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BASIC STUDY ON HIGH PERFORMANCE HEAT PUMP SYSTEMS ACCOMPANYING TWO-PHASE COMPRESSION PROCESS

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ABSTRACT

Two-phase compression heat pump cycles have been studied from the theoretical point of cycle analysis to clarify the optimum liquid injection conditions as a first and basic step for developing the higher performance high temperature heat pump systems, which recover the heat from a 150°C heat source and supply 300°C temperature thermal output.

In the case where the working fluid is water, the COP is improved by utilizing two-phase compression process and there exists an optimum gas-liquid ratio which brings the maximum value of COP.

Two-phase injection cycle, in which the liquid partially extracted from the outlet of the condenser and the gas separated in the economizer are injected into a intermediate point of the compression process, can effectively improve the COP and reduce the discharge superheat.

INTRODUCTION

Heat pump systems have a noticeable effect on energy conservation. Though they are coming into wide use for industrial heat recovery systems their output temperature is limited to approximately 120°C [1].

The high temperature process heat (300°C or more) is common and in great demand for industry, and is usually supplied by boilers. If heat pump systems which can supply such high temperature output heat are developed and come into wide use, they will have much effect on energy conservation and will contribute toward overcoming current global environmental problems.

Water is suitable as the working fluid for high temperature compression heat pumps because of its thermal stability. Technology to treat high pressure/temperature steam has progressed in the field of steam power equipment. With foundations in this technology, if the most appropriate cycles are utilized, we will be able to develop high temperature steam compression heat pumps with high performance.

As is well known, adiabatic compression of steam from a saturated condition induces a material degree of discharge superheat, and it brings about problems in heat pumps, i.e., it causes not only the coefficient of performance (COP) to be reduced, but in some cases operation itself to be impossible.

Some steam recompression heat pumps [2], which are some the highest temperature heat pumps used in industrial fields, and a test plant for a 300°C output heat pump [3], which is the one being developed under the Japanese national R&D project "Super Heat Pump Energy Accumulation System" [4], use water as the working fluid and control the discharge temperature by coolant water injection into the compression process. This two-phase compression process is a key technology in realizing high temperature compression heat pumps.

Though the heat pump cycle using the two-phase compression process is not a novel idea, there are few systematic studies on clarifying its optimum conditions. Because, the two-phase compression process have no or little good effect on conventional heat pumps which use common CFC, HCFC refrigerants [5] and it tends to cause decreasing of compressor performance [6].

This paper considers a high temperature heat pump recovering heat from a 150°C heat source (matching with output temperature of heat pumps actualized in the industrial fields these days) and producing 300°C heat (matching the boilers). Two-phase compression heat pump cycle is studied from the theoretical point of cycle analysis to clarify its potentiality on increasing the COP and the optimum conditions of basic liquid injection cycles. There, two-phase compression process is assumed to be quasi-static.

Cycle analyses are based on the thermophysical properties of real working fluids [7,8,9].

NOMENCLATURE

P : Pressure
T : Temperature
m : Wetness
h : Specific enthalpy
s : Specific entropy
x : Parameter for pressure at the injection point
y : Injection ratio (ratio of injection mass flow rate to total mass flow rate)

Subscript

1 : Suction of compressor
2 : Discharge of compressor
3 : Outlet of condenser
4 : Inlet of evaporator or outlet of expansion valve
5 : Saturated liquid at the evaporation pressure
6 : Saturated vapor at the condensation pressure
01 : Saturated vapor at the evaporation pressure
H : Heat output side
L : Heat source side
I : Injection point

EFFECTIVENESS OF TWO-PHASE COMPRESSION

Condition To Increase COP

For working fluids to increase the degree of superheat by adiabatic compression, the most basic heat pump cycle by single stage two-phase compression is considered (cycle: 1-2-6-3-4-1 in **Figure 1**). The wet fluid from an intermediate point of the evaporation process is led to the compressor and compressed adiabatically with the vapor phase and liquid phase in equilibrium. The compressor suction wetness is represented by m_1, and the compression power per unit working fluid mass flow rate is indicated by the area: 1-2-6-3-s_3-s_4-4-1 in **Figure 1**.

The COP of this cycle is smaller than that of the Carnot heat pump cycle for following reasons. One is the influence of the irreversible change at the expansion valve (area **A**: 3-s_3-s_4-7-3 in **Figure 1**), and the other is the superheat of the compressed fluid (area **B**: 6-8-2-6). As m_1 increases, the influence of **A** becomes relatively larger, and that of **B** becomes smaller. Therefore, the effect of the two-phase compression process depends on the properties of the working fluids and temperature range where the heat pumps operate.

Denoting heat input and output per unit working fluid mass flow rate by q_L and q_H respectively, we can write the following expressions in the case of $0<m_1<m_6$,

$$q_L = (s_1-s_4)T_L \quad , \tag{1}$$

$$q_H = (s_6-s_3)T_H + \int_{s_6}^{s_1} T_{P=PH}\, ds \quad , \tag{2}$$

$$COP = q_H \,/\, (q_H-q_L) \quad . \tag{3}$$

From these expressions, we can obtain the next criterion of increasing the COP by two-phase compression as a necessary and sufficient condition for the existence of s_1 which gives the maximum *COP* value satisfying $s_6<s_1<s_{01}$,

$$(s_6-s_3)T_H - (s_{01}-s_4)T_{02} + \int_{s_6}^{s_{01}} T_{P=PH}\, ds < 0 \quad . \tag{4}$$

Equation (4) means

$$\text{area } \mathbf{A} < \text{area: } 6\text{-}02\text{-}9\text{-}7\text{-}6 \quad , \tag{5}$$

and the COP takes the maximum value when the m_1 satisfies

$$\text{area A} = \text{area: } 6\text{-}2\text{-}10\text{-}7\text{-}6 \quad . \qquad (6)$$

Under this condition, the COP corresponds to that of the Carnot cycle operating between temperatures T_L and T_2.

Figure 2 shows the change of COP with the value of m_1 in the case where the working fluid is water; the heat source temperatures are 150, 200, 250°C; and the output temperature is 300°C. Here, the COP are expressed in the quantities standardized in each case by the COPo that is the COP when m_1 is 0. The COPo is 2.83, 4.52, 9.74, where the heat source temperature is 150, 200, 250°C respectively.

To compare with the case of water, **Figures** 3 and 4 show the change of COP with the value of m_1 where the working fluid is R22 and R502 respectively. The heat source temperatures and output temperatures in **Figures** 3 and 4 have been chosen to be almost the same temperatures as in **Figure** 2 when they are expressed by the reduced dimensionless temperature standardized by the temperatures at the critical points of each working fluid.

Figure 5 shows the change of discharge temperature with the value of m_1 where the working fluid is water.

In the cases where water is used, the latent heat and the degree of superheat in the compression process are relatively large, and the COP takes the maximum value with an optimum value of m_1. On the other hand, in the cases of R22 and R502, the COP doesn't increase by two-phase compression because the influence of irreversible pressure drop (A in **Figure** 1) dominates that of superheat (B in **Figure** 1).

In the case where the working fluid is water, the heat source temperature is 150°C, and the output temperature is 300°C, then the optimum suction wetness is 14%. In this condition, the rate of COP improvement is 5.2% and the discharge superheat is 60deg.

Comparison With Gas Injection (Economizer) Cycle

Economizer cycles, which may be considered as the gas injections into the compression processes, are effective in improving the COP. To compare with the two-phase compression cycles, the most basic economizer cycle (**Figure** 6) is theoretically studied. **Figures** 7 and 8 show the COP improvement and the change of discharge temperature by the pressure at the injection point (P_I). The temperature conditions of heat source and output, and consequently, the COPo are the same as in the case of **Figure** 2. P_I is represented by a parameter x defined by the following equation,

$$x = \ln(P_I/P_L) \;/\; \ln(P_H/P_L) \quad . \qquad (7)$$

Comparing **Figure** 7 with **Figure** 2 on each optimum point where the COP takes the maximum value, we find the COP improvement by gas injection is about two times that by two-phase compression. Comparing **Figure** 8 with **Figure** 5, the effect to reduce the discharge superheat of gas injection is much smaller, however.

In gas injection cycles, since the cooling effect for the vapor phase is small, the COP improvement decreases in the heated compression process as in the compressors whose adiabatic efficiency is low. To show it quantitatively, we use a parameter *Ead* representing the degree of heating in the compression process. *Ead* is defined by the following equation,

$$Ead = (h_2-h_1) \;/\; (h_{2ad}-h_1) \quad , \qquad (8)$$

where, h_{2ad} is the discharge specific enthalpy with isentropic compression. Considering the compressors which keep the same values of *Ead* in any compression process, we get **Figure** 9 showing the COP improvement, depending on the values of *Ead*. Here, the values of COPo are COP when vapor is compressed from a saturated condition with each *Ead* value.

In the case of two-phase compression, **Figure** 10 shows the COP improvement, depending on the values of *Ead*.

Comparing **Figure** 10 with **Figure** 9, we find that the COP improvement by two-phase compression is maintained even in the small value of *Ead*, while that by gas injection is much reduced with decreasing *Ead*.

LIQUID INJECTION CYCLE

Liquid Injection Into Suction

Isobaric Mixing Injection

In the previous section, we considered the cycles where the wet fluid from an intermediate point of the evaporation process is led to the compressor. The cycle with injecting atomized liquid into compression process (**Figure 11**) is much more practical for promoting evaporation of liquid phase and stabilizing the system operation.

Here we consider the cycle where the liquid from the outlet of the condenser is injected into the suction of the compressor. If the liquid phase is considered to be injected into the gas phase in the isobaric mixing process following an iso-enthalpic pressure drop at the atomizer, the specific enthalpy of the wet vapor after the mixing process h_1 is,

$$\begin{aligned} h_1 &= (1-y)\ h_{01} + y\ h_3 \\ &= (1-m_4\ y)\ h_{01} + m_4\ y\ h_5 \quad , \end{aligned} \tag{9}$$

because,

$$\begin{aligned} h_3 &= h_4 \\ &= (1-m_4)\ h_{01} + m_4\ h_5 \quad , \end{aligned} \tag{10}$$

where, y is the injection ratio, i.e., the ratio of the injection mass flow rate to the total mass flow rate.

Equation (10) shows that the liquid injection cycle with the injection ratio y corresponds to the cycle shown in **Figure 1**, m_1 of which is $m_4 y$. Therefore, when the optimum suction wetness is m_{1opt} in the cycle shown in **Figure 1**, the optimum injection ratio is given by the following equation,

$$y = m_{1opt}/m_4 \quad . \tag{11}$$

Figure 12 shows the change of COP with the injection ratio in the case where the working fluid is water; the heat source temperatures are 150, 200, 250°C; and the output temperature is 300°C. When the heat source temperature is 150°C, the optimum injection ratio is 21%. In this condition, the rate of COP improvement is 5.2% and the superheat is 60deg.

Effect Of Injection Liquid Temperature

Here we consider the case where a low temperature liquid is injected for promotion to cool the gas phase. The basic methods of continuously supplying the low temperature injection liquid in the closed cycles are:

- cooling a part of the liquid from the outlet of the condenser by the heat exchange with the heat source,
- extracting a part of the liquid phase after a partial pressure drop at the expansion valve.

In both methods the temperature of the injection liquid is intermediate between heat source and output temperatures. To indicate the temperature of the injection liquid, we use the wetness when the liquid expands to the evaporation pressure in the iso-enthalpic process, i.e. $m_{4'}$. In this case, we obtain the following expression for the optimum injection ratio in the same way as Equation (11) when the mixing process is considered to be isobaric,

$$y = m_{1opt}/m_{4'} \quad . \tag{12}$$

The lower the temperature of the injection liquid, the smaller is the value of $m_{4'}$. We can get the optimum cycle by the smaller injection ratio with the lower temperature of the injection liquid.

Influence Of Difference In Mixing Process

Injection accompanies mixing of two phases of the same kind of fluid but of different conditions. Varied mixing process brings the different conditions of the fluid to be compressed and have different effects on the COP.

The power needed to mix and adiabatically compress two phases of the same kind of fluid is minimized when the phases are mixed in isentropic process. We can theoretically consider the isentropic mixing for the most ideal case of the liquid injection. In this case, the values of enthalpy and entropy of the fluid after mixing determine the compression process and the compression power.

Isochoric (constant-volume) mixing, which may be realized in ideal displacement compressors, includes intermediate degree of irreversible change between isentropic and isobaric mixing. A model of isochoric mixer is shown in **Figure 13**. It works as the following. Initially, gas phase from the evaporator is sucked; the plate, i.e. the gate valve is closed; liquid phase from the condenser is sucked; the plate is opened and the two phases are mixed without displacement of the piston; mixed fluid is discharged into the compressor suction with constant pressure; all process progresses adiabatically. In this case, the values of specific volume and internal energy of the fluid after mixing are prescribed and they determine the compression process. Compression power is obtained as the difference in enthalpy between the discharge from the compressor and the total suction into the mixer.

Figure 14 shows the change of COP with the injection ratio by each mixing process mentioned above. Where, the working fluid is water, the heat source temperatures are

150 and 250°C, and the output temperature is 300°C. In the range of the injection ratio smaller than 20%, where liquid injection should be practical, isobaric mixing brings COP improvement which is more than half the improvement of isentropic mixing. That is noticeable when we consider the isobaric mixing is much easier to realize. Isochoric mixing improves the COP more than isobaric mixing, because the evaporation of the injected liquid follows the expansion and it works to compress the gas phase in the mixing process. In the actual systems, when the liquid is injected keeping higher pressure than the gas phase, an intermediate mixing condition between isobaric and isochoric mixing will occur and we can expect the COP improvement higher than that of isobaric mixing, theoretically.

Optimum Conditions Of Liquid Injection

Here we consider the liquid Injection into an intermediate point of the compression process with isobaric mixing process. The pressure at the injection point is P_X and it is represented by the parameter x defined by Equation (7). The compression power is calculated considering the compression process divided into two stages, i.e., the pressure range is from P_L to P_X and from P_X to P_H, and considering the mass flow rate and the initial condition of the working fluid at each stage.

In this case, there is the optimum injection ratio which makes the COP maximum when the working fluid is water, in the same manner as the injection to the suction. **Figure 15** shows the COP change with the injection ratio, in the case where the x value is 0.3 for an example. There, the heat source temperatures are 150, 200, 250°C, and the output temperature is 300°C. The maximum value of COP and the optimum injection ratio which brings the maximum COP change with the change of x value as shown in **Figure 16**. There, the temperatures of the heat source and the output are 150°C and 300°C, respectively. In this temperature condition, liquid injection cycle is optimized when the liquid phase is injected into the point where the x value is 0.30 with the injection ratio to be 25%, then, the rate of COP improvement is 6.3%.

In the case where the liquid is injected into the suction, i.e., where the x value is 0, the maximum COP improvement is 5.2% and the optimum injection ratio is 21%. Compared with the optimum injection condition mentioned above, the COP improvement in this case is smaller by only 1.1%, with the optimum injection ratio being smaller by 4%. In addition, from the point of view of the margin in the pressure drop at the atomizer and in the period of phase changing, which we did not consider in the analysis, injection into an earlier stage of the compression process is advisable. In the above respects, the injection into the suction or at the very beginning of the compression process is practical in actual systems.

TWO-PHASE INJECTION CYCLE

Liquid injection cycles are effective to improve the COP and to avoid excessive discharge superheat, gas injection cycles, i.e. economizer cycles, are very effective to improve the COP. To utilize the effects of both cycles, here, we consider a two-phase injection cycle (**Figure 17**). The liquid phase partially extracted from the outlet of the condenser and the gas phase separated in the economizer are injected into an intermediate point of the compression process with isobaric mixing process.

In the same way as in **Figure 16**, we show the maximum value of COP and the optimum injection ratio changing with the x value in **Figure 18**. There, the temperatures of the heat source and the output are 150°C and 300°C, respectively, the injection ratio means the ratio of the liquid injection mass flow rate to the total mass flow rate of the working fluid. In this temperature condition, the two-phase injection cycle is optimized when the liquid phase is injected into the point where the x value is 0.52, with the liquid injection ratio to be 27%. The rate of COP improvement is then 16%; the gas injection ratio is 25%; the mass flow ratio passing through the evaporator is 48%.

CONCLUSIONS

Two-phase compression heat pump cycles have been studied from the theoretical point of cycle analysis to clarify the optimum liquid injection conditions as a first and basic step for developing the higher performance high temperature heat pump systems, which recover the heat from a 150°C heat source and supply 300°C output heat. As conclusions, we have obtained the following:

(1) In the case where the working fluid is water, the COP is improved by utilizing two-phase compression process and there exists an optimum gas-liquid ratio which brings the

maximum value of COP.

(2) Though the COP improvement by two-phase compression cycles is smaller than that by economizer cycle in the case of isentropic compression, it does not decrease materially in the case of the heated compression process, and it is very effective in reducing the superheat of the discharge vapor.

(3) When the heat source and output temperatures are 150°C and 300°C, respectively, and when the injection is isobaric, the most basic two-phase compression cycle is optimized when the liquid phase is injected into the intermediate point of the compression process. There, the compression ratio standardized by the total compression ratio is 0.30, the injection mass flow ratio is 25%. Then, the rate of COP improvement is 6.3%.

(4) Two-phase injection cycle, in which the liquid partially extracted from the outlet of the condenser and the gas separated in the economizer are injected into a intermediate point of the compression process, can effectively improve the COP and reduce the discharge superheat.

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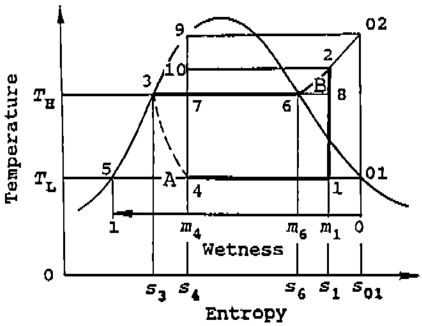

Figure 1 Two-phase Compression Heat Pump Cycle, Temperature-Entropy Diagram.

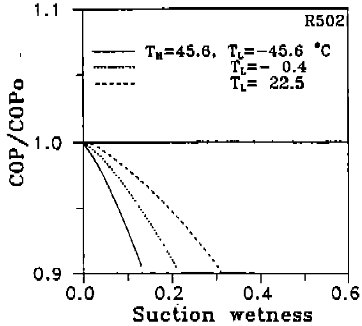

Figure 4 Change of COP with Suction Wetness, Working Fluid: R502.

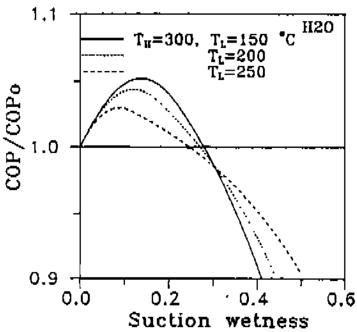

Figure 2 Change of COP with Suction Wetness, Working Fluid: Water.

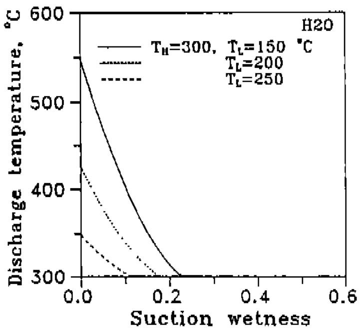

Figure 5 Change of Discharge temperature with Suction Wetness, Working Fluid:Water.

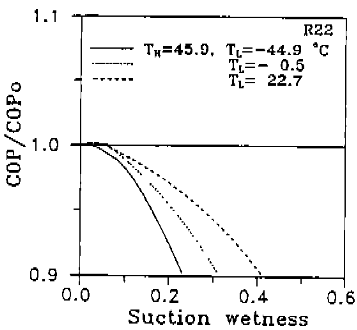

Figure 3 Change of COP with Suction Wetness, Working Fluid: R22.

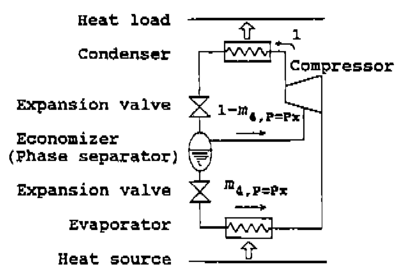

Figure 6 Economizer (Gas Injection) Cycle.

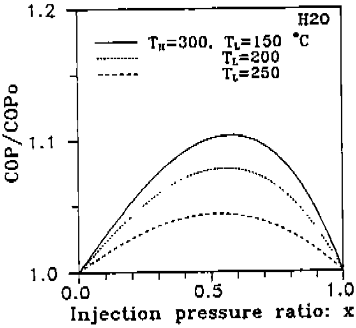

Figure 7 Change of COP with Injection Point, in Gas Injection cycle.

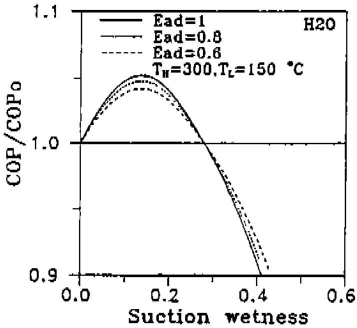

Figure 10 Change of COP Improvement with Heating in Compression Process, in Two-Phase Compression Cycle.

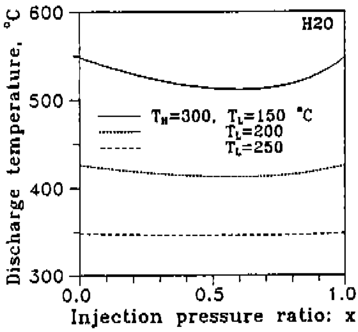

Figure 8 Change of Discharge Temperature with Injection Point, in Gas Injection cycle.

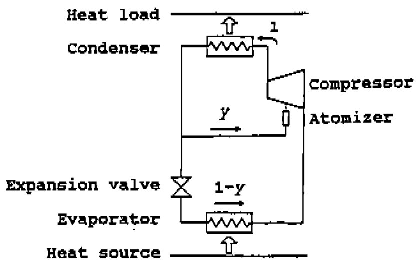

Figure 11 Liquid Injection Cycle.

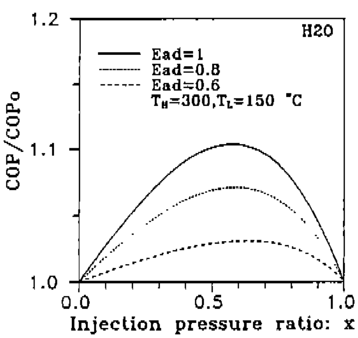

Figure 9 Change of COP Improvement with Heating in Compression Process, in Gas Injection Cycle.

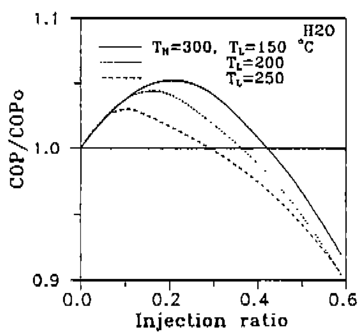

Figure 12 Change of COP with Injection Ratio, in Liquid Injection cycle.

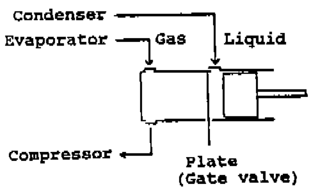

Figure 13 Model of Isochoric Mixing Process.

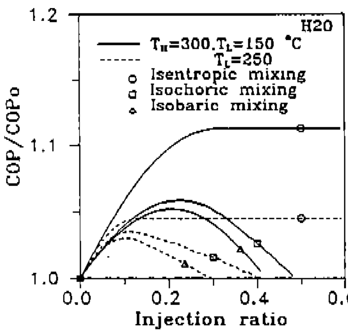

Figure 14 Change of COP Improvement with Difference in Mixing Process.

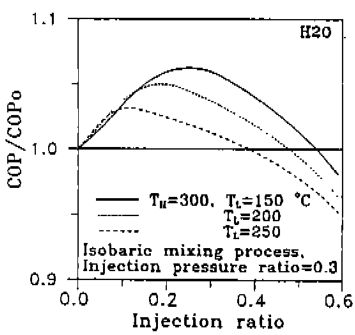

Figure 15 Change of COP with Injection Ratio, in Liquid Injection cycle, x=0.3 .

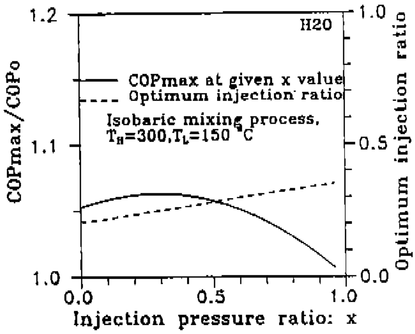

Figure 16 Change of Maximum COP and Optimum Injection Ratio with Injection Point, in Liquid Injection Cycle.

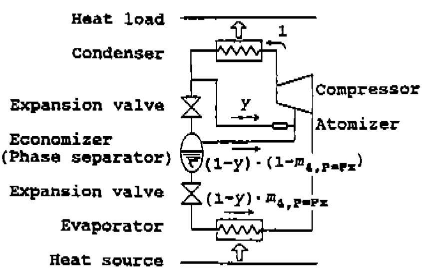

Figure 17 Two-phase Injection Cycle

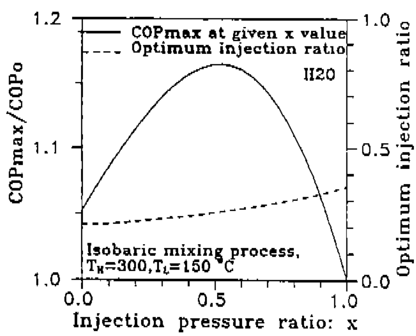

Figure 18 Change of Maximum COP and Optimum Injection Ratio with Injection Point, in Two-phase Injection Cycle.