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Experimental study of a series-connected two-evaporator refrigerating system with propane (R-290) as the refrigerant

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Abstract

The performance of a refrigerating system with an environment-friendly refrigerant, propane (R-290) as the refrigerant, was experimentally studied. There were two evaporators connected in series within the system under study.

The results show that with both lengths of the two capillary tubes fixed, both the mass flow rate of the refrigerant and the suction pressure of the system increase with the condensing pressure. In addition, the cooling capacity of the high-temperature evaporator decreases, but that of the low-temperature evaporator increases. As the condensing pressure is fixed and the length of the capillary tube for the high-temperature evaporator is increased while that for the low-temperature evaporator is fixed, the cooling capacity of the high-temperature evaporator increases while that of the low-temperature evaporator decreases. On the other hand, as the capillary tube for the low-temperature evaporator is lengthened while that for the high-temperature evaporator is fixed, the variations in the cooling capacity of these two evaporators reverse. The enthalpy changes of the refrigerant within the evaporators are strongly affected by the length of the high-temperature capillary tube, while the evaporating pressures are influenced mainly by the length of the low-temperature capillary tube.

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Keywords: Propane; R-290; Two-evaporator; Refrigerating system

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Nomenclature

A_i	inside heat transfer area (m ²)
A_o	outside heat transfer area (m ²)
c	specific heat (kJ/(kg K))
d	inside diameter of capillary tube (mm)
$D_{i,i}$	inside diameter of the inner tube (mm)
$D_{o,i}$	outside diameter of the inner tube (mm)
$D_{i,o}$	inside diameter of the outer tube (mm)
$D_{o,o}$	outside diameter of the outer tube (mm)
H	enthalpy (kJ/kg)
h	heat transfer coefficient (W/(m ² K))
L	length of capillary tube (m)
L_e	length of the evaporator (m)
LMTD	logarithmic-mean temperature difference (°C)
M	mass flow rate (kg/h)
P_c	condensing pressure (kPa)
P_{suc}	suction pressure (kPa)
Q	cooling capacity (W)
R	conduction resistance (K/W)
T	temperature (K)
U	overall heat transfer coefficient (W/(m ² K))
W	power input (W)

Subscripts

H	high-temperature evaporator
L	low-temperature evaporator
r	Refrigerant
t	Total
hm	heating medium
in	inlet
out	outlet

1. Introduction

It is well known that the impacts of the refrigerant on the ozone depleting potential (ODP) and the global warming potential (GWP) are determined by the chlorine and fluorine, respectively, contained in the refrigerants. Refrigerants containing chlorine or fluorine, such as R-22, HFC-134a, etc., should therefore be replaced by environment-friendly substances, containing neither chlorine nor fluorine, hydrocarbons (HCs) represent a zero ODP and an extremely low GWP solution. From technical and thermodynamic points of view, some HCs are good alternatives. The refrigerating properties of propane (R-290) are close to those of R-22. R-290 is, therefore, a

potential alternative to R-22. In addition, the compatibility of R-290 with the conventional mineral oil used in R-22 systems is very good.

The performance of the refrigerating system with HCs as the refrigerant has been investigated (e.g. [1–5]). However, the investigation was concentrated on the conventional system with only one evaporator. A refrigerating system with two or more evaporators connected in series may give better performance than that with only one evaporator. From the thermodynamic viewpoint, decreasing the temperature difference between evaporation and condensation increases the energy efficiency. The advantage may be availed with two evaporators, in which the temperature difference for the high-temperature evaporator is smaller. For example, a two-evaporator refrigerating system charged with nonazeotropic refrigerant R22/R11 (50%/50% by weight) shows a power saving of 20% compared with that of R-12 with only one evaporator [6]. Similar results were obtained by Rose et al. [7]. The computer simulation of a two-evaporator refrigerating system charged with pure and mixed refrigerants, conducted by Jung and Radermacher [8], shows a significant increase in COP. The cycle proposed by Lorenz and Meutzner [6] was modified by Simmons et al. [9,10] with some bypasses to the compressor. Again the experimental results showed that the energy saving potential is high with two evaporators in series.

It should be pointed out that although there are two evaporators in both the original cycle proposed by Lorenz and Meutzner [6] and the modified one [9,10], there is only one capillary tube in each system. Different evaporating temperatures are achieved through the nonazeotropic characteristics of the mixed refrigerants. With a single refrigerant, different evaporating temperatures can only be obtained by the use of multiple capillary tubes and evaporators. To this end, an experimental system was established to explore the performance of the two-evaporator refrigerating system with pure R-290 as the refrigerant.

2. Experimentation

Fig. 1 shows the schematic of the experimental system setup for specific purpose. The experimental apparatus can be divided into three subsystems: one refrigerant loop and two heat-exchange fluid loops. The states of the working fluids at appropriate locations were monitored with T-type thermocouples and pressure gauges as shown.

As shown in Fig. 1, the refrigerant loop is composed of a reciprocating compressor (L51B562DBL B, Bristol), a condenser, a filter–dryer, a refrigerant flow meter, a sight glass, an electromagnetic valve, two capillary tubes, two evaporators, and some valves. A capillary tube is arranged before each evaporator.

The power input of the three-phase, 220 V, reciprocating compressor is controlled by the frequency converter. The output of the converter can be adjusted between 0 and 120 Hz, but the range for stable operation is from 40 to 80 Hz. The lubricant of the compressor is a 150 SSU mineral oil. The condenser is a finned-tube heat exchanger. The condensation of the refrigerant in the tube is effected by the cooling air, which is induced by the action of a fan inside the condenser. The airflow rate is controlled by the rotating speed of the fan through a voltage transformer.

Both evaporators are of double-tube type in which the refrigerant flows in one direction through the inner tube while the heating medium flows in the opposite direction through the annular space between the inner and outer tubes. The heating medium of the high-temperature

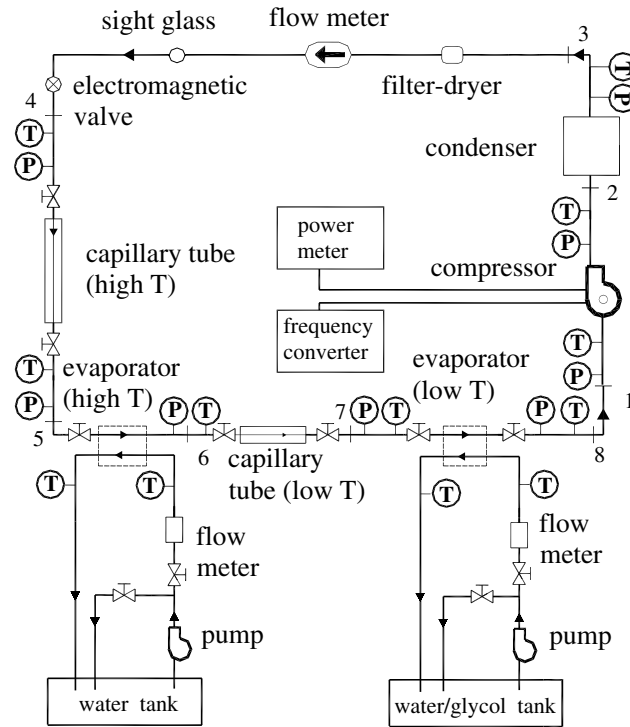


Fig. 1. Schematic diagram of the experimental system.

evaporator is water while that of low-temperature evaporator is water/glycol solution to avoid icing in the annular space between the tubes.

The selection of the capillary tubes involves some preliminary calculations and a trial-and-error test procedure for appropriate flow rates and operating temperatures. The cooling capacity of the low-temperature evaporator, Q_L , may cover that of freezer of small to medium size. Note that the capillary tube for the high-temperature evaporator is not only smaller but also longer than that for the low-temperature counterpart. This situation is induced by the effect that part of the refrigerant flowing within the high-temperature capillary tube is in liquid state, while that within the low-temperature capillary tube is all in two-phase. The evaporators and the capillary tubes are all made of copper and are all insulated outside. The dimensions of the capillary tubes and the evaporators are listed in Table 1.

Table 1
Dimensions of capillary tubes and evaporators

d_H (mm)	d_L (mm)	L_e (m)	$D_{i,i}$ (mm)	$D_{o,i}$ (mm)	$D_{i,o}$ (mm)	$D_{o,o}$ (mm)	
1.1	1.4	2	8.7	9.525	14.85	15.875	
L_H (m)			L_L (m)				
1.5	1.75	2	2.25	0.2	0.4	0.6	0.8

Both the heat-exchange loops are composed of a pump, a flow meter, and an electrically heated unit within the tank. Depending on the operation conditions, the refrigerant within the high-temperature evaporator is controlled between 15 and 20 °C, while that within the low-temperature evaporator is below -10 °C.

T-type thermocouples with monitor (APL7000&S.B767, Jenco) with accuracy ± 0.1 °C and analogical pressure meters (316 SS TUBE&SOCKET, Imperial) with accuracy $\pm 1\%$ of the reading are used. Both the temperatures and pressures of the refrigerant before and after the compressor, both capillary tubes, and both evaporators are monitored. In addition, the temperatures of the water and the water/glycol solution of the heat-exchange loops are also recorded. The flow rates of the refrigerant, water, and water/glycol solution are measured with flow meters. The accuracy of the refrigerant flow meter (102-5T, Macmillan) is $\pm 3\%$ in the range of the reading.

The controlled variables are the condensing pressure, the frequency of compressor, and the lengths of both capillary tubes, while the flow rates of both heating media are kept constant.

3. Results and discussion

3.1. Pressure-enthalpy diagram

Based on some test data and the thermodynamic properties of the refrigerant [11] and the heating media, the state of the refrigerant at different locations along the system is shown in Fig. 2,

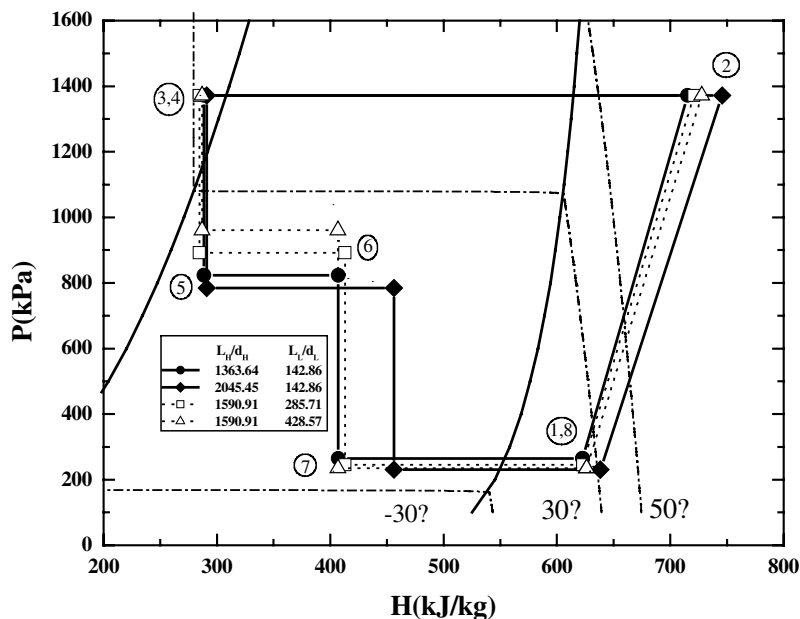


Fig. 2. P-h diagram of the refrigerant within the system and the effect of lengths of capillary tubes on states of refrigerant ($P_c = 1372$ kPa).

in which the encircled numbers indicate the corresponding locations as shown in Fig. 1. The details of Fig. 2 will be discussed later. To determine the states 6 and 7, the following equations for heat transfer are adopted:

$$Q_r = M_r \cdot (H_{r,out} - H_{r,in}) \tag{1}$$

$$Q_{hm} = M_{hm} \cdot c_{hm} \cdot (T_{hm,in} - T_{hm,out}) \tag{2}$$

Eq. (1) gives the actual heat absorbed by the refrigerant. As both evaporators are insulated, Q_r is taken as equal Q_{hm} . $H_{r,in}$ within the high-temperature evaporator, i.e. H_5 , equals H_4 which is achievable through the measured P_4 and T_4 . $H_{r,out}$ within the high-temperature evaporator, i.e. H_6 , can thus be obtained through Eq. (1). The location of point 6 can then be fixed by H_6 and the measured P_6 . Knowing state 6, state 7 can be allocated by the measured P_7 and the fact that $H_7 = H_6$. Similar situations occur for all other running conditions in the study.

3.2. Effect of condensing pressure

Fig. 3 shows the variation of the mass flow rate of the refrigerant (M_r) and the suction pressure (P_{suc}) with the condensing pressure (P_c). It can be seen that both M_r and P_{suc} increase with P_c for different length of high-temperature capillary tube, L_H . For a given L_H , the total pressure drop available for the refrigerant flowing through the system increases with P_c . However, the pressure ratio across the compressor, P_c/P_{suc} , varies with M_r , which is a characteristic of the reciprocating compressor. Therefore P_{suc} increases with P_c . On the other hand, for a given P_c , M_r decreases with L_H since more pressure drop is required for a longer capillary tube.

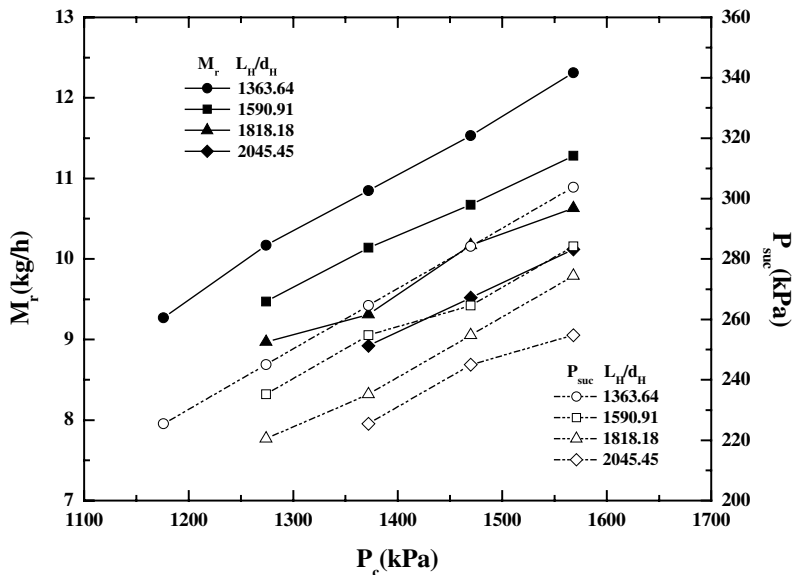


Fig. 3. Effects of condensing pressure on mass flow rate of refrigerant and suction pressure ($L_L/d_L = 142.86$).

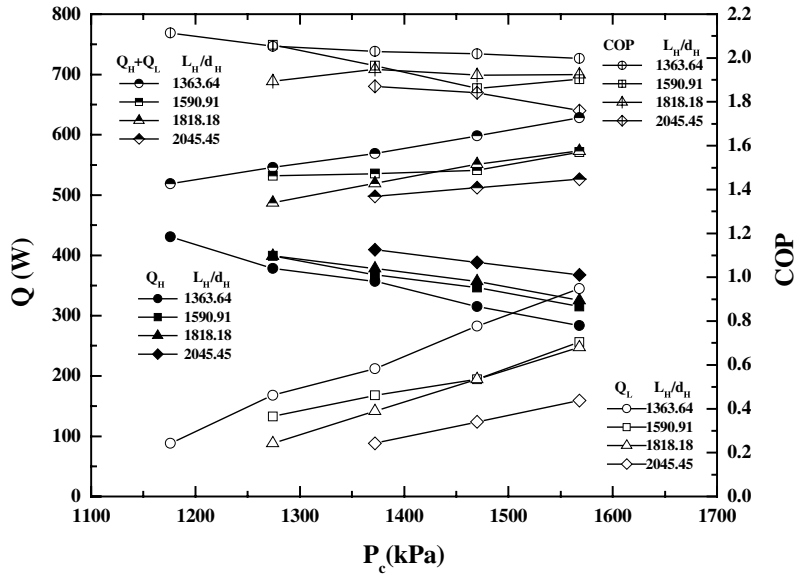


Fig. 4. Effects of condensing pressure on cooling capacities and COP ($L_L/d_L = 142.86$).

The effect of P_c on the cooling capacity of the two evaporators and COP is shown in Fig. 4. The heat-exchange rate between the working media within an evaporator may be obtained with Eqs. (1) and (2). However, it may also be expressed as

$$Q = (U \cdot A) \cdot (\text{LMTD}) \tag{3}$$

where the products of the overall heat transfer coefficients and the heat transfer areas and LMTD are defined as

$$(U \cdot A) = \left(\frac{1}{h_r \cdot A_i} + R + \frac{1}{h_{hm} \cdot A_o} \right)^{-1} \tag{4}$$

$$(\text{LMTD}) = \frac{(T_{hm,out} - T_{r,in}) - (T_{hm,in} - T_{r,out})}{\ln \left(\frac{(T_{hm,out} - T_{r,in})}{(T_{hm,in} - T_{r,out})} \right)} \tag{5}$$

Q and LMTD in the high-temperature evaporator are taken as equal Q_H and LMTD_H , respectively, while those in the low-temperature evaporator are taken as equal Q_L and LMTD_L . $T_{r,in}$ in the high- and low-temperature evaporators are the saturation temperatures of the refrigerant corresponding to the measured P_5 and P_7 , respectively, while $T_{r,out}$ in the low-temperature evaporator is measured directly.

In the study, the conduction resistance, R , is fixed. In addition, the heat transfer coefficients of the heating media, h_{hm} , is more or less constant since the flow rates of the heating media are kept constant and the variation in the temperatures of the heating media entering the evaporator is negligible. It is clear that both Q_H and Q_L are affected by LMTD and the heat coefficient of the refrigerant, h_r , which is dominated by M_r .

Note that the average quality of the refrigerant in the high-temperature evaporator is lower than that in the low-temperature counterpart as shown in Fig. 2. With the criteria suggested by Shin et al. [12], the phase change in the high-temperature evaporator is thus mainly in the nucleate boiling region, while that in the low-temperature evaporator is mainly in the film boiling region. It has been pointed out that for nucleate boiling h_r does not change with M_r , while for film boiling, h_r increases with M_r [12]. Q_H is, therefore, dominated completely by $LMTD_H$, while Q_L is affected by $h_{r,L}$ along with $LMTD_L$.

With L_H and L_L fixed, the refrigerant enters each evaporator with higher pressure when P_c is increased. Therefore, both $LMTD_H$ and $LMTD_L$ decrease with increasing P_c . The decrease of Q_H with P_c shown in Fig. 4 is then induced entirely by the decreasing $LMTD_H$. On the other hand, the increase of Q_L with P_c indicates that the effect of $h_{r,L}$ dominates that of $LMTD_L$. It is noticeable that the total cooling capacity of the two evaporators, Q_t , increases with P_c as shown in Fig. 4.

The variation of COP with P_c is shown in Fig. 4. COP is defined as

$$COP = \frac{Q_t}{W_r} \quad (6)$$

where

$$W_r = M_r \cdot \Delta H_{12} = M_r \cdot (H_2 - H_1) \quad (7)$$

The specific enthalpies, H_1 and H_2 , are determined through the states of the refrigerant just before and after the compressor, i.e. states (P_1, T_1) and (P_2, T_2) . Note that since the study is concentrated on the effect of two evaporators with individual capillary tubes, isentropic compression is assumed in obtaining H_2 . The increases of M_r , Q_t and W_r with P_c are shown in Figs. 3–5, respectively. The net effect of Q_t and W_r makes COP of the system also decrease slightly with P_c .

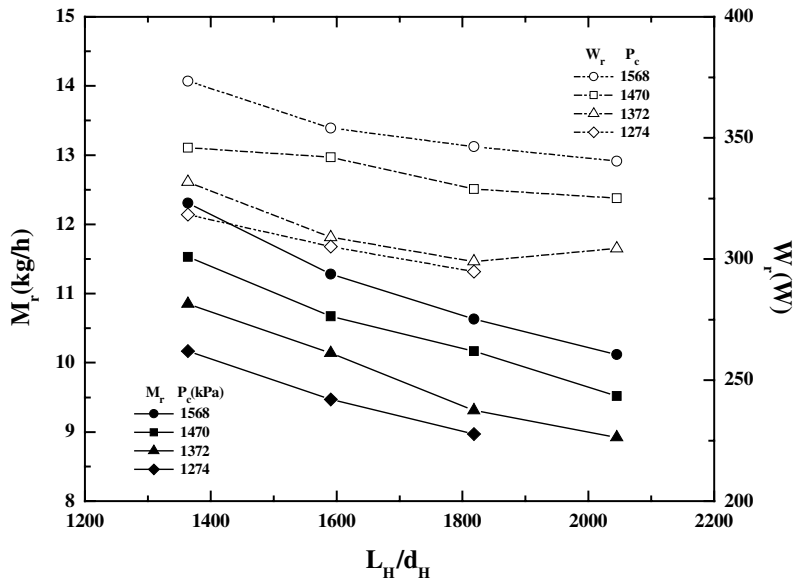


Fig. 5. Effects of length of capillary tube for high-temperature evaporator on mass flow rate of refrigerant and compression work ($L_L/d_L = 142.86$).

3.3. Effect of the length of capillary tubes

The physical properties of the refrigerant at the inlet of the high-temperature capillary tube are more or less the same for different lengths of capillary tubes as shown in Fig. 2. M_r is therefore affected mainly by the length of capillary tube. Fig. 5 shows that for a given P_c , both M_r and W_r decrease with increasing L_H . Similar variation in M_r versus L_L can be found in Fig. 6. However, W_r seems to increase with L_L at high P_c . The increase of M_r versus P_c is discussed in the previous subsection. For the case of high-temperature capillary tube, higher frictional loss is expected for longer tube, so M_r decreases with L_H . For the case of low-temperature capillary tube, a check has been made to see whether the flow is choked with the criterion suggested by Pate and Tree [13]. The results indicate that indeed choke occurs within the low-temperature capillary tubes for all of the four different P_c . With choke occurring at the exit of the tube, the flow speed of the refrigerant there is more or less fixed. Lower density of the refrigerant at the exit of the low-temperature capillary tube can thus be expected at lower P_{suc} induced by increasing L_L . Therefore, M_r decreases with L_L . On the other hand, ΔH_{12} increases with L_H . Since W_r is a product of ΔH_{12} and M_r , its variation versus L_H is thus more gradual than that of M_r . It can be seen from Fig. 6 that a similar phenomenon occurs for the variation of W_r versus L_L .

For a given P_c , Q_H increases, while Q_L decreases with L_H as shown in Fig. 7. Q_H decreases and Q_L increases with L_L as shown in Fig. 8. As explained earlier, $LMTD_H$ increases with L_H and $h_{r,H}$ is not affected by M_r . On the other hand, $h_{r,L}$ decreases with decreasing M_r . However, when L_H is increased, the increasing $LMTD_L$ makes heat transfer higher. It seems that the effect of $h_{r,L}$ on Q_L is stronger than that of $LMTD_L$.

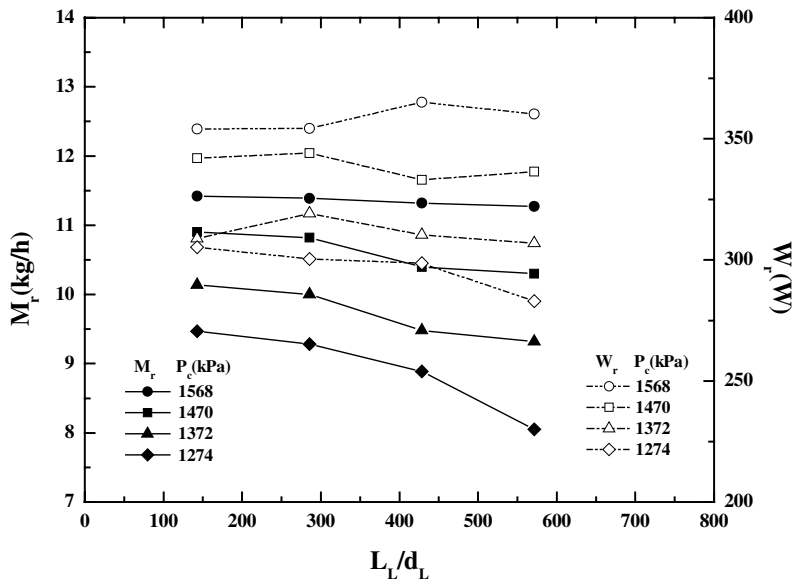


Fig. 6. Effects of length of capillary tube for low-temperature evaporator on mass flow rate of refrigerant and compression work ($L_H/d_H = 1590.91$).

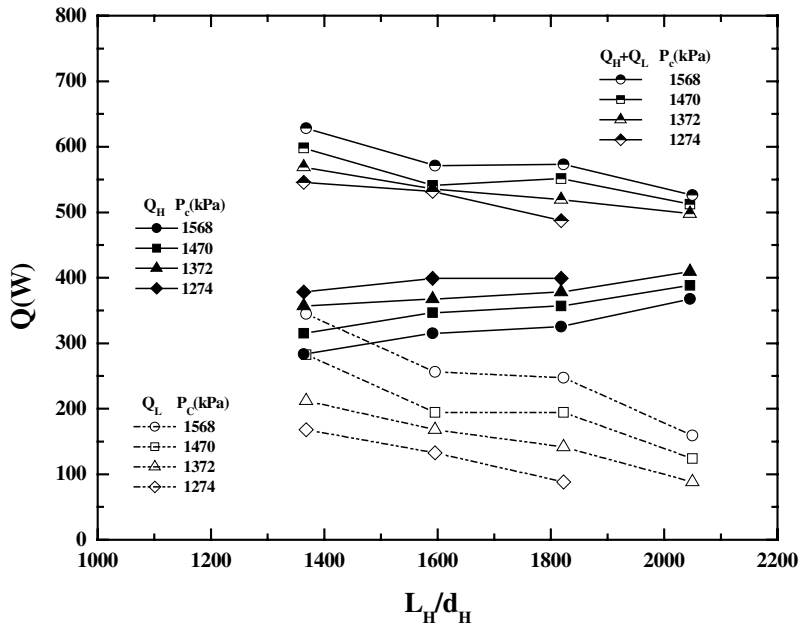


Fig. 7. Effects of length of capillary tube for high-temperature evaporator on cooling capacities ($L_L/d_L = 142.86$).

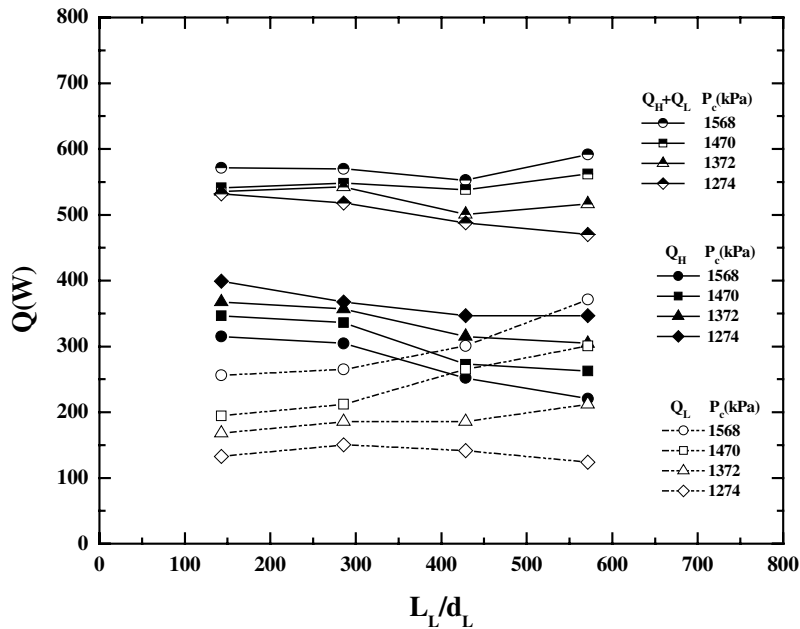


Fig. 8. Effects of length of capillary tube for low-temperature evaporator on cooling capacities ($L_H/d_H = 1590.91$).

For a given L_L , the increasing L_H does not lower the pressure in the high-temperature evaporator, P_5 shown in Fig. 2. For an increase of 50% in L_H (from 1.5 to 2.25 m), the pressure drop

induced by the high-temperature capillary tube increases by 39.2 kPa, while that by the low-temperature capillary tube decreases by 4.9 kPa. The relative changes of the pressure drops in comparing with their own original values within the high- and low-temperature capillary tubes are thus +7.1% and −0.9%, respectively, while the total pressure difference between states 4 and 7 increases by only 3.1%. The decreasing M_r with the increasing L_H may be the main reason for such a small increase in pressure drop. However, the effect of increasing L_L on the pressures of states 5 and 6 is more dramatic. With L_H fixed, an increase of 50% in L_L (from 0.4 to 0.6 m) enlarges the pressure drop induced by the high- and low-temperature capillary tubes by −14.3% and 12.1%, respectively. However, the total pressure difference increases by only 0.9%. It seems that the changes of the pressure difference in the individual capillary tubes induced by varying L_L are more than those by varying the high-temperature counterpart.

Although lengthening L_H does not induce much pressure drop between states 5 and 6, it does bring about an apparent increase in the enthalpy of state 6 as shown in Fig. 2. Compared with the original enthalpy change in the high- and low-temperature evaporators, the increases induced by lengthening L_H by 50% are 39.5% and −15.5%, respectively, while those induced by lengthening L_L are −6.9% and 4.7%, respectively. The details of the variations may be worthy of further investigation.

4. Conclusions

With P_c , L_H and L_L as the varying factors, the following conclusions may be drawn from the experimental study of the series-connected, two-evaporator refrigerating system with R-290 as the refrigerant.

1. Both M_r and P_{suc} increase with P_c for fixed L_H and L_L . In addition, Q_H decreases but Q_L increases with P_c .
2. With other factors fixed, Q_H increases with L_H , while Q_L and M_r decrease. On the other hand, Q_L increases with L_L , while Q_H and M_r decrease.
3. Q_H and Q_L are mainly affected by the two effects: h_r and LMTD. For the case of the high-temperature evaporator, LMTD is the dominant factor. However, h_r also plays an important role in the case of the low-temperature evaporator.
4. The enthalpy changes of the refrigerant within the evaporators are strongly affected by L_H . However, the evaporating pressures within the evaporators are influenced mainly by L_L .

The variations of M_r , W_r , Q_H , Q_L and COP versus L_H , L_L and P_c studied in this work may be applied to the design of a two-evaporator refrigerating system. However, the operating ranges of P_c and the dimensions of the capillary tubes may be enlarged for clearer understanding of the effect of these factors. In addition, works on other environment-friendly refrigerants or the blends of them are also worthy of further investigation.

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