A New Method for Increasing COP of Heat Pump Cycle

Tsutomu YAMAGUCHI*

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1. Introduction

The removal of water by heating from solutions or liquids is one of the important operations of the evaporation, distillation, and concentration processes in the chemical plants or in the food industry. Especially the evaporation becomes now most necessary step of producing water for industrial or drinking use. In these cases large amounts of heat energy is usually necessary to evaporate water. For example, in the case of a simple apparatus as shown in Fig. 1, required heat energy is a sum of the latent heat of evaporation and the sensible heat of liquid. These energies are not recovered. Although the apparatus shown in Fig. 2 is able to economize the sensible heat of liquid, the vapor-compression evaporating system1) shown in Fig. 3 is still better than that of Fig. 2 from the economical point of view.

This method is a kind of heat pump system, because the system increases the pressure and the temperature by compressing the vapor evaporated at a lower temperature, and makes use of the heat energy obtained by the condensation of the vapor for heating. However, this heat pump system is somewhat different from the usual one in the point that the discharged heat energy at a higher temperature is applied to the lower temperature side again and that substances of a heat source and a heat medium are quite the same. Accordingly the equation for the coefficient of performance is also somewhat different.

Now, usual methods of lowering the evaporation temperature have a disadvantage that the coefficient of performance decreases rapidly with a fall of the evaporation temperature so long as the condensation is carried out at the atmospheric pressure. If an additional energy is needed to lower the condensation pressure, we cannot expect an improvement in the coefficient of performance. The author worked out a new system of cycle which can supress the decrease in the coefficient of performance by means of reducing the condensation pressure in accordance with the fall of the evaporation temperature. This paper presents

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* Associate Professor, Department of Mechanical Engineering

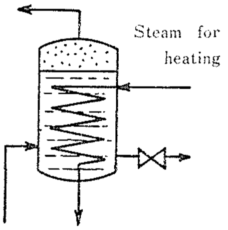

Fig. 1 Simple evaporator

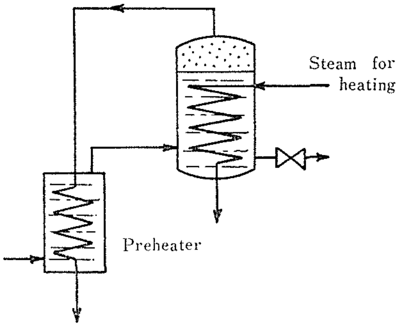

Fig. 2 Simple evaporator with preheater

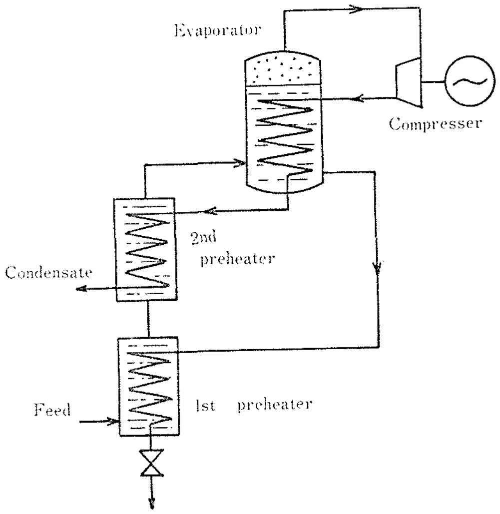

Fig. 3 Vapor-compression evaporator

some theoretical considerations and expermental results to show the prospect of its practical use.

2. Theoretical Considerations

The vapor-compression cycle of heat pump is shown $3'\ 2''\ 3''\ 4''\ 5\ 6\ 5\ 4'\ 3'$ in the T-s diagram of Fig. 4, and the area of closed cycle shows the necessary energy. In the case of vacuum evaporation, that is, an evaporation at a temperature below 100°C accompanied by a condensation at the atmospheric pressure, the cycle is shown as $3\ 8\ 3\ 3'\ 4'\ 5\ 6\ 4\ 3$ in Fig. 4. In the case of evaporation at a much lower temperature, the cycle changes to $1\ 2'\ 3'\ 4'\ 5\ 6\ 7\ 1$ and the necessary energy increases rapidly. Hence it is evident from the T-s diagram

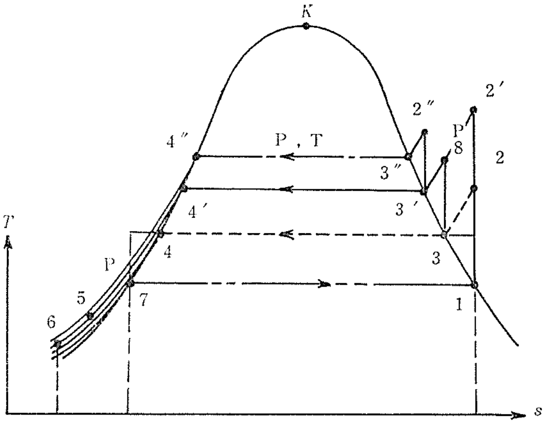

Fig. 4 T-s diagram of heat pump cycle

that the coefficient of performance decreases considerably.

If we reduce the condensation pressure according as the evaporation temperature, on the other hand, we have the cycle 1 2 3 4 5 6 7 1, so that the required energy can be saved remarkably. If, however, an extra energy is required to lower the condensation pressure (by driving a vacuum pump, say) at the same time, we cannot get an economical coefficient of performance. Nevertheless, when we use a siphon, the decrease in the coefficient of performance would be small and we shall be able to expect an improvement in economy, since the cycle 1 2 3 4 5 6 7 1 merely displaces as it is towards the s-axis side.

3. Theoretical calculations

The coefficient of performance is given by the equation

$$\varepsilon = \frac{T_c}{T_c - T_e}$$

for the cycle of a usual heat pump, but it is given by the equation

$$\varepsilon = \frac{Q_e}{Q_c - Q_e} = \frac{T_e}{T_c - T_e} = \frac{T_e}{\Delta T}$$

for a vapor-compression cycle based upon the inverse-Carnot cycle. Here $T_c - T_e = \Delta T$ is the temperature difference necessary to transfer heat energy from condensing steam to evaporating liquid. Since the amount of heat energy required for the purpose of giving the temperature difference is comparatively small in this case, the coefficient of performance of a vapor-comporession evaporating system is considerably better than that of a usual heat pump.

The notations and units to be used in the calculations are as follows:

t = temperature, °C,
$T = t + 273.16$ = absolute temperature, °K
P = pressure (absolute), kg/m^2; p = pressure, kg/cm^2,
i = enthalpy, kcal/kg,
r = latent heat of vaporization, kcal/kg,
s = entropy, kcal/kg°K

Together with the steam table and the Mollier chart (i-s diagram) published by the J. S. M. E.[2] the equations of state given by Tanishita are employed for calculations of the thermodynamic functions of steam, and the equation by Sugawara is used for calculations of the relation between the temperature and the pressure of saturated steam.

The calculated results of the coefficient of performance of the inverse-Carnot cycle obtained from the above equation $\varepsilon = T_e/\Delta T$ are shown in Fig. 5, 6, and 7 by solid lines. The equation to be used for the calculation of the coefficient of performance for the inverse-Rankine cycle is

$$\varepsilon = \frac{i_1 - i_6}{i_2 - i_1},$$

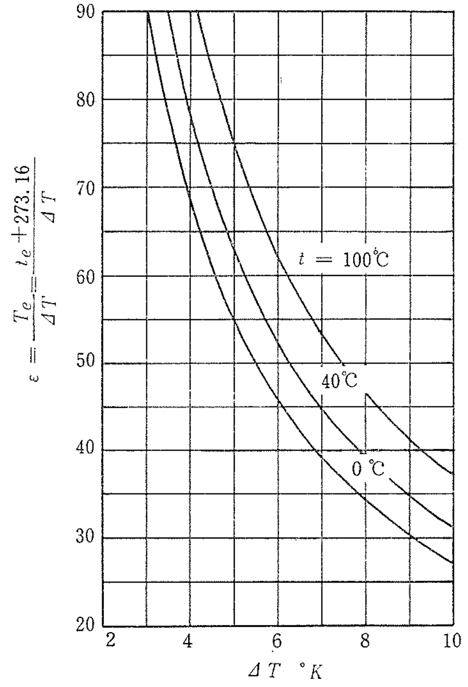

Fig. 5 Coefficient of performance for inverse-Carnot cycle

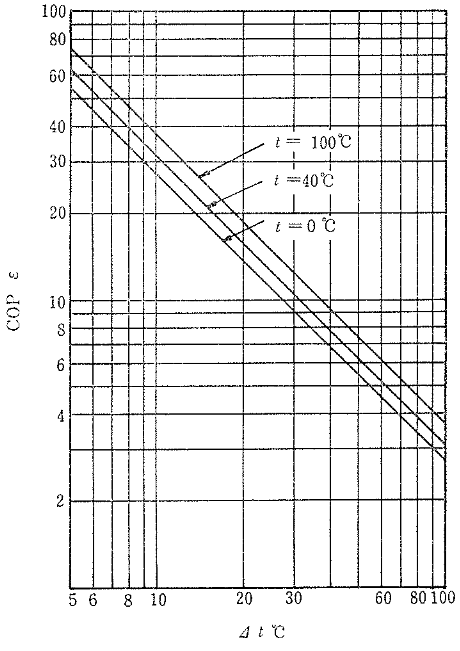

Fig. 6 Coefficient of performance for inverse-Carnot cycle

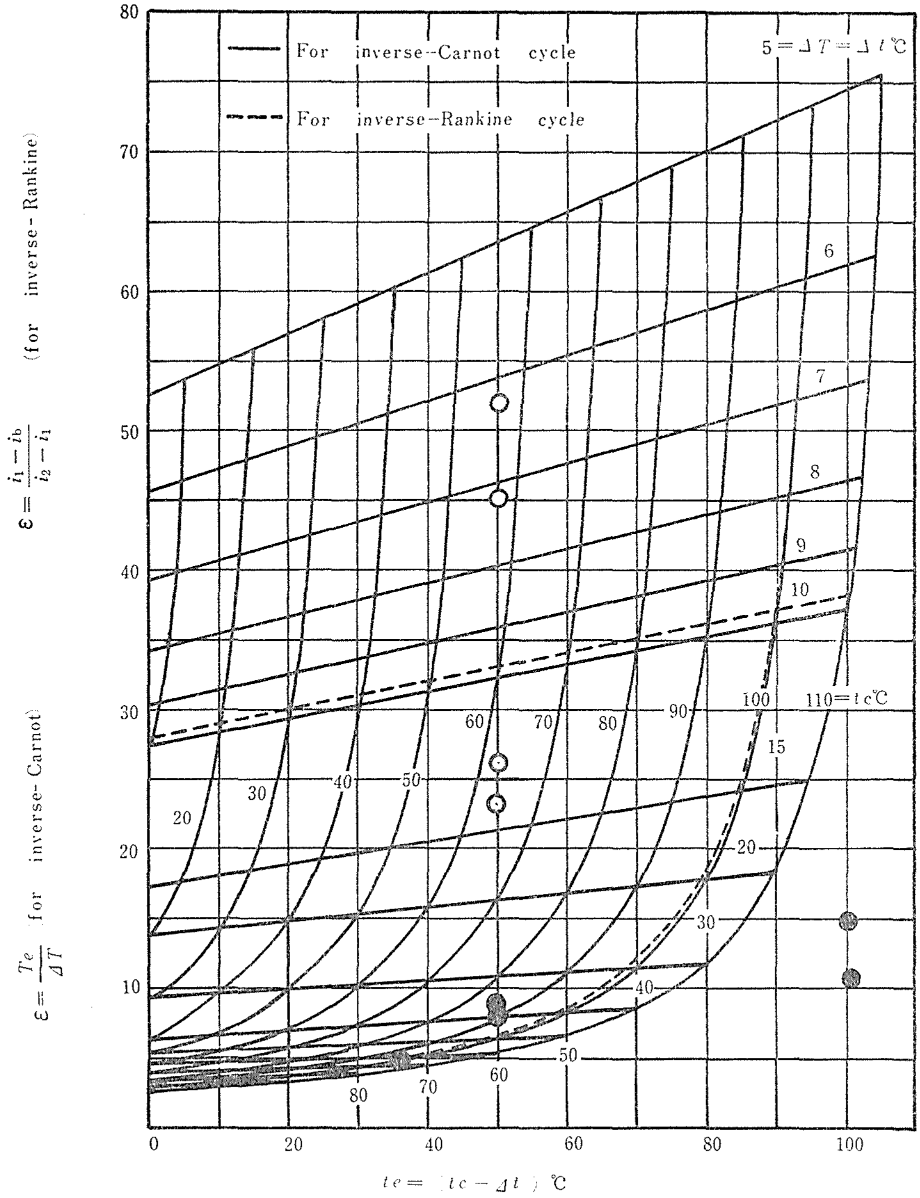

Fig. 7 Calculated results

where the suffices to enthalpy i correspond to the point shown in Fig. 4. In these calculations the process $1 \sim 2$ is assumed theoretically to be adiabatic, $2 \sim 3$ isobaric, $4 \sim 5$ polytropic ($n = 1 \sim 0$), $5 \sim 6$ isobaric, and $6 \sim 7$ polytropic ($n = 1 \sim 0$), respectively, so that the coefficient of performance can be calculated according to the foregoing equations, the steam table, and the Mollier diagram. The calculated results of the coefficient of performance for the inverse-Rankine cycle are shown in Fig. 7 by dashed lines. As is seen from the figure the solid and dashed lines are in approximate agreement.

4. Experimental Results

A few experiments were performed with the apparatus shown in Fig. 8 as

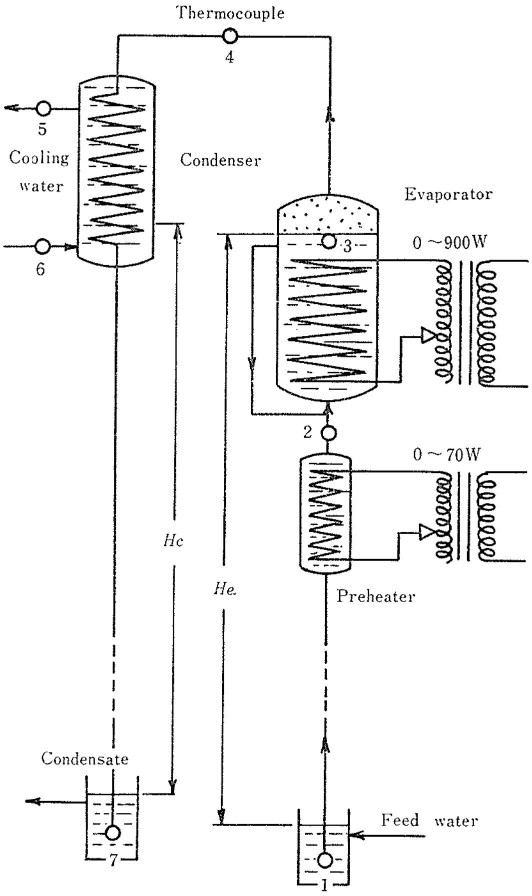

Fig. 8 Experimental apparatus

preliminaries for practical design. From those experiments the following results were obtained.

1. By controlling the heating and the cooling energies, a stable balance of conditions is realized easily.
2. Spacial considerations are needed for the design of an evaporator.
3. In the case of an indirect system, air should be removed from liquid.
4. Prevention of heat losses from an apparatus is very important.
5. In the case of evaporation at a low pressure (or a low temperature), where the specific volume of steam usually becomes very large, we have to apply an indirect method as shown in Fig. 9. When the shaft sealing of a compressor in a high vacuum is difficult, it may be suitable to use a waterproof or a submerged motor.

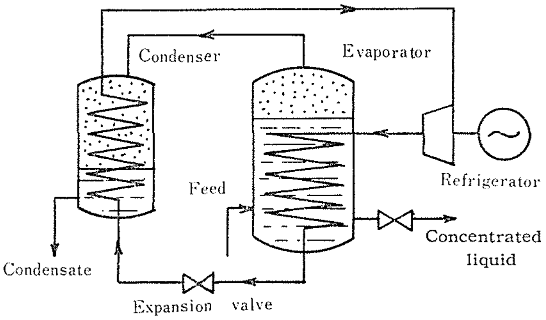

Fig. 9 Indirect vapor-compresson evaporator

5. Conclusions

So long as the evaporation pressure is reduced with keeping the condensation pressure at the atmospheric one, the coefficient of performance decreases remarkably as is shown in Fig. 7. In contrast with this the coefficient of performance decreases gently as is also shown in Fig. 7 for the proposed cycle which lowers the condensation pressure according to the fall of the evaporation pressure.

It is evident from the T-s diagram that the values of the coefficient of performance for the inverse-Rankine cycle are very close to those of the inverse-Carnot cycle. The main reason why the values of the coefficient of performance for the inverse-Rankine cycle, within a small temperature difference between the condenser and the evaporator, are better than those for the inverse-Carnot cycle is the fact that the sensible heat from the normal temperature to the saturation temperature is utilized here.

Black circles in Fig. 7 show examples of the data from actual apparatus[3). when the compression efficiency of a compressor is 70 ~ 80%, the mechanical efficiency 85 ~ 90%, and other heat losses are negligible, the coefficient of performance is estimated to be 45~52 in the case of $\varDelta T = 5$°K (for a direct method), and 23~26 in the case of $\varDelta T = 10$°K (for an indirect method). These are shown by white circles in Fig. 7.

Since the white circles appear far above the black ones in Fig. 7, the practical use of the proposed cycle seems to be well promissing.

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