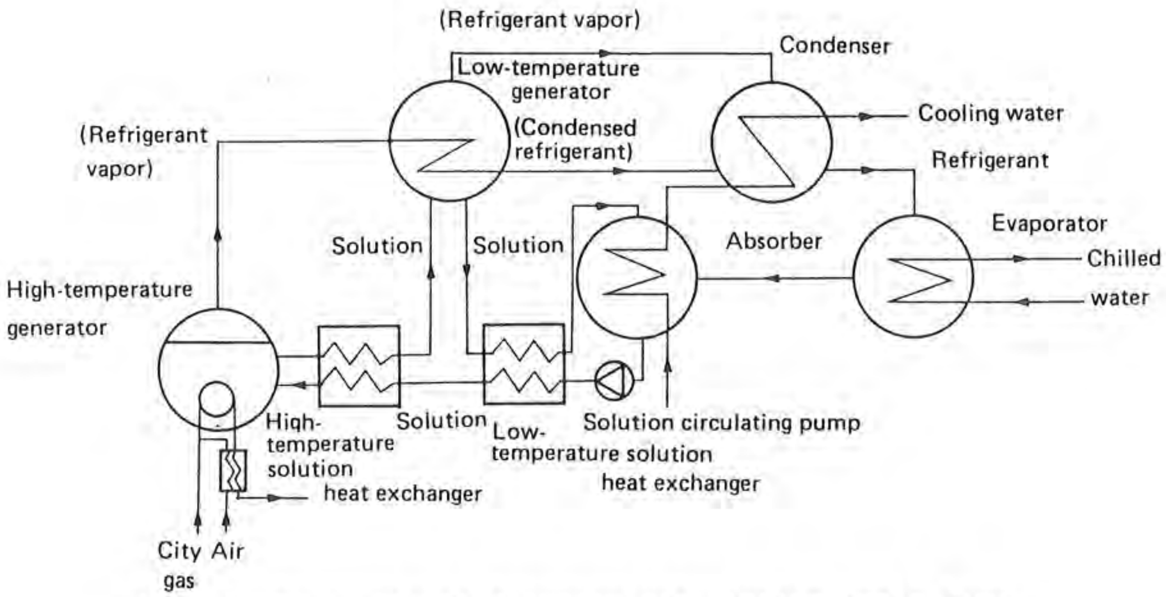


Fig. 1 Conventional Solution Circulation Cycle Flow

3. Components to be Developed

Fig. 1 illustrates the conventional absorption cycle flow.

The major components are: high- and low-temperature generators, condenser, evaporator, absorber, and high- and low-temperature solution heat exchangers. Fig.2 shows the new cycle that we developed by reevaluating and modifying the various components including the flow of the solution.

What follows is a description of each of components improved.

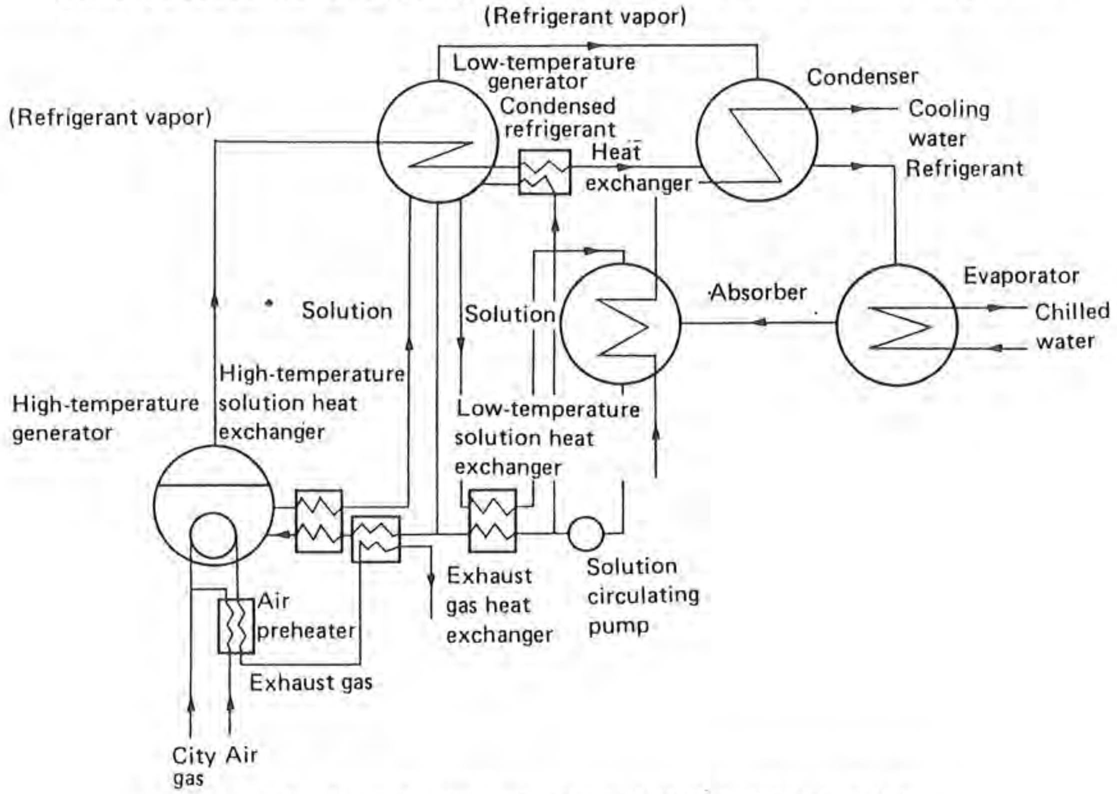


Fig. 2 New Solution Circulation Cycle Flow

Table 1 Development Items

Items	Improvements	
(1) Cooling COP	Improvement of heating efficiency	Exhaust gas heating with low-temperature generator
		Re-heating of condensed refrigerant in low-temperature generator
		Pre-heating of combustion air
	Improvement of cycle efficiency	Less heat loss between components
		Refrigerant orifice (between low-temperature generator and condenser)
		Higher efficiency of absorber
		Higher efficiency of evaporator
(2) Heating efficiency	Improvement of heating efficiency	Exhaust gas heating with low-temperature generator
		Re-heating of condensed refrigerant in evaporator
		Pre-heating of combustion air
(3) Power consumption	Chilled and hot water circulating pumps	VWV method and operation control method
	Cooling water pump	VWV method and operation control method
		Less piping loss (through packaging)
		Pump optimized
	Solution circulating pump	Pump output reduced
		Operation control method
	Blower for combustion	
(4) Amount of water replenished	Cooling water circulation rate	Blow-down rate reduced
		Carry-over loss reduced
	Higher radiation coefficient	Evaporation loss reduced

3.1 Solution Circulation Cycle

The working fluid of absorption water chiller-heater is the combination of water and a lithium bromide solution. The solution circulation cycle comes in two types: series flow, and parallel flow.

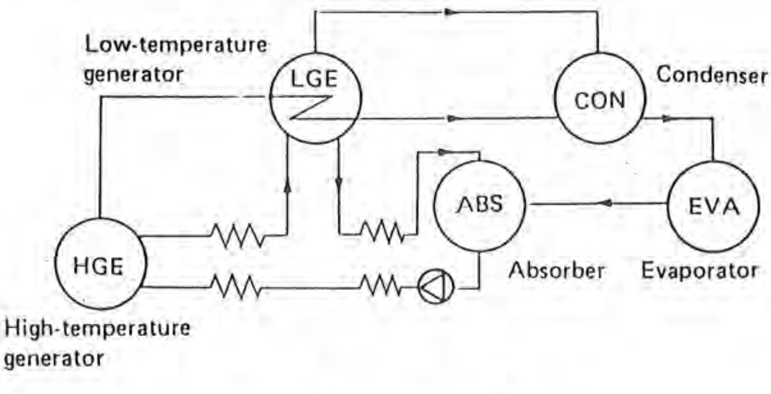


Fig. 3 Series Flow

As shown in Fig. 3, the series flow is a flow in which a weak solution from the absorber is first led to the high-temperature generator where the solution is heated and condensed. When forwarded to the low-temperature generator, the solution is further condensed there before returning to the absorber.

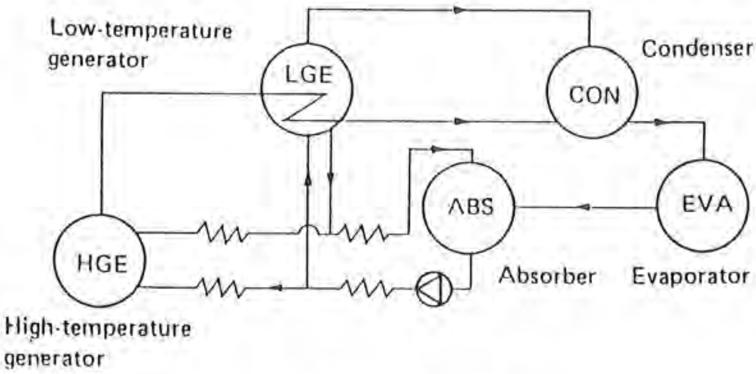


Fig. 4 Parallel Flow

The parallel flow involves, as shown in Fig. 4, having the weak solution from the absorber branch halfway to the high- and the low-temperature generator. After being condensed in both generators, the solutions is sent back to the absorber.

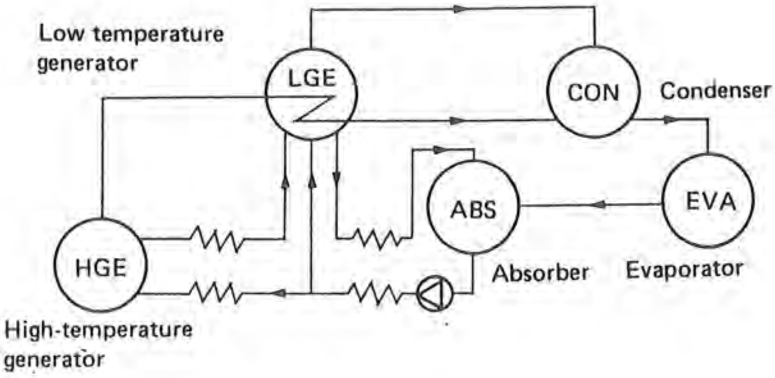


Fig.5 New Flow

The amount of the weak solution flowing to the high-temperature generator in the parallel flow structure is lower than in the series flow structure. That is, the amount of sensible heat in the high-temperature generator is lower with the parallel flow than with the series flow. To the extent the amount of sensible heat is reduced, to that extent refrigerant vapor can be increased. On the other hand, the amount of sensible heat in the solution is lower in a series flow setup than in a parallel flow setup. This feature increases the amount of refrigerant generated in the series flow setup.

Fig. 5 shows the new flow of solution circulation which was developed this time. In this new flow, the solution from the absorber branches (as in parallel flow) to the high- and the low-temperature generator. The solution that emerges from the high-temperature generator passes through the low-temperature generator before returning to the absorber (as in series flow). The new flow keeps a good balance of refrigerant generation between the high- and the low-temperture generator for higher efficiency.

Our study has verified through simulation and actual tests that the new flow provides higher COP levels than conventional flow schemes.

3.2 High-performance Heat Transfer

As shown in Fig. 1, the absorption water chiller-heater is a collection of heat exchangers. We set out to inprove the heat transfer performance of the evaporator and absorber, the two key components that affect efficiency significantly.

When heat is transferred through an evaporator, heat transfer by annular-type forced convection occurs inside the pipes (cooling water side), and boiling heat transfer by natural convection takes place outside the pipes (refrigerant side).

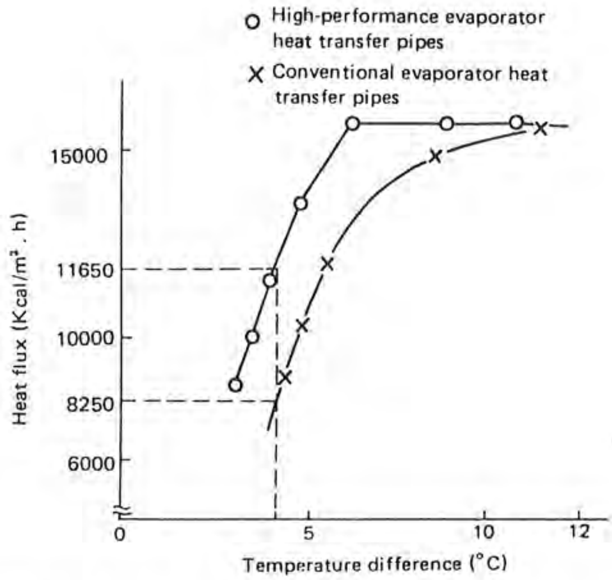
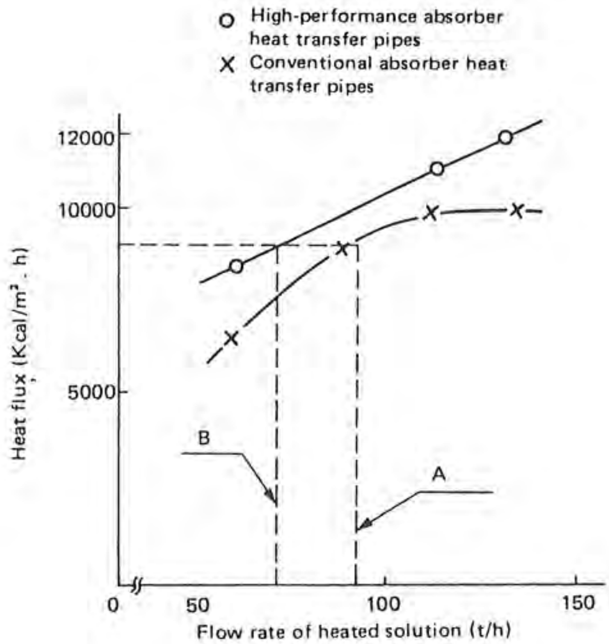


Fig. 6 Comparing Evaporators in Heat Transfer Performance



A:Flow rate of solution in conventional setup
B:Flow rate of solution in high-performance setup

Fig. 7 Comparing Absorbers in Heat Transfer Performance

In our experiments, the pipe exterior was specially treated to improve the heat transfer performance outside the pipes, whereby boosting the heat flux. That in turn contributed to eliminating the ineffective refrigerant, realizing higher efficiency.

Meanwhile, when heat is transferred through the absorber, heat transfer by annular-type forced convection occurs inside the pipes (cooling water side), and film condensation heat transfer with absorption takes place outside the pipes (solution/refrigerant side).

We treated the pipe exterior specifically to boost the heat flux so that with the same flow rate of solution maintained outside the pipes, more refrigerant vapor could be absorbed. The rate of solution circulation was lowered and the amount of sensible heat in the generators was reduced for higher efficiency.

Fig. 7 compares conventional and high-performance absorber heat transfer pipes in heat transfer performance. The new pipes are about 20 percent higher in performance than conventional heat transfer pipes.

3.3 Heat Exchangers for Heat Recovery

We developed two new heat exchangers, condensed refrigerant heat exchanger and exhaust gas heat exchanger, to improve the system efficiency through heat recovery (see Fig. 2).

The condensed refrigerant heat exchanger was designed to recover the heat of the refrigerant. In a conventional setup, the refrigerant condenses at 95℃ in the low-temperature generator and is cooled by the condenser down to 38℃; the remaining heat is discarded unused out of the cooling water circuit. In our new arrangement, the refrigerant exchanges heat with part of the weak solution from the absorber. With the condensed refrigerant feat exchanger in place, the condensed refrigerant is cooled to around 45℃, while the weak solution is heated from 38℃ to about 80℃. This reduces the amount of sensible heat in the solution that occupies the low-temperature generator for higher efficiency.

The exhaust gas heat exchanger was also designed to recover heat through heat exchange with part of the weak solution. In operation, as shown in Fig. 2, the weak solution coming out of the low-temperature solution heat exchanger is heated before being led into the high-temperature solution heat exchanger. With the exhaust gas heat exchanger in place, the solution is heated from 80℃ to about 85℃, While the exhaust gas is cooled from 185℃ to about 85℃. This reduces the amount of sensible heat in the solution that occupies the high-temperature generator for higher efficiency.

3.4 Heating Efficiency

The baffle plate arrangement in the passage of hot combustion gas was modified to improve the high-temperature generator in heating efficiency. As shown in Table 2, the modified baffle plate setup has boosted the heat transfer performance by 1.5 percent.

4. Results

We combined the techniques and components developed this time

Table 2. Performance Characteristics of the High-temperature Generator with Baffle Plates

	For air-conditioning	
	With conventional baffle plates	With modified baffle plates
Temperature of solution in high-temperature generator(℃)	150	
Combustion gas outlet temperature(℃)	220	185
Efficiency of high-temperature generator	0.828	0.843

into an actual system to attain the target performance envisaged. The results from the development that we engaged in were all verified on actual machines.

Fig. 8 and 9 compare, in performance, conventional machines with a machine of the comparable class which we developed this time. Comparisons were made for both air conditioning and heating applications. GASMAX is the name of the machine developed this time.

GASMAX has attained a COP level of 1.2 at the rated point of air conditioning (7℃). The machine is 26 percent and 6 percent more efficient in cooling and heating, respectively, than conventional models.

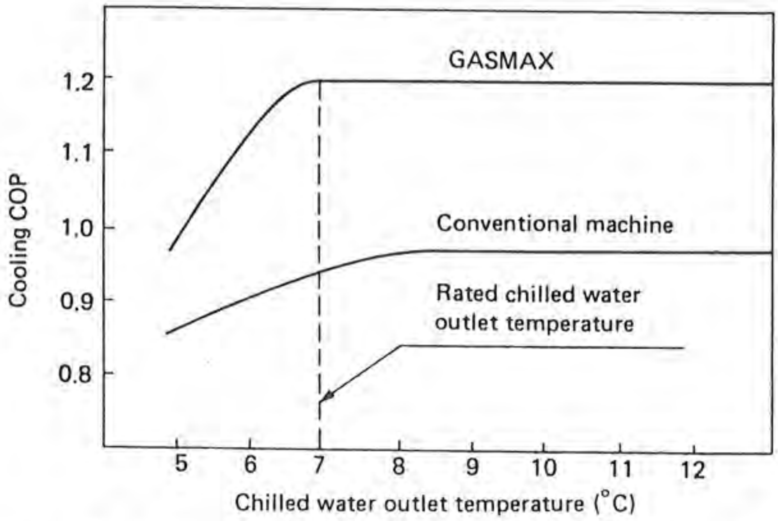


Fig. 8 Comparing Performance Characteristics (Cooling)

5. Summary

This chapter has described our efforts to improve the efficiency

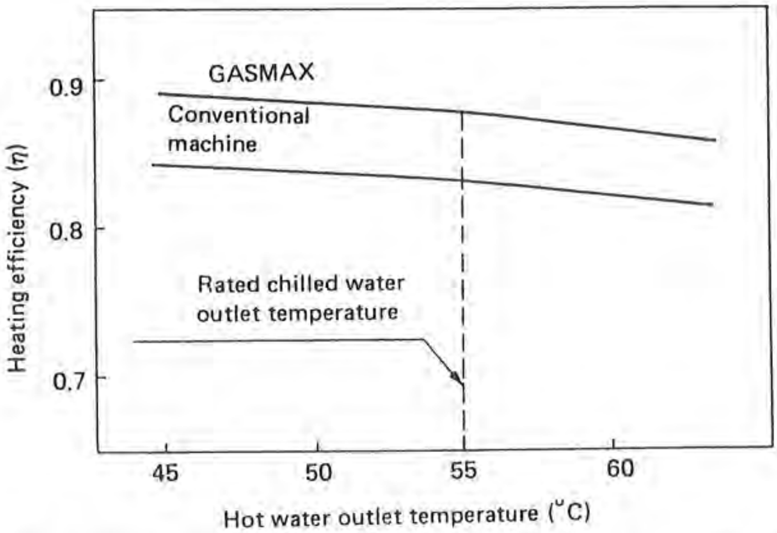


Fig. 9 Comparing Performance Characteristics (Heating)

of the gas absorption chiller-heater proper. We are now addressing what may be called a chilled/hot water and cooling water variable flow rate method, or VWV method. From now on, we intend to focus our attention on how to improve the efficiency of the system as a whole through such means as reductions in medium transport power.

12. DEVELOPMENT OF ULTRA EFFICIENCY LARGE-SIZE GAS-FIRED ABSORPTION WATER CHILLER-HEATER

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1. Overview

In large buildings, a large volume of energy has to be consumed for air conditioning and heating. This is where the calls for saving more energy are most pronounced. Meanwhile, motor-driven machines such as enegy-saving centrifugal refrigerators and screw heat pumps have been getting more and more efficient, emerging as a formidable competition threatening gas-fired machines. Under the circumstances, we in the gas business decided to launch a joint project to develop high-efficiency large-size gas absorption chiller-heaters whose efficiency would approximate theoretically optimum levels.

2. Targets of Development

Our project aimed higher. Based on computer-simulated high-efficiency cycles, our target machines were to be more efficient than conventional standard machines by 30 percent or more, and more efficient than the most effficient type at that time by 17 percent.

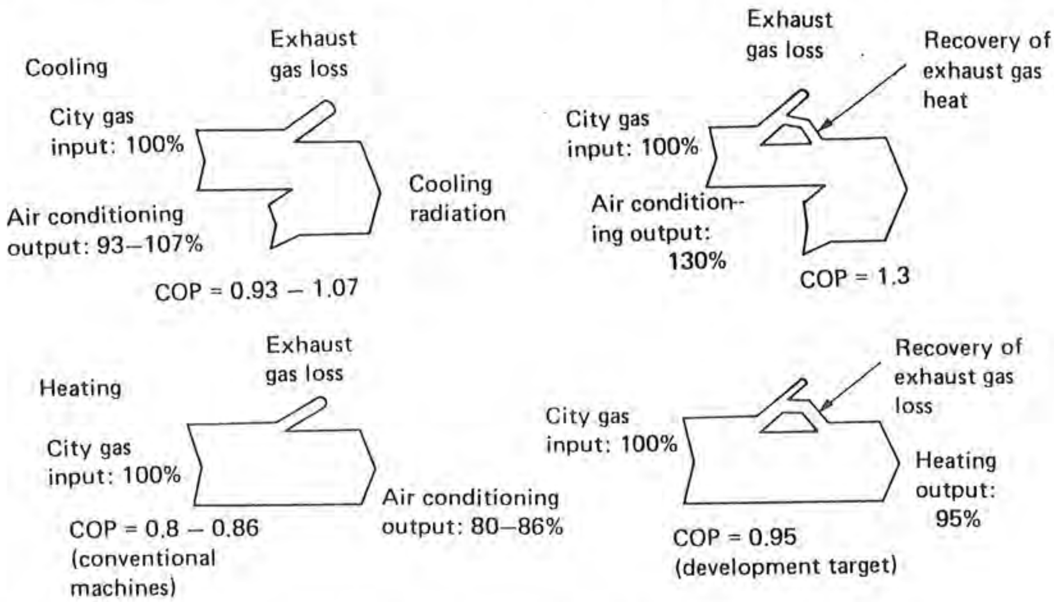

Fig. 1 Comparing Conventional and New Machines in Efficiency

3. Key Points in Developing Techniques

To achieve drastic improvements in efficiency requires developing various components and carrying out a number of tests. The key points in fulfilling these requirements are as follows.

3.1 Determining the High-efficiency Cycle through Programming of Cooling and Heating Cycles

A heating value computation program was prepared to determine a new cycle for higher system efficiency. The program was run to simulate various cycles under both ideal and actual conditions. The efficiency calculations thus performed showed the COP level we envisaged to be 1.44 under ideal conditions and 1.3 or higher under actual conditions.

3.2 Developing the Combustion Exhaust Gas Latent Heat Recovery Method

An important factor in making the system more efficient is the recovery of latent heat from combustion exhaust gas. To recover the latent heat requires reducing the exhaust gas temperature to around 50℃. The key to successful recovery of the latent heat is to determine how to make the recovery in what part of the cycle. The procedures involved are different between air conditioning and heating applications.

(1) Air conditioning cycle

To cool the exhaust gas to the 50℃ level for heat recovery calls for considering how to exchange heat in the low-temperature section during a cycle flow. We decided to have the weak absorbing solution exchange heat with exhaust gas between absorber and low-temperature generator to preheat the absorbing solution. Fig.2 shows a typical air conditioning cycle operating on this method.

(2) Heating cycle

In heating applications, the hot water outlet temperature is about 50℃, which cannot be reduced appreciably. However, to improve efficiency, the exhaust gas temperature should preferably be reduced as mush as possible. Given these considerations, we decided to adopt a setup of exhaust gas heat recovery using a heat pump cycle.

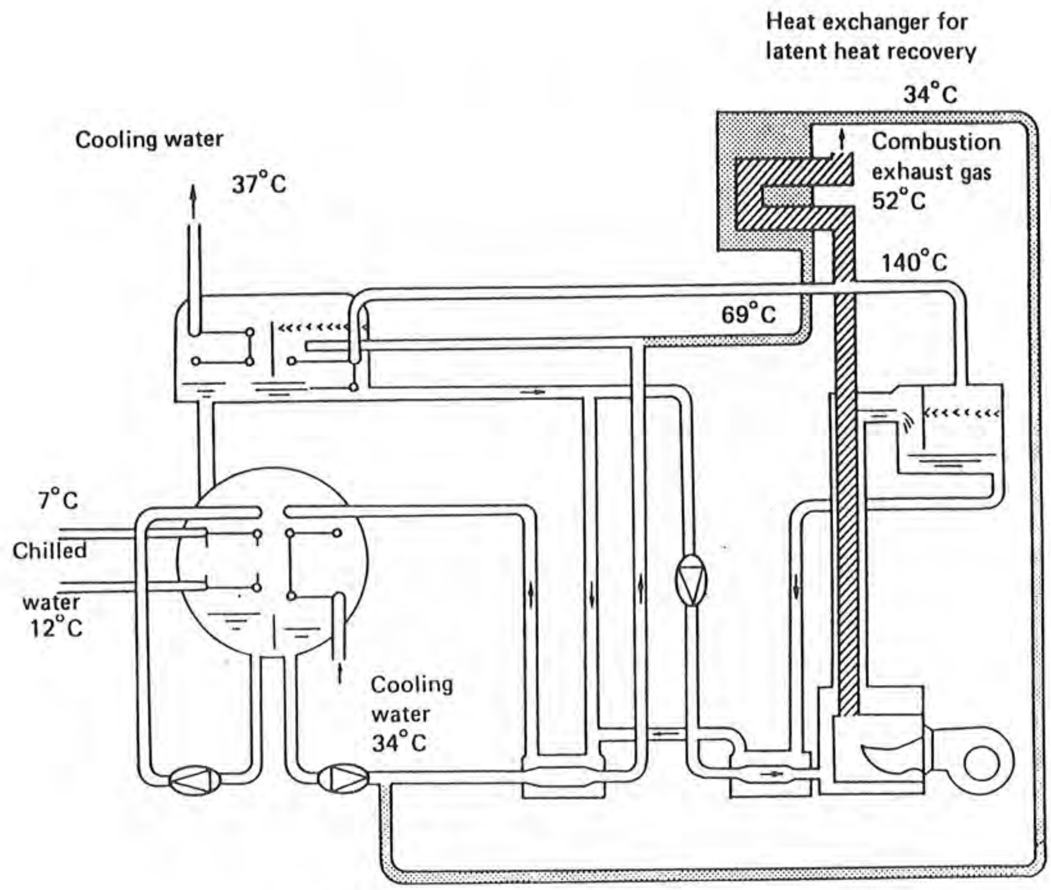

Fig. 2 Cooling Cycle

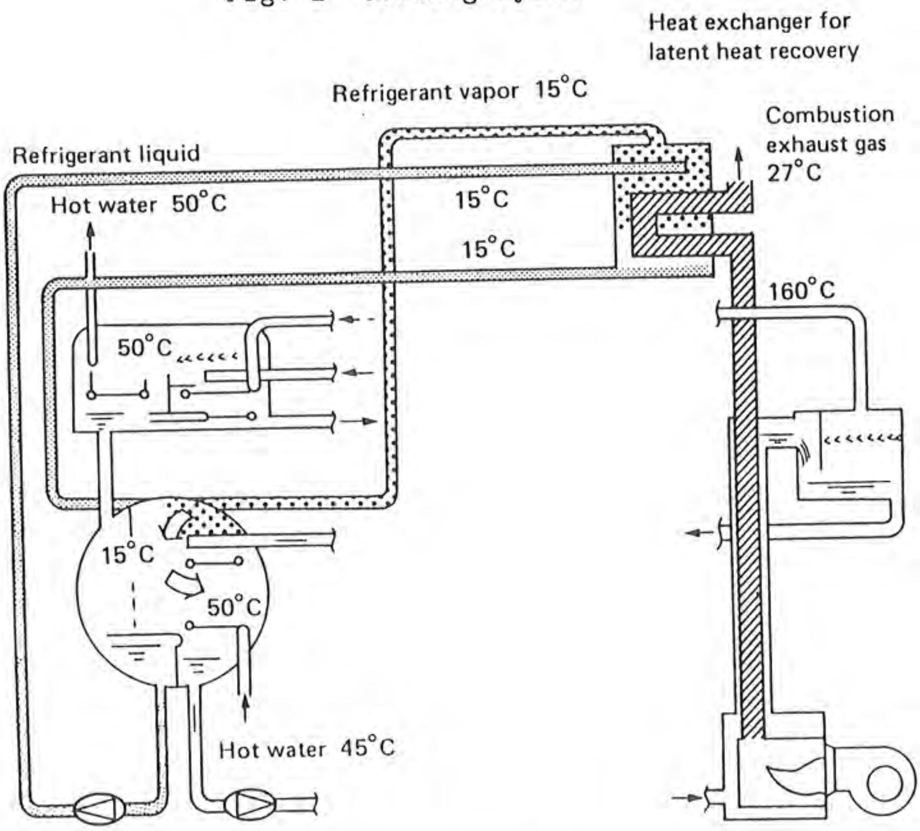

Fig. 3 Heating Cycle

In operation, the refrigerant liquid at about 15℃ from the evaporator is evaporated by exhaust gas heat and sent to the absorber, where the liquid is absorbed in the solution inside. The absorption temperature at this time is around 57℃. The heat pump effect raises the temperature by about 50℃ from the refrigerant vapor temperature of 15 ℃.

(3) Endurance tests on the exhaust gas latent heat recovery device

It is necessary to take into consideration the corrosion of heat transfer of pipe materials due to the drain from the exhaust gas latent heat recovery process. Laboratory level tests have been conducted in this area, but they may not adequately reflect the actual conditions that are far serverer. That is, heat transfer pipes are dried and get wet repeatedly during starting and stopping operations as well as during load fluctuations.

We built test equipment by which to verify the durability of the material under actual operating conditions. Exhaust gas was actually condensed for the testing, and optimum materials were selected during endurance tests that lasted 10,000 hours. Table 2 lists the test conditions adopted.

(4) Making heat exchangers more efficient

The absorption water chiller-heater may be said to be a collection of heat exchangers. An important factor that increases the cycle efficiency is the use of heat transfer pipes whose heat transfer characteristics are optimum when used in the respective components such as an evaporator and absorber. We tested various high-performance heat transfer pipes on actual machines, and selected optimum pipe types for use in the evaporator, absorber and low-temperature generator.

Table 2 Conditions for Endurance Tests on the Latent Heat Recovery Device

City Gas	Natural Gas
Heat transfer pipe material tested	STB-35E Cu+Sn plating SUS430TB YUS-190 YUS-170 SUS316TB SHOMAC30-2 SUS304TB
Burner input	18,000 kcal/h
Exhaust gas temperature	150 - 200℃ → 15 - 30℃
Test interval	10,000 h (4 hours of combusion followed by 1 hour of rest, repeated for the duration)

(5) Solution circulation rate control by inverter

Under this project, we aimed to save more energy in both rated-value operations and partially loaded operations. To achieve that, an inverter arrangement was used whereby the absorbing solution pump was controlled in revolutions. What was pursued specifically was too boost the COP level in partially loaded operations and to reduce the pump power required.

4. Testing Procedures and Results

4.1 Performance Tests on the Prototype

We built a 100 RT prototype by which to prove the efficiency values obtained from cycle simulation and to implement the components contemplated. A variety of tests were conducted on this prototype. Fig.4 illustrates the structure of this equipment.

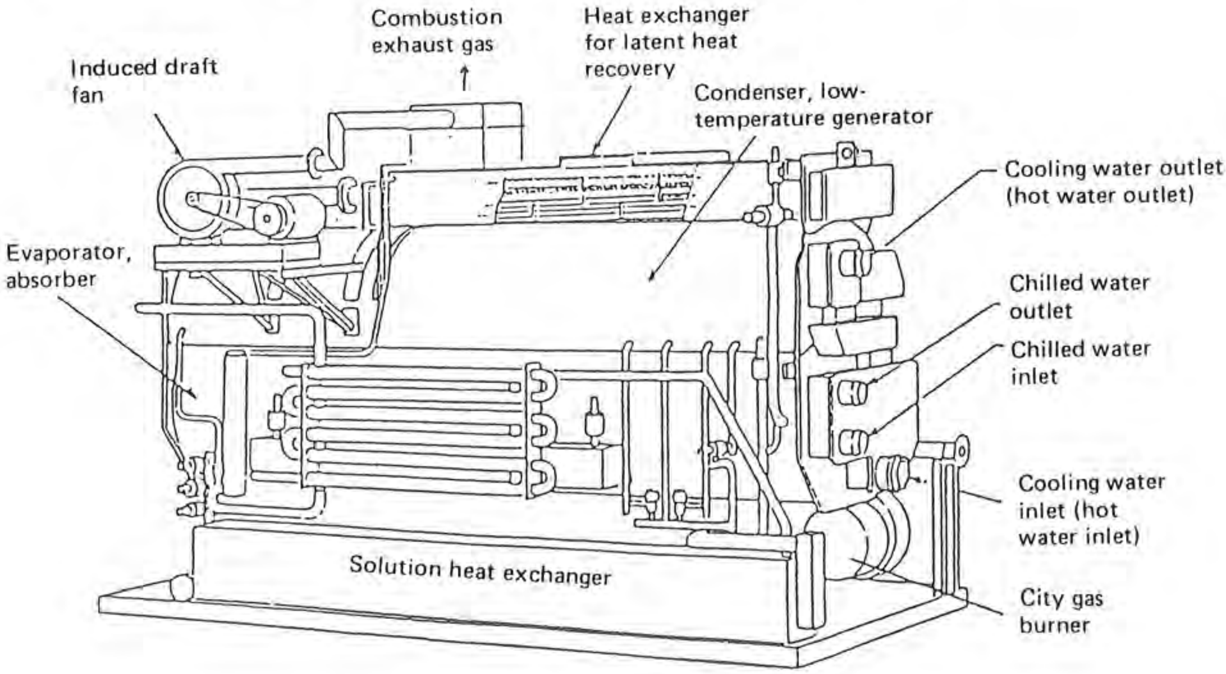

Fig. 4 Structure of the 100 RT prototype

(1) Rated performance
Rated performance tests were carried out in air conditioning and heating cycles on the 100 RT prototype. The tests revealed that the target COP for air conditioning and that for heating, 1.3 and 0.95 respectively, were both cleared. Figs. 5 and 6 show the data obtained from the tests.

(2) Partially loaded performance
Absorption water chiller-heaters are generally higher in partial load efficiency than compression type machines. To further boost the efficiency, absorption water chiller-heaters should preferably have optimum control over the absorbing solution circulation rate at partial load.

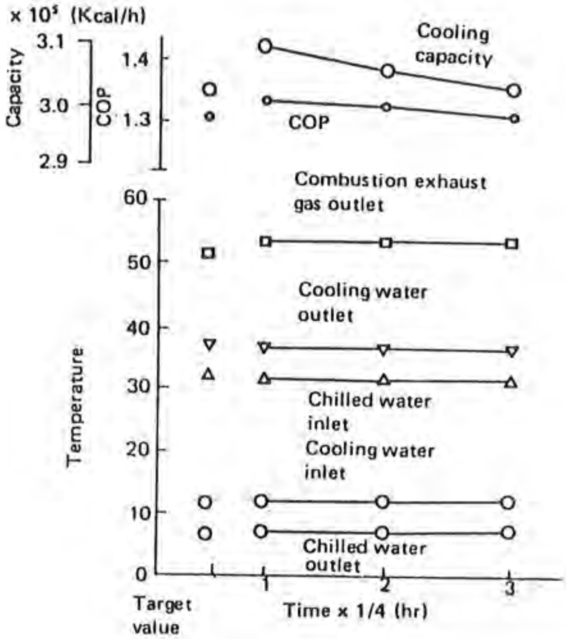

Fig. 5 Rated Performance Tests for Cooling

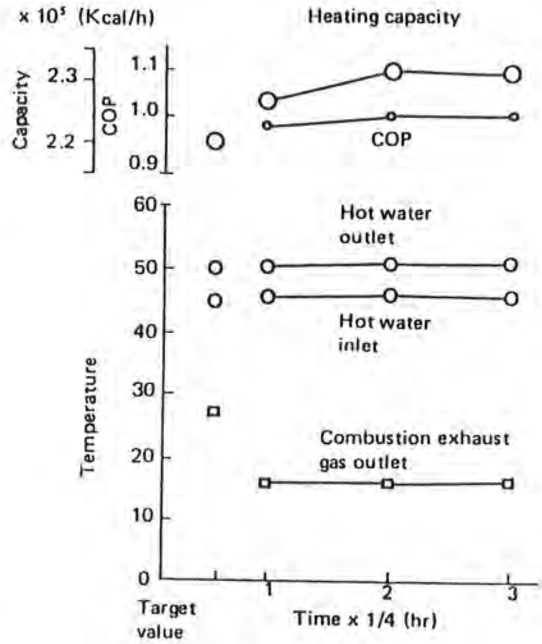

Fig. 6 Rated Performance Tests for Heating

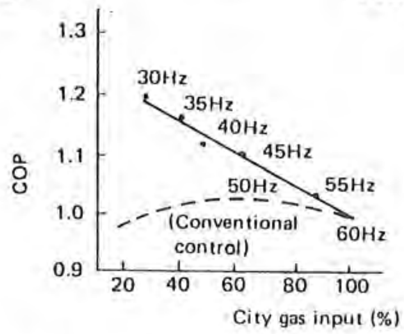

Fig. 7 Energy-saving Characteristics at Partial Load under Inverter Control

We put into use the method of controlling the absorbing solution pump in revolutions by inverter in keeping with load fluctuations. Our tests revealed that the efficiency was improved by up to 20 percent at partial load, and that the pump power required was reduced by about 50 percent. (See Fig.7)

4.2 Endurance Tests on the Latent Heat Recovery Device

During the tests, city gas was burned and the resulting drain was allowed to condense on the surface of various metals. The surface condition was kept under observation, and the drain was analyzed periodically for ingredients.

It turned out that of the materials tested, tubes made of STB and of Cu with Sn coating developed corrosion at earlier stages; they were clearly unfit for use. Stainless steel tubes showed no sign of corrosion.

5. Summary

The results of the development may be summarized as follows.

(1) The target COP for air conditioning and that for heating, 1.3 (0.93-1.07 at present) and 0.95 (currently 0.85) respectively, were attained .

(2) The system characteristics at partial load were improved under inverter-based optimum control over the absorbing solution circulation rate.

(3) The latent heat recovery device that also covers the cooling cycle was developed.

(4) Endurance tests on the latent heat recovery device showed that SUS304 is fit as the tube material.

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13. INCREASE OF THE EFFICIENCY OF A POWER PLANT WITH ITS FEED WATER HEATED BY AN ABSORPTION HEAT PUMP

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Résumé

A lot of heat is released to cooling water in a steam turbine power plant, though improvement of it's efficiency has been realized by preheating feed water to the boiler heated by steam extracted from the steam turbine.

In this paper the author describes that it is possible to improve the power generating efficiency by heating the feed water by a heat pump which is driven by steam extracted from the turbine and recovers heat released to the cooling water, and also the system like this can be put to practical use.

1. The efficiency of the steam turbine power generation

One of the most typical power generating system by steam turbines is a form using a condensing turbine and we can see many examples like that in steam power stations for utility.

To improve the efficiency of power generating, elevation of temperature and pressure of steam, addition of a reheating process, adoption of multi-stage preheating of feed water etc. have been incorporated into the system. However, the increase of efficiency by those ideas is approaching to its limitation. The further improvement by those ways needs a lot of cost and must not be advantageous on profit.

Fig. 1 shows a general idea of a power station using a condensing steam turbine. In Fig. 1 for simplicity the reheating process was abbreviated. As far as on-site power stations are concerned, a simply condensing steam turbine system with no reheating is usually adopted.

Besides that, Fig. 1 shows only two feed water heaters, each one before and after the deaerator, though in a practical power station many feed water heaters are provided as shown in Fig. 2 to try to increase its power generation as much as possible.

The output of a simply condensing steam turbine can be shown by Fig. 3 using I-s(Enthalpy-Entropy) diagram. In Fig. 3 the theoretical adiabatic heat drop in a steam turbine is $i''_o - i''_1$ kcal/kg and its practical heat drop is given by $i''_o - i''_e$ kcal/kg. When mechanical and generator's efficiencies are applied, its effective generator's output can be given by $(i''_o - i''_e) \times \eta m \times \eta g$ kcal/kg.

The feed water to a boiler is heated by steam extracted from the turbine during its expansion, which means that the latent heat of the exhaust steam from the turbine can be used effectively without being released to the cooling water and results in the improvement of the efficiency.

2. A steam turbine power station provided with an absorption heat pump

Fig. 4 shows a steam turbine system with a heater of the feed water heated by an absorption heat pump operated by steam extracted from the turbine and using the heat released to the cooling water in the condenser as its low temperature heat source.

This system can increase its power output for a given steam rate to the turbine because of less steam rate to a feed water heater extracted from the turbine for the same feed water temperature.

That is to say, the amount of steam from the steam turbine can be decreased to 1/C of that of a conventional system when C is the figure of the coefficient of performance of a heat pump.

Consequently, a heat pump with a big figure of COP becomes advantageous.

The system like one in Fig. 4 has a merit that a heat pump packaged in a manufacturer can be used as it is. However, temperature differences for heat transfer in a low temperature heat source line and in a heater of feed water are required, which cause the heat pump deteriorate its COP and increase it's initial cost owing to the doubled heat transfer surfaces.

Fig. 5 shows the way to improve this point. In this system the exhaust from the turbine is directly led to the evaporator of the absorption heat pump and the vapor of the refrigerant (heating medium) generated in the generator of the absorption heat pump can be directly applied to the heater of feed water.

This system can raise its temperature by 5 to 6°C compared with the case of Fig. 4.

The increase of the output of electricity in the case of Fig. 5 to the conventional system like Fig. 1 can be expressed by Eq. 1.

$$\Delta e = \left\{\frac{T_2 - T_D}{i_3 - T_S} \times (i''_3 - i''_e) - \frac{(i''_2 - i''_e) \times (T_2 - T_D)}{(t_{g1} - t_{g2}) + C \times (i''_2 - t_{g1})} - \frac{(C-1) \times (i''_s - i''_e) \times (i''_2 - t_{g1}) \times (T_2 - T_D)}{\{(t_{g1} - t_{g2}) + C \times (i''_2 - t_{g1})\} \times (i''_s - T_2)}\right\} \times \eta m \times \eta g \qquad \text{Eq. 1}$$

where

Δe	: Increase of the specific output	kcal/kg
T_s	: Temperature of the feed water after the low pressure feed water heater	°C
T_2	: Temperature of the feed water after the heat pump	°C
T_D	: Temperature of the feed water from the main condenser	°C
t_{g1}	: Temperature of the drain from the generator of the heat pump	°C
t_{g2}	: Temperature of the drain after the feed water preheater	°C

i''_3 : Enthalpy of steam to the low pressure feed water heater kcal/kg
i''_2 : Enthalpy of steam to the generator of the heat pump kcal/kg
i''_s : Enthalpy of steam to the evaporator of the heat pump kcal/kg
i''_e : Enthalpy of exhaust of the turbine kcal/kg
C : Coefficient of performance of the heat pump
ηm : Mechanical efficiency
ηg : Generator's efficiency

As the temperature T_2 which could be raised by a heat pump is dependent on the feature of the heat pump itself, the temperature of the low temperature heat source (in this case the heat source is the exhaust of the turbine) and the saturated temperature of the steam extracted from the turbine, it is necessary to study and make clear the feature of temperature rise in the absorption heat pump.

3. Temperature rise in an absorption heat pump

Among many expressions for characteristics of absorption heat pumps Fig. 6 is a Dühring chart puting stress on the temperatures for H_2O-LiBr (water-Lithium Bromide) system. In Fig. 6 the horizontal line is for the saturated temperature (evaporating or condensing temp.) of the pure water, the vertical line means the LiBr solution temperature and the parameters are constant concentrations of Lithium Bromide.

$$\text{Concentration} = \left(\frac{\text{weight of Lithium Bromide}}{\text{weight of Lithium Bromide solution}} \times 100\% \right)$$

Fig. 7 shows the cycle of the single stage (single effect) absorption heat pump written on the Dühring line like Fig. 6.

te is the evaporating temperature of the refrigerant in the evaporator and a little lower than the temperature of the low temperature heat source.

tc is the condensing temperature of the refrigerant in the condenser and a little higher than the temperature of the recovered heat Tc in the condenser, which usually gives the highest temperature of the heat pump.

ta is the attainable highest temperature of the solution in the absorber which also gives the highest temperature got by the heat pump, and the limited highest temperature Ta is a little lower than ta.

The heating by the heat pump is acted in the absorber and condenser.

Either the absorber or the condenser heats the medium first gives different limitations on the highest temperature.

When the heating temperature in the generator is constant, generally speaking, in order to get higher temperature it is advantageous to heat the medium in the absorber in advance for rather low temperature of low temperature heat source from the point of cristalization. On the contrary, it seems to be useful to heat the medium first in the condenser for rather high tempeature of low temperature heat source.

Anyway the difference of the highest temperature is not so big between the two systems.

tg is the highest temperature of the LiBr solution in the generator, and Tg, the heating steam temperature, is required to be a little higher than tg.

When the temperature differences required for heat transfer in the evaporator, condenser, absorber and generator are expressed by Δte, Δtc, Δta, Δtg respectively, practical figures of Δte, Δtc, Δta, and Δtg are 3 ~ 4, 2 ~ 3, 6 ~ 7 and 7 ~ 10 respectively.

The heat transfer coefficient in the H_2O-LiBr absorption machine is rather small because of the low working pressure in the system and the effect of the air leaked into the system and hydrogen generating in the system, when the machine is used as a refrigerator.

On the contrary, the higher heat transfer coefficient can be selected to the heat pump because of its higher working pressure, and those figures Δte, Δtc, Δta and Δtg can be made smaller.

We can get the following expression for a single stage absorption unit theoretically concerning the relation among te, tc, ta and tg.

$$(273 + te) \times (273 + tg) = (273 + tc) \times (273 + ta)$$

From this relation Eq. 2 can be derived.

$$\begin{aligned} &(Te + 273 - \Delta te) \times (Tg + 273 - \Delta tg) \\ &= (Tc + 273 + \Delta tc) \times (Ta + 273 + \Delta ta) \end{aligned} \qquad \text{Eq. 2}$$

For the case the medium is led to the absorber first, and with an assumption $Tc = Ta + \Delta T$

Eq. 2 is transformed to Eq. 3.

$$Tc = -\left(273 + \frac{\Delta tc + \Delta ta - \Delta T}{2}\right) + \sqrt{\frac{(\Delta tc + \Delta ta - \Delta T)^2}{4} - \Delta tc \times (\Delta ta - \Delta T) + (Te + 273 - \Delta te) \times (Tg + 273 - \Delta tg)} \qquad \text{Eq. 3}$$

On the other hand the author assumes following relations for practical ranges of concentrations and temperatures from Fig. 6.

$$ta = te + \alpha \cdot \xi$$
$$tg = tc + \beta \cdot \xi$$

where ξ : concentration of Lithium Bromide in %

uses the relation $Tc = T_A + \Delta T$, as mentioned above, and gives Eq. 4 for Tc.

$$Tc = \frac{1}{(\frac{1}{\alpha} + \frac{1}{\beta})} \times \left(\frac{Te}{\alpha} \frac{Tg}{\beta} \frac{\Delta T}{\alpha} - \{\frac{1}{\alpha}(\Delta te + \Delta ta) + \frac{1}{\beta}(\Delta tc + \Delta tg)\}\right) \qquad \text{Eq. 4}$$

Two equations of Tc give almost the same figures for practical ranges of usage. This has been examined by the author.

Fig. 8 shows calculation results using Eq. 4 relating to realistic temperatures heated by an H_2O-LiBr absorption heat pump,

where $\alpha = 0.72$, $\beta = 0.85$, $\Delta T = 5°C$, $\Delta te = 3°C$, $\Delta tc = 2°C$, $\Delta ta = 7°C$, $\Delta tg = 10°C$ are applied.

From this graph we can see the relation among the temperatures of the low temperature heat source, the heating temperature in the generator and the attainable level of the temperature as a heat pump.

For the constant temperature of the low temperature heat source, the pressure of steam extracted from the turbine should be raised for the higher outlet temperature of the heat pump (Tc).

4. Increase of the output of electricity by combination of a heat pump

To raise the temperature of feed water the pressure of steam heating the heat pump should be elevated, as mentioned above. The pressure can be got as saturated pressure corresponding to the temperature Tg in Fig. 8.

Fig. 9 shows the results of calculation of the increase of the electricity output for the steam rate of 1 kg to the steam turbine.

Though calculations have been conducted to coefficients of performance of 1.4, 1.6 and 1.8 for an absorption heat pump, the value of 1.6 is very popular and practical figure for a single stage H_2O-LiBr absorption heat pump.

The increase of the output of electricity is 2.25 kcal/kg for T_2 of 70°C and COP of 1.6.

In this case the heating capacity as a heat pump becomes 37.6 kcal and for COP of 1.6 the heating capacity of the generator is 37.6/1.6 kcal from the definition of COP.

The steam rate of the simply condensing steam turbine for a 30 x 10^3 kW power station is, generally speaking, about 100 T/h.

The increase of the electricity, ΔE, of the power plant with the steam rate of 100 T/h, by adding a heat pump heating the feed water up to 70°C, amounts to:

$$\Delta E = 2.25 \times 10^5 \times 0.95 \times \frac{1}{860} = 249 \text{ kW}$$

when the power consumption for auxiliaries of the power plant is assumed to be 5% of the output.

Judging from the heating capacity in the generator the size of the absorption heat pump corresponding to this will be

$$23.5 \times 10^5/4{,}500 = 522 \text{ USRT}$$

and it is agreeable that an abosption refrigerator with a capacity of 500 to 550 USRT usually used for air-conditioning can be applicable.

5. Profitability

The annual increase of electric power will reach 1.988×10^6 kWh for a 100 T/h power plant, because a plant like this runs about 8,000 hours a year.

The annual profit becomes $1.988 \times 10^4 \times a$ \$ for a ¢ of the evaluation cost for 1 kWh.

Consequently, the allowable initial cost will be $1.988 \times 10^4 \times a \times n$ \$, simply being calculated, for the redemption of n years.

In Fig. 10 the horizontal line gives the evaluation cost a ¢ for 1 kWh and the vertical line gives the allowable initial cost \$ of a heat pump for the steam rate of 100 T/h with parameters of n.

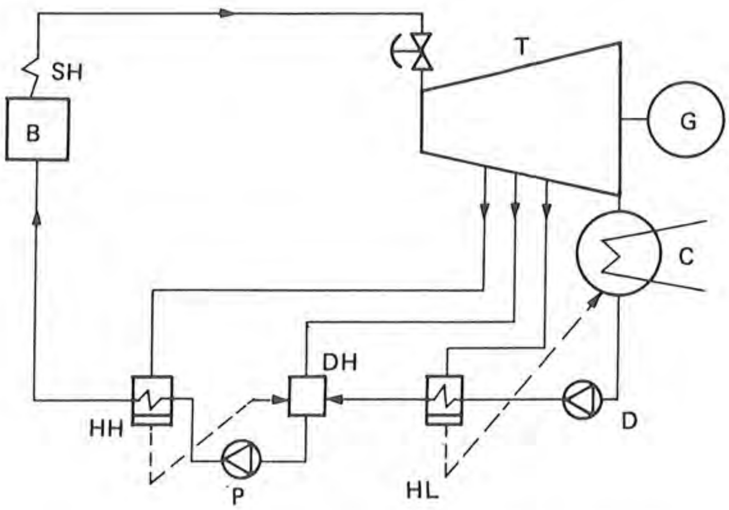
For example, the allowable initial cost can be 2.78×10^5 \$ for 5 ¢ evaluation cost of power generation and 3 years redemption and this initial cost will be reasonable for the single effect absorption refrigerator with capacity 500 USRT.

Especially, the case of Fig. 5, in which the absorber and condenser of the absorption heat pump are directly used as a heater for feed water and the evaporator of the heat pump used as a part of the main condenser for the steam turbine, will be more advantageous because of being able to eliminate some parts from the conventional system.

Conclusion

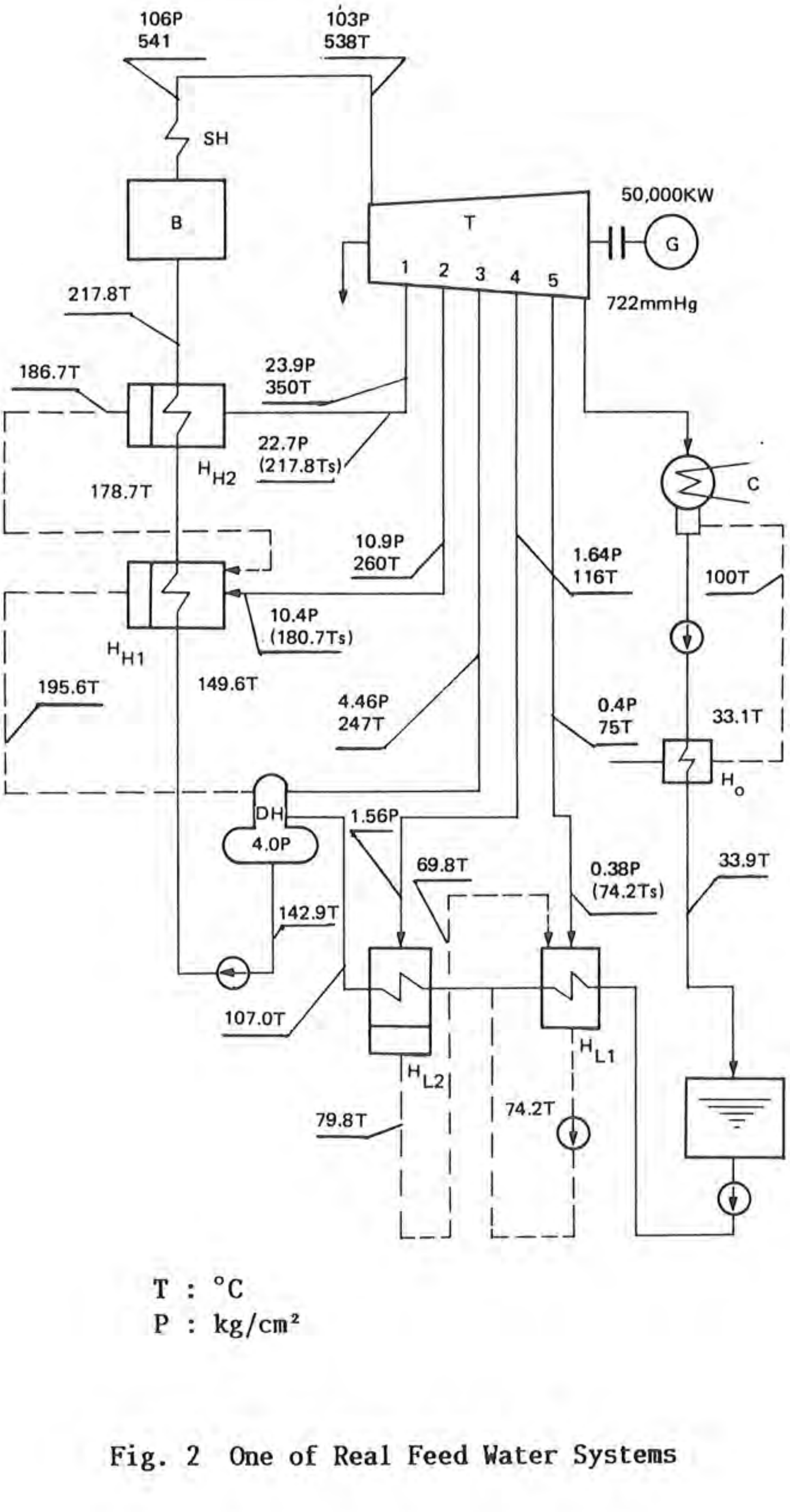
A calculating system of the increase of the output of a steam turbine power plant using an absorption heat pump to heat the feed water is concretely presented.

When an absorption heat pump using a combination H_2O and LiBr, which is very popular in air-conditioning, is applied, and the heater of the feed water and a part of the main condenser are inserted into the absorption heat pump, the initial cost can be brought down and it has been examined that the system is profitable and can be applied to practical use.



B : Boiler
SH : Superheater
G : Generator
T : Turbine
C : Condenser
HH : High pressure water heater
HL : Low pressure water heater
D : Condensate pump
P : Feed water pump
DH : Deaerator

Fig. 1 A General Diagram of a Steam Turbine Power Plant



T : °C
P : kg/cm²

Fig. 2 One of Real Feed Water Systems

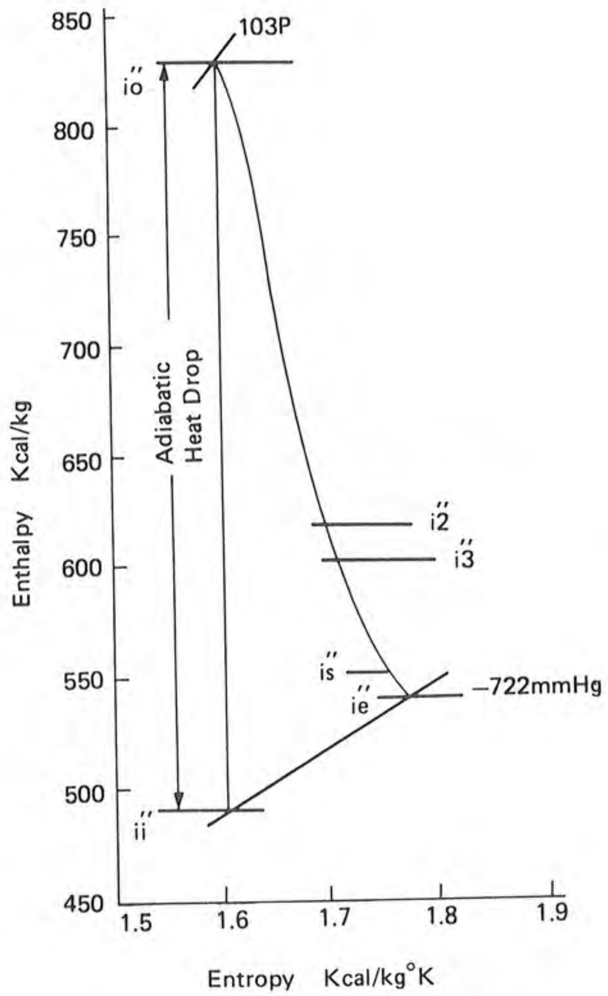
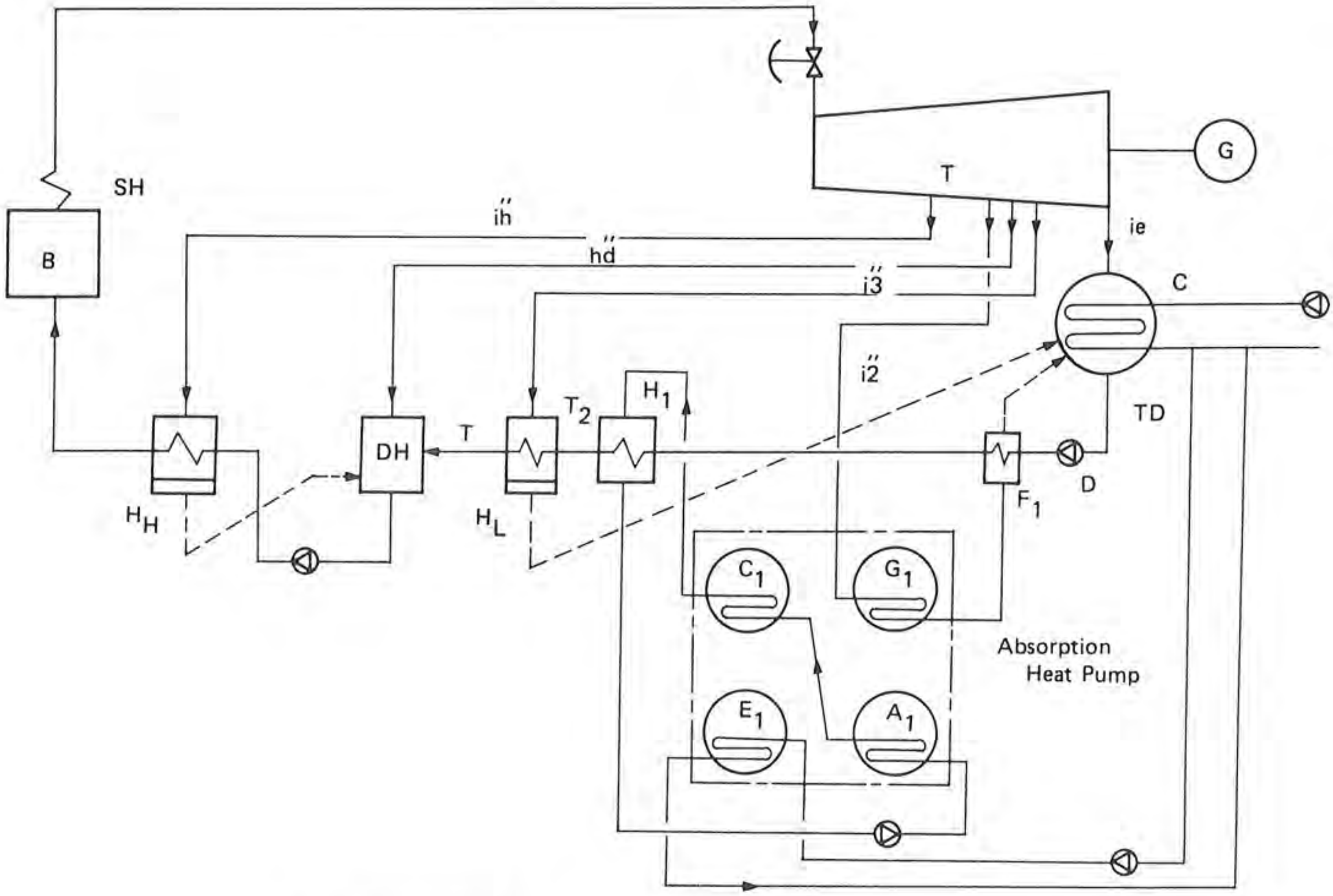


Fig. 3 Expansion Process in a Steam Turbine



G_1 : Generator
A_1 : Absorber
C_1 : Condenser
E_1 : Evaporator
of an absortion unit
H_1 : Feed water heater by heat pump
F_1 : Feed water heater by drain

Fig. 4 A Feed Water System Heated by an Absorption Heat Pump

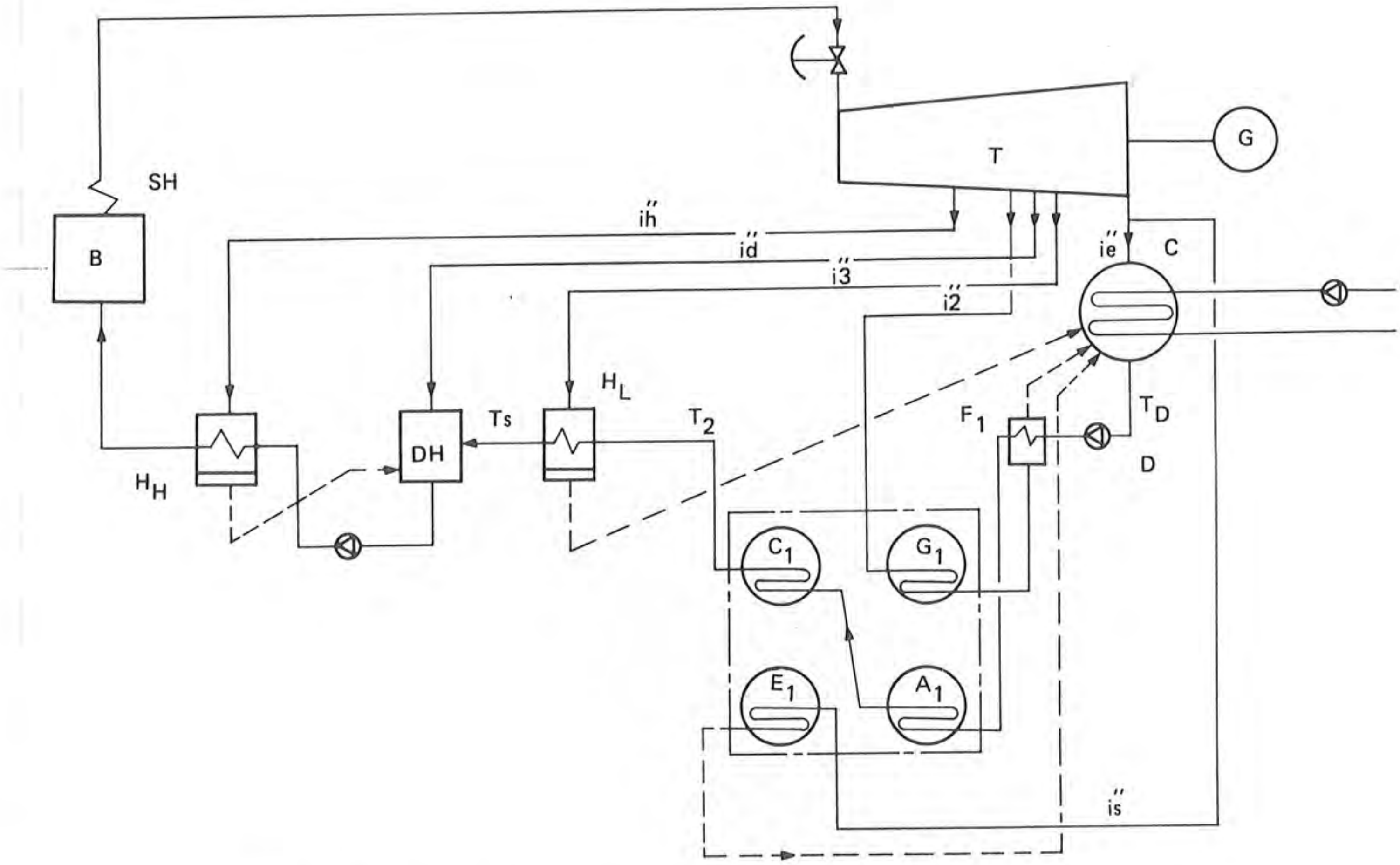


Fig. 5 An Improved Feed Water System Heated by an Absorption Heat Pump

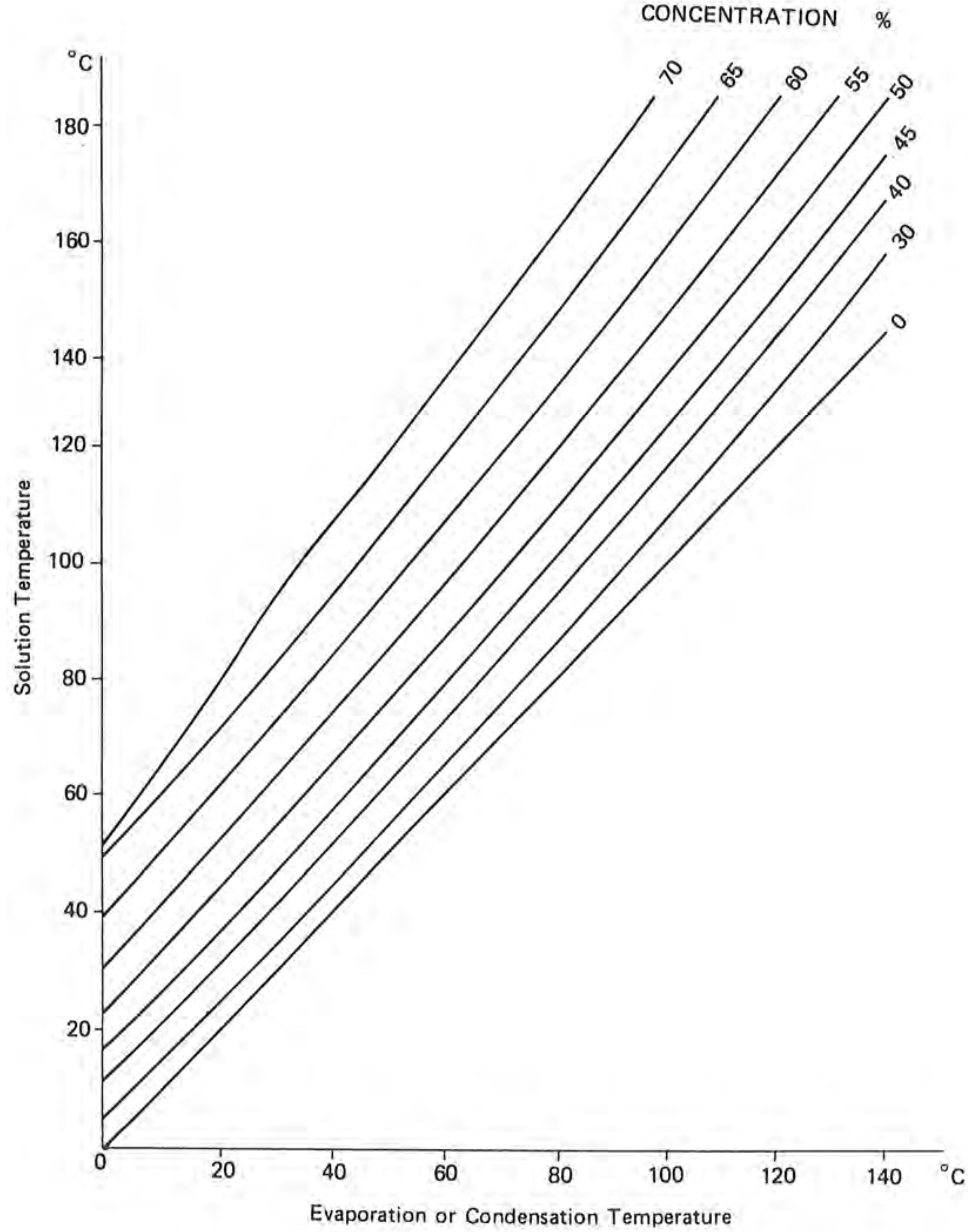


Fig. 6 Dühring Diagram of Lithium Bromide Solution

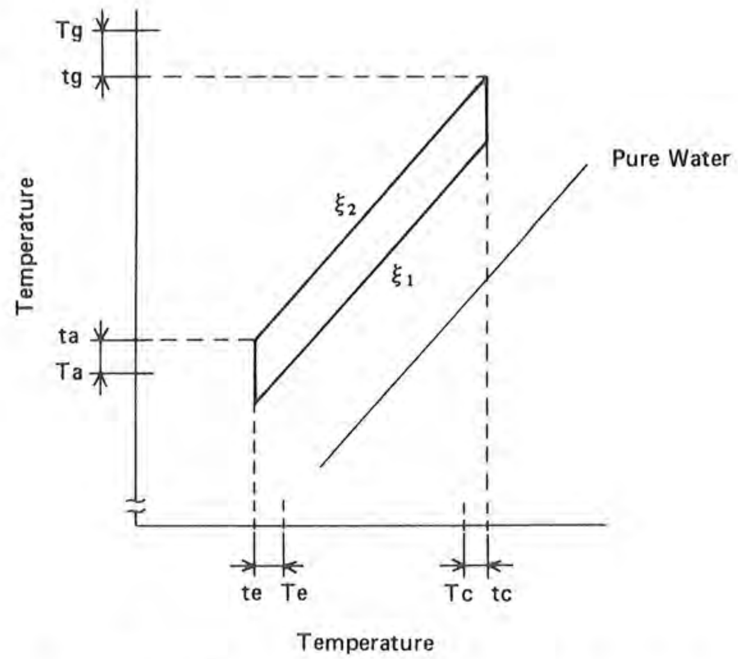


Fig. 7 A Single Stage Absorption Cycle

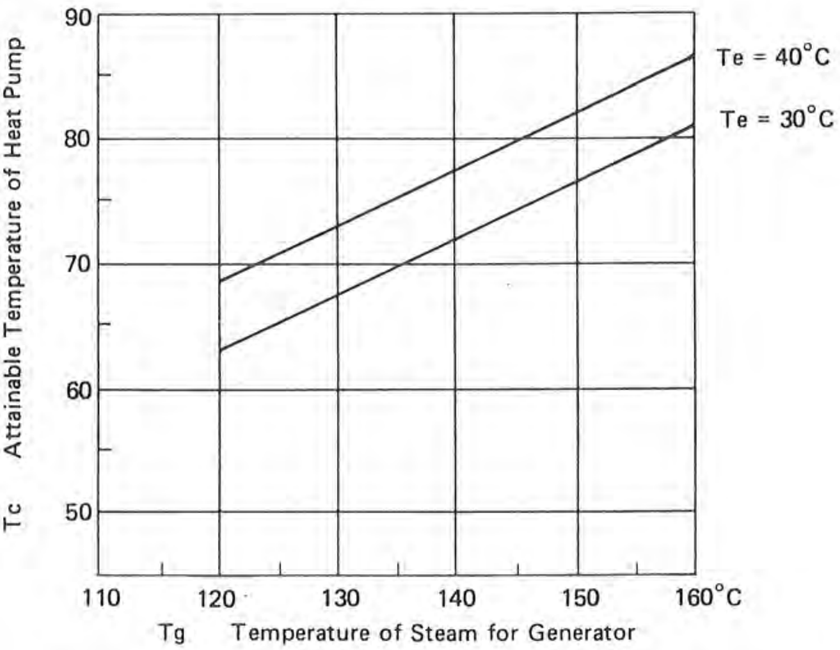


Fig. 8 Relations between Tg and Te for a Practical Absorption Heat Pump

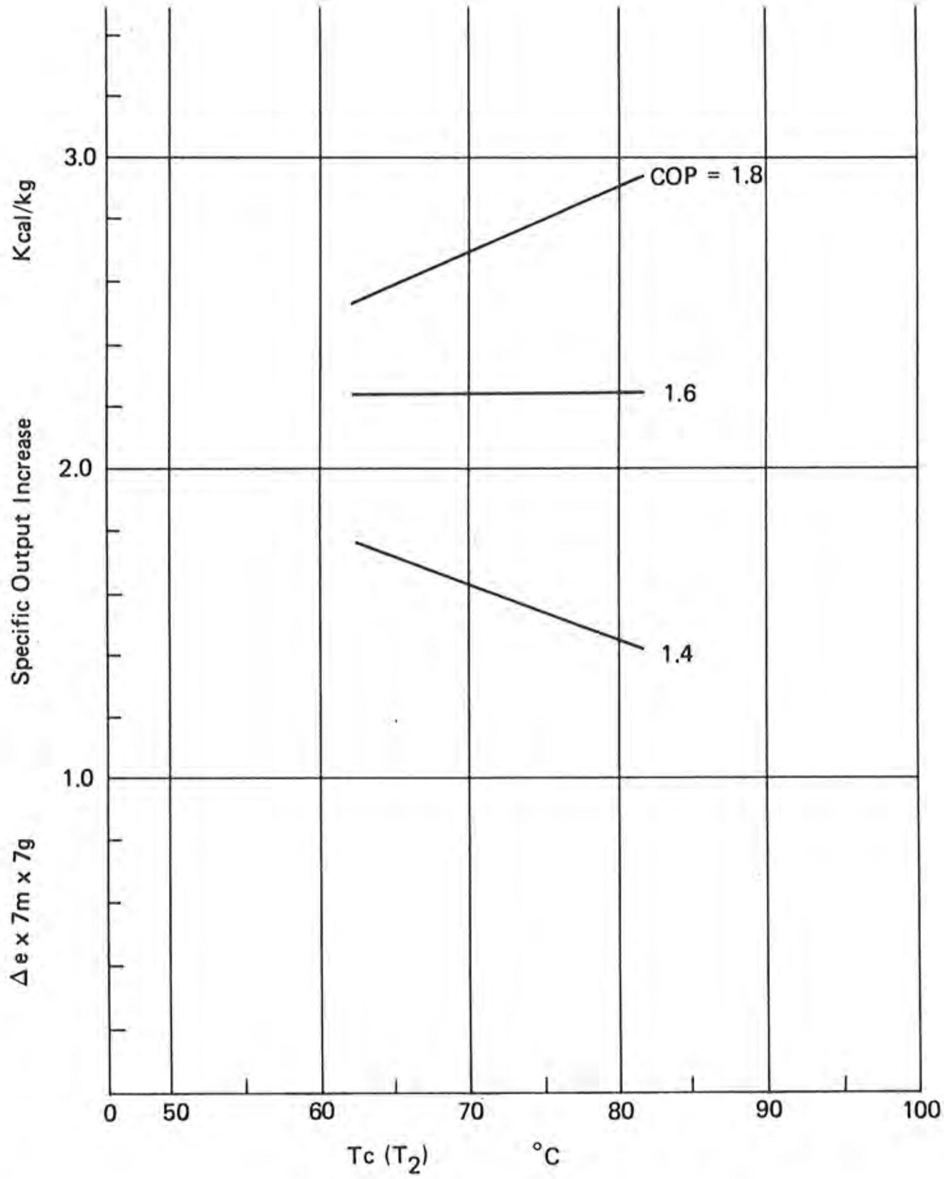
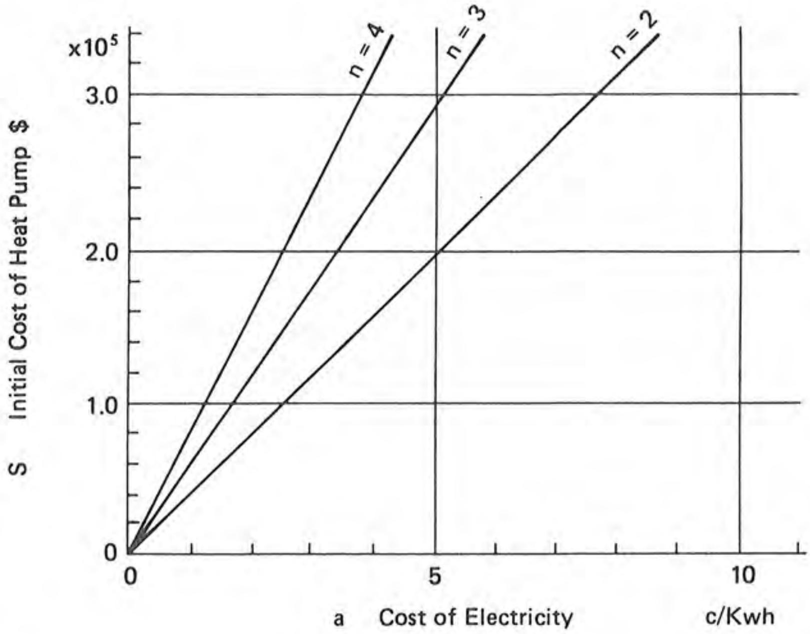


Fig. 9 Specific Output Increase in Case of Improved Feed Water System Heated by an Absorption Heat Pump



HP capacity : 550 USRT Equivalent
HP operation hours : 8,000 H/y
Boiler steam rate : 100 T/H

Fig. 10 A Diagram for Estimation of Initial Cost

Appendix 1

Derivation of equations for increase of electricity output of a steam turbine power plant by combination of a steam driven absorption heat pump.

1. Flow sheet

Fig. 1 and Fig. 2 show a flow diagram of a conventional steam turbine power plant and that of a combination respectively.

2. A Conventional System

An equation to calculate the output of electricity for a conventional steam turbine system is given by Eq. 1.

$$\begin{aligned} E_1 &= \{(G - G_h - G_d - G_3) \times (i''_o - i''_e) + G_h \times (i''_o - i''_h) \\ &\quad + G_d \times (i''_o - i''_d) + G_3 \times (i''_o - i''_3)\} \times \eta m \times \eta g \\ &= \{G \times (i''_o - i''_e) - G_h \times (i''_h - i''_e) - G_d \times (i''_d - i''_e) \\ &\quad - G_3 \times (i''_3 - i''_e)\} \times \eta m \times \eta g \end{aligned} \qquad \text{Eq. 1}$$

The heat balance in the low pressure feed water heater gives Eq. 2.

$$G_3 \times (i''_3 - T_S \times C_{PW}) = G \times (T_S - T_D) \times C_{PW} \qquad \text{Eq. 2}$$

where

E_1	: Output of electricity	kcal/h
G	: Steam rate to a steam turbine	kg/h
G_h	: Steam rate to a high pressure feed water heater	kg/h
G_d	: Steam rate to a deaerator	kg/h
G_3	: Steam rate to a low pressure feed water heater	kg/h
i''_o	: Enthalpy of steam to a turbine	kcal/kg
i''_h	: Enthalpy of steam to a high pressure feed water heater	kcal/kg
i''_d	: Enthalpy of steam to a deaerator	kcal/kg
i''_3	: Enthalpy of steam to a low pressure feed water heater	kcal/kg
i''_e	: Enthalpy of steam to a condenser	kcal/kg
T_S	: Temperature of feed water after a low pressure feed water heater	°C
T_D	: Temperature of condensate	°C
C_{PW}	: Specific heat	kcal/kg°C
ηm	: Mechanical efficiency	
ηg	: Generator's efficiency	

3. A combination system

1) Equation for output

Eq.3 gives the equation of output of electricity for the combination system.

$$E_2 = \{(G - G_h - G_d - G'_3 - G_{G1} - G_S) \times (i''_o - i''_e) + G_h$$

$$\times (i''_o - i''_h) + G_d \times (i''_o - i''_d) + G'_3 \times (i''_o - i''_3)$$

$$+ G_{G1} \times (i''_o - i''_2) + G_S \times (i''_o - i''_S)\} \times \eta m \times \eta g$$

$$= \{G \times (i''_o - i''_e) - G_h \times (i''_h - i''_e) - G_d \times (i''_d - i''_e)$$

$$- G'_3 \times (i''_3 - i''_e) - G_{G1} \times (i''_2 - i''_e) - G_S \times (i''_S - i''_e)\}$$

$$\times \eta m \times \eta g \qquad \text{Eq. 3}$$

where

E_2	: Output of electricity	kcal/h
G'_3	: Steam rate to a low pressure feed water heater	kg/h
G_{G1}	: Steam rate to a generator of absorption heat pump	kg/h
G_S	: Steam rate to an evaporator of absorption heat pump	kg/h
i''_2	: Enthalpy of steam to a generator of an absorption heat pump	kcal/kg
i''_S	: Enthalpy of steam to an evaporator of an absorption heat pump	kcal/kg

2) Equations of heat balance for a heater and a heat pump

Eq. 4 is for heat balance in the feed water preheater.

$$G_{G1} \times (t_{g1} - t_{g2}) \times C_{PW} = G \times (T'_D - T_D) \times C_{PW} \qquad \text{Eq. 4}$$

When feed water is heated by an absorption heat pump, the capacity of a low pressure feed water heater shall be diminished for a given outlet temperature and the steam rate shall be shifted from G_3 to G'_3.

Eq. 5 gives the equation for this case.

$$G'_3 \times (i_3 - T_S \times C_{PW}) = G \times (T_S - T'_D) \times C_{PW} \qquad \text{Eq. 5}$$

Equations of heat balancve for the absorption heat pump are given by Eq. 6 and Eq. 7.

Eq. 6 is for the feed water line and Eq. 7 for the heat pump itself.

$$C \times G_{G1} \times (i''_2 - t_{g1} \times C_{PW}) = G \times (T_2 - T'_D) \times C_{PW} \qquad \text{Eq. 6}$$

$$(C - 1) \times G_{G1} \times (i''_2 - t_{g1} \times C_{PW}) = G_S \times (i''_S - T_2 \cdot C_{PW}) \qquad \text{Eq. 7}$$

where

G''_3	:	Steam rate to a low pressure feed water heater	kg/h
T_D	:	Temperature of feed water from a condenser	°C
T'_D	:	Temperature of feed water after a preheater	°C
t_{g1}	:	Temperature of drain from a generator of an absorption heat pump	°C
t_{g2}	:	Temperature of drain after a preheater	°C
T_S	:	Temperature of feed water after a low pressure feed water heater	°C
T_2	:	Temperature of feed water after a heat pump	°C
C_{PW}	:	Specific heat of water	kcal/kg°C
C	:	Coefficient of performance	

4. Increase of output in a combination system

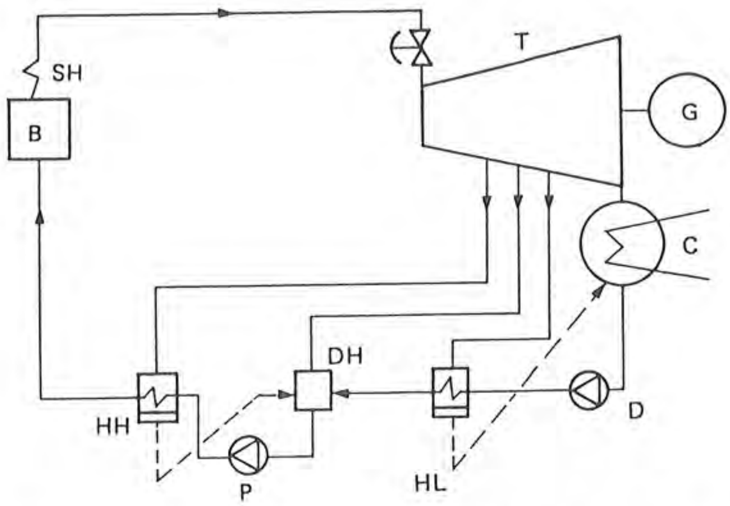
Eq. 8 can be derived from Eq. 1 through Eq. 7 for output increase with assumption of C_{PW} = 1.0 kcal/kg°C

$$\frac{1}{G} \times (E_2 - E_1) = \Delta e = \{\frac{T_S - T_D}{i''_3 - T_S} \times (i''_3 - i''_e) - \frac{T_S - T_2}{i''_3 - T_S} \times (i''_3 - i''_e)$$

$$- \frac{(i''_2 - i''_e) \times (T_2 - T_D)}{(t_{g1} - t_{g2}) + C \times (i''_2 - t_{g1})}$$

$$- \frac{(C - 1) \times (i''_s - i''_e) \times (i''_2 - t_{g1}) \times (T_2 - T_D)}{\{(t_{g1} - t_{g2}) + C \times (i''_2 - t_{g1})\} \times (i''_s - T_z)}\}$$

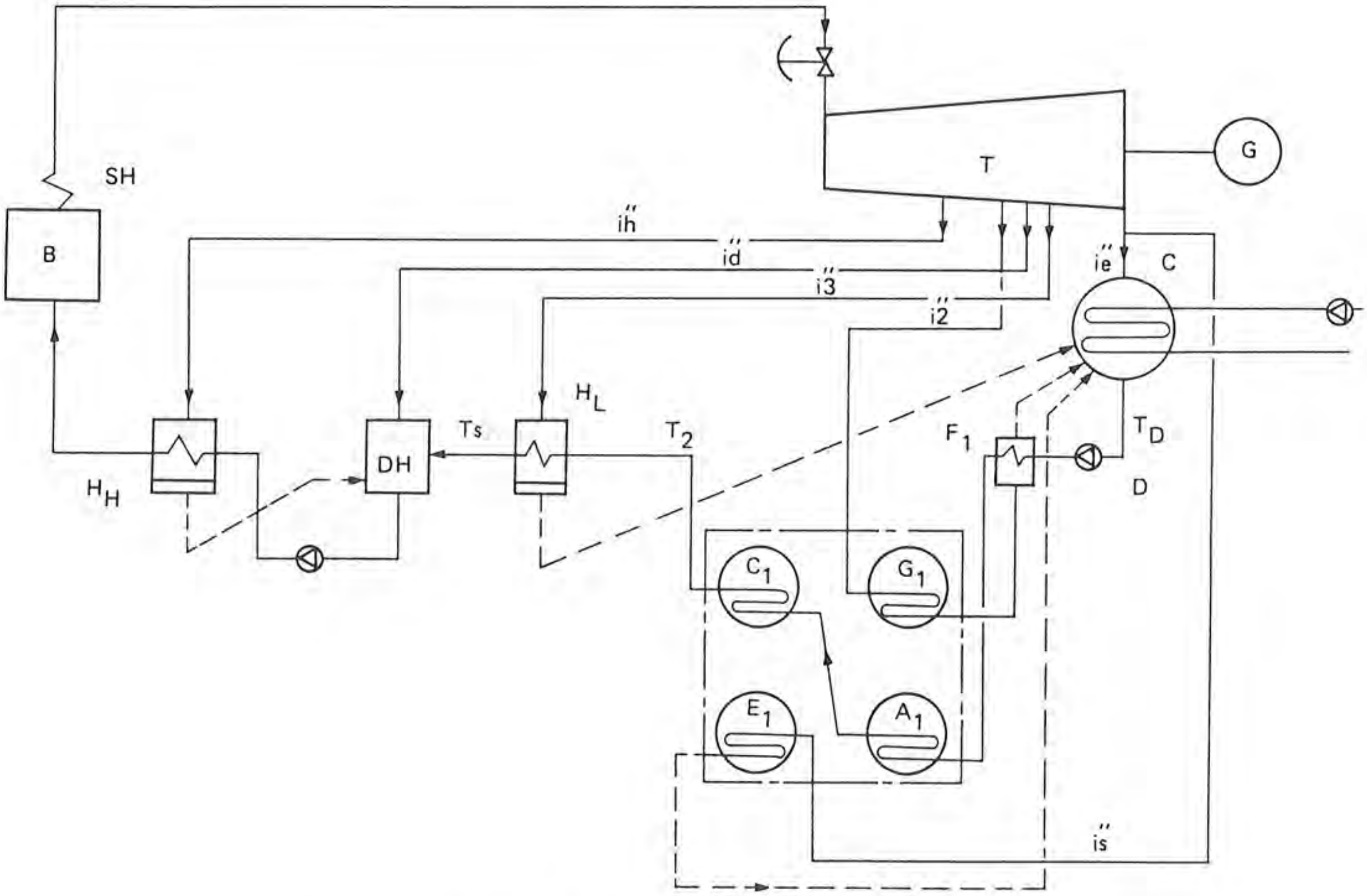
$$\times \eta m \times \eta g \qquad \text{Eq. 8}$$



B : Boiler
T : Turbine
C : Condenser
H_L : Low pressure feed water heater
H_H : High pressure feed water heater
D : Condensate pump

SH : Super heater
G : Generator
DH : Deaerator
P : Feed water pump

Fig. 1 General Diagram of a Steam Turbine Power Plant



G_1 : Generator   of absorption heat pump
A_1 : Absorber
C_1 : Condenser
E_1 : Evaporator
F_1 : Feed water preheater

Fig. 2 General Diagram of a Steam Turbine Power Plant whose Feed Water is heated by an Absoprtion Heat Pump

14. RESEARCH ON A TWO ABSORPTION HEAT TRANSFORMER

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1. Introduction

The development of an energy saving technology has relaxed in the last decade due to the low crude oil price. But recently the environment of the earth, especialy the global warming by CO_2 gas and the reduction of an ozone layer by CFC gas (Chloro Fluoro Carbon), becomes issues.

Absorption heat transformer can be driven by a waste heat at a temperature of more than 60℃, and it can produce a usable heat at a higher temperature than the heat source temperature. The application of this system to an industry brings out the reduction of CO_2 gas exhaustion by saving a primary energy, and furthermore it does not use CFC gas as a working fluid. Therefore, the development of this system has been paid attention.

A newly absorption heat transformer, which is called the "Mitsubishi Heat Transformer -MHT-", has developed. In a conventional one, the boost temperature, which is defined as the temperature difference between the heat source and the delivered temperature, is about 30℃, and the delivery temperature is usually less than 130℃. The newly developed MHT can boost a waste heat temperature by approximately 60℃ which is twice compared with a conventional system, and can provide 120℃ to 150℃ steam from a waste heat of temperature from 60℃ to 90℃. The two proto-types of MHT, which are No.1 MHT and No.2 MHT, have developed and tested (1) (2). The No.1 MHT was developed for the purposes of studying the operations, the characteristics of a two-stage heat transformer system, and of confirming a large boost temperature. The system is manually controlled at the starting, running, and stopping. The No. 2 MHT was developed for the purposes of studying the dynamic characteristics, its automatical operation method for the application to an industrial process, and the long-term reliability.

This report presents the outline of the design and performance of two proto-type MHTs.

2. Absorption Heat Transformer System

A two-stage absorption cycle is considered to achieve a high boost temperature as twice as a conventional one. The typical two-stage systems are shown in Figure 1. In the system-A and system-B, the heat output at the first-stage absorber is utilized as the heat source at the second-stage evaporator. But in the system-C, the heat output at the first-stage absorber is utilized as the heat source at the second-stage generator. The solution flow is also shown in Figure 1. An output solution temperature at the second-stage absorber depends on a variation width of the concentration across the absorber, and the smaller the variation width of that is, the higher the output temperature is. Since the variation width of that in system-A is smaller than that in system-B, the higher temperature output is obtained in system-A. In system-C, the second-stage generator has a limiting state point related to the concentration, at which the solution experiences a crystallization of a LiBr salt.

From the above discussion, system-A is selected in order to achieve the highest temperature boost.

3. Newly Developed Heat and Mass Transfer Tube

A new type of tube has been developed to enhance a heat and a mass transfer. It is named the "Fir Finned Tube". This tube consists of a helical segmented fin around the tube. Therefore, LiBr solution flows slowly and spreads all over the surface area of the fins when it falls down along the outer surface of the vertical tube.

Heat and mass transfer characteristics were tested for the design optimization of heat exchanger. The test were carried out for the fin height of 8 mm to 11.3 mm and fin pitch of 5 mm to 10.6 mm under the fixed tube length of 1 m. Some examples of the tested Fir Finned Tube are shown in Figure 2. Inlet LiBr solution concentration, its temperature, its flow rate, and refrigerant vapor pressure were changed. An analytical model of the absorption process in the falling film flow was developed, and heat and mass transfer rate for long Fir Finned Tube was simulated. The whole tube is divided into some segments along its axis. In each segment, heat and mass balance are calculated by using the mass transfer coefficient in a layer of the film flow, and heat transfer coefficient on the outer surface of the tube. The simulation results were checked by the test of 4.8 m Fir Finned Tube. The test apparatus is shown in Figure 3. These results were used for the design of No.1 MHT and No.2 MHT. This tube is used in the generator and absorber for the No.1 MHT and is used in the generator, the first-stage evaporator, the first- and the second-stage absorber for the No.2 MHT.

4. System Design and Performance of No.1 MHT

4.1 Construction

A schematic diagram of the No.1 NHT is shown in Figure 4 and its photograph is shown in Figure 5. This heat transformer consists of three major components, two heat-recovery heat exchangers, refrigerant and solution pumps, valves, and connecting pipes.

The generator and the condenser are housed in the righthand component in Figure 4. The generator is composed of a vertical Fir Finned Tube bundle. The condenser tubes are also perpendicularly arranged at the outside of the generator. Waste hot water flows in the generator tubes and cooling water flows in the condenser tubes.

The first- and the second-stage evaporator, and the first stage absorber are housed in the middle component in Figure 4. The outside surface of the vertical Fir Finned Tube bundle, placed at the center of this component, is used for the first-stage absorber, and the inside surface is used for the secondstage evaporator. The first-stage evaporator is located at the outside of the first-stage absorber shell. The upper section of the second-stage evaporator is connected to the first-stage evaporastor through the flow regulation valve (V11) and to the second-stage absorber through hte valve (V12).

The secondo-stage absorber is shown in the left-hand component in Figure 4. It also consists of the perpendiculary arranged Fir Finned Tube bundle and is connected to the inlet and the outlet of the process water or steam line.

The two heat-recovery heat exchangers, called the recuperator, are shell-and-tube type heat exchangers. Solution and refrigerant pump are canned-moter centrifugal pump. The main specifications for each heat exchanger are shown in table 1.

The four valves, V1～V4, are used for start-up procedure. Solution and refrigerant flow are regulated through each valve by detecting the liquid levels of each component. An aqueous LiBr solution is used as a working fluid.

4.2 Description of the cycle

As shown in Figure 4, waste hot water is introduced into the generator and the first-stage evaporator, while cooling water is required at the condenser. The process steam is generated at the second-stage absorber. The temperature, pressure, and solution state at each component are plotted on an equilibrium chart for aqueous LiBr solution in Figure 6.

The diluted solution (Pt.7) from the first-stage absorber enters into the bottom of the generator and about half of it circulates the generator. It folws down along the generator tubes and refrigerant vapor is desorbed from the solution by the heat from the waste hot water flowing in the tubes. The refrigerant vapor goes to the condenser and condenses at temperature Tc (Pt.8). The condensate is introduced into the second-stage evaporator (Pts.8 to 10). At this evaporator, about half of the condensate evaporates at temperature T_{e2} (Pt.10) by the heat output from the first-stage absorber. This refrigerant vapor raises the pressure up to a high pressure P_H ,in the second-stage evaporator and the second-stage absorber. Unvaporized refrigerant at the second-stage evaporator enters to the first-stage evaporator and is vaporized at temperature T_{e1} by the heat from the waste hot water flowing in the tubes (Pt.9). This refrigerant vapor raises the pressure up to the intermediate pressure, Pm, in the first stage evaporator and the first-stage absorber.

The concentrated solution in the generator (Pt.1) is introduced into the top of the second-stage absorber. On the way to this absorber, the solution is preheated by the diluted solution through the first recuperator (Pts.1 to 2) and by the intermediate solution through the second recuperator (Pts.2 to 3). At the second-stage absorber, the solution flows down along the tubes and its temperature rises by absorbing the refrigerant vapor from the second-stage evaporator. The useful heat at temperature T_{a2} is transfered from the heated solution to the process water flowing in the tubes. And then the concentration of the solution becomes intermediate (Pt.4). This solution goes to the first-stage absorber and it absorbs the refrigerant vapor from the first-stage evaporator. The solution temperature is raised by the heat of the absorption process and this heat is transfered to the second-stage evaporator. As a result, the intermediate concentration solution becomes dilute (Pts.5 to 6) and it returns to the generator.

4.3 Operating Procedure

When the system starts up, the waste hot water is introduced into the generator and the first-stage evaporator, and cooling is introduced into the condencer.

At first, the solution in the generator is concentated by the circulation through the valve V6 shown in Figure 4 and the refrigerant vapor desorbed from the solution is condensed at the condenser. Then, the single-stage cycle is produced. The concentrated solution is sent to the first-stage absorber through the two recuperators, valve V2, and valve V9. The condensate is sent to the first-stage evaporator through the second-stage evaporator. After the vapor pressure in the second-stage evaporator rises enough, system changes to the two-stage cycle.

4.4 Experimental Results

The first purpose of the No.1 MHT's performance is to confirm the high temperature boost. The results are shown in table 2. This system achieved the output temperature of about 151 ℃ with the heat source temperature of about 89℃ and the cooling water temperature of about 23℃。

Figure 7 shows the results of raising the hot water temperature at the generator and the first-stage evaporator. In this case, the condensing temperature is fixed at 35℃ by regulating the cooling water temperature. The desorbed refrigerant vapor at the generator increases with the rise of the hot water temperature, and the refrigerant flow rate through each component increases. The solution concentration at the generator outlet increases. The intermediate pressure, P_m , at the first-stage evaporator is largely influenced by the hot water temperature, and it increases with the rise of that. The absorbing temperature, T_{a1}, at the first-stage absorber also rises because of higher absorbing pressure, P_m , and higher solution concentration. With the rise of the first-stage absorber temperature, T_{a1}, the second-stage evaporator temperature, T_{e2}, and its pressure, P_h , become high. Therefore the output steam temperature becomes high and the output heat also increases with the increased refrigerant flow rate.

The results of raising the cooling water temperature are shown in Figure 8. The hot water temperature of the generator and that of the first-stage evaporator are 84.5℃ and 92℃. The condensing temperature and pressure rise with the rise of the cooling water temperature and the refrigerant flow rate decreases. Therefore, the solution concentration at the generator outlet, shown at Pt.1 in Figure 6., becomes dilute under the constant generating temperature. The pressure in the first-stage evaporator is approximately constant because of the constant hot water temperature, but the heat transfer rate decreases due to the decrease in the refrigrant flow rate. At the second-stage absorber, mass transfer rate decreases due to the decrease of the inlet solution concentration and it causes to the rise of this pressure slightly. But, the output heat and temperature decrease with the rise of the cooling water temperature.

5. System Design and Performance of the No.2 MHT

5.1 Construction and Cycle Description

A schematic diagram of the No.2 MHT is shown in Figure 9 and a photograph of it is shown in Figure 10. This system has the differdent construction from the No.1 MHT except for the second-stage absorber. The condenser is divided from the generator, and the first-stage evaporator is also divided from the first-stage absorber and the second-stage evaporator. Two solution pumps are added between the second-stage absorber and the generator, and one refrigerant pump is added between the first- and the second-stage evaporator. These pumps are controlled by the signal of liquid level meter installed each component. Each heat exchanger except for the condenser consists of the vertical Fir Finned Tubes as shown in Figure 2. The main specifications of heat exchangers are shown in Table 3. The component's shells and pipes are made from carbon steel. The heat transfer tubes and fins are made from copper. The improvements above-mentioned, make it easy to scale up the system compared with the No.1 MHT, and allow the automatic operation.

Corrosion of the materials is a great problem for reliability. For high temperature operation at more than 150℃, it is said that the corrosion effect of LiBr solution upon the carbon steel is particularly enhanced. Anticorrosion technology has been studied for long-term reliability of the machine. Based on these studies, it was found that Li_2MoO_4 with LiOH was an effective inhibitor. For copper some additions functioned well.

The solution cycle is as same as the No.1 MHT but the refrigrant flow differs from that. At first, liquid refrigerant from the condenser goes to the first-stage evaporator (Pts. 8 to 9) and then goes to the second-stage evaporator (Pts. 9 to 10).

5.2 Automatic Control and Operation

The starting procedure is shown in Figure 11. At first, the degree of vacuum in each component is checked. If the pressure is not enough low, the purge unit is operated automatically. Second, an open/shut state of each valve is checked. After those checks, the unit starts automatically. The solution, refrigerant, heat source, and cooling water pumps start in fixed order.

In a steady state, the heat output is controlled by the automatic flow rate control valves (V2, V3, and V4) and the rotational speed control of pump P4, which are controlled by the signals of the liquid level meters in each component.

In a stop control, the concentration of the solution is checked by detecting the liquid levels in the condenser and the first-stage evaporator. If the concentration is high, the solution in the generator is diluted to prevent a crystallization from the solution.

5.3 Experimental Results

The steady-state performance tests were carried out and its results are shown in Table 4. It is found that the No.2 MHT can generate the output temperature of 151.2℃ with a heat source temperature of 91.0℃, and a cooling water temperature of 32.5℃. Its COP is 0.31. These values are similar to those of the No.1 MHT. The actual heat output was found to be as large as 90% of the designed value. This seems to be due to the low thermal effectiveness of the recuperator.

The starting test at a room temperature was carried out. The MHT reached steady state approximatelly 40 minutes later. This time is relatively short, so that this system is sufficient for a practical use.

In the actual processes, flow rate and temperature fluctuations of the heat source and the cooling water may occur. Dynamic responce of the No.2 MHT to such fluctuations becomes important in the installation. Figure 12 shows an example of the response to an abrupt decrease of 40% of the heat source flow rate. The abrupt decrease causes the operating state of the cycle to the changes and it takes about 30 minutes to return to steady state condition.

6. Conclusion

A tow-stage absorption heat transformer has been developed for the first time. This system is named the MITSUBISHI HEAT TRANSFORMER -MHT- and it has a newly developed Fir Finned Tube in order to enhance a heat and mass transfer. It can be driven by waste heat ranging from 60℃ to 90℃, and can deliver a process heat up to 150℃. This boost temperature of 60℃ is twice that of a conventional one. And it has a COP of 0.31, close to a theoretical value. The starting, steady state and stopping operation are controlled automatically. This system reaches a steady state about 40 minutes later after the starting and 30 minutes later after the abrupt change of operating condition.

7. Reference

(1) M. Ikeuchi, et. el., ASHRAE Transaction, Vol.91, Pt2B, (1985), pp2081-2094.

(2) This paper will be presented at ASHRAE Annual Meeting in Vancouver in 1989.

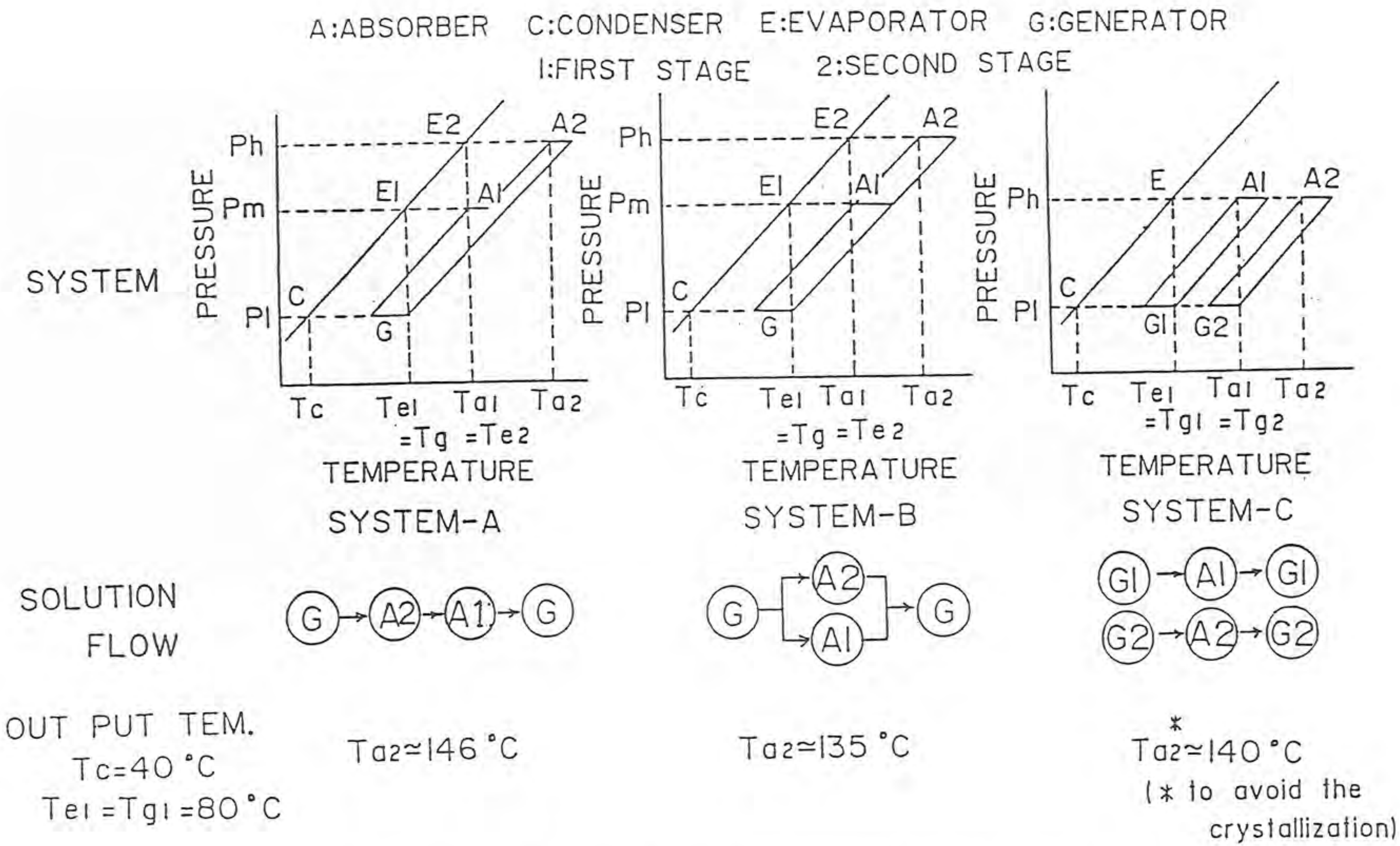


Figure 1. Three Types of Two-Stage Absorption Heat Pump System

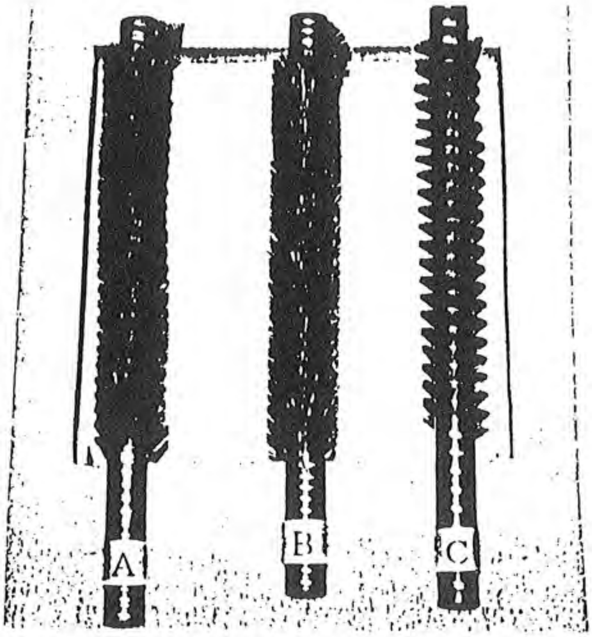


Figure 2. Some Examples of the Tested Fir Finned Tube

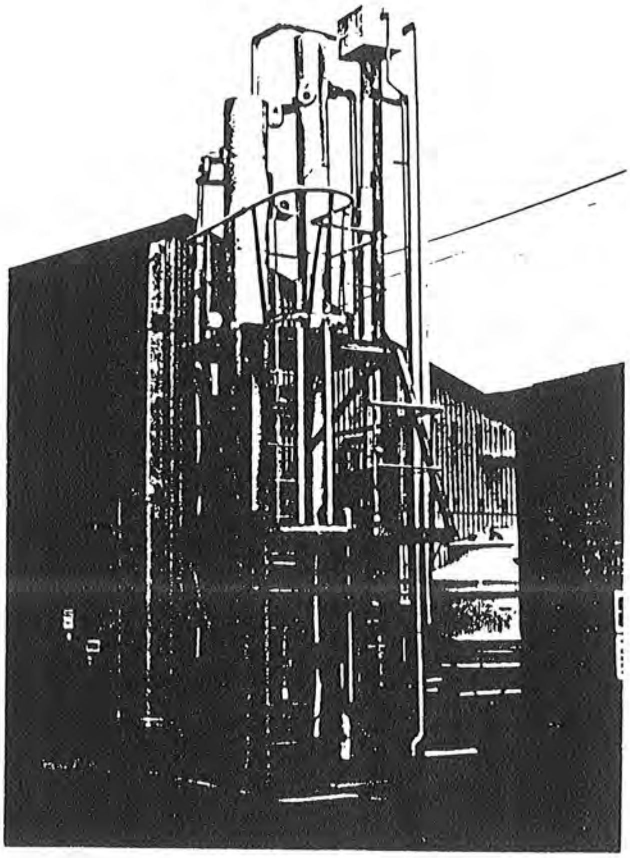


Figure 3. Test Apparatus of the Long Fir Finned Tube

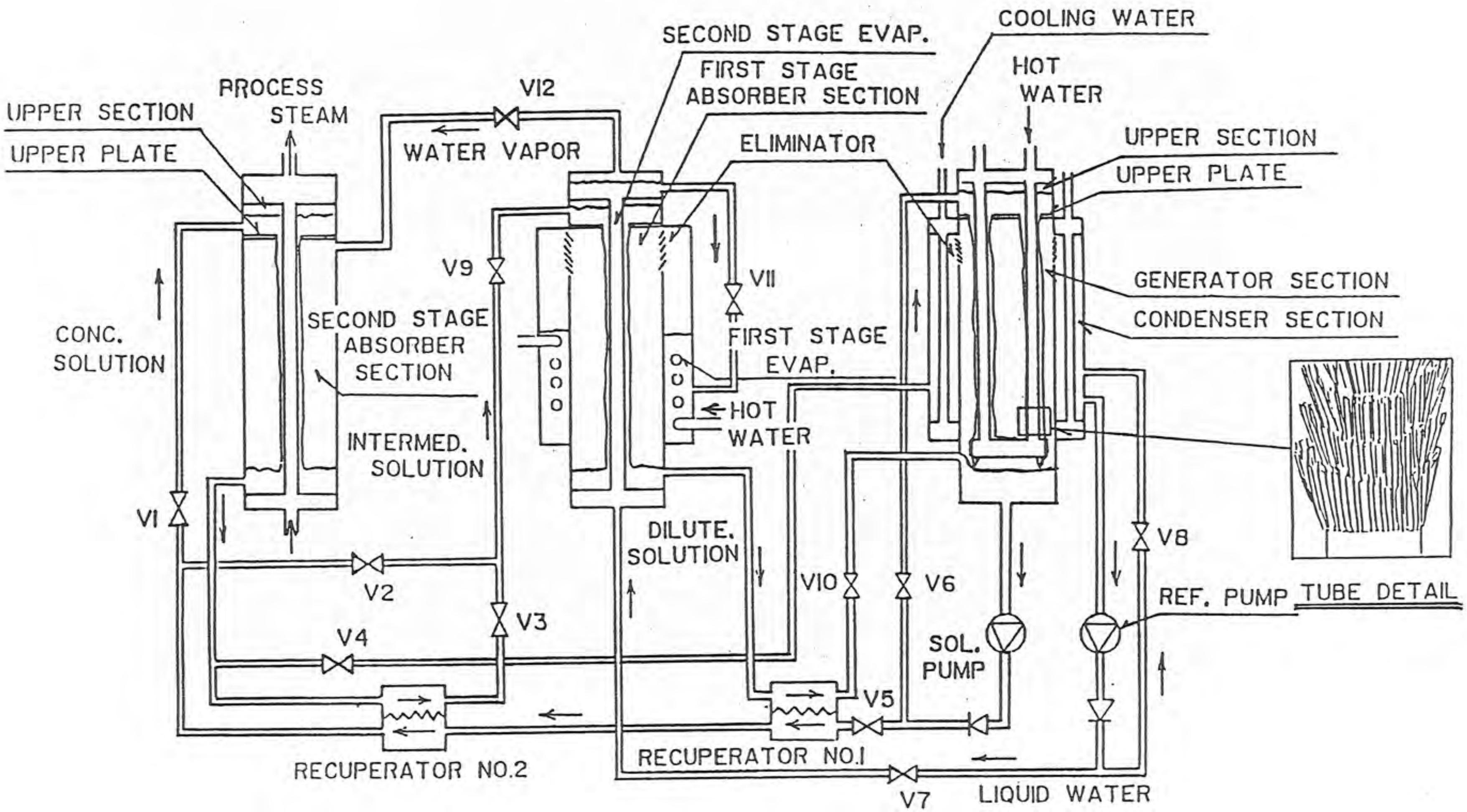


Figure 4. Schematic Diagram of the No.1 MHT

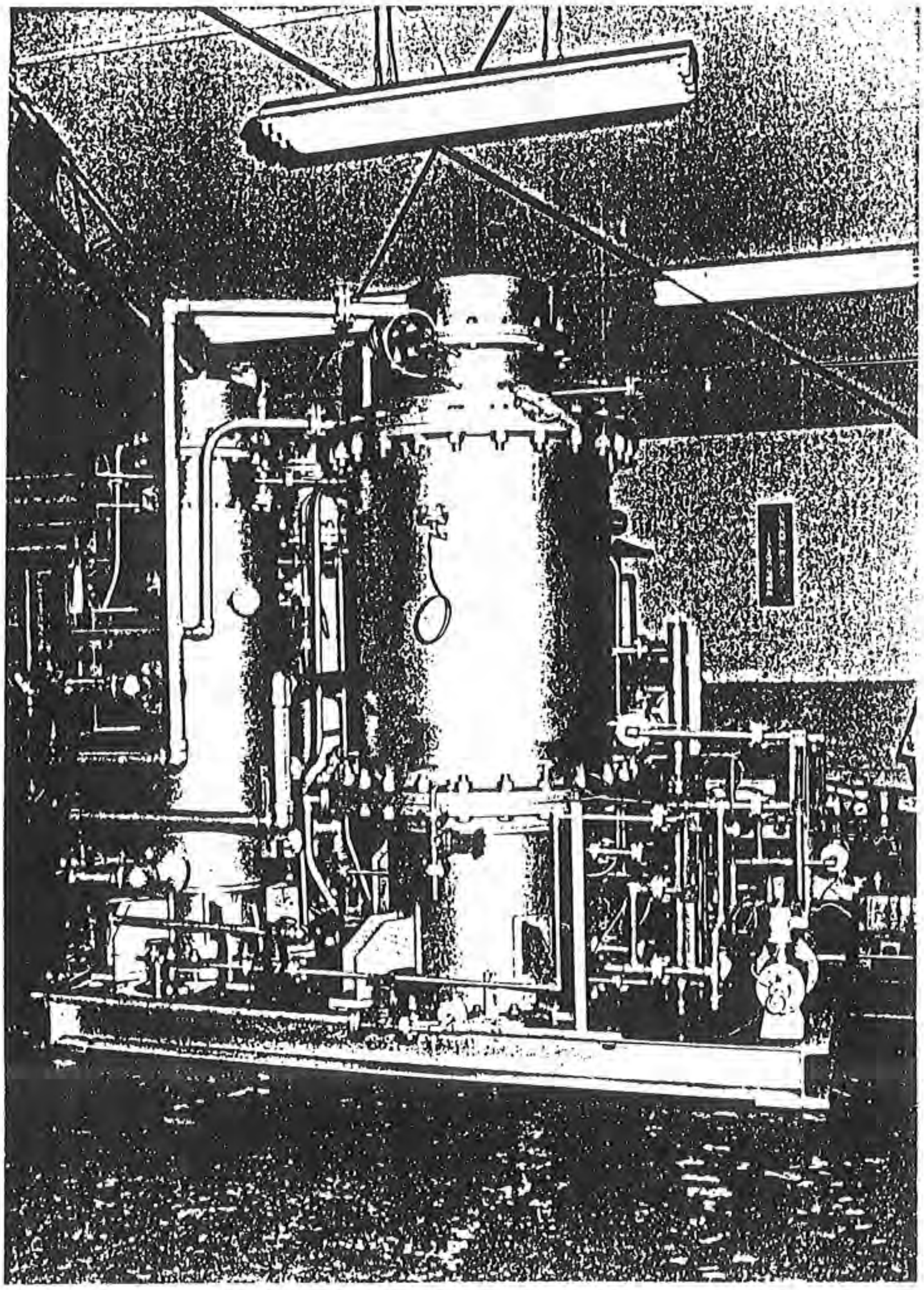


Figure 5. Photograph of the No.1 MHT

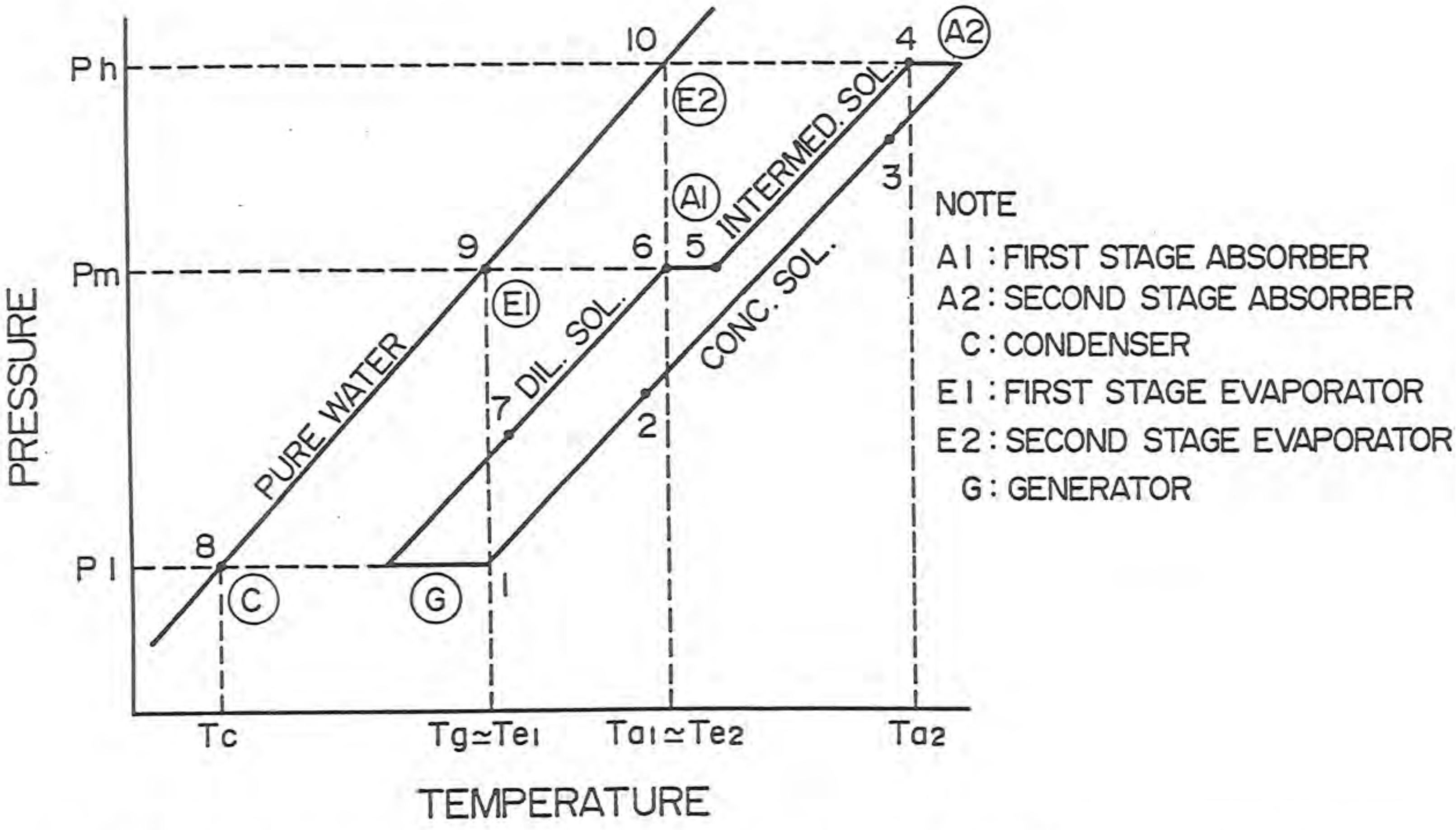
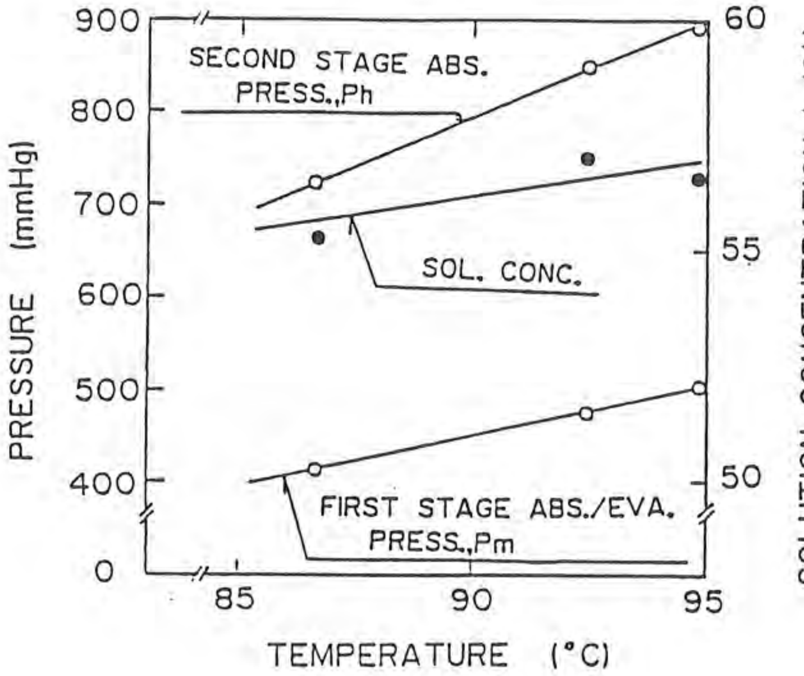
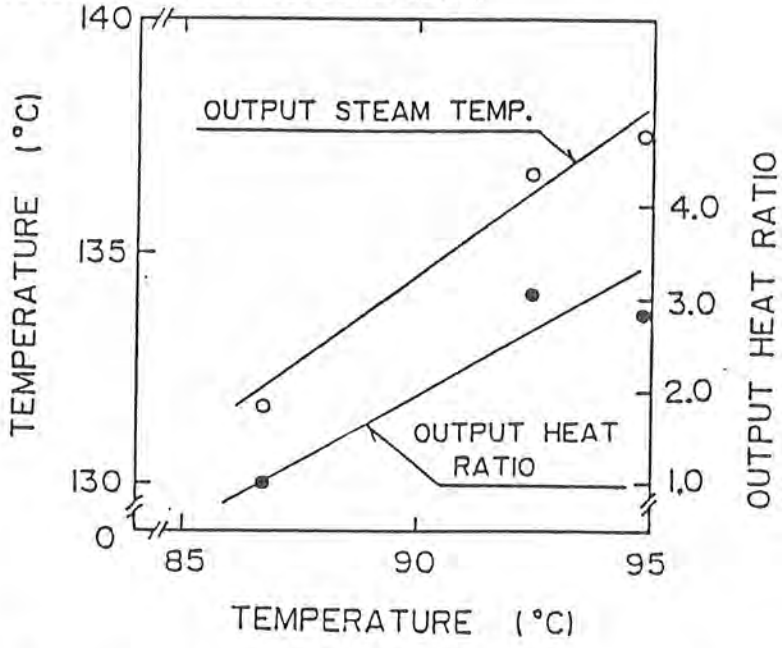


Figure 6. High Temperature Boost Cycle on an Equilibrium Chart for Aqeous LiBr Solution

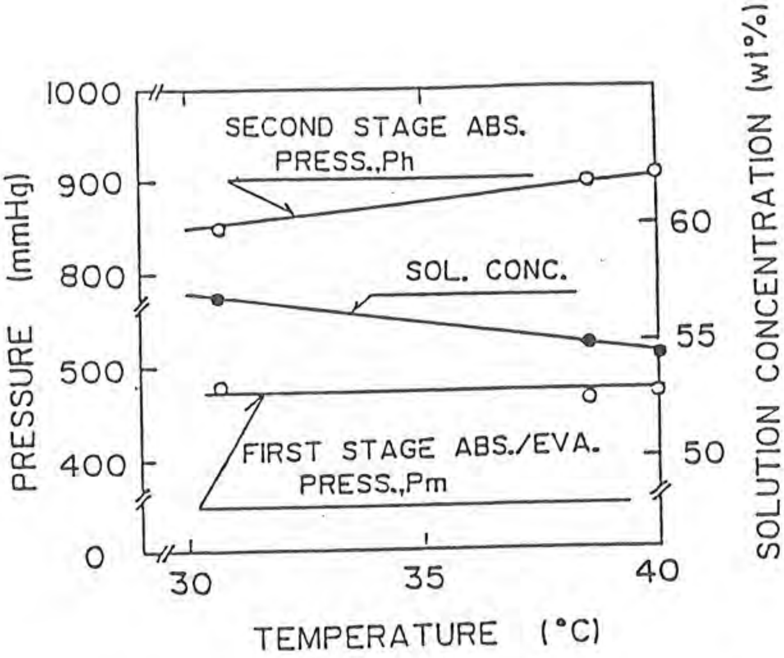


(a) Pressures of Second Stage Absorber and First Stage Absorber/Evaporator, and Generator Outlet Solution Concentrtaion

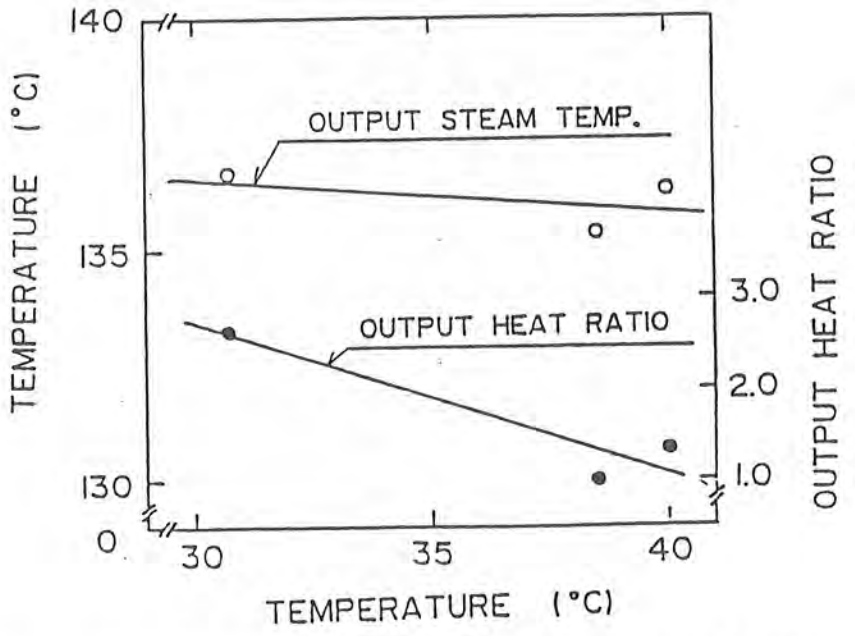


(b) Output Steam Temperature and Output Heat Ratio

Figure 7. Heat Pump Performances versus Heat Source Temperature



(a) Pressures of Second Stage Absorber and First Stage Absorber/Evaporator, and Generator Outlet Solution Concentration



(b) Outlet Steam Temprature and Outlet Heat Ratio

Figure 8. Heat Pump Performances versus Cooling Water Temperature

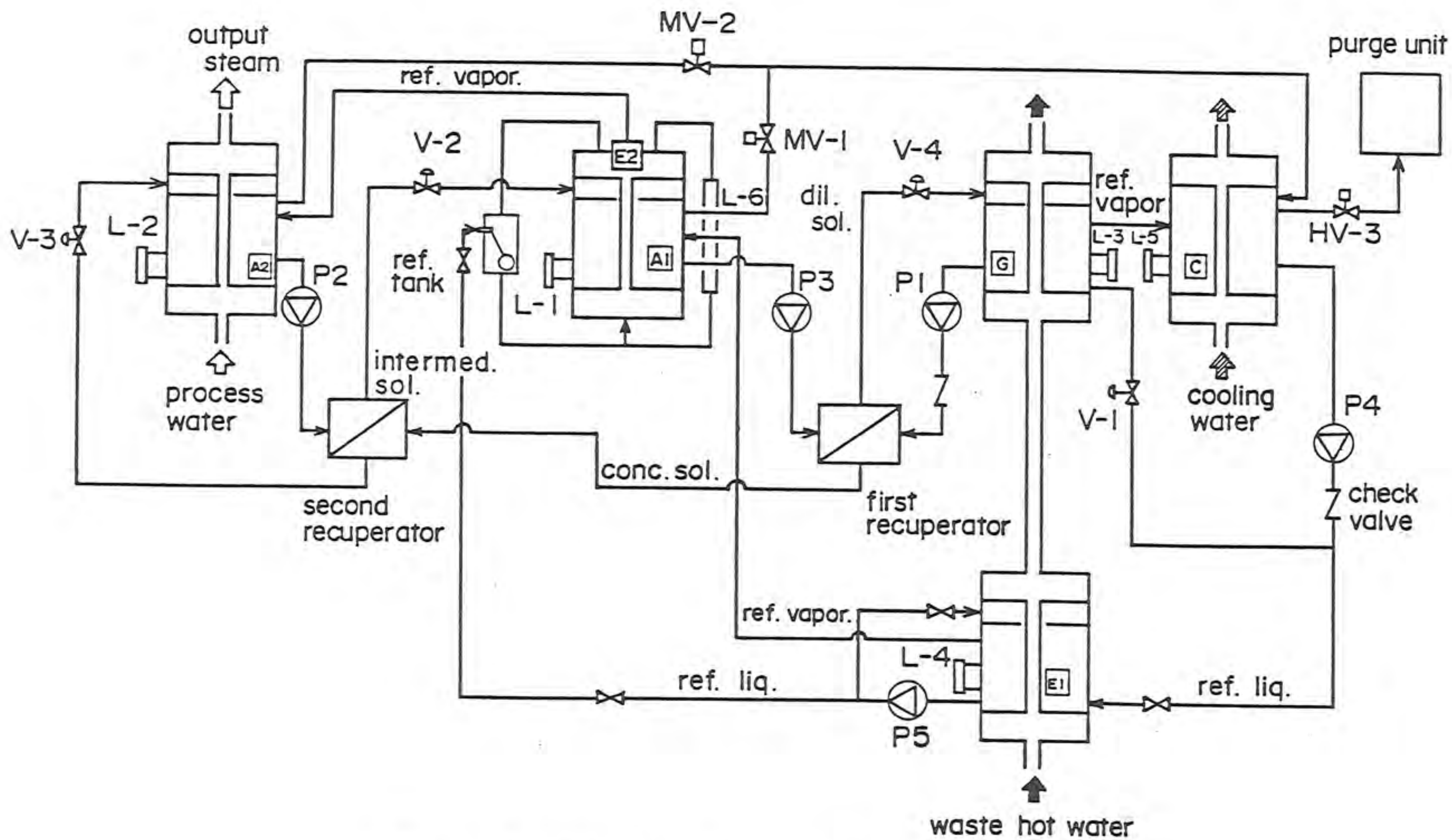


Figure 9. Schematic Diagram of the No.2 MHT

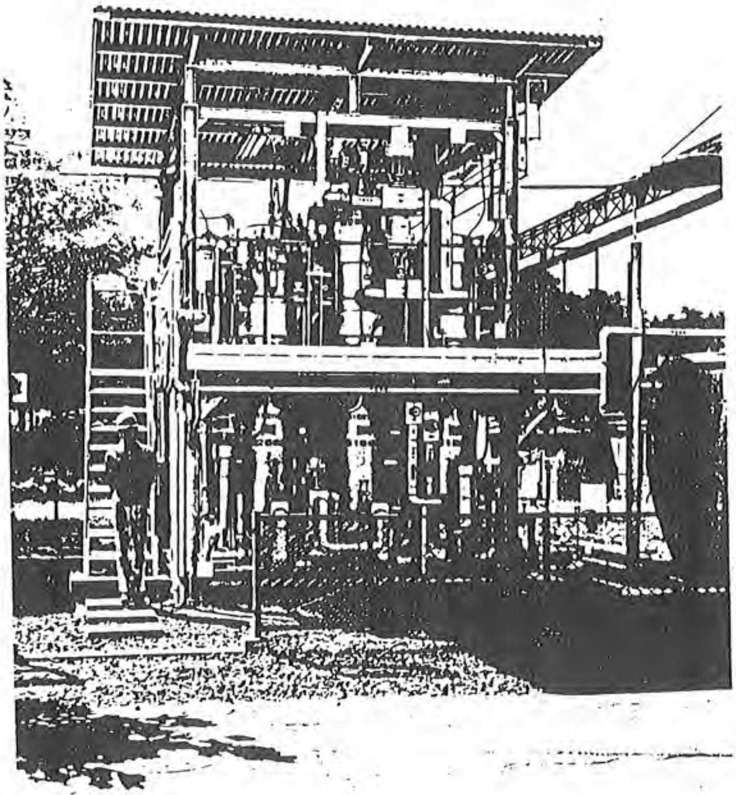


Figure 10. Photograph of the No.2 MHT

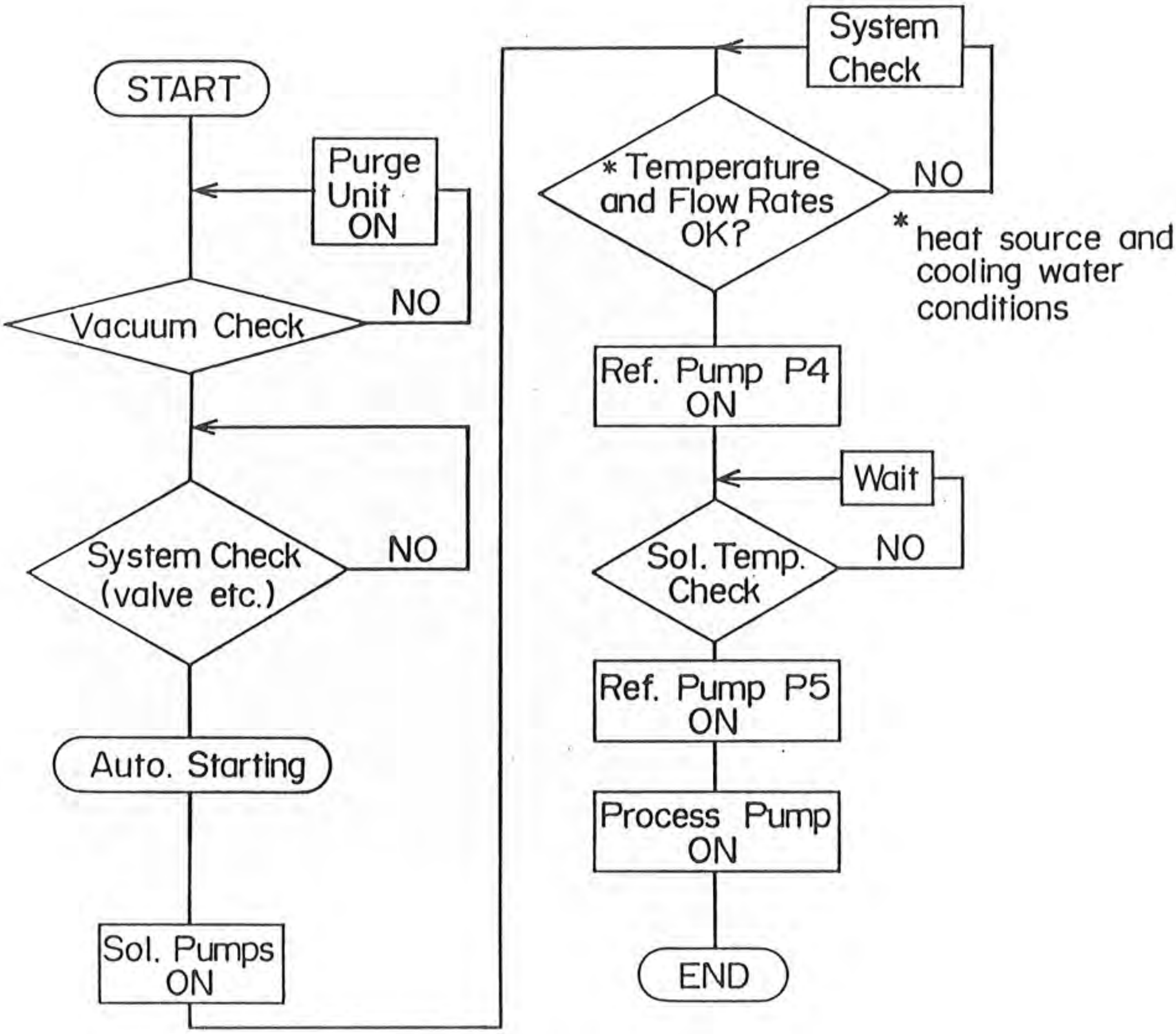
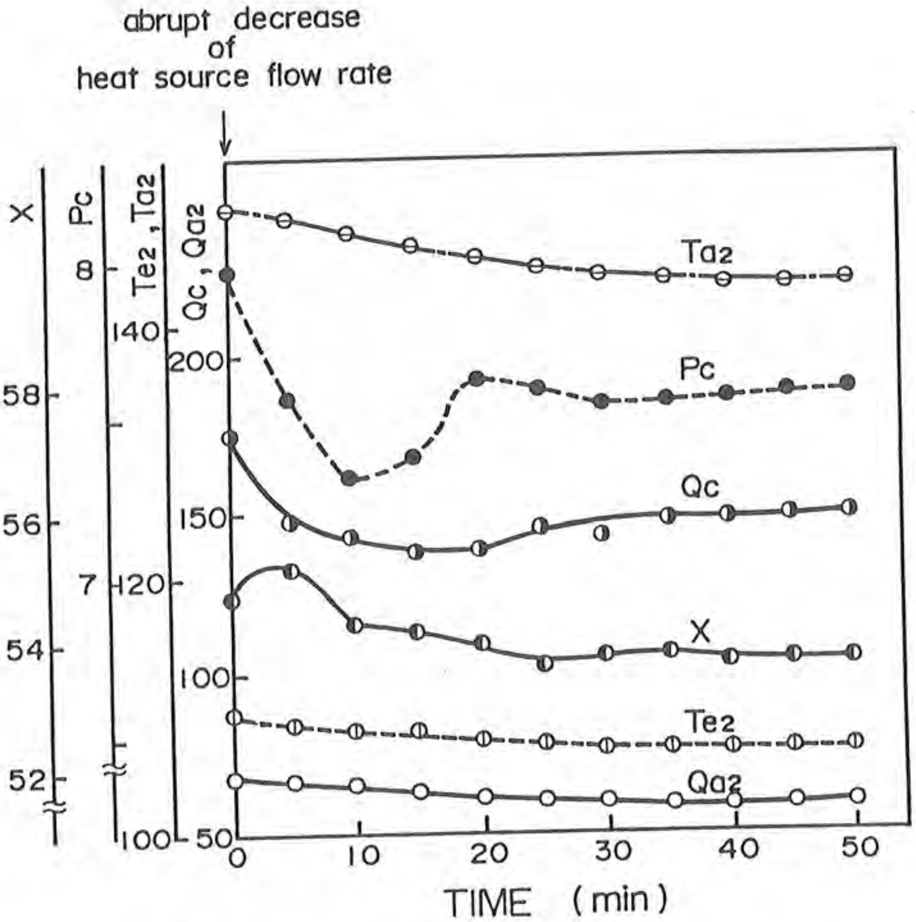


Figure 11. Control Flow Chart at Starting



T_{a2} : output steam temp. (℃)
P_c : condensing pressure (kPa)
Q_c : heat flow rate at condenser (kW)
X : intemediate sol. concentration (WT%)
T_{e2} : temp. of 2nd-stage evaporator (℃)
Q_{a2} : heat output (kW)

Figure 12. Transient Characteristics

Table 1. Main Specifications of No.1 MHT Heat Exchangers

		Shell	Tube	
		Dia × Thickness × Length mm	Type	Dia × Length × Number mm
Absorber	First Stage	355.6 × 6.4 × 2100	Inner Fin Tube with Helical Segmented Fin	12.00 × 1760 × 69
	Second Stage	406.4 × 6.4 × 2060	The Same Type as Above	12.00 × 1800 × 89
Generator		612.0 × 6.0 × 2340	The Same Type as Above	12.00 × 1540 × 150
Evaporator	First Stage	508.0 × 6.4 × 1230	Tube with Treated Surface	19.05 × 12730 × 1 19.05 × 14600 × 1
	Second Stage	355.6 × 6.4 × 2100	Inner Fin Tube	12.00 × 1760 × 69
Condenser		812.0 × 6.0 × 1310	Double Fluted Tube	19.05 × 1110 × 90
First and Second Recuperator		216.3 × 6.4 × 1440	Low Finned Tube	19.05 × 1300 × 76

Table 2. Temperature Boost Results of No.1 MHT

Inlet Heat Source Temp. ℃	Second Stage Evap.	88.3
	Generator	87.5
Inlet Cooling Water Temp. ℃	Condenser	23.2
Outlet Process Steam Temp. ℃	Second Stage Abs.	150.8
Boost Temp. ℃		62.0
Outlet Solution Temp. ℃	Second Stage Abs.	150.2
	First Stage Abs.	103.2
	Generator	69.2
Pressure mmHg	Second Stage Abs.	1119
	First Stage Abs.	434
	Condenser/Generator	37.5
Concentration of Outlet Solution wt%	Generator	56.4

Abs.: Absorber, Evap.:Evaporator

Table 3. Main Specifications of No.2 MHT Heat Exchangers

		Shell	Tube	
		Dia × Thickness × Length mm	Type	Dia × Length × Number mm
Absorber	First Stage	450 × 6 × 3400	Inner Fin Tube with Fir Fin	19 × 2000 × 69
	Second Stage	400 × 6 × 3200	The same Type as Above	19 × 2000 × 55
Generator		600 × 9 × 3600	Bare Tube with Fir Fin	19 × 2400 × 92
Evaporator	First Stage	450 × 6 × 3150	The Same Type as Above	19 × 2000 × 60
	Second Stage	450 × 6 × 3400	Inner Fin Tube	19 × 2000 × 69
Condenser		450 × 6 × 2750	Double Fluted Tube	19 × 2000 × 104

* a length of tube shows an effective heat transfer length
* materials : shell; carbon steel
tube and fin; copper

Table 4. Temperature Boost Results of No.2 MHT

Inlet Heat Source Temp.	91.0 ℃
Inlet Cooling Water Temp.	32.5 ℃
Outlet Temp.	151.2 ℃
Boost Temp.	60.2 ℃
Output Steam Pressure	0.492 MPa
Heat Output	69.4 kW
COP(=Heat Output/Heat Input)	0.31
Sol. Concentration Diff.	5.0 wt%
Recuperator Temp. Effect.	0.86

15. SIMULATION AND PRACTICAL APPLICATION OF HEAT TRANSFORMER

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1. Introduction

Recently "global environmental issues" have become one of the worldwide key-words. In order to prevent green house effect, it is propsed that mitigation of cargbon dioxide (CO_2) gas must be made with an improved efficiency of energy utilization.[1]

In Japan, technological innovations concerning energy conservation have brought almost completion of waste heat recovery at high temperature level. However, unused waste heat at medium and low temperature levels still remains. From the reuse point of view. It is important to convert waste heat into high temperature level energy using heat pump technology.

As for the environmental conservation, destruction of the ozone layer due to the discharge of CFCs is also a serious problem. Since absorption heat pumps are expected to contribute a great deal to the solution of these global environmental issues, they are increasingly being expected.

Of the two types of absorption heat pump currently in commercial use, a heat transformer which is also calld "type 2" was announced in 1918 by Altenkirch concerning its concept[2]. In Japan, it was developed under the "Moonlight Project" that started from 1976, and from the 1980's it has been commercialized as a waste heat recovery unit for industrial processes. So far, about 20 units have been manufactured and installed.

Here, through a heat transformer simulation system developed by the authors, the characteristics and installation examples are introduced.

Subscript

- P : Saturated vapor pressure (Pa)
- T : Temperature (°K)
- t : Temperature (°C)
- X_1: Concentration of diluted absorbent (mass%)
- X_2: Concentration of concentrated absorbent (mass%)
- C_P: Specific heat capacity
- S_X: Integral heat of solution
- Q : Exchange heat (kW)
- a : Circulation ratio of diluted absorbent $X_2/(X_2-X_1)$
- D : Flow rate of refrigerant (kg/h)
- G : Flow rate of external fluids (kg/h)
- K : Overall heat transfer coefficient (kW/m²°C)
- F : Heat transfer area (m²)

h : Enthalpy of external fluids (kJ/kg)
i : Enthalpy of refrigerant and absorbent (kJ/kg)
j : Enthalpy of refrigerant vapor (kJ/kg)

Suffix
a : Absorber
c : Condenser
e : Evaporator
g : Generator
h : Solution heat exchanger
i : Inlet
o : Outlet
p : Pump

2. Concept of Heat Transformer

A heat transformer is based on "refrigerant-absorption phenomenon of absorbent" caused by a saturated vapor pressure difference between refrigerant and absorbent, as its working principle. In almost all cases, water is used as refrigerant and lithium bromide water solution is used as absorbent. The heat transformer is specially characterized so that it does not require high quality energy, such as fuel and steam, as its driving heat source with recovering medium/low temperature (80 to 100°C) waste heat as the driving heat input by using cooling water as the heat sink. It can deliver high temperature energy about 40°C higher than the driving heat source, in the form of hot water or low pressure steam. Thus, due to its capability to produce reusable energy from the waste heat and cooling water, it will become increasingly gain in important.

Chemical plants are perhaps the best suited for introducing the heat transformer for recovery of the waste heat. Where heat is recovered from the tower top vapor of a distillation tower, and is used to heat the reboiler [3)~8)].

3. Simulation Model

External conditions of a heat transformer comprise waste heat conditions (type, temperature and flow rate of fluids), available heat conditions (temperature and flow rate of hot water, steam, etc.), cooling water conditions (temperature and flow rate), their calorific ratio and etc. These are complicatedly related to many parameter that determine the internal cycle of the transformer, and finally affect the economic effect of its introduction, i.e. payback period, conspicuously. Therefore, it is important to determine the main specifications most suitable for the user's service conditions. Additionally, since the transformer is often used in an industrial process, it is indispensable to conduct characteristics prediction when external conditions change.

These important values are required by the user as a feasibility study to be made when introducing the heat transformer, so that they must be presented quickly and accurately. For this purpose, simulation by a mathematical model of heat transformer has proved very effective.

3.1 Calculation Model and Physical Property Values

Fig. 1 shows the cycle and calculation model of a heat transformer. The heat transformer consists of four main heat exchangers (evaporator, absorber, regenerator and condenser), a solution heat exchanger and pumps.

Approximate equations for physical property values of working fluids[7] (1) to (5) are shown below. The saturated vapor pressure of absorbent was derived from the data of I.C.T.[8] By these equations, it is possible to calculate the saturated vapor pressure of refrigerant alone. The equation (2) for enthalpy of absorbent consists of specific heat and integral heat of solution. For the former, an approximate equation was prepared by using the equation of Hasaba et al[9], and for the latter, an approximate equation was also prepared by using the measured values of Hasaba et al[10]. Enthalpy of refrigerant is calculated by the equation (4), assuming that the value at 0°C is 100 kcal/kg. Enthalpy of refrigerant vapor is calculated by the equation (5) prepared by treating the equation of Tanishita[11] as with the equation (4).

$$\log P = (A + B/T + C/T^2)*133.22 \qquad (1)$$

$$A = 8.0509*10^0 - 6.3601*10^{-2}X + 1.1915*10^{-3}X^2$$

$$B = -1.6851*10^3 + 4.7677*10^1X - 9.9522*10^{-1}X^2$$

$$C = -9.0991*10^4 - 7.3963*10^3X + 1.4776*10^2X^2$$

where $0 \leq t \leq 100,\ X = 0,\ 44.44 \leq X \leq 66.67$

$$i = (100 + \int C_P dt - S_X)*4.184 \qquad (2)$$

$$C_P = A + Bt + Ct^2$$

$$A = 1.098*10^0 - 1.529*10^{-2}X + 6.22*10^{-5}X^2$$

$$B = -3.651*10^{-3} + 4.204*10^{-5}X$$

$$C = 3.576*10^{-5} - 4.239*10^{-7}X$$

$$S_X = -20.6 + 3.025x - 2.988*10^{-2}X^2 \quad (X \leq 30) \qquad (3)$$

$$i = (199 + t)*4.184 \qquad (4)$$

$$j = (h_5 - (h_1 + h_2 + h_3 + h_4)*4.184 \tag{5}$$

$$h_1 = 5.78843*10_{-3}P/(T/100)^{2.7}$$

$$h_2 = \frac{9.62037P + 6.9456*10^{-5}P^2 - 6.64106*10^{-32}P^6}{(T/100)^{2.4}}$$

$$h_3 = 1.96726*10^{-12}P^8/(T/100)^{30.5}$$

$$h_4 = 4.346*10^{-45}P^{26}/(T/100)^{147}$$

$$h_5 = 100 + 481.78 + 34.618*T/100$$

$$+ 0.916*(T/100)^2 + 14.07*\ln(T/100)$$

3.2 Calculation Flow

Shown below are equations related to the quantity of heat exchanged for an absorber as representing the four main heat exchangers.

For the heat balance of working fluids:

$$Qa = D\{(a - 1)i_8 + j_1 - aj_2\} \tag{6}$$

For the heat balance of hot water:

$$Qa = Ga(hao - hai) \tag{7}$$

For the heat transfer between the absorbent and hot water:

$$Qa = KaFa\frac{(t_6 - tao) - (t_2 - tai)}{\ln\{(t_6 - tao)/(t_2 - tai)\}} \tag{8}$$

Where the absorbent shall be at the saturation temperature, to be obtained from the equation (1) in the form of

$$t = f(P,X) \tag{9}$$

Since the pump power is small compared to power required for other components, it can be neglected, so that the heat balance and COP are expressed by the following equations.

$$Qe + Qg = Qa + Qc \tag{10}$$

$$COP = Qa/(Qe + Qg) \tag{11}$$

The calculation is carried out by assuming the concentration of absorbent, and repeated by modifying the concentration until the quantity of heat exchanged in the working fluids in each heat exchanger coincides satisfactorily with the quantity of heat exchanged in the heat transfer side.

4. Simulation Result

The simulation system consists of program groups as shown in Fig. 2, which is related to actual business operations.

Now, assuming that we received an order for design specifications as shown in Table 1, let's describe how the simulation system is used concretely.

4.1 Determination of Design Points

In the first place, based on design conditions to be achieved, the quantity of absorbent circulation (differential of concentration) and coefficient of performance (COP), which are optimum for deriving the capacity that satisfies the external conditions, are determined.

Fig. 3 shows the calculation result of relations between the solution concentration difference and material cost of main components under the constant COP condition. From Fig. 3, it is known that the cost becomes lowest when the solution of concentration difference is about 4% (CAMOL).

Also, by checking the internal cycle when the heat transfer area and etc. of main heat exchangers have been changed, the design point is determined. The results can be indicated on Dühring diagram as shown in Fig. 4.

4.2 Characteristics Prediction

These two programs are intended for selecting the heat transformer model and determining the design point. As such, they cannot predict possible states of operation when external conditions have changed. Additionally, since the heat transformer's operational conditions are not limited to those at the design point, characteristics prediction has been made by "ANALYSIS" according to changes in external conditions. Fig. 5 shows the characteristics predicted for changes of hot water outlet temperature. It is seen that, although the COP does not change much, the heat output from the absorber is affected by the hot watcr outlet temperature.

4.3 Economy

As shown in Fig. 6 of typical temperature rise characteristics, a heat transformer has a very interesting characteristic that the lower the cooling water temperature, the higher the temperature of hot water that can be obtained. The heat transformer's economy is most affected by the cooling water temperature.

Although the waste heat temperature and hot water inlet temperature can be controlled in many cases, since river or sea water is directly used as the cooling water, its temperature is often affected by air temperature even when a cooling tower is used. Thus it subjected to relatively large yearly temperature fluctuations naturally. At "SALES AID", by inputting each month's average cooling water temperature, annual gained steam can be obtained as shown in Fig. 7. The lower bar graph in Fig. 7 shows an average output per hour for each month. In winter or spring when the cooling water temperature is low, an output larger than in summer, is obtained. In the middle of Fig. 7, integration of monthly gained steam calculated by taking account of monthly operation time, is shown. Above this graph are shown cooling water inlet/outlet temperatures. If we know the unit steam cost at the place where the heat transformer is installed, the yearly value of saved steam can be calculated, and presented to the user as data on payback period. Usually, as requirements for design specifications, severe conditions in summer, where the cooling water temperature is high output reduces, are selected. So, based on the output value in that state, economic calculation is made to decide whether it be introduced or not. In view of the year-round output characteristics shown in Fig. 7, it is thought that the heat transformer can have a larger investment effect than that derived from simple economic calculation.

4.4 Dimensional Drawing

The dimensional drawing of a heat transformer which is designed based on the simulation results according to the design specifications given in Table 1.

5. Examples of Practical Application

As described above, the heat transformer simulation system is used as a business operations supporting tool from the feasibility study stage. After an agreement is performed, the detailed design, manufacture and inspection stages follow. Finally, it is installed on the spot, and a system shown in Fig. 9[12] (flow sheet) is completed and put into commissioning.

The system shown in Fig. 9 has been designed for recovery of heat from alcohol vapor discharged from an ethyl alcohol distilling plant. So far, at distilling plants, almost all the latent heat of tower top vapor has been through into the cooling water, and its use has been limited to heating water fed into the boiler. If the latent heat of the tower top vapor is recovered and reused to heat the distilling tower, effective energy utilization and saving can be achieved. This system, which consists of a distillation tower that produces alcohol vapor, a heat transformer to obtain hot water, and a flash tank for changing the hot water obtained into steam, has been in operation since 1983. The tower top vapor discharged from the distilling tower is condensed in the heat transfer tubes of the evaporator and regenerator, and used as a heat transformer driving source. Almost all of alcohol which is condensed and liquefied is returned to the distilling tower. Meanwhile, in the condenser through which the cooling water is passed, refrigerant vapor brought by a differential pressure from the regenerator is cooled and condensed, and turned into refrigerant liquid. This refrigerant, fed into the evaporator, evaporates, and is absorbed by absorbent that is flowing down outside the absorber tubes. By the heat of absorption generating at this time, hot water being pressure-fed from the hot water pump is heated. Since the hot water returned to the flash tank is relieved from the pump pressure, part of it flashes into steam. This steam is led into the reboiler, and utilized to heat the alcohol. By this series of cycles, about 1/2 the quantity of heat of the tower top vapor is recovered, and recycled to heat the reboiler.

Thus, as a result of introduction of this heat transformer, a great deal of energy saving effect has been achieved. Coupled with the small power consumption, no modification has been required for the conventional plant, with relatively simple connection/piping installations. Thus, the convenient installation after introduction has been evaluated.

6. Conclusion

From the specifications for installed heat transformers, data on waste heat temperatures, served hot water temperatures and output can be obtained as shown in Fig. 10[6] and Fig. 11[6]. In many cases, waste heat temperatures are 90 to 100°C, and served hot water temperatures are 120 to 140°C. While the output per one unit is 1,000 to 2,000 kW for about half the units supplied. From these results it may be said that heat transformers are effective tools for recovering medium/low temperature waste heat.

Temperature levels of waste heat from various industrial processes are distributed between 60 and 90 °C, and steam higher than 110°C is often required from the user side. Thus, it is necessary to develop heat transformers with which a larger temperature boosting can be achieved. In West Germany[13], Japan[14~15] and the U.S.[16], those using TFE(trifluoroethanol)/DTG(dimethylethertetraethylenglycol) system or TFE/Pyr(pyrrolidone) system as a working fluids pair have been proposed and are under progressive development. Through the pursuit of new-type working fluids and absorption cycles, absorption heat pumps are achieving higher efficiency, higher temperature boosting, and more compactness. They are also expected to contribute to the environmental preservation on the global scale.

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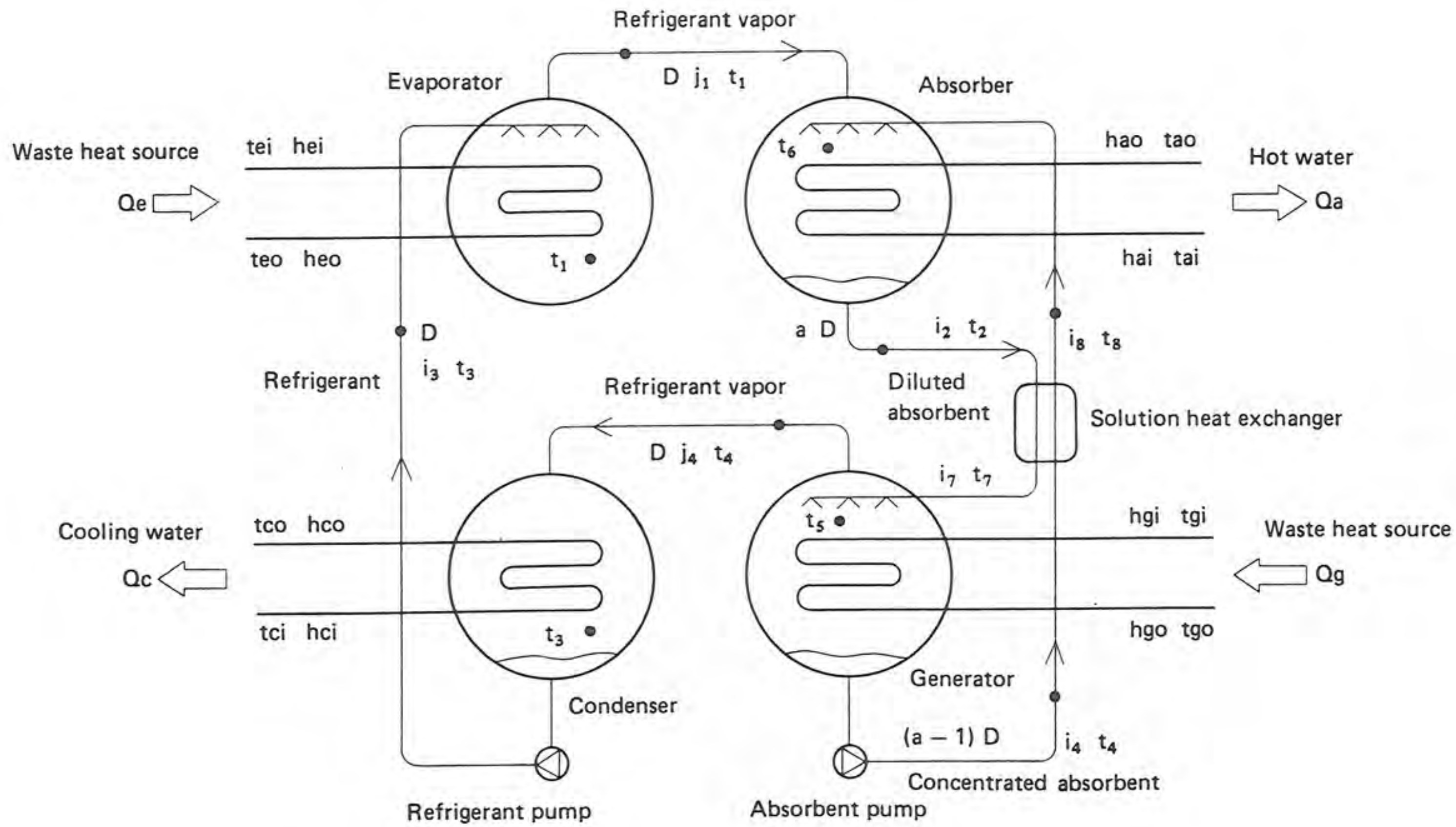
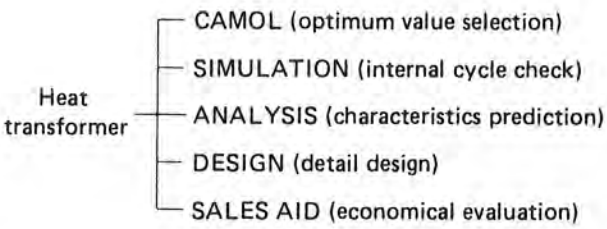


Fig. 1 Flow Diagram and Simulation Model of Heat Transformer



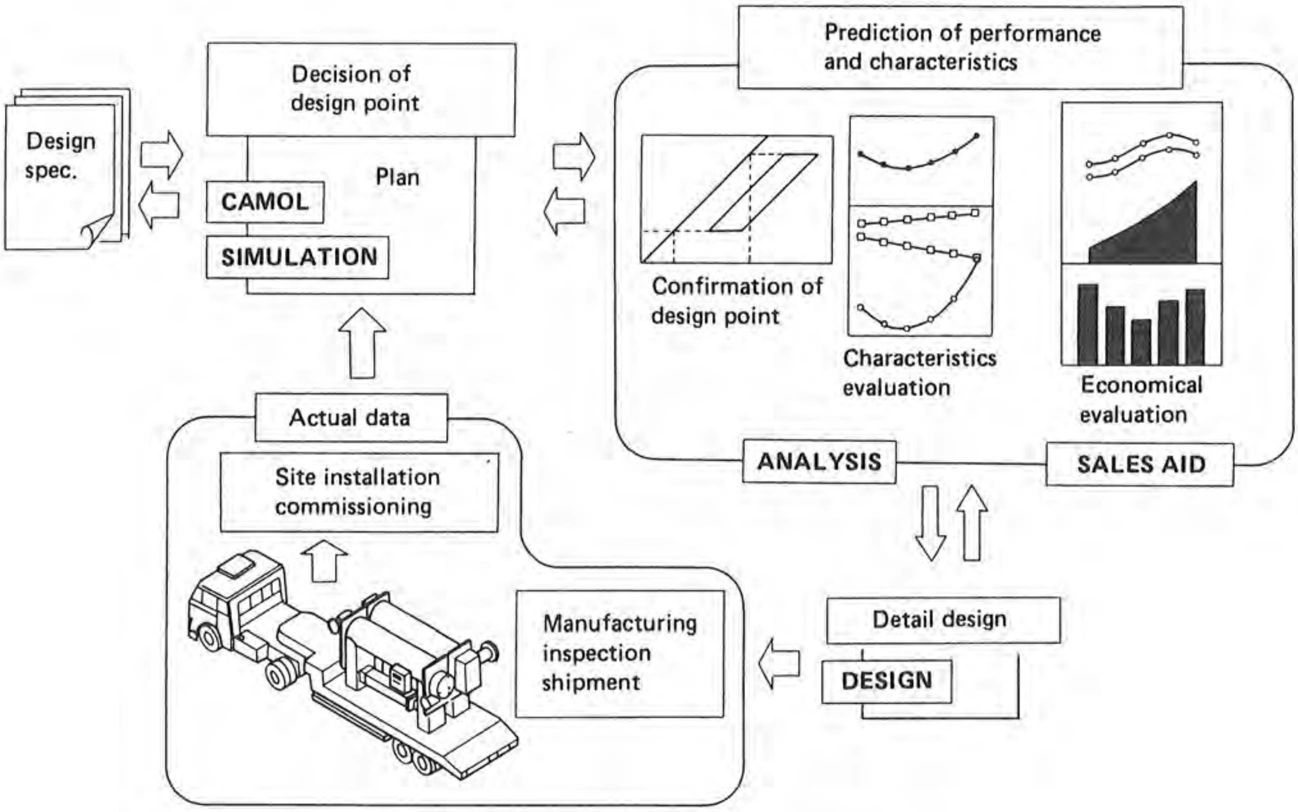


Fig. 2 Construction of Simulation System

Table 1 Example of Design Specification from Customer

Waste Heat Source	Organic Vapor	Condensing Temp. 90°C, 4.8 MW
Cooling water	Cooling tower	Design Temp. 30°C
Available heat	Steam	248 kPa, 3.5 ton/h

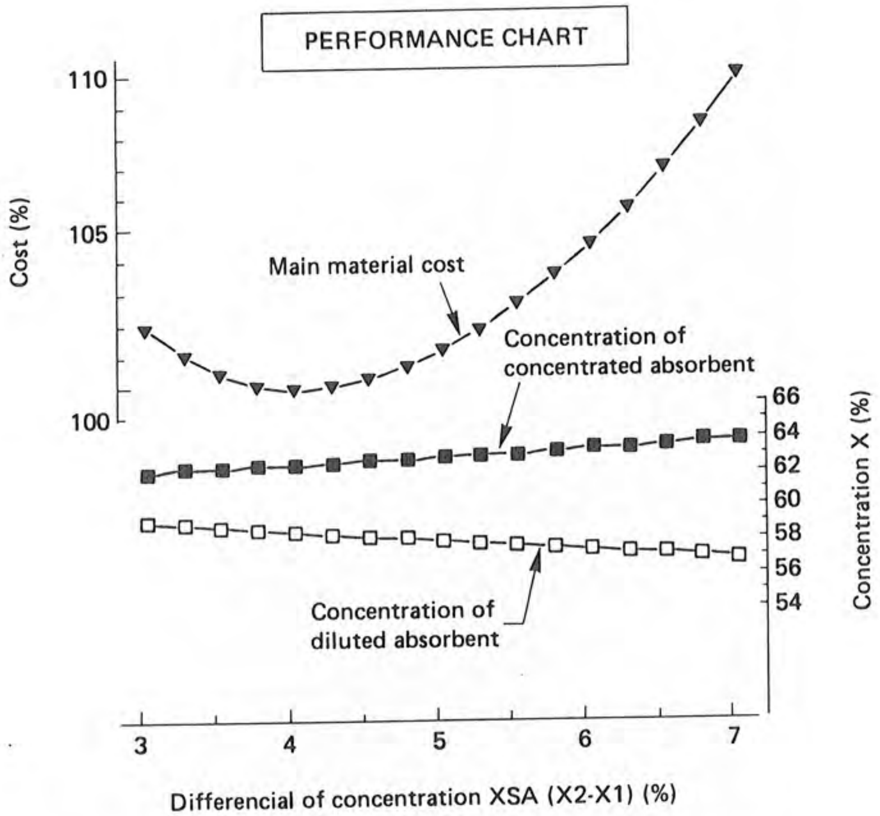


Fig. 3 Main Material Cost vs. Quantity of Absorbent Circulation

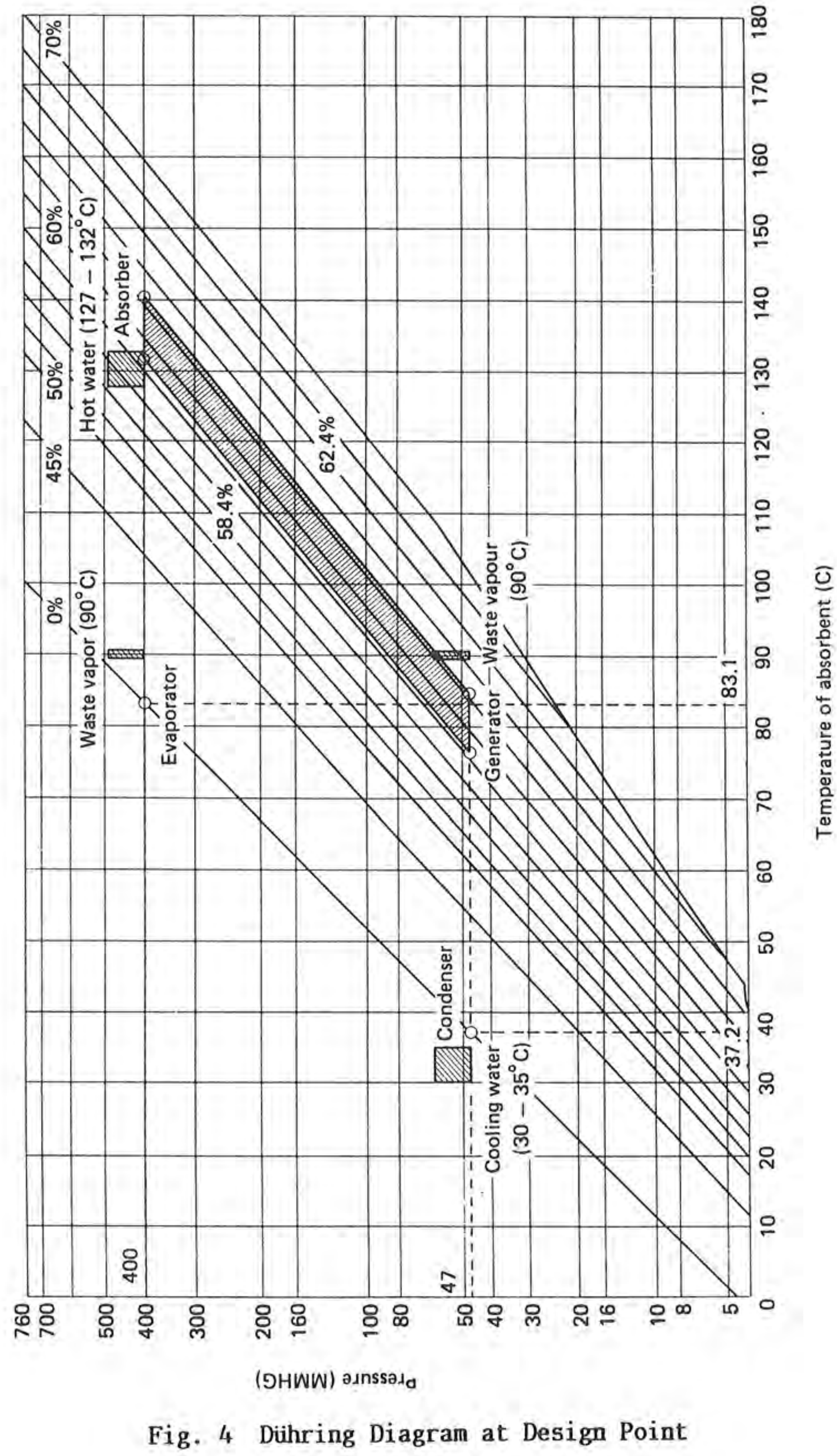


Fig. 4 Dühring Diagram at Design Point

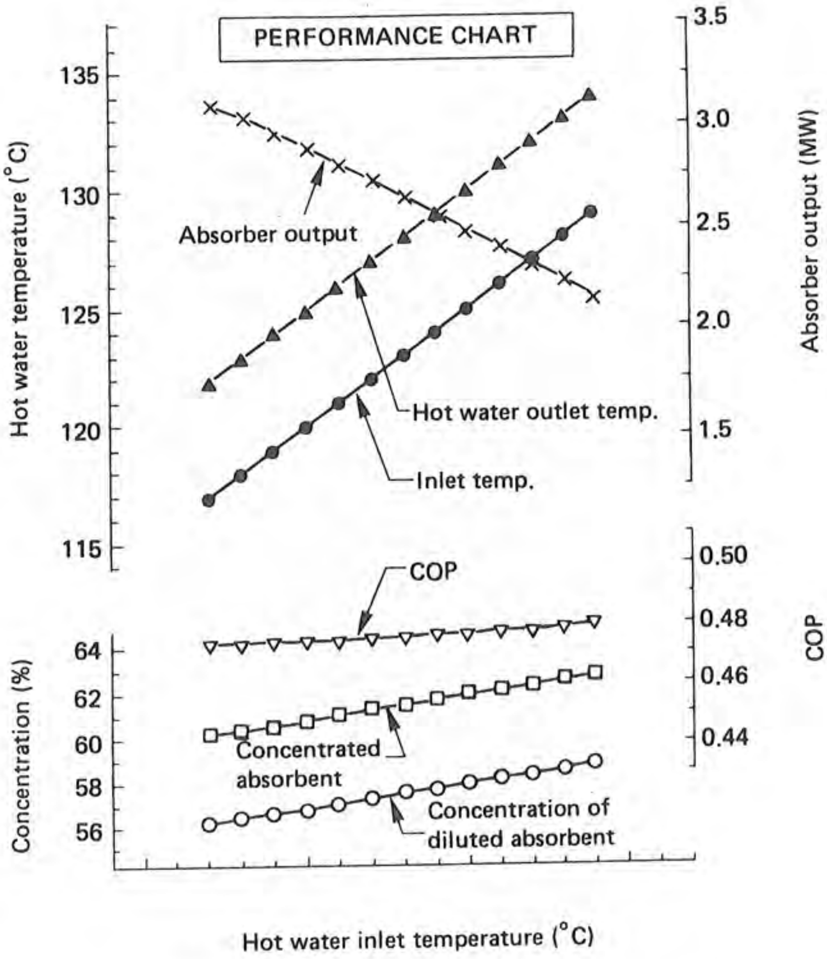


Fig. 5 Characteristics Prediction
– the change of hot water outlet temperature –

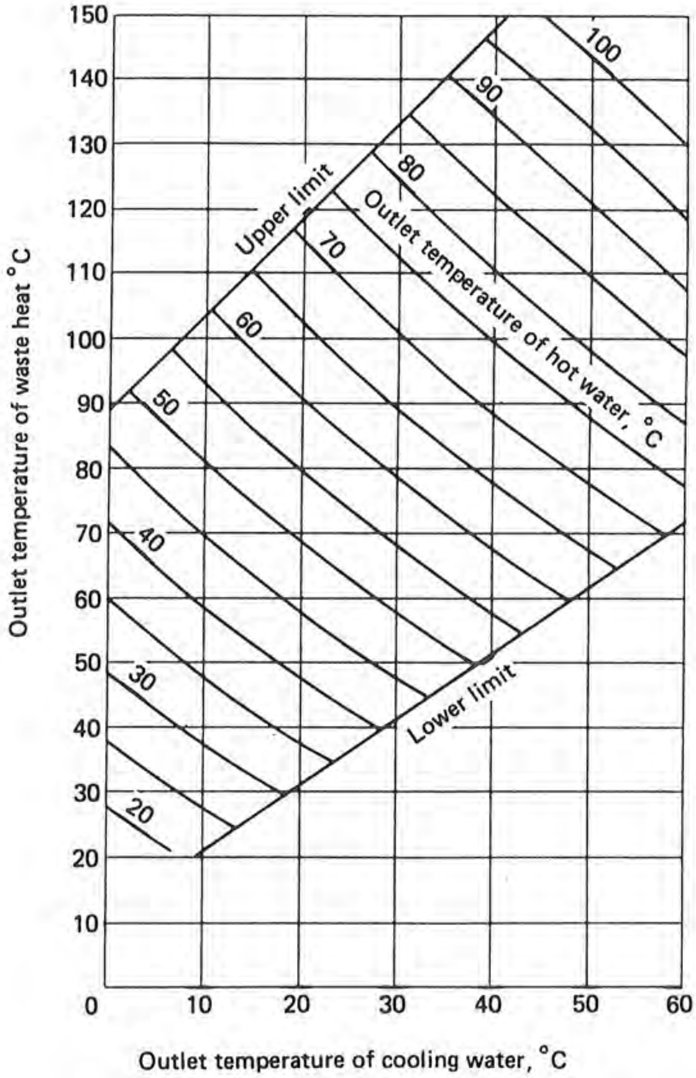
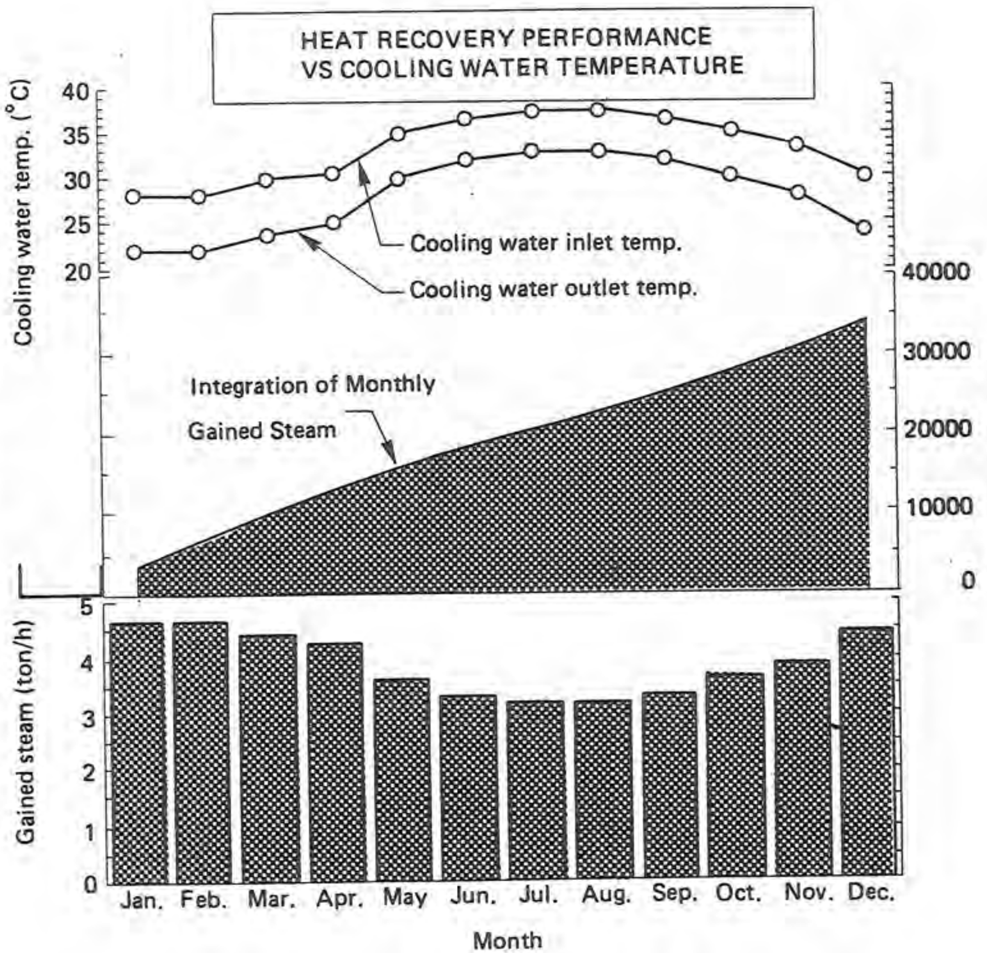


Fig. 6 Typical Temperature Rise Characteristics of Heat Transformer



HEAT RECOVERY COST

MON	TCI	GS	GS	TIME
	C	T/H	T/M	H/M
Jan.	22	4.66	3467.8	744
Feb.	22	4.66	3132.2	672
Mar.	24	4.39	3266.9	744
Apr.	25	4.26	3066.5	720
May	30	3.59	2670.2	744
Jun.	32	3.31	2385.4	720
Jul.	33	3.18	2367.4	744
Aug.	33	3.18	2367.4	744
Sep.	32	3.32	2389.0	720
Oct.	30	3.59	2670.2	744
Nov.	28	3.85	2774.9	720
Dec.	24	4.39	3264.7	744

Total stm = 33822 (T/Y)

Steam cost = _____yen/ton
Total recov. = _____yen/year

Fig. 7 Prediction of Annual Gained Steam

Main Specifications			
Hot Water	Output	Qa	2.3 MW
	Outlet Temp.	tao	132 °C
	Inlet Temp.	tai	127 °C
	Flow Rate	Ga	400 tons/h
Waste Vapor	Input	Qg+Qe	4.8 MW
	Cond. Temp.		90 °C
Cooling Water	Exchange heat		2.5 MW
	Outlet Temp.	tco	35 °C
	Inlet Temp.	tci	30 °C
	Flow Rate	Gc	434 tons/h
Total Weight*			60 tons

* : Contained water is not included.

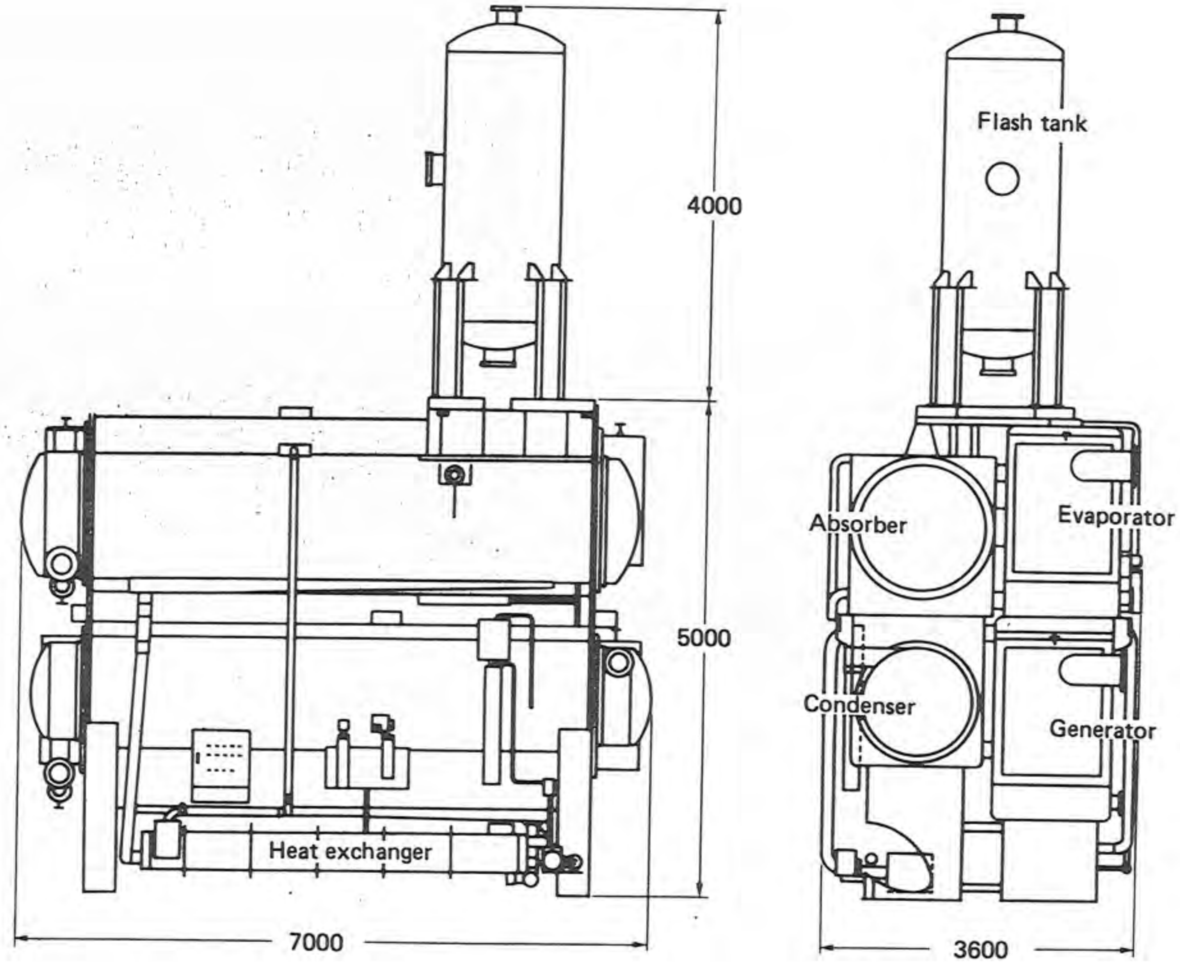


Fig. 8 Dimensional Drawing of Heat Transformer Designed

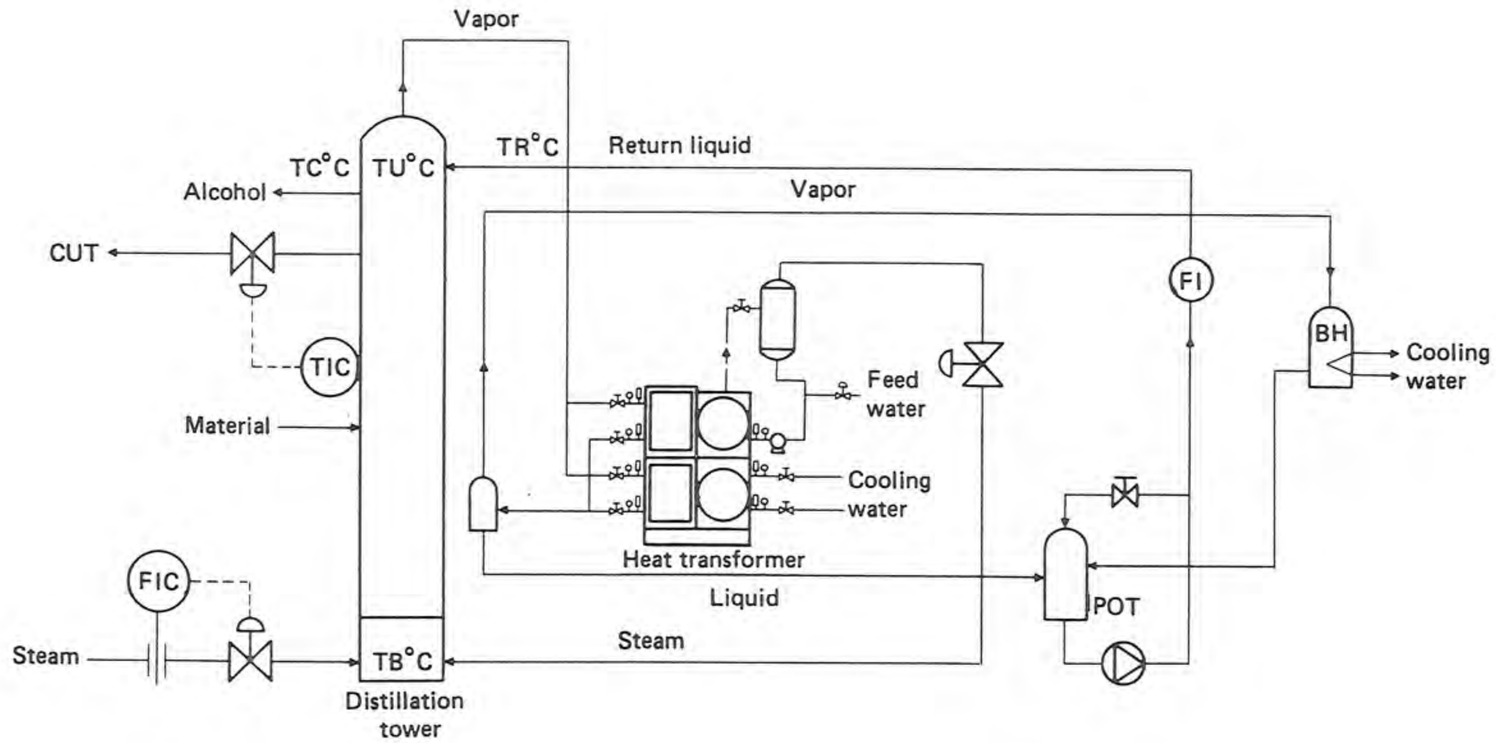


Fig. 9 Practical Application of Heat Transformer

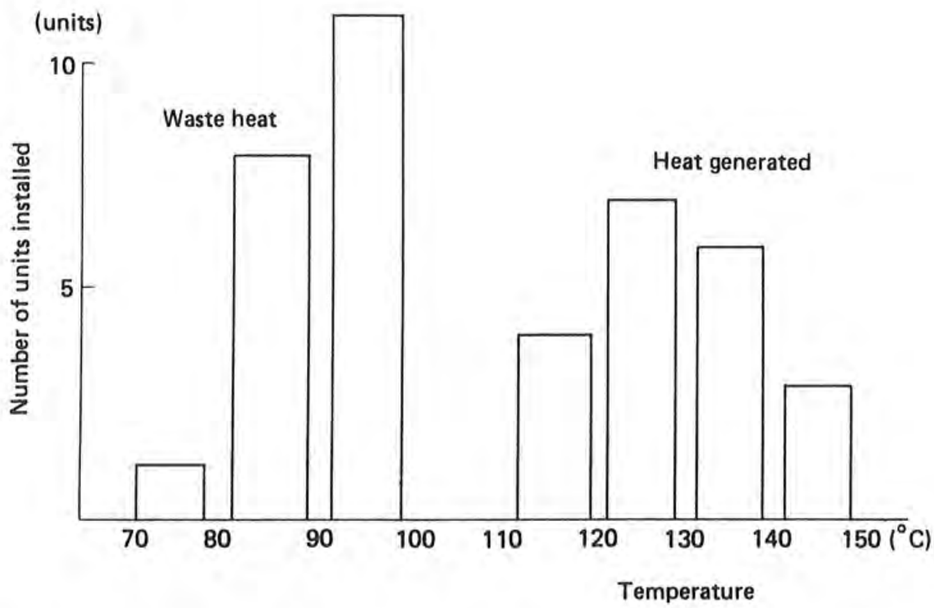


Fig. 10 Temperature Distribution of Waste Heat and Output

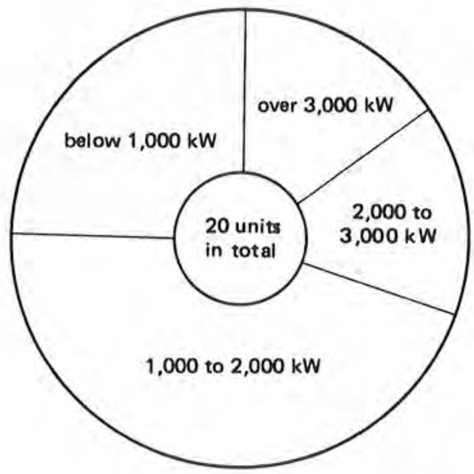


Fig. 11 Distribution of Singular Machine's Output Capacity

The National Team of Sweden

16. HEAT TRANSFER AREAS AND HEAT ADDITIVES IN FALLING FILM ABSORBERS

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SUMMARY

An experimental plant for determination of heat and mass transfer rates in falling films is constructed at the Department of Chemical Engineering at KTH. Two alternative techniques to improve the transfer rates in a falling film will be investigated. One is to induce the formation of large mixing waves by profiling the tubes. Another is to use "mass-additives".

I. HEAT TRANSFER AREAS AND HEAT ADDITIVES IN FALLING FILM ASBSORBERS

I.I. Introduction

Absorbers and generators in absorption heat pumps are preferably constructed using the falling film technique. Film instabilities have a large influence on the heat and mass transfer rates (1). The instabilities at the gas-liquid interface, for example caused by the Marangoni effect may contribute to the enhancement of the transfer rate.

The Department of Chemical Engineering at KTH has recently completed the construction of an experimental plant, consisting of an evaporator and an absorber. The height of the absorber is 4 m. The absorber tubes are interchangeable and several design may be tested. Two alternative techniques to improve the transfer rates in a falling film will be investigated. One is to induce the formation of large mixing waves by profiling the tubes. Another is to use so-called " mass transfer additives" which influence the transfer processes at the gas-liquid. As absorbents in the pilot-plant, Lithium-Bromide /water and Sodiumhydroxide/water are chosen. As model additive we use 2-heptanol.

I.I. Falling film tubes

Large mixing waves may be induced by profiling the heat transfer area. Horizontal, finned tubes have been thoroughly studied (9), but much less work of a systematic nature has been devoted to vertical finned tubes.

By attaching wires to the tube surface, Thomas (1) and Schröder et al (6) have achieved longitudinal fins. The liquid flows parallelly to the fins and is drawn into the cavity between the wire and the tube surface by capillary forces.

Thomas used wires with diameters 0.7 - 1.5 mm in his experiments. The wires were fastened with single loop of 0.1 mm wire to the surface every 30 cm. Using eight wires on a 12.5 mm tube, which corrensponds to a ridge to space ratio between 2.2 and 6, the heat transfer coefficients were increased 4.5 times for a heat flux of 20 kW/m and doubled for a heat flux of 10 kW/m. In another series of experiments four wires were wound in a spiral around the tube with a 10 cm pitch for each wire. Independently of heat flux magnitude the heat transfer coefficient was roughly doubled.

Scröder et al found that the optimum spacing is 8.6 mm for plates in sea water evaporators and the fins should be at least 3 mm.

Davies and Warner (2) have used horizontal ridges on vertical plates in CO_2 absorption system. The result is summarized in figure 1 which shows the optimum space to height ratio for the ridges.

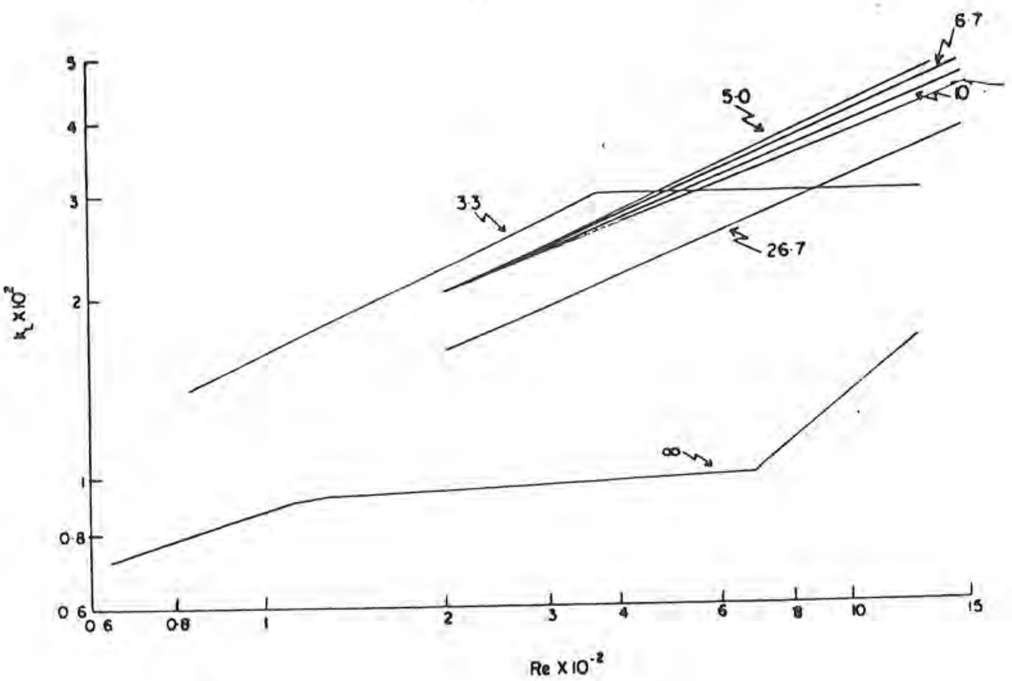

Figure 1: Mass transfer coefficients versus Reynolds numbers for CO_2 in water.

In our application Reynolds number is ca 200. This corrensponds to the ratio 3.3 in the figure. In the application a ridge height of 1.8 mm is used and therefore the ridge space is 5.9 mm.
In the present investigation we intend to use horizontal profiles on vertical tubes to increase waviness. In order to accelerate developement of large waves, vertical profiles will be used in the initial part of the tube. These will thin out the film, but also collect fluid near the ridges, creating thicker lumps of liquid that will develop more rapidly into the desired large waves. The optimal length of these initial profiles must be determined experimentally. The amplitude of natural waves may be calculated using the wave equations derived by Maron et al (7) and experimentally determined wave freqiencies (10). The large waves, ca one meter below the inlet of the flow, have a frequency of 8 Hz, an amplitude of 0.8 mm and a distance between the wave peaks of ca 10 cm. The corrensponding space to height ratio is 125. The

turbulent wave front is 13 mm yielding a geometric ratio of 11. Similar profile of a tube, 0.8 mm high ridges spaced 13 mm, might reduce the ineffective laminar area.

The tubes utilized by Hitachi Zosen in their absorption heat pumps have space to height ratio of 11 -16. The fins are between 0.5 and 0.9 mm spaced 8 - 10 mm apart. The heat transfer coefficient is increased 2.4 times relatively to smooth tubes. The increase reported by Thomas is similar.

The effect of mass additives

The enhancement of mass transfer during absorption of a vapour caused by surface-active additives has been amply demonstrated (3, 4, 8). Burnett and Himmelblau (3), Zawacki, Leipziger and Weil (4), and Kashiwagi el al. (8) reported experiments on absorption of vapour in a static pool of absorbent at temperatures close to ambient. The enhancement of absorption rate observed was attributed to interfacial turbulence, i.e. Marangoni convection. Experiments with an absorption heat pump show some improvement of absorption rates when an additive is introduced into the unit, but the results are less clear (4, 8). Kashiwagi et al. has shown that enhancement occurs only if an excess of alcohol is present at the surface (8). The n-alcohols were found more efficient than 2-alcohols or 3-alcohols. However, Zawacki et al. found the opposite effect (4). In the present investigation we hope to resolve this contradiction and intend to extend the work to higher temperatures. The effect of additives on absorption in a flowing system will be studied.

Presently, equilibrium properties are being collected for LiBr solution and a variety of additives. The surface tension of LiBr solution (50% wt) has been measured as a function of temperature by the ring method. The results agree with literature data. The surface tension has also been measured for the LiBr (50% wt) and 500 ppm, 1000 ppm n-octanol system, see Fig. 2. Results from Kashiwagi's report are shown in Fig. 3.

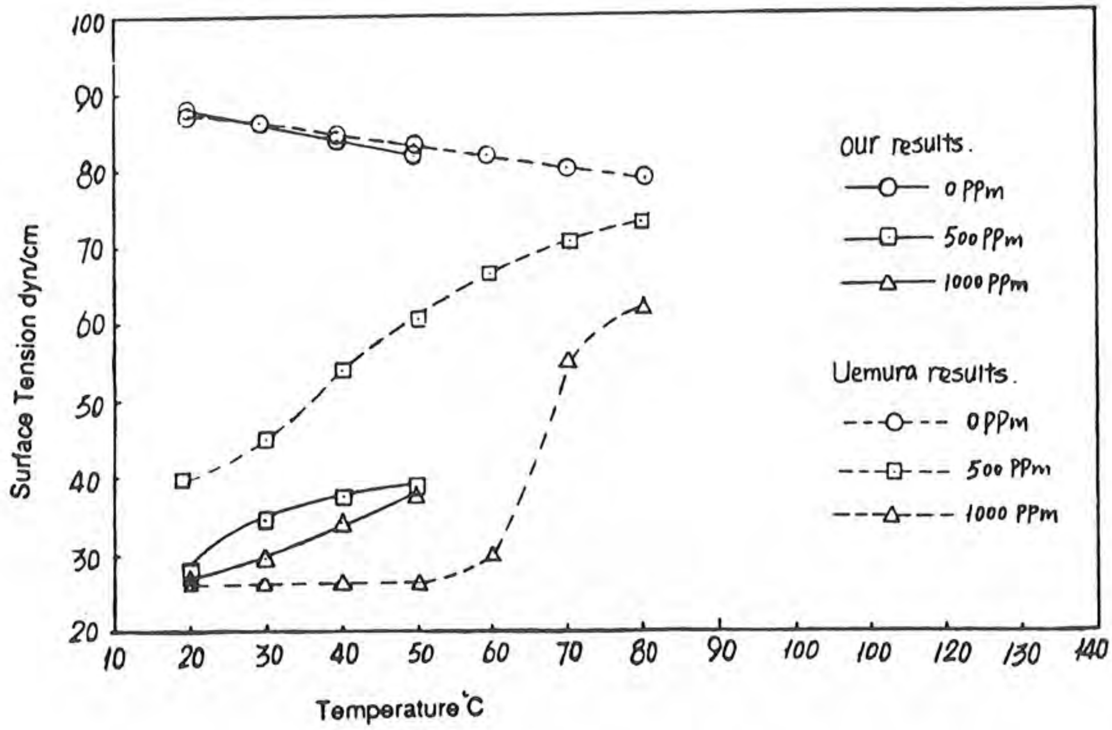

Figure 2: Surface tension of LiBr (50 % wt) adding n-octanol at different temperatures

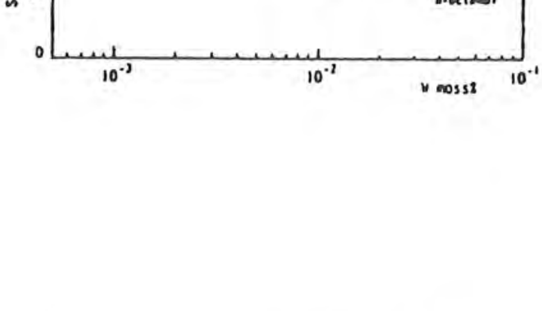

Figure 3: Surface tension of LiBr (50% wt) adding n-octanol at 20 ^{0}C

Our results at ambient temperature are in agreement with those of Kashiwagi (8), but in disagreement with the data reported by Iyoki and Uemura (5). Data could not be obtained above 50 ^{0}C with the ring method. Interfacial tensions between LiBr solution and n-octanol have also been measured, see Fig. 4.

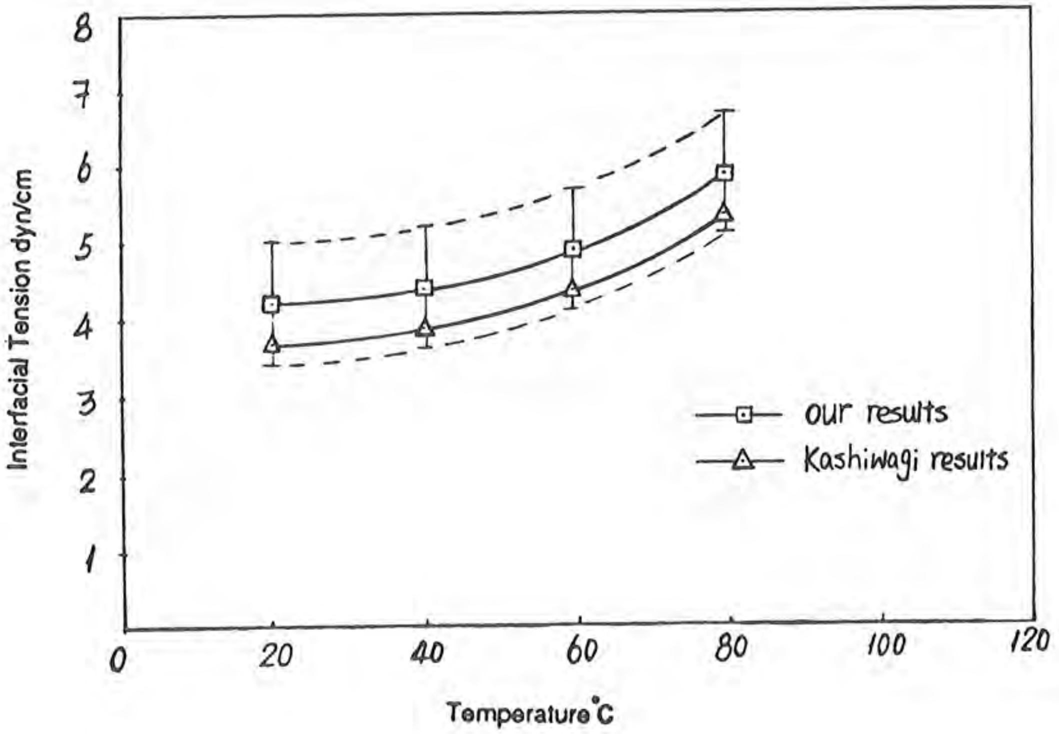

Figure 4: Interfacial tension between LiBr (50 %wt) and n-octanol

We are now constructing an experimental set-up for studying the absorption of water vapour in static LiBr solutions. In this apparatus, the influence of additive concentration and of temperature will be investigated for several proposed surface-active agents. Later, a laboratory absorber will be used to extend these studies to flowing system.

I.IV. Description of the experimental plant

A scheme of the pilot plant is shown in figure 5. The plant constitute of an absorber/generator and an evaporator/condenser. Cold water will circulate in the absorber and hot water will circulate in the evaporator when the plant is used for absorption purposes. When the absorbent will be distilled, hot water will circulate in the generator and cold water in the condenser.

The diameter of the absorber/generator is 500 mm and the height is 5 m, which allows tube lengths of 4 m. The material is SMO 254 which contains about 6 % molybdenum. The component is insulated with 100 mm mineral wool. The tubes to be tested are made of sanicro, titan or any other high-alloy steel. The evaporator/condenser has a diameter of 500 mm and a total height of 1500 mm. The material is reinforced glass fibre. Intalox saddles constitute the contact surface.

In the absorbent circuit two 500 l tanks of SMO 254 are installed, in which the chemical is collected.

The positions of the measuring devices are shown in figure 5. The temperatures are measured with PT100 sensors. The flows are measured with inductive flowmeters, type magnetic flowmeter flowtube, model 8711 from Rosemount. The pressure in the absorber is measured by a manometer.

The absorbent is weighted continuously by eight units mounted on each collecting tank, which makes it possible to determine the absorption rate and the regeneration rate. The type of the units are KIS 3-2 k, Nobel Electronics. The concentrations in the outlet absorbent streams are also measured continuously by two parallell connected densityr meters type DMS, Density Monitoring System with D25-D300 from Micro Motion.

The measurement test results are stored in a computer.

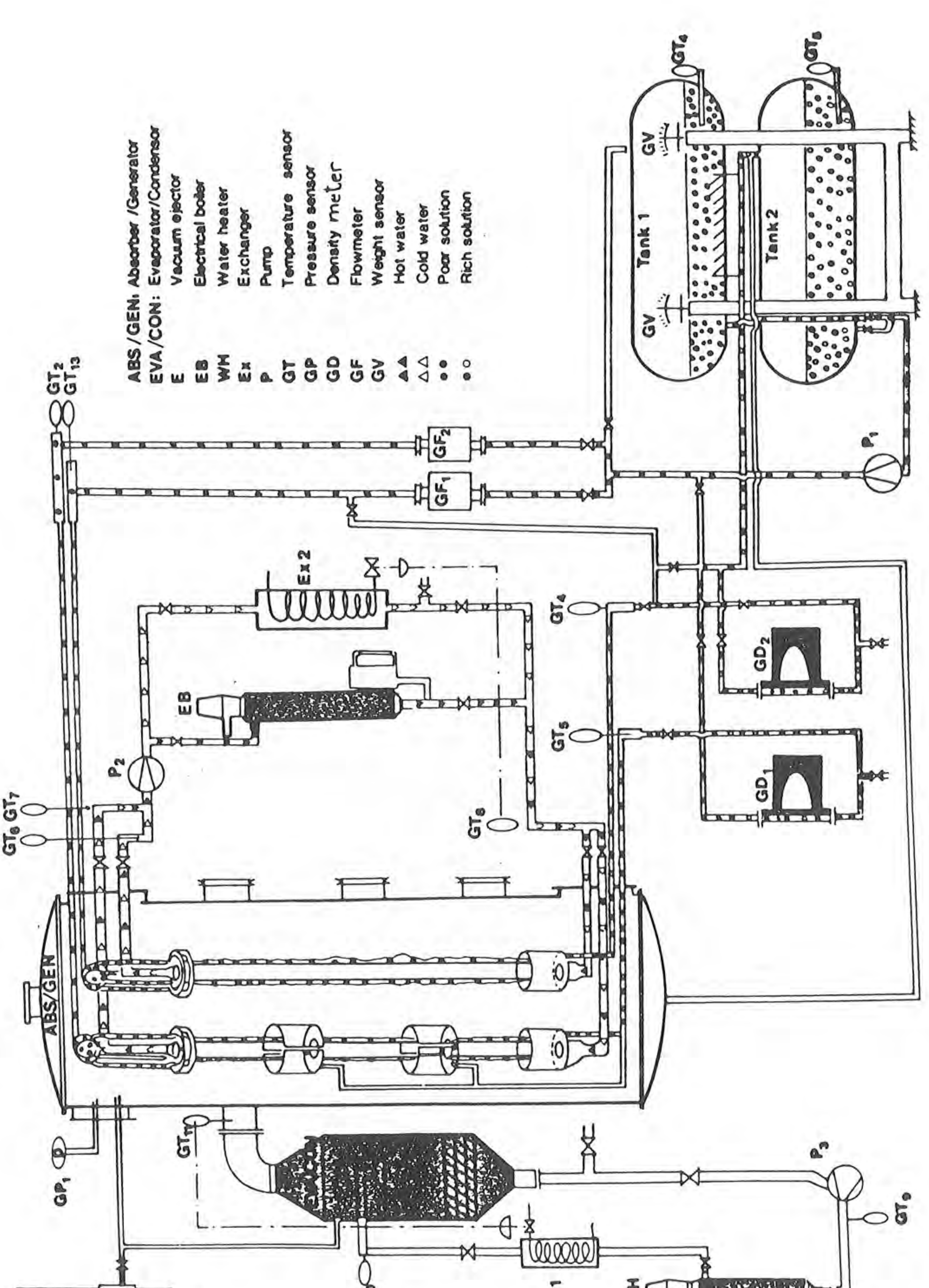

Figure 5: A scheme of the experimental plant

I.V Acknowledgement

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Studsvik Energy

17. RESEARCH ACTIVITIES AT STUDSVIK ENERGY IN THE FIELD OF ABSORPTION HEAT PUMPS

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SUMMARY

Past and present research activities at Studsvik are presented shortly. The program is financed by grants from the National Energy Administration and the Swedish Building Research Council.

1. STUDSVIK ENERGY

Studsvik AB was founded in 1947 as AB Atomenergi, a national laboratory for research and development in the field of nuclear power. The present name was taken in 1978 when development work on non-nuclear energies was started. Today Studsvik AB consists of four divisions: Nuclear, Energy, Alnor and MPC. Studsvik Nuclear carries on work servicing the nuclear power industry, Studsvik Energy develops non-nuclear technologies, Alnor and MPC manufacture and sell technical and scientific instruments.

2. PREVIOUS WORK

Activities in the domain of absorption heat pumps were initiated with an investigation of opportunities for thermochemical heat storage in industrial plants. In the report, it was concluded that there were possibilities for heat transformers rather than heat stores. In other nomenclatures heat transformers are called temperature boosters or heat pump type II. A prototypal 10 kW unit was therefore built and tested. Ammonia - water was used as working pair. Commercial shell-and-tube heat exchangers were used for all components except for the absorber, which was of the falling film type. The temperatures were 45°C for the heat source, 10°C for the condensor and 55°C for the absorber. Work was not pursued beyond this single unit, as Japanese heat transformers became available at that time.

In cooperation with a manufacturing company, a heat pump prototype was then built (type I). It was designed for 40 kW heat output, the falling film technique was used for a-l components and the working pair was water and LiBr. Falling energy prices interrupted work in 1985, when the design had been verified experimentally.

Studsvik was also engaged in work on hydrogen energy technology at the same time. As one of the most promising applications a metal hydride heat pump was built: 10 kW heat delivered at 120°C from a source at 80°C, the cold sink being at 10°C. It has been operated in the laboratory, with a performance a little lower than design.

3. PRESENT WORK

Work on heat pumps was restarted in 1988 with research grants from the Swedish National Energy Administration and the Swedish Council for Building Research. There are several topics in the program. Concrete results are not yet available, so only a general orientation will be given here.

a) high temperature lifts

The temperature lift or boost available from a working pair with a liquid absorbent is limited by the evaporation of the absorbent or by the crystallization of a salt. Solid absorbents allow much larger temperature increases but the heat transfer properties of powders are unsatisfactory. By suspending a solid absorbent in an inert liquid it is thought that heat transfer properties of liquids and absorption properties os solids could be achieved at the same time.

A search for candidate systems yielded the following results: metal hydride systems are well suited as working system, leaving ammoniates as a close second. Silicone oils seem to be appropriate solvents: low vapor pressure, stable and inert. The kinetics of the absorption and desorption processes are currently being investigated.

b) absorption compression

The use of a non-azeotropic system such as ammonia - water in a compression heat pump cycle offers some advantages. Absorption and desorption in a temperature range which can be made large, lower pressures in the system are some of those. Absorption-compression gives also the possibility to avoid the controversed CFC's. Studsvik has built a small experimental unit, rated at ca 2 kW heat. The heat source is cooled from 20°C to -5°C and the heat sink is heated from 40 to 60°C. These temperatures correspond to a heat pump using exhaust air from a ventilation system as heat source and a hydronic heating system as heat sink. The characteristics that are special to this design are the compression system and the very small amount of ammonia in the system.

c) heat additives

This is a cooperation between Studsvik Energy and the Department of Chemical Engineering at KTH. We intend to investigate the conditions under which the absorption of water vapour in a LiBr solution is accelerated. The surface tension of mixtures is determined using a drop volume tensiometer at KTH. The kinetics of absorption of water vapour in a static pool are studied in Studsvik using a set-up modelled after that used by Zawacki et al. at IGT. A laminar jet apparatus is currently being built at KTH.

d) non-absorption projects

Beside these projects concerned with absorption techniques, a few projects are also pursued on other heat pump technologies. Preliminary studies have been done of Peltier systems, of magnetic heat pumps. Studsvik has a cooperation with a foreign company on two-phase expansion of working fluids. Such components could be inserted into Sweden's heat pumps with a power output above 10 MW. The cold air cycle is also investigated.

The National Team of U.S.A.

18. DEVELOPMENTS IN GAS-FIRED ABSORPTION HEAT PUMPS IN NORTH AMERICA

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U.S.A.

ABSTRACT: Gas fired absorption heat pumps for space conditioning (heating and air conditioning) are being developed by several U.S. organizations. Single-effect cycle absorption heat pumps for residential applications were developed in the 1970's, but proved to be uneconomic and were not manufactured. Advanced cycle absorption heat pumps with substantially higher efficiencies are now being developed, and have the potential to be economically attractive to consumers.

BACKGROUND

Residential and commercial buildings account for 37% of the total energy used in the United States primarily for heating, cooling, and hot water. Natural gas and oil are major fuel sources for building heating, and almost all air conditioning is done with electricity. Figure 1 shows energy consumed by fuel type in the U.S. for heating and cooling.

Air conditioning is a major energy use in residential and commercial buildings. Currently in the U.S., about 85% of all new houses are air-conditioned, and about 30% of new houses use electric heat pumps for heating and air-conditioning [1].

Figure 2 shows the relative operating fuel cost to the consumer for electric heat pumps, gas furnaces, and gas fired heat pumps for a range of electricity to gas price ratios. As can be

*Research sponsored by the Office of Buildings and Community Systems, U.S. Department of Energy, under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

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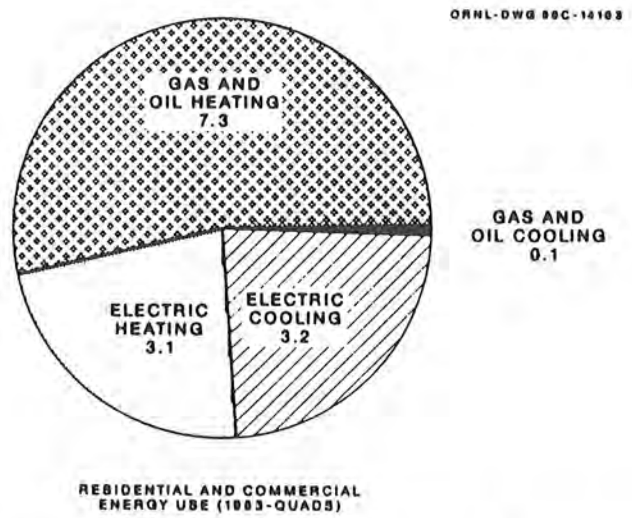

Figure 1. Natural gas and oil are major fuel sources for building heating.

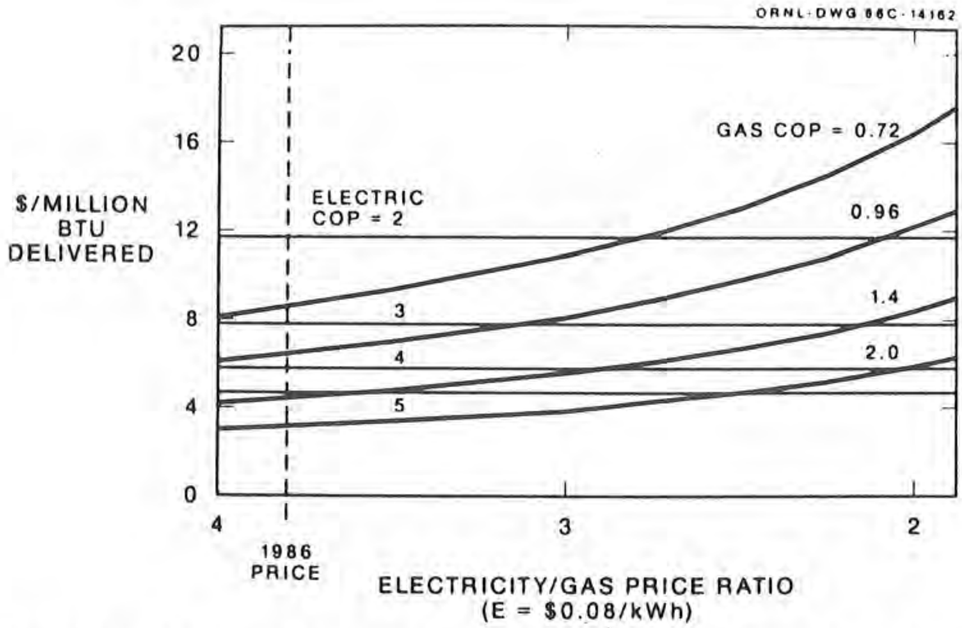

Figure 2. Advanced absorption heat pumps will cost less to operate than electric heat pumps.

seen, with modestly efficient gas fired heat pumps (coefficient of performance (COP) > 1.4), heating fuel costs should be lower than electric heat pumps for any credible electric to gas price ratio.

The U.S. Department of Energy (DOE) and the Gas Research Institute (GRI) have been supporting research and development for absorption heat pump systems since the late 1970's. The goals for development of gas-fired absorption heat pumps are to produce equipment that will save primary energy compared to existing technologies, be economical to the customer to purchase and operate, and also provide for the use of natural gas for air-conditioning as an alternative to electricity.

PROJECT SUMMARIES

Initial projects, first funded in the 1970's and now completed, were for the development of prototype single-effect absorption heat pumps for residential applications.

Arkla Industries, a manufacturer of ammonia-water air conditioners, developed a single-effect ammonia-water heat pump for heating only with steady state COPs of 1.25 at 8.3°C (47°F) and 1.12 at -8.3°C (17°F) [2]. These performance levels are fuel based efficiencies including flue losses using the higher heating value of natural gas.

Allied Corporation, under sponsorship of DOE, the American Gas Association (AGA), and GRI developed a single-effect heat pump for both heating and cooling for residential applications, using a fluorocarbon refrigerant and an organic absorbent. Prototype heat pumps were developed to achieve a heating COP of 1.2 at 8.3°C (47°F) and a cooling COP of 0.5 at 35°C (95°F) [3]. The prototype development used R-123a (1,2-dichloro 1,2,2-trifluoroethane) for the refrigerant and ETFE (ethyl-tetrahydro furfuryl ether) for the absorbent. Further refinement of the Allied prototype, including a condensing flue, showed potential for about a 1.5 heating COP and a 0.6 cooling COP.

Both the Arkla and Allied heat pump prototypes were successfully operated and achieved their performance goals. However, the single-effect performance levels were not sufficient to offer reasonable economic incentive for customer acceptance, so neither machine was further developed or manufactured.

Columbia Gas Service Corporation started the development of a residential size double-effect absorption heat pump using a proprietary absorption fluid pair in the mid-1970's. Their performance target was a heating COP of 1.55 and a cooling COP of 0.80. In 1980, GRI joined Columbia Gas in funding development of this double-effect concept. In 1984, Columbia Gas reached their original performance goals with laboratory breadboard prototype equipment [4]. Recently, in October 1986, they have operated a packaged prototype machine and demonstrated gas-fired COPs of 1.7 in heating and 0.85 in cooling, exceeding the original 1980 program goal [5].

Based on the achieved performance results of the Arkla and Allied single-effect cycle prototypes, and the need to provide economic payback to customers of less than three years against conventional heating and air conditioning systems, the DOE absorption heat pump program goals were re-evaluated in late 1981. The decision was made to evaluate advanced absorption cycles capable of producing higher efficiency heat pumps than conventional single-effect cycles while still using existing absorption fluid combinations. Higher efficiency levels were needed, especially for air-conditioning operation, in order to save enough money in fuel costs to offset the assumed high initial cost of absorption machines.

In 1982, the DOE absorption program, managed by Oak Ridge National Laboratory's (ORNL) Building Equipment Research Program, issued a Request for Proposal (RFP) for the development of advanced absorption cycles with performance goals of COPs of 1.6 in heating and 0.7 cooling, as compared with current state-of-the-art single-effect values of 1.2 and 0.5. The RFP was distributed to more than 50 potential bidders within the U.S. Three contractors were selected for the work under the advanced cycles program, and work was started in late 1982 and early 1983. The three contractors are Carrier Corporation, the Trane Company, and Phillips Engineering Company.

The work in the advanced absorption cycles program is divided into three phases. Phase I is the analytical evaluation of advanced cycles, with a preferred advanced cycle being selected for further development. Phase II is the development and testing of a laboratory breadboard prototype of the selected advanced cycle. Phase III is the development and testing of a packaged proof-of-concept prototype after successful completion of Phase II.

The goal of the Phase I selection of a preferred advanced cycle was not necessarily to pick the highest efficiency cycle, but rather to select cycles with a combination of features, such as high efficiency, low manufacturing cost, high inherent reliability, etc., that appeared to offer the most likely path to a manufactured product by the 1990's.

In Phase I analysis work, Phillips Engineering examined a variety of cycles including: (1) double-effect cycles, (2) resorber augmented cycle, (3) generator-absorber heat exchange (GAX) cycle, (4) double-effect regenerative cycle, (5) variable-effect cycle, and (6) two-stage generator-absorber heat exchange cycle [6]. Phillips evaluated each cycle on the basis of (1) potential efficiency, (2) heat exchanger surface requirements, and (3) ability to operate over a broad range of temperatures. The GAX cycle was selected as having the best combination of high potential efficiency and low manufacturing cost for residential applications. Both ammonia-water and ammonia-lithium bromide-water were selected as candidate absorption fluid systems for this cycle.

The Trane Company and Carrier Corporation each evaluated advanced cycles for commercial applications. The Trane Company conducted a Phase I analysis similar to Phillips Engineering and also selected a GAX cycle, but with additional features and internal heat exchange for higher efficiency [7]. Economic analysis predicts customer payback of less than 3 years in commercial buildings while simultaneously giving a better utility load profile for both gas and electric utilities.

Carrier's study of advanced cycles resulted in selection of a dual-loop cycle based on the use of methylamine-lithium bromide-water in the lower (temperature) loop and lithium bromide-water in the upper loop [8]. The DOE has been granted six patents based on the Carrier Phase I work. Figure 3 shows a schematic of the Carrier dual loop heat pump.

Each of these advanced cycles selected by Trane, Carrier, and Phillips offers potential for a rated heating COP in the range

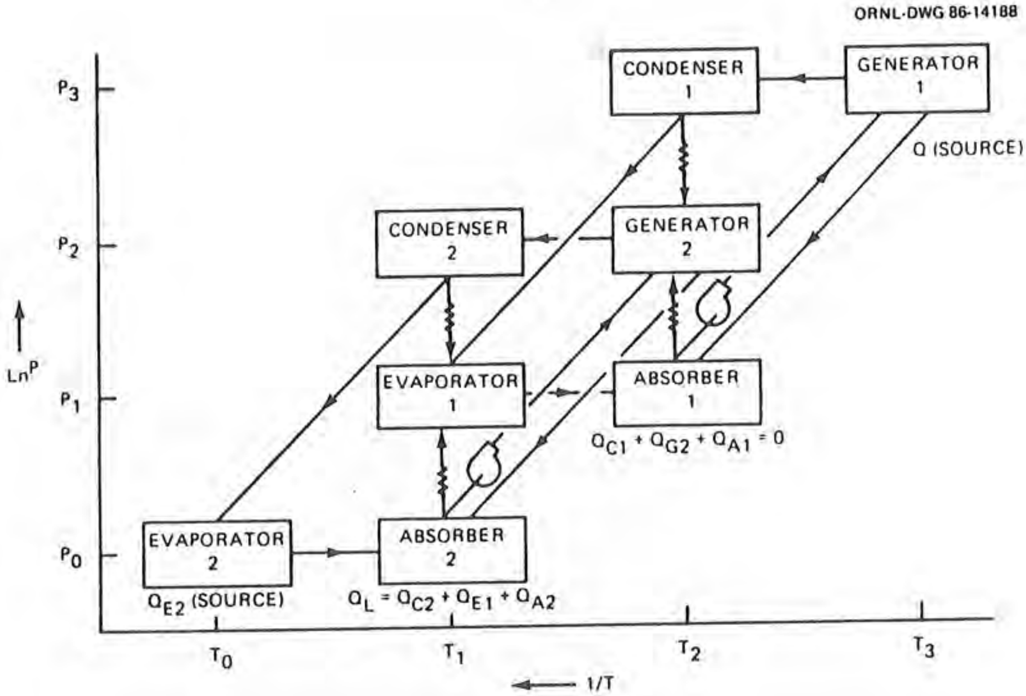

Figure 3. The Carrier dual loop cycle uses different fluids in each loop for best performance.

of 1.8 to 2.0 and cooling COP in the range of 0.8 to 1.2, significantly exceeding the original program goals set in 1982.

Phase II of the projects, the laboratory breadboard demonstration of the cycles identified as most promising, is underway. During 1985, Phillips Engineering successfully operated a proof-of-principle breadboard prototype of a residential-sized unit with performance exceeding the original advanced cycle RFP goals. The GAX cycle breadboard achieved COPs of over 1.8 in heating at 8.3°C (47°F) and 0.8 in cooling at 35°C (95°F). An independent cost analysis of the Phillips Engineering prototype concept has predicted a customer-installed cost equal to existing mid-line gas furnace/electric air-conditioner combinations or to electric heat pumps, indicating the potential for advanced absorption cycle residential heat pumps to become a major consumer product in the future.

Trane expects to demonstrate proof-of-principle with a laboratory breadboard prototype model of a commercial-sized unit in 1987.

Carrier is currently scheduled to build and test two separate laboratory breadboard prototypes. The first prototype, a single-stage commercial-sized absorption heat pump capable of below-freezing operation is scheduled for operation in 1987. The second prototype using the dual-loop concept is scheduled for testing in 1988.

In mid-1983, GRI funded a dual-cycle advanced absorption residential size heat pump concept proposed by Battelle-Columbus. The objective is to develop a residential-size gas heat pump with 1 1/2 to 3 tons cooling capacity and COPs of 1.80 heating and 0.94 cooling [4]. During 1985, Battelle operated an integrated breadboard of the dual-cycle absorption heat pump concept and achieved the target COPs [9]. Phase II work to design, build, and test a series of packaged prototypes for field testing is currently underway, with a manufacturing participant. Large scale field evaluation of the Battelle dual-cycle advanced absorption heat pump is scheduled to start June 1, 1987, with subsequent commercialization projected for early 1989.

The American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) is sponsoring absorption supporting research on heat and mass transfer in absorbers to address absorber performance at previously unexplored operating conditions applicable to advance cycle systems [10].

DOE has also sponsored the development of a modular user-oriented absorption cycle computer model for use in evaluating advanced absorption cycles for heat pumps and heat transformers. The basic model has been completed and is available for use on main-frame computers and for IBM compatible personal computers [11].

The Institute of Gas Technology (IGT), under DOE sponsorship, has undertaken the cataloging of absorption fluids properties data and references for all available U.S. data [12] and also for available foreign data [13].

In support of the advanced heat pump work, ORNL has developed a unique instrument using fiber optics to measure precisely the refrigerant concentration in the refrigerant-absorbent mixture in an operating absorption device [10]. This fiber-optic refrigeration cycle monitor measures the refractive index of fluids. Because most refrigerant-absorption fluids are binary mixtures, the refractive index of the fluid is directly related to the concentration of its individual components. In tests with $LiBr/H_2O$, the instrument has demonstrated the ability to measure local refrigerant concentration to an accuracy of 0.1% inside an operating absorber. Combining measurements of local concentration and temperature allows on-line monitoring and control of the refrigeration system, as well as a detailed study of heat and mass transfer processes occurring within absorption systems. This new measurement capability should help in the design of new absorption systems for increased operating efficiency.

Figure 4 shows primary energy consumption in heating and cooling for existing technologies, the prior single-effect cycles and the current advanced cycle concepts. Based on recent progress in the DOE and GRI programs, advanced absorption cycle heat pumps show the potential for significant energy savings with acceptable consumer economics for both residential and commercial applications.

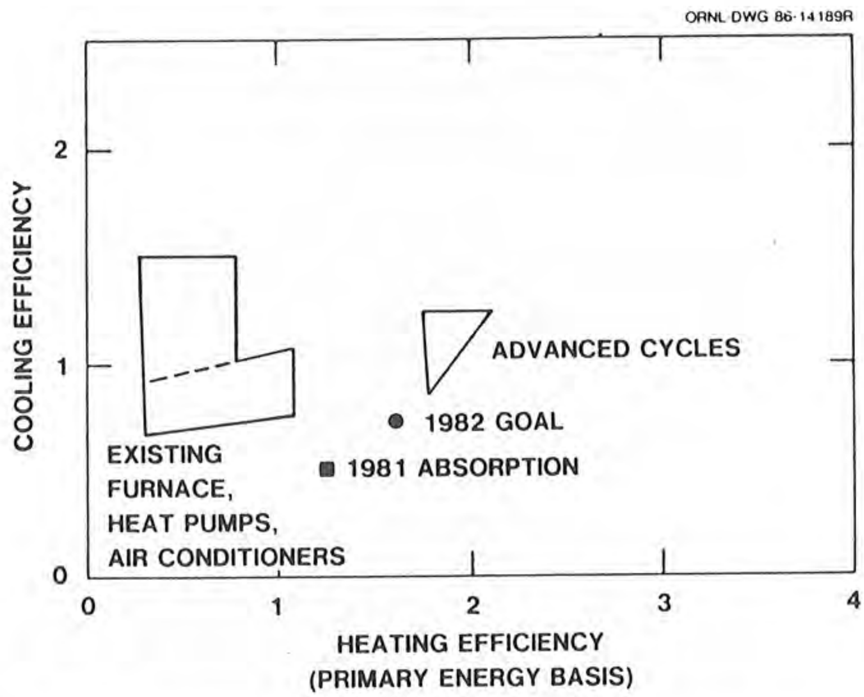

Figure 4. Advanced cycles have performance potential beyond current technology.

Note: Assumes electricity delivered to customer at 33% efficiency including generation and distribution losses. Heating performance based on 47°F standard steady state rating for all technologies, cooling performance based on 95°F ambient steady state rating for all technologies.

Notes:

All performance figures are based on high heating value of natural gas.

Studies of corrosion ratios aid absorption fluids thermal stability in advanced absorption cycles as included in each project listed.

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ENERGY

CONSERVATION

ORNL/Sub/84-47989/3

19. ABSORPTION FLUIDS DATA SURVEY: FINAL REPORT ON WORLDWIDE DATA

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Prepared by
INSTITUTE OF GAS TECHNOLOGY
Chicago, Illinois 60616

under
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for

OAK RIDGE NATIONAL LABORATORY
operated by
MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the

U.S. DEPARTMENT OF ENERGY

OFFICE OF BUILDINGS AND COMMUNITY SYSTEMS

ABSORPTION FLUID SYSTEMS IN THE SURVEY

500	TOTAL REFERENCES
278	PRIMARY REFERENCES
38	REFRIGERANTS
131	SINGLE ABSORBENTS
42	BINARY ABSORBENTS
14	TERNARY ABSORBENTS

REFRIGERANTS

INORGANIC	3
ORGANIC	35
• AMINES	4
• ALCOHOLS	5
• HALOGENATED	25
• HYDROCARBONS	1

ABSORBENTS

INORGANIC	48
ORGANIC	83
• ALCOHOLS	9
• ETHERS	5
• ALCOHOL-ETHERS	3
• AMIDES	9
• AMINES	4
• AMINE-ALCOHOL	1
• ESTERS	21
• KETONES	5
• ACIDS	4
• ALDEHYDE	1
• OTHERS	20

MIXTURE PROPERTIES

1. VAPOR-LIQUID EQUILIBRIA
2. CRYSTALLIZATION TEMPERATURE
3. CORROSION CHARACTERISTICS
4. HEAT OF MIXING
5. LIQUID-PHASE DENSITIES
6. VAPOR-LIQUID-PHASE ENTHALPIES
7. SPECIFIC HEAT
8. SURFACE TENSION
9. STABILITY
10. VISCOSITY
11. MASS TRANSFER RATE
12. HEAT TRANSFER RATE
13. THERMAL CONDUCTIVITY
14. FLAMMABILITY
15. TOXICITY
16. ENTROPY
17. REFRACTIVE INDEX

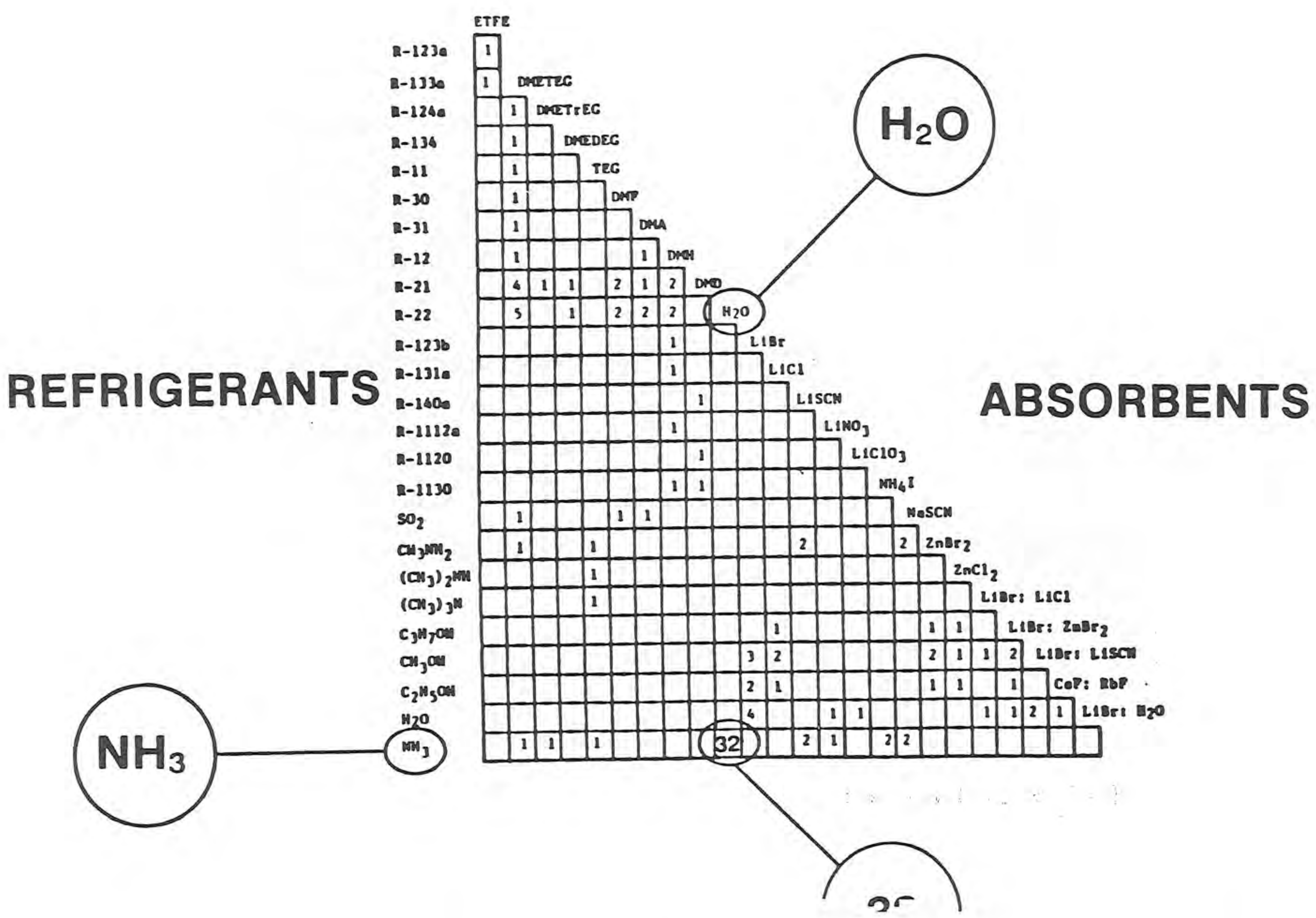
DATA RANK

FORM OF DATA	SYMBOL
RAW EXPERIMENTAL	(E)
• TABULAR	(T)
• GRAPHICAL	(G)
SMOOTHED EXPERIMENTAL	(S)
• TABULAR	(T)
• GRAPHICAL	(G)
EMPIRICAL POLYNOMIAL	(P)
EQUATIONS OF STATE	(C)

THERMODYNAMIC, TRANSPORT AND OTHER DATA

FLUID PROPERTY	NUMBER OF REFERENCES
VAPOR-LIQUID EQUILIBRIUM	132
ENTHALPY	53
DENSITY	46
CORROSION (AND INHIBITORS)	17
STABILITY	25
TRANSPORT PROPERTIES	79
HEAT OF MIXING	49
CRYSTALLIZATION	66
TOXICITY	1
FLAMMABILITY	1

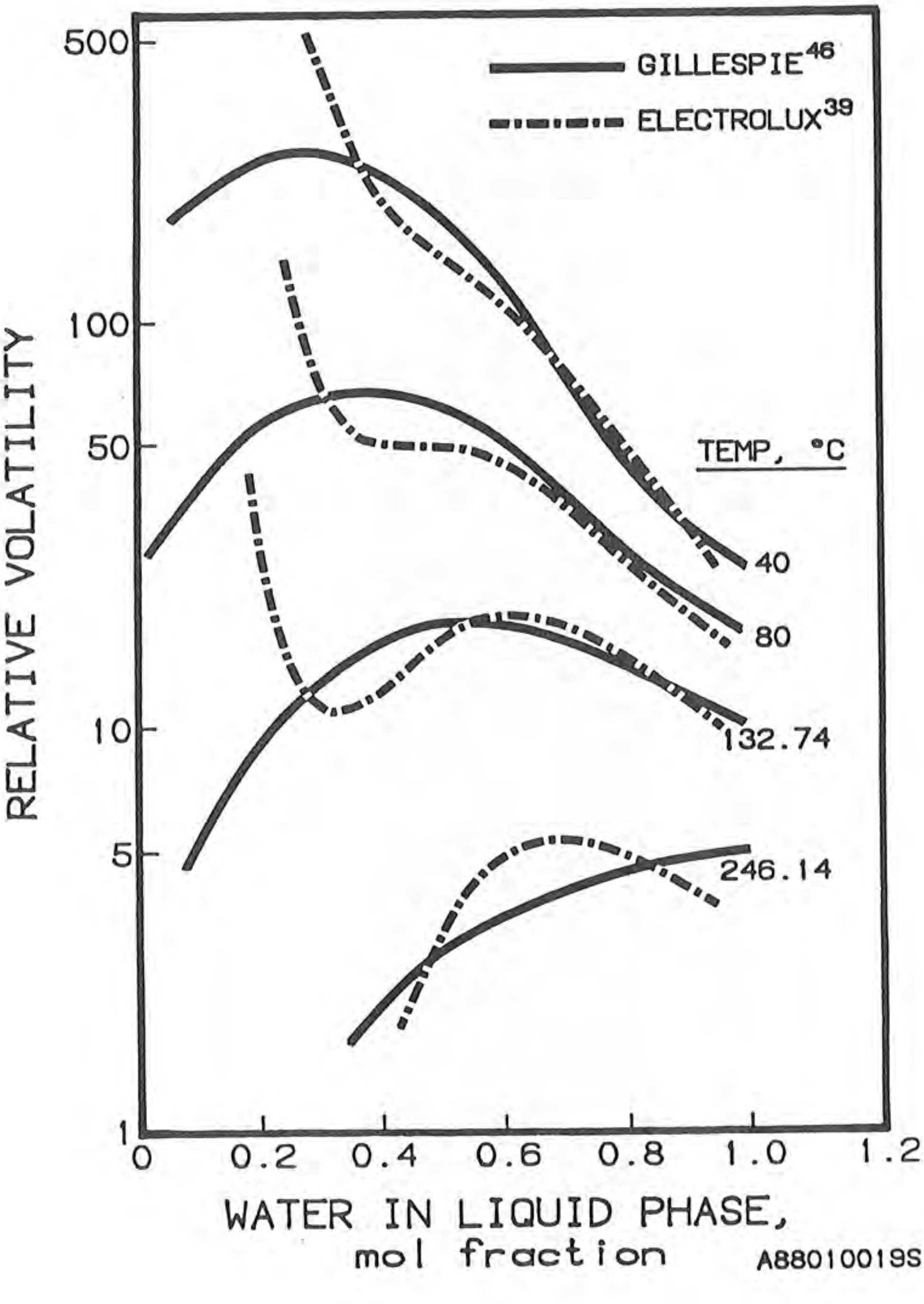
REFERENCES FOR VLE DATA



DATA SCREENING METHODS

AUTHOR'S STATEMENTS
MEASUREMENT TECHNIQUE AND APPARATUS
AUTHOR'S REPUTATION IN MEASUREMENTS
DATA PLOTS FOR INTERNAL CONSISTENCY
TRENDS IN HOMOLOGOUS FLUIDS

AMMONIA + WATER



KEY FLUIDS

NH_3-H_2O
NH_3-H_2O-LiBr
NH_3-H_2O-$LiNO_3$
NH_3-NaSCN

H_2O-LiBr
H_2O-LiBr-LiCl
H_2O-LiBr-LiSCN
H_2O-LiBr-$ZnBr_2$

H_2O-NaOH-KOH-CsOH
H_2O-$LiNO_3$-$NaNO_3$-KNO_3
CH_3OH-LiBr
CH_3OH-$ZnBr_2$

CH_3OH-LiBr-$ZnBr_2$
TFE-NMP
TFE-DMEDEG
TFE-DMETEG

CH_3NH_2-H_2O-LiBr
R21-DMETEG
R22-DMEDEG
R22-DMETEG

R22-DMF
R123a-ETFE
R124-ETFE
R133a-NMP

DETERMINATION OF PRIORITY LEVELS FOR FILLING GAPS IN FLUID PROPERTIES DATA

1. RESOLUTION OF CONFLICTS AND NEW DATA ARE IMPERATIVE
2. RESOLUTION OF CONFLICTS ARE NEEDED
3. NEW DATA ARE DESIRABLE

BLANK – NO ACTION REQUIRED

MOST CRITICAL DATA GAPS

- DIFFUSION RATES
- MASS TRANSFER ADDITIVES
- THERMAL STABILITY
- CORROSION RATES
- CORROSION INHIBITORS

INTERNATIONAL COOPERATION GROUP ON ABSORPTION HP FLUIDS DATA

UNIVERSITY OF ESSEN

UNIVERSITY OF STUTTGART

FRENCH PETROLEUM INSTITUTE

GAZ DE FRANCE

BEN GURION UNIVERSITY

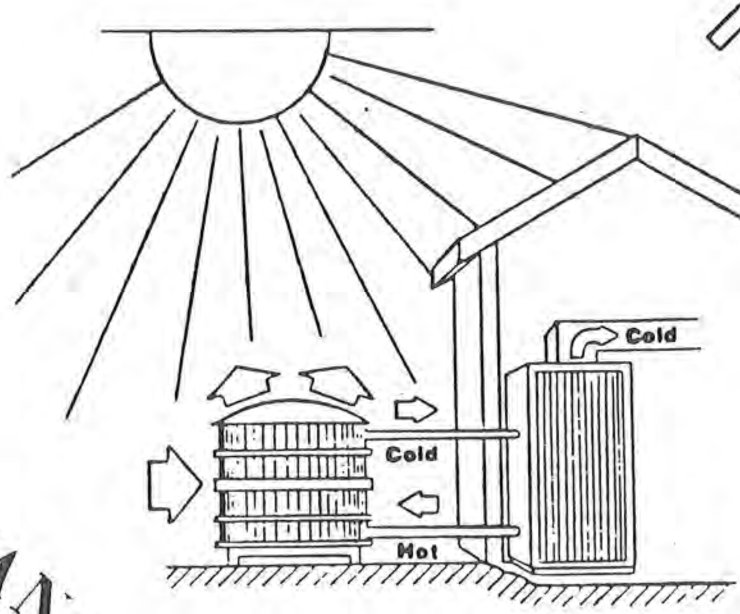
KANSAI UNIVERSITY

OAK RIDGE NATIONAL LABORATORY

INSTITUTE OF GAS TECHNOLOGY

ABSORPTION FLUIDS

NH_3-H_2O



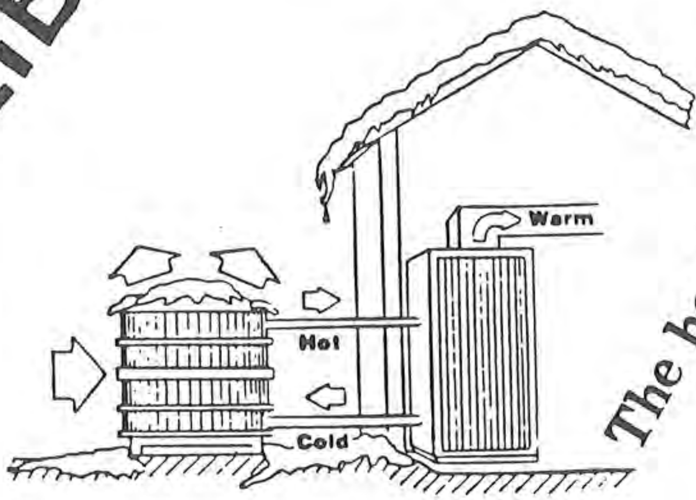
Air Conditioners

ADVANCED SYSTEMS

CH_3NH_2/H_2O-LiBr

MEAM / H_2O - LiBR

NH_3/H_2O-LiBr



The heat pump

R22/DMETEG

LiBr - H_2O

20. TRANANE, GRI DEVELOPING NEW ABSORPTION CHILLER

CHICAGO — The Trane Co. and the Gas Research Institute (GRI) are joining forces to develop a new absorption cooling technology patented by Oak Ridge National Laboratory.

Commercial energy users will benefit from development of the advanced technology, which will be applicable in large gas-fired chillers, typically, several hundred tons of cooling capacity, Trane said.

The cycle could reach a 50% greater efficiency than current absorption technology, making gas systems competitive with electric chillers in more regions of the country, the company added.

The concept to be developed is called a "triple-effect" absorption cycle because it uses heat more efficiently than a single cycle — up to three times more effectively, in theory. In practice, the advanced cycle should reach a gas coefficient of performance of about 1.5.

Also, the advanced cycle has lower material requirements per ton of cooling capacity. The technology is based in research funded by the U.S. Department of Energy.

In the triple-effect system, two refrigerant loops are used. Fluid in the first loop is heated by natural gas to about 400°F, and the heat is rejected to the second loop at about 200°. Water in the building's distribution system is chilled to 44° by both loops.

In addition to this new project, GRI is addressing other research and development issues related to absorption technology. For example, lower-cost materials and fabrication techniques and methods for preventing corrosion are being developed.

For more information, contact Bill Dolan, GRI manager, Commercial Space Conditioning 312-399-8359.