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Investigation of application of suction line heat exchanger in R290 air conditioner with small diameter tube

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ABSTRACT

The use of small diameter copper tube is an effective way to reduce refrigerant charge for R290 air conditioner, but may reduce system performance. To improve the performance of R290 air conditioner with small diameter copper tube, this paper presents an investigation of application of a suction line heat exchanger. A theoretical analysis is proposed to investigate the effect of the suction line heat exchanger on capacity and coefficient of performance, and a simulation based analysis is processed to investigate the effect of suction line heat exchanger on refrigerant charge of condenser. To verify the results of analysis, experiments on a prototypeR290 air conditioner are carried out, and the results show that the suction line heat exchanger improves the cooling capacity by 5.3%and system efficiency by 4.5%, and reduces the refrigerant charge by 6%, which agree well with that of the theoretical and simulation based on analysis.

1. INTRODUCTION

R290 is a potential refrigerant replacing existing R22 systems because of zero Ozone Depression Potential (ODP) and virtually zero Global Warming Potential (GWP). However, using R290 has great risk of firing due to its inflammability. Promoting the use of small diameter copper tube (5 mm or even smaller one) is an effective way to obviously reduce refrigerant charge and thus avoid the risk of firing (Ding et al. 2012). However, employing small diameter copper tube will increase pressure drop and consequently reduce system performance. (Wei et al. 2007; Ding et al. 2009). In order to promote the application of R290, it is necessary to improve the performance of air conditioner with small diameter copper tube.

The performance of air conditioner with small diameter copper tube can be improved by following ways: 1) optimizing refrigerant circuitry to reduce pressure drop; (2) optimizing fin configurations for 5 mm diameter copper tube to enhance air side heat transfer; 3) optimizing distributor to achieve a better refrigerant flow distribution; 3) applying suction line in the heat exchanger. For refrigerant circuitry optimization, the investigation shows that using simulation-based design method is an effective way to optimize refrigerant circuitry and to get a good compromise of heat exchange capacity and pressure drop of small diameter tube heat exchanger (Ding et al. 2012). For fin configuration optimization, a design principle for heat exchanger with smaller diameter tubes has been proposed by Wu et al. (2012), and it indicates that the air conditioner with 5mm diameter tubes can get good performance. For distributor optimization, the design method has been studied experimentally among different structures, and the optimization principle has been proposed by Gao et al. (2013). To the best of the authors' knowledge, there is no researches on application of suction line heat exchanger (SLHX) in R290 air conditioner with small diameter copper tube.

A SLHX employs the low temperature refrigerant in suction line to cool down the refrigerant upstream expansion value. It is widely used in refrigeration system, such as refrigerator, freezer and etc. (Ding et al. 2007, Lin et al. 2011, Islamoglu et al 2005, Gholap et al 2007). However, the SLHX is seldom used in air conditioner with HFC refrigerant (eg. R22, R410A), because it will make the air conditioner circuits much more complicated but only has very limited improvement on system performance. R290 has low discharge temperature compared with HFC refrigerants used in air conditioner (Park et al. 2007), so it has more potential to improve system performance while employing SLHX in R290 air conditioner. Further, the SLHX can increase the inlet temperature of condenser due to higher suction temperature of compressor, resulting in a potential to reduce refrigerant charge of whole system. As a result, it is valuable to investigate the application of SLHX in a R290 air conditioning system.

The purpose of this paper is to investigate the system performance and refrigerant charge while using a SLHX in a R290 air conditioner with small diameter tube, including the theoretical system cycle analysis, simulation based analysis of refrigerant charge in condenser while using SLHX and experimental investigation.

2. THEORETICAL SYSTEM CYCLE ANALYSIS

2.1 Refrigerant cycle with and without SLHX

SLHX is a heat exchanger which is added between the condenser and evaporator in air conditioner. With the aids of SLHX, the refrigerant came out from condenser is further cooled down by the refrigerant from the outlet of evaporator. The refrigerant cycle with and without SLHX for R290 air conditioner are shown in Fig. 1. The SLHX heat exchanger increases the subcooling and superheat. The cooling and heating capacities and their COPs of air conditioner without SLHX is evaluated by Equation (1), and with SLHX is evaluated by Equation (2). In Equations (1) & (2), the compressor power is evaluated by the ten coefficient equation provided by compressor manufacturer. The superheat and subcooling of the refrigerant cycle without SLHX (see point #1 and point #3 in Fig.1) are 5 ^{o}C and 10 ^{o}C respectively.

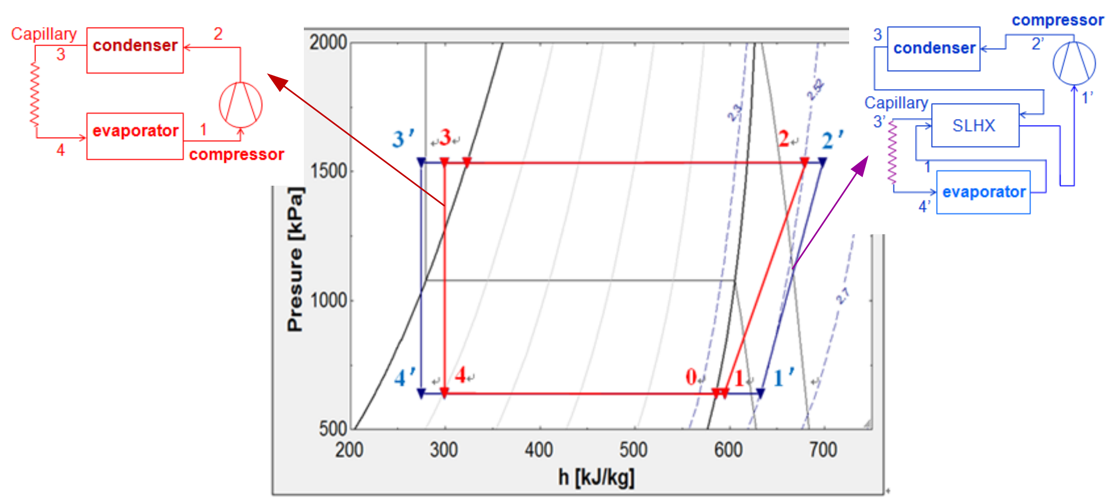


Figure 1: Refrigerant cycle w/o SL HX for R290 air conditioner

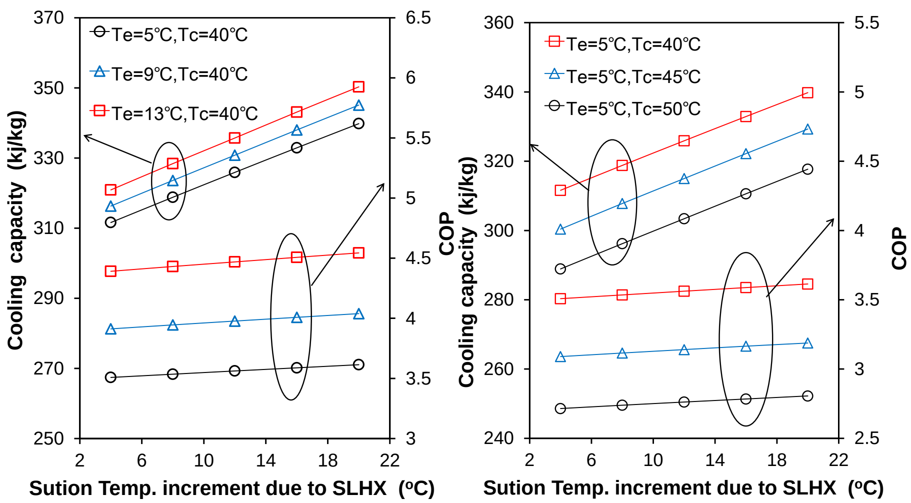
$$\begin{cases} COP_{\text{cooling}} = \dfrac{h_1 - h_4}{W}, \quad EER_{\text{heating}} = \dfrac{h_2 - h_3}{W} \\ Q_{\text{cooling}} = h_1 - h_4, \quad Q_{\text{heating}} = h_2 - h_3 \end{cases} \tag{1}$$

$$\begin{cases} COP_{\text{SLHX,cooling}} = \dfrac{h_1 - h_4{}'}{W}, COP_{\text{SLHX,heating}} = \dfrac{h_2{}' - h_3}{W} \\ Q_{\text{SLHX,cooling}} = h_1 - h_4{}', Q_{\text{SLHX,heating}} = h_2{}' - h_3 \end{cases} \tag{2}$$

Where, *Q* is the capacity; *COP* is the coefficient of performance; *h* is the specific enthalpy; *W* is energy consumption; subscript cooling and heating means cooling condition and heating condition respectively; subscript 1, 2, 3, 4 is the point in the refrigerant cycle, as shown in Fig. 1.

2.2 The effect of SLHX on system performance under cooling model

In a real air conditioning system, the evaporating temperatures under cooling condition are generally varied from 5 ºC to 13 ºC, and the condensing temperatures are varied from 40 ºC to 50 ºC. Thus, such ranges of working conditions are applied in present study in order to investigate the effect of SLHX on system performance under cooling model. Furthermore, when the effect of evaporating temperature is investigated, the condensing temperature is fixed at 40 ºC, and when the effect of condensing temperature is studied, the evaporating temperature is fixed at 5ºC.



(a) The effect at different Evap. Temp. (b) The effect at different Evap. Temp**.**

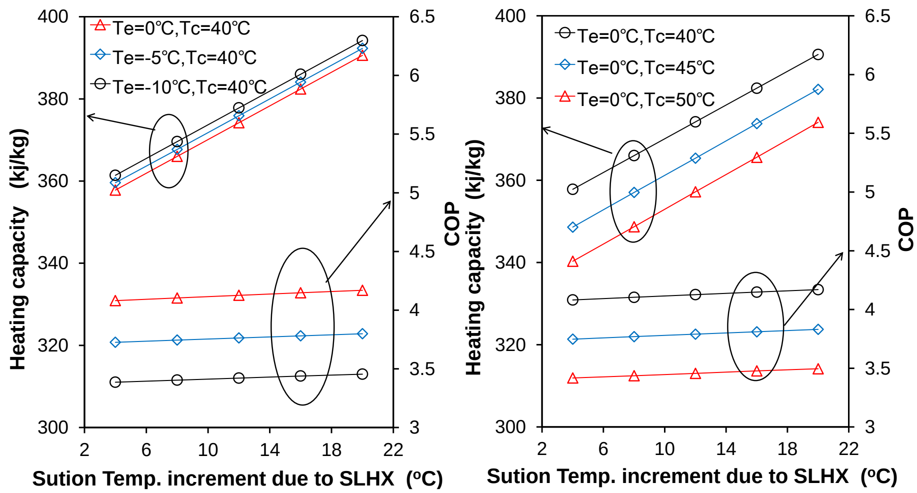
Figure2: The effect of SLHX on system performance under cooling model

Figure 2 shows the effect of SLHX on the air conditioning system under different working temperatures in cooling condition. It shows that as the outlet suction line temperature increment due to the heat exchange of SLHX increases from 4 ºC to 20 ºC, the cooling capacity and COP improves linearly. The maximum improvement for cooling capacity and COP approaches to 12% and 4%, respectively. Further, this trend keeps the same under different evaporating temperatures (see Fig. 2(a)) and condensing temperatures (see Fig. 2(b)).Based on above results, we can say that SLHX theoretically has ability to improve the performance of R290 air conditioner under cooling condition.

2.3 The effect of SLHX on system performance under cooling model

In a real air conditioning system, the evaporating temperatures under heating condition are generally varied from -10 ºC to 0 ºC, and the condensing temperatures are varied from 40 ºC to 50 ºC. Thus, such ranges of working conditions are applied in present study in order to investigate the effect of SLHX on system performance under heating condition. Furthermore, when the effect of evaporating temperature is investigated, the condensing temperature is fixed at 40 ºC, and when the effect of condensing temperature is studied, the evaporating temperature is fixed at 0ºC.

Figure 2 shows the effect of SLHX on the air conditioning system under different working temperatures in heating condition. It shows that as the outlet suction line temperature increment due to the heat exchange of SLHX increases from 4 ºC to 20 ºC, the heating capacity and COP improves linearly. The maximum improvement for heating capacity and COP approaches to 11.7% and 4%, respectively. Further, this trend keeps the same under different evaporating temperatures (see Fig. 3(a)) and condensing temperatures (see Fig. 3(b)).Based on above results, we can say that the SLHX theoretically has the ability to improve the performance of R290 air conditioner under the heating condition.



(a) The effect at different Cond. Temp. (b) The effect at different Evap. Temp.

Figure 3: The effect of SLHX on system performance under heating model

3. SIMULATION BASED ANALYSIS OF REFRIGERANT CHARGE AND SUBCOOLING IN CONDENSER W/O SLHX

Most of refrigerant in air conditioner is stored in condenser, and refrigerant charge is a key factor in R290 system. This section employs a heat exchanger simulation model to investigate the changes of refrigerant charge in condenser when SLHX is applied.

3.1 Heat exchanger simulation model

In the present study, Liu's distributed-parameter model based on graph theory (Liu et al. 2004) is adopted to predict heat exchanger performance. Liu's model is a three dimensional distributed-parameter model, and it has the ability to simulate heat exchanger or heat exchanger combinations of different tube diameter and structure with a high accuracy. The predicted heat exchange capacity of Liu's model agree with experimental ones within a maximum error of ±10% (Liu et al., 2004).

Liu's model divides heat exchanger into several control volumes along length, width and height direction, as shown in Figure 4. Each single control volume includes three objects (i.e., refrigerant, air and fin-tube), and the governing equations of each object are established. The governing equations of refrigerant include energy equation and momentum equation as shown in Equations (2) and (3), respectively; the governing equations of air include energy equation and momentum equation as shown in Equations (4) and (5), respectively; the governing equation of fin-tube is energy balance equation as shown in Equation (6).

$$Q_r = M_r(h_{r,out} - h_{r,in}) = \alpha_r A_i \left(\frac{T_{r,in} + T_{r,out}}{2} - T_{wall} \right) \tag{3}$$

$$\Delta p_r = \Delta p_{r,f} + \Delta p_{r,acc} + \Delta p_{r,g} \tag{4}$$

$$Q_a = M_a(h_{a,out} - h_{a,in}) = \alpha_a A_o \left(\frac{T_{a,in} + T_{a,out}}{2} - T_{wall} \right) \tag{5}$$

$$\Delta p_a = \frac{G_{a,\max}^2}{2\rho_{a,i}} \left[\frac{A_o \rho_{a,in}}{A_c \rho_{a,m}} f_a + (1+\sigma^2)\left(\frac{\rho_{a,in}}{\rho_{a,m}} - 1 \right) \right] \tag{6}$$

$$Q_r + Q_a + Q_{front} + Q_{back} + Q_{top} + Q_{bottom} = 0 \tag{7}$$

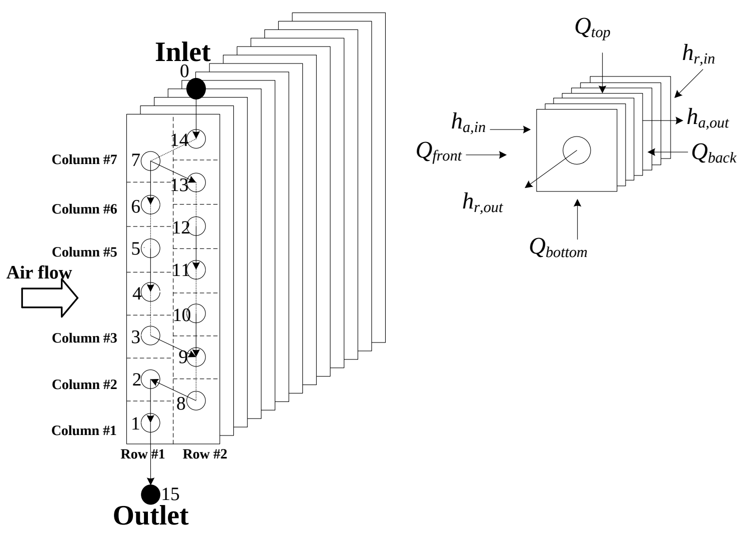


Figure 4: Schematic diagram of heat exchanger and a single control volume

where, Q_r is heat exchange of refrigerant side; α_r is heat transfer coefficient of refrigerant side; A_i is heat transfer area of refrigerant side; $T_{r,in}$ and $T_{r,out}$ are inlet and outlet temperature of refrigerant, respectively; T_{wall} is tube wall temperature; Δp_r is pressure drop of refrigerant side; $\Delta p_{r,f}$ is frictional pressure drop; $\Delta p_{r,acc}$ is acceleration pressure drop; $\Delta p_{r,g}$ is the pressure drop caused by gravity; Q_a is heat exchange of air side; α_a is heat transfer coefficient of air side; A_o is heat transfer area of air side; $T_{a,in}$ and $T_{a,out}$ are inlet and outlet dry bulb temperature, respectively; Δp_a is

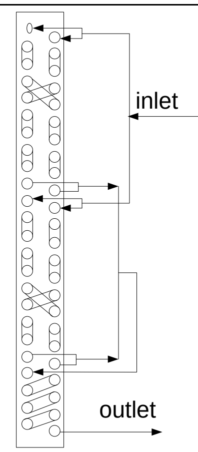
pressure drop of air side; $G_{a,max}$ is air mass flux at minimum cross-sectional area; f_a is friction factor of air; σ is contraction ratio of cross-sectional area; Q_{front}, Q_{back}, Q_{top} and Q_{bottom} are heat conductions through fins from front row, back row, upper column and bottom column, respectively.
The void fraction model (Zivi et al. 1979) is employed to calculate the refrigerant charge in the simulation model.

3.2 Results and discussion

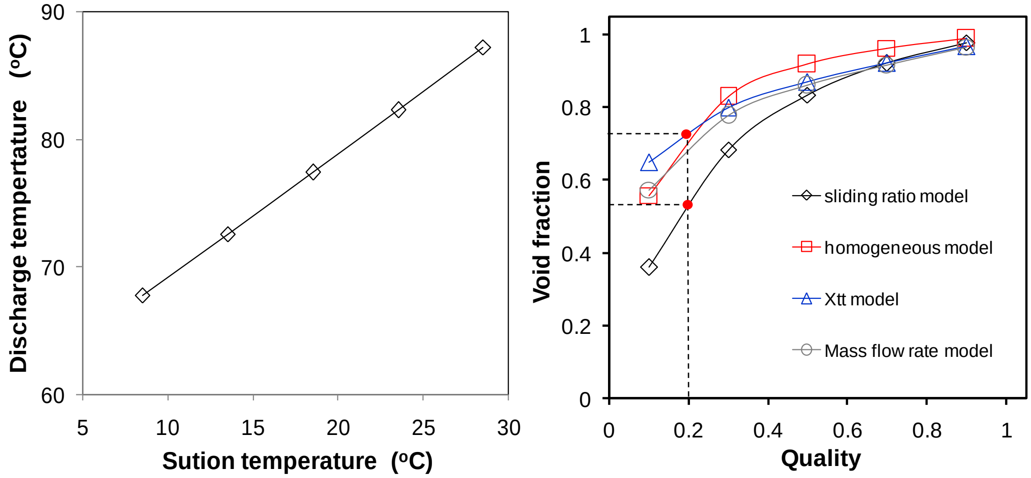
Due to the heat exchange of SLHX, the suction temperature (compressor inlet) increases will result in the increase of discharge temperature. As the suction temperature increases from 8 ^{o}C to 28 ^{o}C, the discharge temperature increases from 68 ^{o}C to 88 ^{o}C, as shown in Fig. 5(a), which have great impact on refrigerant charge. A typical working condition of refrigerant cycle without SLHX, namely superheat 4.25 ^{o}C , evaporator inlet pressure 636 kPa and polytropic index of compression 0.7, is employed to analyze the impact of SLHX on refrigerant charge in condenser.

In this section, a typical condenser of a 2600 W air conditioner is employed to quantitatively investigate the impact of heat exchange of SLHX on refrigerant charge in the condenser. The detailed geometry of the condenser is listed in Table 1.

Table 1: The detailed geometry of condenser

Structure	**Value**	**Note**
Length (mm)	691	
Depth (mm)	27.2	
Height (mm)	456	
Row	2	
Column	24	
Row Space (mm)	13.6	
(Column Space(mm)	19	
FBS (mm)	6.8	
BBS (mm)	14.25	
Fin Space (mm)	1.2	

The refrigerant in condenser exists gas, two-phase and liquid. The refrigerant near the condenser inlet section is almost gas, and the refrigerant near the condenser outlet section mostly becomes liquid, and the refrigerant in the rest of condenser is two-phase flow. The two-phase refrigerant in condenser can be divided into two states: gas rich two-phase flow where the proportion of gas is over 50%, and liquid rich two-phase flow where the proportion of liquid is over 50%. Fig. 5 (b) shows the predicted void fraction of two phase R290 refrigerant under different quality by the existing four kinds of void fraction models, such as sliding ration model, homogeneous model, X_{tt} model and mass flow rate model. It shows that the void fraction of refrigerant predicted by all the models is over 50% when quality is 0.2. As a result, the refrigerant is considered as gas rich liquid when the quality is larger than 0.2. Fig.6 show the gas flow, gas rich two-phase flow, liquid rich two-phase flow, and liquid flow in the condenser where the inlet pressure is 1740 kPa and inlet temperature is 67.8 ^{o}C.



(a) Discharge temperature (b) Void fraction

Figure 5: Void fraction of condenser and discharge temperature

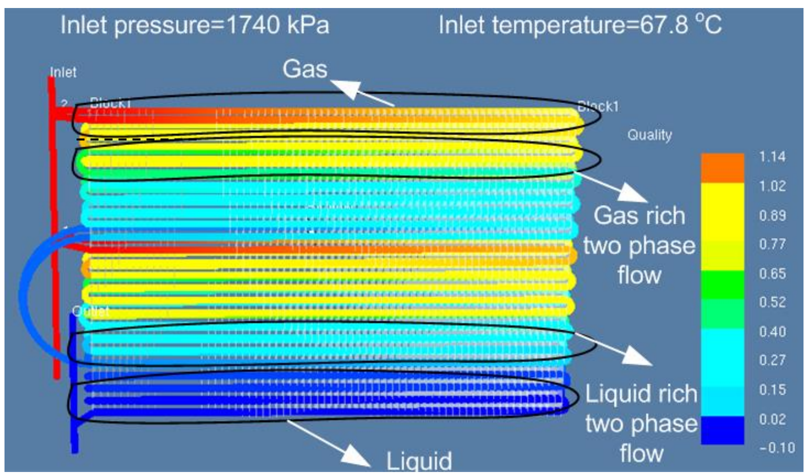
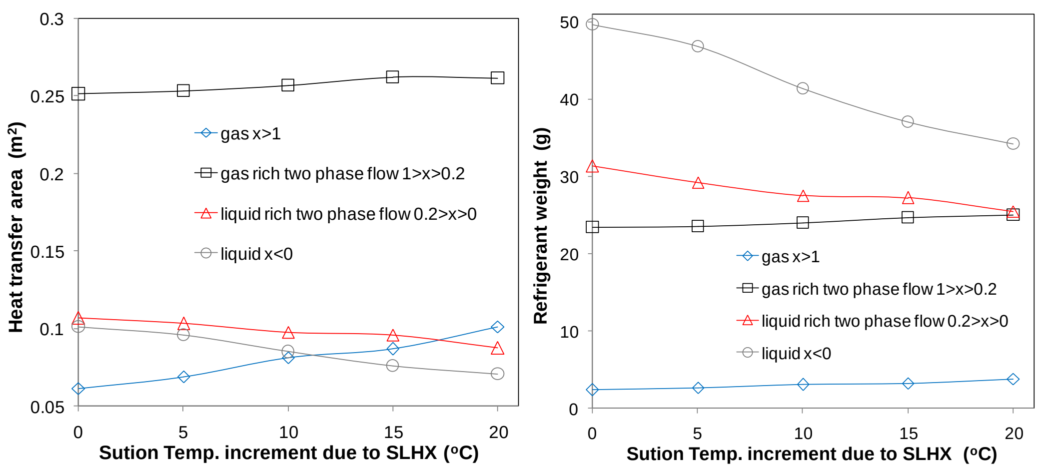


Figure 6: The tendency of quality in condenser



(a) The tendency of heat transfer area with Δt (b) The tendency of refrigerant weight with Δt

Figure 7: The effect of suction Temperature increment due to SLHX on condenser

Figure 7 shows the heat transfer area and refrigerant weight of condenser are the functions of suction temperature increment due to heat exchange of SLHX. Fig. 7 (a) indicates that as the suction temperature increases from 0 ºC to 20 ºC, the heat transfer area of gas and that of gas rich two-phase flow increases up to 64.7% and 4.1%, respectively, but the heat transfer area of liquid rich two-phase flow and liquid decreases up to 18.1% and 30.4% respectively. Fig. 7 (b) indicates that as the suction temperature increases from 0 ºC to 20 ºC, the weight of liquid and liquid rich two-phase flow decreases as much as 59.4% and 6.8% , respectively, but the weight of gas and gas rich two-phase flow almost keeps the same. This is due to that the density of liquid is 20 times higher than that of gas. Therefore, the refrigerant charge of condenser can be significantly reduced when the suction line heat exchanger is employed in R290 system, resulting in less refrigerant charge in whole system.

4. EXPERIMENT

To verify the performance of SLHX, an experiment is processed. The experimental equipment of R290 air conditioner with SLHX consists of a compressor, an evaporator, a condenser, a capillary and a SLHX, as shown in Fig.8.In order to validate the optimization result, the air conditioner with SLHX is tested in a standard air conditioner lab, which follows GB/T 7725-2004(ISO 5154:1994). The uncertainty of dry bulb temperature and wet bulb temperature is within ±2% under the Celsius scale. The heat exchanger capacity is tested by air-enthalpy test method with the uncertainty of ±2%. The uncertainty of system COP is within ±5%

Table 2: Experimental data of air conditioner with SLHX

Experiment group	System with SLHX						System without SLHX	
Experimental No.	1	2	3	4	5	**6**	7	**8**
Cooling capacity (W)	2446	2551	2589	**2625**	2367	2620	2481	**2492**
Power (W)	786	792	796	**800**	787	802	797.5	**798.2**
COP	3.11	3.22	3.25	**3.28**	3.01	3.27	3.11	**3.12**
Airflow (M^3/h)	541	533.7	528.3	**526.4**	522.9	525.6	536.1	**536.2**
Dry bulb Temperature (ºC)	16.76	16.13	15.86	**15.59**	15.83	15.49	15.88	**15.83**
Wet bulb Temperature (ºC)	14.24	13.95	13.81	**13.71**	14.23	13.71	14.24	**14.22**
Inlet of evaporator (ºC)	9.5	10.4	10.8	**11.3**	13.1	11.2	12.2	**12.4**
Outlet of evaporator (ºC)	17.6	16.6	16.3	**16**	11.6	15.8	7.3	**7.4**
Temp. before capillary(ºC)	31	29. 9	29. 3	**28. 9**	28. 5	13	38. 9	**38. 7**
Discharge temperature (ºC)	91.2	89.5	88.8	**88**	67.6	87.5	63.9	**62.6**
Suction temperature (ºC)	39.7	38.6	38	**37.4**	19.4	37.1	13	**12.2**
Condensing Temperature (ºC)	42.5	43	43.4	**43.7**	43.3	43.9	46.2	**46.2**
Refrigerant charge (g)	285	295	305	**315**	335	325	325	**335**

The experimental results are shown as Table 2, and it shows that the SLHX can improve the system performance and reduce the refrigerant charge. The cooling capacity and COP of the system with SLHX is increased by 5.3% and 4.5% compared to the one without SLHX, respectively. This result agrees well with that of the theoretical cycle analysis. The subcooling at the inlet of capillary tube and the discharge temperature increases 10.2º C and 25.4ºC, respectively. The total refrigerant charge of air conditioner is reduced by 6% due to the increase of discharge temperature, which is the same as the results of simulation based analysis discussed in section 3.

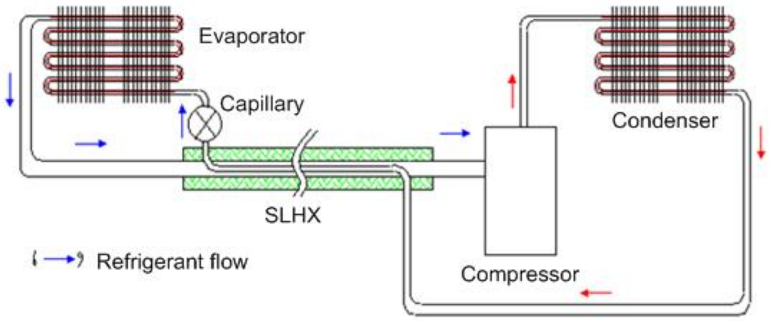


Figure 8: The equipment of R290 air conditioner experiment

5. CONCLUSION

The system performance and refrigerant charge while using a suction line heat exchanger in a R290 air conditioner with small diameter tube is investigated, and the conclusion is listed as follows:

- SLHX can improve the performance of R290 air conditioner with small diameter tube. The experiment data shows that the cooling capacity and COP of the system with SLHX is increased by 5.3% and 4.5%, respectively.
- SLHX can increase discharge temperature and reduce subcooling. The subcooling and discharge temperature increases 10.2^{o} C and 25.4^{o}C, respectively. SLHX can reduce the refrigerant charge. The experimental data shows that the refrigerant charge of the system with SLHX is reduced by 6%.

NOMENCLATURE

A	heat transfer area	(m^2)
Δt	suction temperature increment due to SLHX	(^{o}C)
Q	cooling capacity	(kJ/kg)
h	Enthalpy	(kJ/kg)
COP	coefficient of performance	
p	pressure	(kPa)
$G_{a,max}$	air mass flux at minimum cross-sectional area	(kg/m^2s)
f_a	friction factor of air	
T	temperature	(^{o}C)
Δp_r	pressure drop of refrigerant side	(kPa)
$\Delta p_{r,f}$	frictional pressure drop	(kPa)
$\Delta p_{r,acc}$	acceleration pressure drop	(kPa)
$\Delta p_{r,g}$	the pressure drop caused by gravity	(kPa)
W	energy consumption	(kJ/kg)
subscript		
cooling	cooling condition	
heating	heating condition	
r	refrigerant	

in	inlet
out	outlet
a	air
front	front row of fin
back	back row of fin
bottom	bottom row of fin
top	top row of fin
wall	tube wall
m	middle of tube

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