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Experimental Study on the Performance of Refrigeration System with an Ejector

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Key words: Ejector, COP, Motive fluid, Suction fluid, Dual-evaporator refrigeration system

Abstract

Experimental investigation on the performance of dual-evaporator refrigeration system with an ejector has been carried out. In this study, a hydrofluorocarbon (HFC) refrigerant R134a is chosen as a working fluid. The condenser and two-evaporators are made as concentric double pipes with counter-flow type heat exchangers. Experiments were performed by changing the inlet and outlet temperatures of secondary fluids entering condenser, high-pressure evaporator and low-pressure evaporator at test conditions keeping a constant compressor speed. When the external conditions (inlet temperatures of secondary fluid entering condenser and one of the evaporators) are fixed, results show that coefficient of performance (COP) increases as the inlet temperature of the other evaporator rises. It is also shown that the COP decreases as the mass flow rate ratio of suction fluid to motive fluid increases. The COP of dual-evaporator refrigeration system with an ejector is superior to that of a single-evaporator vapor compression system by 3 to 6%.

Nomenclature

b	: ejector outlet
COP	: coefficient of performance
C_p	: specific heat [kJ/kgK]
h	: enthalpy [kJ/kg]
i	: mixing section inlet
j	: constant area section inlet
k	: diffuser section inlet
m	: ejector inlet of motive fluid
$\dot{m}$	: mass flow rate [kg/s]
P	: pressure [kPa]
$\dot{Q}$	: capacity [kW]
s	: ejector inlet of suction fluid
T	: temperature [°C]
$\dot{W}$	: work [kW]
x	: quality

Greek symbols

ω	: entrainment ratio

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Subscripts

b : secondary fluid (brine)
c : condenser
$comp$: compressor
$e1$: high-pressure evaporator
$e2$: low-pressure evaporator
ej : ejector
i : inlet
o : outlet
w : secondary fluid (water)

1. Introduction

At present, most of the domestic refrigerators adopt the single-evaporator refrigeration system. A domestic refrigerator is used to hold a freezer temperature at about −18℃ and food compartment temperature near 4℃, however, these two different levels of temperatures are achieved by only one evaporator. A single-evaporator, located in freezer compartment, must cover both the food and freezer compartments. Therefore, it must be operated at a lower pressure required by freezer, which is much lower than that required by food compartment. In this case, due to a high pressure difference between evaporator and condenser, the compressor work is relatively large, which results in degraded performance.

The simplest way to reduce the compressor work and to achieve better performance is to make two independent refrigeration cycles with different evaporation temperatures, which is called dual-loop cycle.[1] Pedersen et al.[2] showed that the compressor work of the dual-loop cycle using R12 decreases by 12%. Simulation results of the Bare et al.[3] showed an improvement of 19% for a dual-loop cycle using R-12. However, in spite of this advantage, the major problem with this system is its high cost. Because of the cost, interest has been given to another refrigeration cycle with two evaporators, single-loop cycle, which has only one compressor and two evaporators.

Dual-evaporator refrigeration cycle with an ejector was mainly focused on in this study, which is called ejector cycle.[4-6] This is a kind of single-loop cycle, which has two evaporators at different pressure levels and an ejector combining the outlet of two evaporators. The role of the ejector is a pre-compressior. In the ejector cycle, the inlet pressure of the compressor increases due to the pre-compression of the ejector. As a result the compressor work reduces as compared to the standard refrigeration cycle.

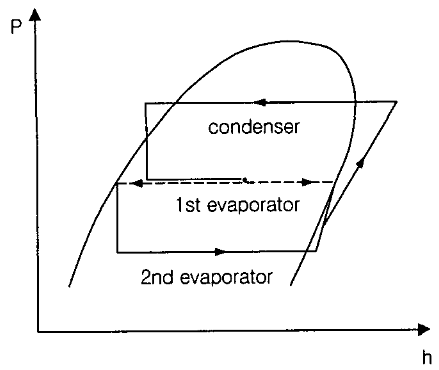
Much attention is given to dual-evaporator refrigeration cycles in these days.[4-8] If there are two-independent evaporators and the temperature of each evaporator is controlled separately, several advantages such as high energy efficiency and low power consumption for defrosting can be taken. One additional advantage is a possibility to achieve a comfortable environment in a refrigerator, because the stink and the moisture are not spread out from the food compartment to freezer compartment.

In this paper, the performance of a dual-evaporator refrigeration system with an ejector, to which the pure refrigerant can be easily adopted, is experimentally investigated using refrigerant R-134a as a working fluid.

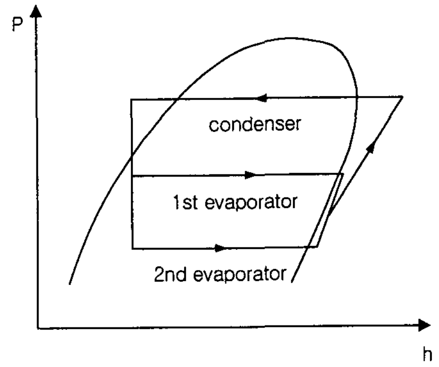
2. Dual-evaporator refrigeration system with an ejector

Two different types of ejector cycles are introduced in this study.

The first system is shown in Fig. 1(a), which is called saturated vapor driven ejector cycle. Subcooled liquid at the exit of condenser is expanded at the first expansion valve and two phase refrigerant mixture is flowing to the first evaporator (high-pressure evaporator). In the first evaporator, the refrigerant is partly evaporated and it is separated into vapor and liquid in the



(a) Saturated vapor driven ejector cycle



(b) Superheated vapor drive ejector cycle

Fig. 1 P-h diagram of superheated vapor driven ejector cycle.

separator. The saturated vapor from the separator becomes a motive fluid of the ejector and the saturated liquid in the separator is expanded at the second expansion valve to the freezer pressure. The vapor evaporated in the second evaporator (low-pressure evaporator) enters the ejector as a suction fluid. In the ejector, the motive flow and suction flow are well mixed in the mixing section of the ejector. this mixed flow goes through the precomrpession process in the constant area section and diffuser section of the ejector and this pre-compressed refrigerant is sent to the compressor. Due to the inflow of the pre-compressed refrigerant to the compressor, reduction of the compressor work and improved performance of the refrigerator are obtained.

The second system called superheated vapor driven ejector cycle is shown in Fig. 2, the cycle is almost the same with the saturated vapor driven ejector cycle except that the separator doesn't exist. After condensation, subcooled refrigerant is distributed into two streams; one is to the first evaporator (high-pressure evaporator) and the other is to the second evaporator (low-pressure evaporator). In the first evaporator, refrigerant is evaporated to a superheated vapor and this superheated vapor is used as a motive fluid in the ejector, while the refrigerant evaporated in the second evaporator becomes a suction fluid in the ejector. In the ejector, through the same processes with the saturated vapor driven ejector cycle, pre-compressed refrigerant flows to the compressor. And this inflow of pre-compressed refrigerant to the compressor makes performance of the refrigerator improved.

The saturated vapor driven ejector cycle has advantages due to the separator such as easy distribution of refrigerant to each evaporator and reliability of inflow of liquid refrigerant to second expansion valve. However, it may have be difficulty in maintaining enough vapor refrigerant to be entered to ejector in the condition of low load of high-pressure evaporator. And it is not desirable situation also that the refrigerant to the low-pressure evaporator should be refrigerant partly evaporated in the high-pressure evaporator.

On the other hand, in the superheated vapor driven ejector cycle low-pressure and high pressure evaporator can be operated separatedly. And the superheated vapor driven ejector cycle has advantages such as simple structure compared with the saturated vapor driven ejector cycle and more effective operation of the ejector due to higher energy of superheated motive fluid from the high-pressure evaporator. However, the superheated vapor driven ejector cycle has difficulty of distribution of refrigerant from the first expansion valve to the high-pressure evaporator and second expansion valve.

3. Experimental apparatus and test procedure

3.1 Ejector

An ejector is a device which entrains the low pressure suction flow by using the high pressure motive flow. Fig. 2 shows a basic structure of a standard ejector and the pressure change inside an ejector. Motive fluid (m) of relatively high pressure enters the nozzle, through which it expands to produce a low pressure at the nozzle outlet (i). The high velocity motive stream entrains the suction fluid (s) of relatively low pressure into the mixing chamber. The velocity of mixed fluid (j) is generally supersonic. Within a constant area section normal shock wave is then generally produced, creating a compression effect, and the velocity of the mixed fluid is reduced to be subsonic. Further compression of the fluid is achieved as the combined streams flow through the diffuser section.

Because the ejector has no moving or rotating part, it has many advantages such as simplicity, reliability and low cost. The ejector can be classified into four types by the flow phase of motive and suction fluid: steam-steam,

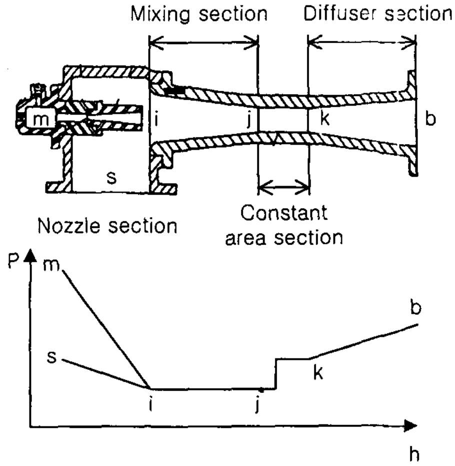


Fig. 2 A structure and pressure change inside an ejector.[9]

liquid-liquid, liquid-gas, liquid-steam ejector.[10]

In this paper, the steam-steam ejector is used in the dual-evaporator refrigeration system to reduce the compressor work by raising the pressure of the compressor suction gas.

3.2 Experimental apparatus and test conditions

Fig. 3 shows an experimental setup for dual-evaporator system with an ejector. The experimental setup is mainly composed of compressor,

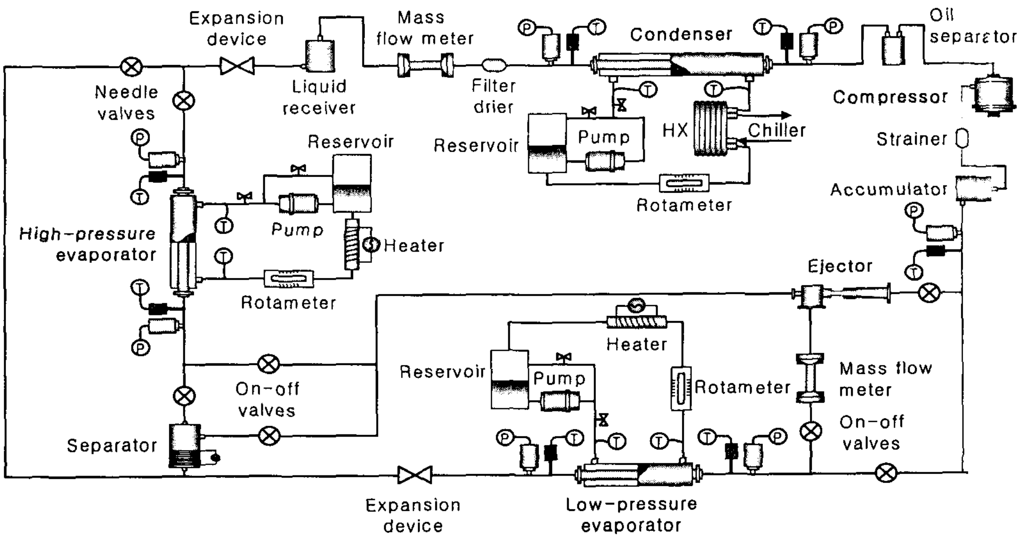


Fig. 3 Experimental setup for saturated and superheated vapor driven ejector cycle.

condenser, two expansion valves, two evaporators and ejector. The compressor is semi-hermetic reciprocating type of capacity of 1 HP with two cylinders. The accumulator is installed before the compressor in order to prevent the inflow of liquid refrigerant to the compressor. The condenser and evaporators are counter-flow-type heat exchangers with concentric dual tubes which are made of copper. Outer diameter and thickness of the heat exchangers are 15.9 mm and 1 mm for outer tube and 7.5 mm and 1 mm for inner tube respectively. And the length of the condenser, high-pressure evaporator and low-pressure evaporator are 15 m, 10 m and 9 m respectively. The secondary-fluid loop which is composed of pump, rotameter, reservoir and heat exchanger which is connected to a chiller is installed in order to control the outlet condition of the each heat exchanger. Water and the mixture of ethylene glycol and water (40/60 by volume) is used as a Working fluid of the secondary-fluid loop for the condenser and evaporator respectively. The expansion device is a metering valve, which can regulate and control the desired degree of superheat at the evaporator exit. And on/off valves are used to switch from one to the other dual-evaporator refrigeration system and two needle valves are used to control the mass flow rate of the system.

Performance tests are carried out to obtain the variation of compressor work and coefficient of performance (COP) with respect to external condition changes of the inlet temperatures of secondary fluids entering two evaporators. Table 1 shows the test conditions in this study for the performance test of dual-evaporator refrigeration system with an ejector. During the tests, degree of superheat and sub-cooling were maintained at 8℃ and the system performances are recorded when the variation of temperature, pressure and mass flow rate of the system is smaller than 0.5℃, 5 kPa and 0.2 g/s respectively.

Table 1 Test conditions in this study

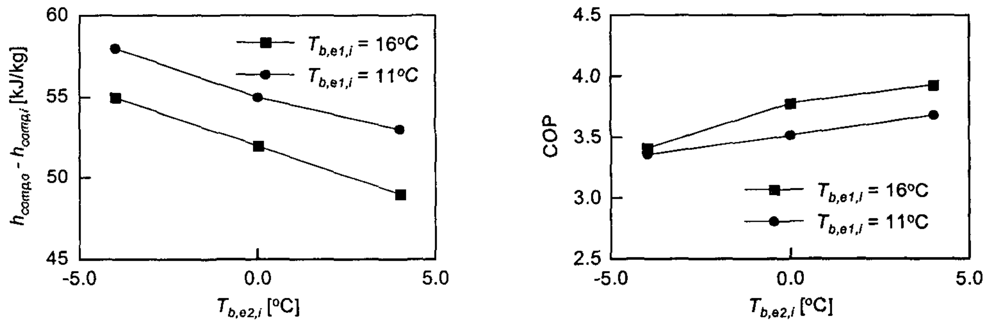
$T_{w,c}$		$T_{b,e1}$		$T_{b,e2}$
Inlet	Outlet	Inlet	Outlet	
26℃	33℃	11℃ 16℃	7℃ 12℃	−4℃, 0℃, 4℃
31℃	38℃	11℃ 16℃	7℃ 12℃	−4℃, 0℃, 4℃

4. Experimental results

4.1 Saturated vapor driven ejector cycle

4.1.1 Performance with respect to inlet temperature of secondary fluid

Fig. 4(a) shows a refrigerant enthalpy change between compressor inlet and outlet as a function of secondary fluid inlet temperature of



(a) Enthalpy change between compressor inlet and outlet (b) COP

Fig. 4 Enthalpy change between compressor inlet and outlet and COP with respect to secondary fluid inlet temperature of low pressure evaporator ($T_{w,c,i}$=26℃, $\Delta T_{w,c}$=7℃, $\Delta T_{b,e1}$=4℃).

low-pressure evaporator. In this test, the secondary fluid inlet temperatures of condenser and high-pressure evaporator remain constant and temperature difference between secondary fluid inlet and outlet is maintained at 7℃.

The enthalpy change between compressor inlet and outlet decreases with an increase of secondary fluid inlet temperature of low-pressure or high-pressure evaporator when the other evaporator temperature remains constant. This phenomenon is resulted from the energy conservation of ejector inlets and outlet, which is shown in equation (1).

$$\dot{m}_m h_m + \dot{m}_s h_s = \dot{m}_b h_b \tag{1}$$

If the mass flow rate and motive or suction flow remain constant, the enthalpy of mixed flow at ejector outlet is proportional to the enthalpy of the other flow (motive or suction flow). The higher the secondary fluid inlet temperature of high-pressure evaporator or low-pressure evaporator is, the higher the enthalpy of the motive flow becomes. Therefore, the enthalpy of the mixed flow at ejector outlet increases and the enthalpy change between compressor inlet and outlet decreases while the enthalpy of the compressor outlet remains constant.

This influences on COP and secondary fluid inlet temperature of high-pressure or low-pressure evaporator. Fig. 4(b) shows the variation of COP. Compressor work and the cooling capacity of high-pressure and low-pressure evaporator is shown in equation (2) and (3).

$$\dot{W} = \dot{m}_{comp}(h_{comp,o} - h_{comp,i}) \tag{2}$$

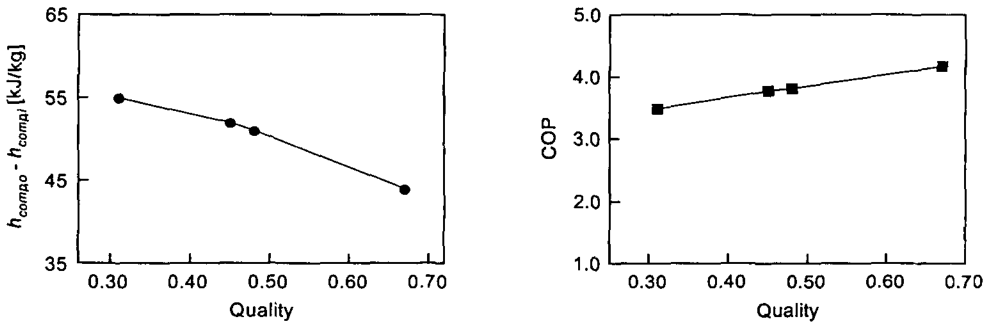
$$\begin{aligned} \dot{Q}_{e1} &= \dot{m}_{e1}\int_{T_{e1,i}}^{T_{e1,o}} C_{p,e1}\,dT \\ \dot{Q}_{e2} &= \dot{m}_{e2}\int_{T_{e2,i}}^{T_{e2,o}} C_{p,e2}\,dT \end{aligned} \tag{3}$$

Since the enthalpy change between compressor inlet and outlet represents the compressor work per unit mass, COP defined as in equation (4) increases when the secondary fluid inlet temperature of high-pressure or low-pressure evaporator gets high.

$$\mathrm{COP} = \frac{\dot{Q}_{e1} + \dot{Q}_{e2}}{\dot{W}} \tag{4}$$

4.1.2 Performance with respect to high-pressure evaporator outlet quality

Fig. 5 shows a variation of the enthalpy change between compressor inlet and outlet and COP with respect to the quality at high-pressure evaporator. As the quality of the high-pressure evaporator outlet increases, the enthalpy change between compressor inlet and outlet decreases and COP increases.



(a) Enthalpy change between compressor inlet and outlet (b) COP

Fig. 5 Enthalpy change between compressor inlet and outlet and COP with respect to quality ($T_{w,c,i}$=26℃, $\Delta T_{w,c}$=7℃, $T_{b,e1,i}$=11℃, $\Delta T_{b,e1}$=6℃, $T_{b,e2,i}$=0℃, $\Delta T_{b,e2}$=4℃).

The meaning of the quality increase at the high-pressure evaporator outlet is that the mass flow rate of vapor phase, which is related with the motive flow of ejector, increases and that of liquid phase, which is connected with the suction flow of ejector, decreases. Equation (5) shows energy balance derived from equation (1) at slightly perturbed condition by the quality increase at the high-pressure evaporator outlet.

$$(\dot{m}_m + \Delta\dot{m}_{ej}) h_m + (\dot{m}_s - \Delta\dot{m}_{ej}) h_s = \dot{m}_b (h_b + \Delta h_{ej}) \quad (5)$$

By subtracting equation (1) from equation (5), equation (6) is obtained.

$$\Delta\dot{m}_{ej}(h_m - h_s) = \dot{m}_b \Delta h_{ej} \quad (6)$$

In equation (6), since $h_m - h_s$ is always positive, Δh_{ej} should be positive when the perturbation of the mass flow rate, $\Delta\dot{m}_{ej}$ has positive value. This means that the enthalpy of ejector outlet and that of compressor inlet increases and compressor work decreases while the enthalpy of the compressor outlet remains constant as the quality at the high-pressure evaporator increases.

However, if the quality at high-pressure evaporator outlet is too high, cooling capacity of low-pressure evaporator will reduce to zero. Thus, cooling capacity can be insufficient to the load of the freezer section because of the decreased mass flow rate of the low-pressure evaporator.

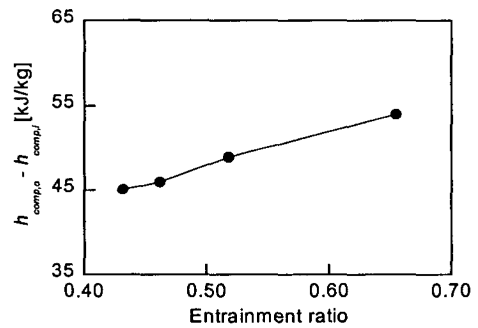
4.2 Superheated vapor driven ejector cycle

4.2.1 Performance with respect to entrainment ratio

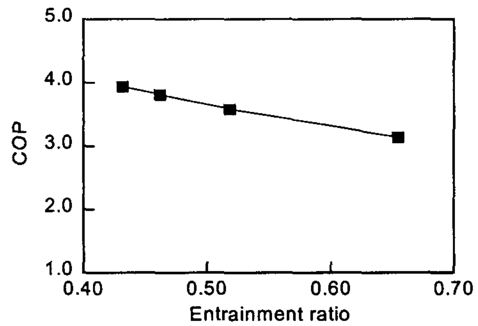
Entrainment ratio, ω, is defined as a ratio of the mass flow rate of suction flow to that of motive flow as shown in equation (7)

$$\omega = \frac{\dot{m}_s}{\dot{m}_m} \quad (7)$$

In the superheated vapor driven ejector cycle, refrigerant flow is divided into two streams after condensation. One stream is sent to the low-pressure evaporator and the other stream is sent to the high-pressure evaporator. As the mass flow rate of refrigerant through the low-pressure evaporator increases, entrainment ratio increases. In the ejector, increase of the entrainment ratio means that the perturbation of the mass flow rate has negative value in the equation (5). Therefore, by the equation (6) the enthalpy perturbation of the ejector outlet, Δh_{ej}, has negative value and the enthalpy of ejector outlet and compressor outlet decrease.



(a) Enthalpy change between compressor inlet and outlet



(b) COP

Fig. 6 Enthalpy change between compressor inlet and outlet and COP with respect to entrainment ratio ($T_{w,c,i}$=31℃, $\Delta T_{w,c}$=7℃, $T_{b,e1,i}$=16℃, $\Delta T_{b,e1}$=6℃, $T_{b,e2,i}$=0℃, $\Delta T_{b,e2}$=4℃).

This effect of the variation of entrainment ratio on the enthalpy change between compressor inlet and outlet and COP are shown in Fig. 6. The enthalpy change between compressor inlet and outlet increases and COP decreases as the entrainment ratio increases. And it is also observed that the trend showed in Fig. 6 is almost inversely proportional to that in Fig. 5. This can be explained by the inverse proportionality between entrainment ratio and quality at the high-pressure evaporator outlet in the saturated vapor driven ejector cycle. And this suggests that the entrainment ratio should be the important control parameter of both saturated vapor and superheated ejector driven ejector cycle.

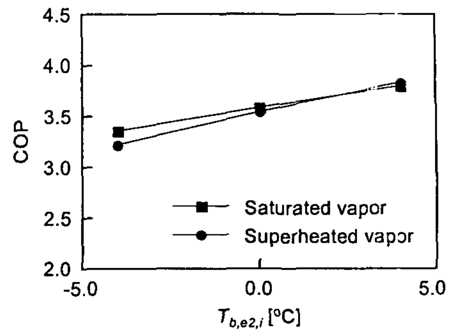


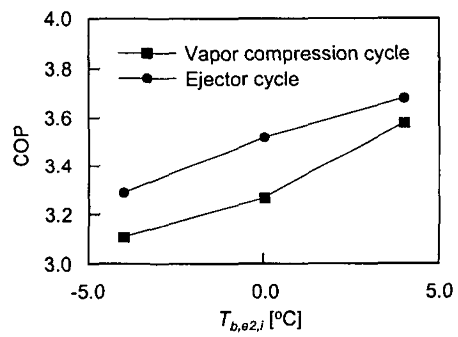
Fig. 7 COP of saturated vapor driven and superheated vapor driven ejector cycle with respect to the secondary fluid inlet temperature of low pressure evaporator ($T_{w,c,i}$=31℃, $\Delta T_{w,c}$=7℃, $T_{b,e1,i}$=16℃, $\Delta T_{b,e1}$=6℃).

4.3 Comparison of saturated vapor driven ejector cycle and superheated vapor driven ejector cycle

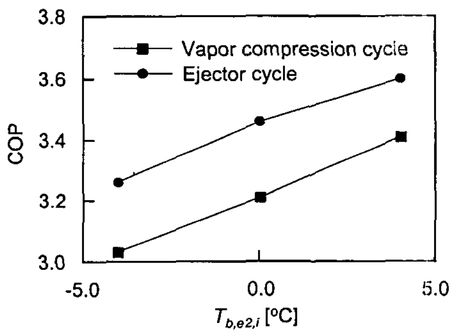
Fig. 7 compares the COP of saturated vapor driven ejector cycle and superheated vapor driven ejector cycle with respect to the secondary fluid temperature entering low-pressure evaporator. In saturated vapor driven ejector cycle, the total mass flow rate is 5.20 g/s, and the quality at the outlet of high-pressure evaporator is 0.4, in superheated vapor driven ejector cycle, the total mass flow rate is 5.65 g/s and the entrainment ratio is 0.5. COP of the saturated vapor driven ejector cycle is slightly higher when the secondary fluid temperature entering low-pressure evaporator is below 2℃ but above 2℃ the trend is reversed.

4.4 Comparison of dual-evaporator refrigeration ejector cycle and sngle-evaporator refrigeration cycle

In Fig. 8, the COP of single-evaporator vapor compression cycle is compared to those of dual-evaporator refrigeration cycle with an ejec-



(a) $T_{w,c,i}$=26℃



(b) $T_{w,c,i}$=31℃

Fig. 8 COP of ejector cycle and vapor compression cycle with respect to secondary fluid inlet temperature of low pressure evaporator ($\Delta T_{w,c}$=7℃, $T_{b,e1,i}$=11℃, $\Delta T_{b,e1}$=4℃).

tor (saturated vapor driven ejector cycle). For these two cycles, the comparison was made when the secondary fluid temperatures at the condenser inlet and at low-pressure evaporator inlet are maintained the same. The secondary fluid inlet temperature of high-pressure evaporator remains constant.

In dual-evaporator refrigeration cycle with an ejector (saturated vapor driven ejector cycle) and single-evaporator vapor compression cycle, evaporator outlet pressure (low-pressure evaporator in case of ejector cycle) is the same, but the compressor inlet pressure of ejector cycle is higher than that of vapor compression cycle due to the pre-compression by the ejector. As a result, the enthalpy difference between compressor inlet and outlet of ejector cycle is 5% lower than that of vapor compression cycle. As shown in Fig. 8 the COP of ejector cycle is superior to that of vapor compression cycle by 3 to 6%.

5. Conclusions

Experimental investigation on the performance of dual-evaporator refrigeration ejector cycle has been carried out and the conclusions of this study can be summarized as follows.

(1) In the saturated vapor driven ejector cycle, as the secondary fluid inlet temperature of low-pressure evaporator or high-pressure evaporator becomes high, the compressor work decreases and the COP increases.

(2) In the saturated vapor driven ejector cycle, the compressor work decreases and the COP increases as the quality of high-pressure evaporator outlet increases.

(3) In the superheated vapor driven ejector cycle, the compressor work increases and COP decreases as the entrainment ratio increases.

(4) COP of the saturated vapor driven ejector cycle is slightly higher than that of the superheated vapor driven ejector cycle when the secondary fluid inlet temperature of low-pressure evaporator is low.

(5) Compressor work of dual-evaporator refrigeration system with an ejector decreases by 5% compared to that of conventional single-evaporator vapor compression cycle without ejector and the COP is higher 3 to 6%.

Acknowledgments

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