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Experimental Investigation of the Refrigerant Flow Distribution Characteristic of Heat Exchangers on the Residential Heat Pump Air Conditioner System Performance

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ABSTRACT

In this study, the refrigerant flow distribution characteristic of fin-and-tube heat exchanger used as outdoor and indoor heat exchanger of a residential heat pump air conditioner was experimentally studied and analyzed. The outdoor heat exchanger included “n shape” 1 -circuit, 2-circuit and 3-circuit arrangements, and the indoor heat exchangers were only “n shape” 2–cirucuit arrangement. It showed that the refrigerant flow distribution of both outdoor and indoor heat exchanger had greatly influenced the system performance change of the residential heat pump air conditioner. The refrigerant flow distribution which was be expressed with the temperature difference value of each circuit exit superheat as evaporator with no significant effect on system performance had been obtained both under cooling mode and heating mode. The temperature difference value was nearly the same at various test conditions and can be considered as 2K. According to the experimental results, the system performance under heating mode especially for rated heating mode was the most sensitive with refrigerant flow distribution characteristic. The degradation of rating heating capacity and COP reached 26% and 14% due to refrigerant flow mal-distribution of indoor and outdoor heat exchanger. On the basis of discussion, the adjustment of refrigerant distribution characteristic of both the outdoor and indoor heat exchanger should mainly be operated on the rated heating mode.

1. INTRODUCTION

Fin-and-tube heat exchangers are widely used in residential heat pump air conditioner system. The performance of heat exchangers is directly related to the capacity and efficiency of system. To effectively improve the system performance, some enhancement techniques are often employed for heat exchangers such as using enhanced fin surfaces, increasing total surface area, and increasing the effective mean temperature difference between the air and refrigerant. In practice, the most common and economical method is increasing the temperature difference via circuitry. The outdoor and indoor heat exchangers of residential heat pump air conditioner system often use several parallel refrigerant circuits to optimize between the benefit of improved refrigerant heat transfer and the penalty of refrigerant pressure drop. Even though all refrigerant circuits have the same inlet and outlet conditions, the refrigerant distribution is not uniform. Wang et al. (1999) conducted an experimental study of a condenser with various refrigerant circuits operating with uniform air flow. The researchers found that the refrigerant flow for two-circuit arrangement with extremely symmetrical style was mal-distribution. One circuit is completely condensed, while the other is still in the two-phase region. When the two circuits are combined at the exit of heat exchanger, it is expected that direct-contact condensation would occur. This would significantly decrease the heat transfer performance of heat exchanger. The mal-distribution of air flow for each circuit can also cause the mal-distribution of refrigerant flow.

Choi et al. (2003) conducted an experimental investigation on a three-circuit and three-depth-row finned-tube evaporator to determine the capacity degradation due to non-uniform refrigerant and air flow distributions, and to assess the potential to recover the lost capacity via controlling refrigerant distribution between individual refrigerant circuits. The study showed that capacity degradation due to refrigerant mal-distribution with uniform air flow can be as much as 30%. The capacity degradation was found to be 8.7% for air mal-distributions. Refrigerant superheat in a given circuit is affected by the refrigerant mass flow rate and the airflow rate over the coil area associated with that circuit. For a given air distribution there is one refrigerant flow rate that results in a desired superheat at the individual circuit exit. When circuits are not well balanced, the target overall superheat is a result of mixing a highly superheated refrigerant and two-phase refrigerant leaving different circuits. This causes significant degradation in evaporator capacity because the circuit with superheated refrigerant transfers less heat.

Payne and Domanski (2003) experimentally studied the effect of flow mal-distribution in fin-and-tube evaporators and had been shown to decrease the performance of the evaporator. Domanski and Yashar (2007) applied a novel optimization system called ISHED (intelligent system for heat exchanger design) to optimize refrigerant circuitry in order to compensate airflow mal-distribution. Kærn et al. (2011) focused on quantifying the effect of a non-uniform airflow distribution for fin-and-tube evaporators and found that the refrigerant distribution was affected by the airflow distribution and that the distribution has an effect on the evaporator performance.

Up to now, to the author's knowledge, the research related to the flow mal-distribution is mainly based in fin-and-tube evaporators, while the experimental study of system performance of residential heat pump air conditioner affected by flow mal-distribution is still lacking. The main purpose of this study is focused on the system performance affected by flow mal-distribution of outdoor and indoor fin-and-tube heat exchanger in both cooling and heating mode.

2. EXPERIMENTAL SETUP AND TEST PROCEDURE

2.1 Experimental Setup

Figure 1 shows a schematic of the experimental setup, which was constructed according to Chinese national standard GB/T 7725-2004, including indoor and outdoor environmental chamber and air flow rate measuring apparatus. The test objection was modified on a 3500W cooling capacity residential heat pump air conditioner with R32 refrigerant. The outdoor unit consists of a constant speed hermetically-sealed reciprocating compressor with a variable speed motor, four-way reserving valve, fin-and-tube heat exchanger, fan and electronic expansion valve. The indoor unit only contained fin-and-tube heat exchanger and fan. The heat exchangers of outdoor unit and indoor unit were separately mounted in outdoor and indoor multi-nozzle air flow chamber in order to keep uniform air velocity. The air flow rate of indoor and outdoor heat exchanger was set the same as the original air conditioner by the exhaust fan installed at the exit of the air flow chamber. Thermocouples and pressure transducers were used to measure the thermodynamic states of each circuit refrigerant entering or/and exiting the indoor and outdoor heat exchangers.

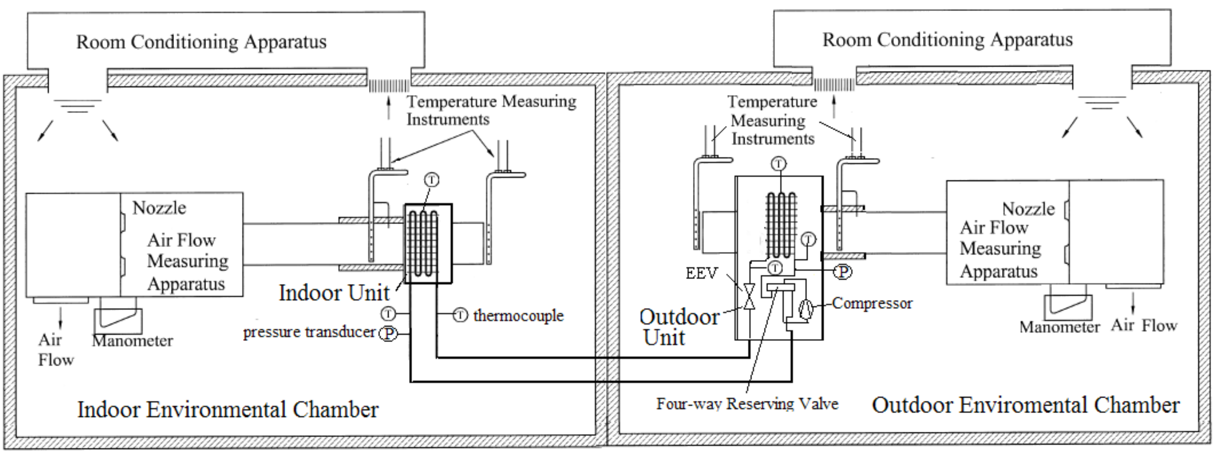

Figure 1: Experimental setup schematic

Figure 2 presents the schematic of the indoor heat exchanger used in this study. The indoor heat exchanger has 24 microfin copper tubes with louvered aluminum fins placed in two depth rows and two parallel circuits. Figure 3 presents the schematic of the outdoor heat exchangers used in this study. The outdoor heat exchanger has 48 microfin copper tubes with wavy aluminum fins placed in two depth rows. The outdoor heat exchangers have three kinds of circuit arrangement, including one circuit, two parallel circuits and three parallel circuits. The extra electronic expansion valves (EEV) were installed in the inlet tube of each circuit which was on the side of facing the wind when the heat exchanger working as evaporator.

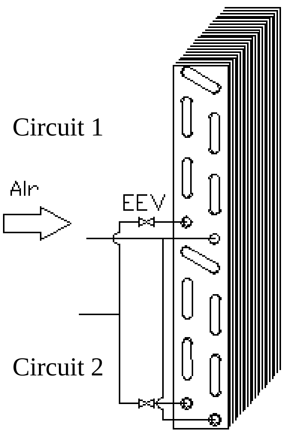

Figure 2: Indoor heat exchanger

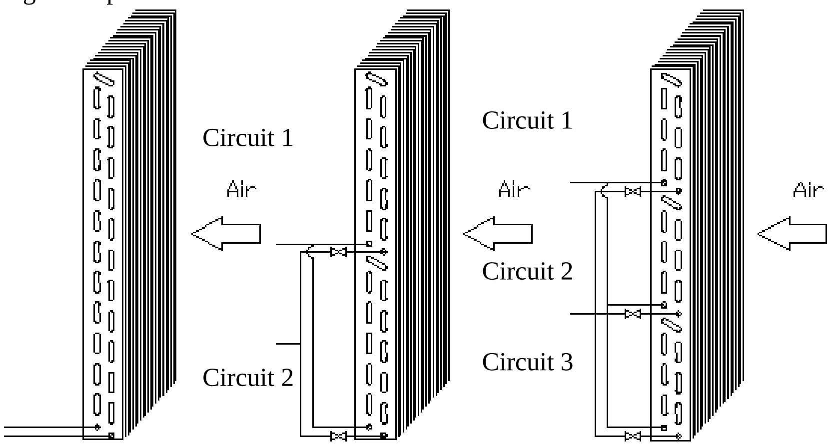

Figure 3: Outdoor heat exchanger

2.2 Test Conditions and Procedure

Test conditions included four working conditions of rating cooling, intermediate cooling (cooling capacity is about 50% rating cooling capacity), rating heating and intermediate heating (heating capacity is about 50% rating heating capacity). The corresponding compressor frequencies were 64Hz, 25Hz, 85Hz and 40Hz, respectively, which was referred to original heat pump air conditioner. For every refrigerant flow distribution, the system performance was adjusted to optimal state according to adjusting the electronic expansion valve. Table 1 presents the tests performed.

Table 1: Test conditions

Conditions	**Rated Cooling**	**Intermediate cooling**	**Rated heating**	**Intermediate heating**
Compressor frequency	64Hz	25Hz	85Hz	40Hz
Outdoor environmental temperature	35℃/24℃		7℃/6℃	
Indoor environmental temperature	27℃/19℃		20℃/15℃	

The optimal refrigerant charge of the air conditioner was determined according charging or discharging the refrigerant of system and adjusting the electronic expansion valve based the outdoor heat exchanger of two parallel circuits carried out only in the cooling condition. The system was running at the simulated conditions described still a quasi-steady operation. During the period, the dry-bulb and wet-bulb temperatures of the two rooms were carefully maintained within ±0.05℃. The optimal refrigerant charge was determined until the maximum EER was obtained.

The cooling capacity or heating capacity was measured according to air enthalpy-difference method. The indoor fan and outdoor fan of original system are working during the whole experiment, so the input power is approximately equal to the system consuming power. Therefore, the EER was defined the ratio of cooling capacity and system input power, and the COP was the ratio of heating capacity and system input power.

3. RESULTS AND DISCUSSION

The difference of subcooling of each circuit for the condenser and the superheat of each circuit for the evaporator was expressed the refrigerant flow mal-distribution characteristic. The difference of subcooling or superheat of each circuit was smaller, the refrigerant flow distribution was more uniform.

3.1 Effect on System Performance under Cooling Mode

The indoor heat exchanger as figure 2 was used as evaporator and the outdoor heat exchanger as figure 3 was used as condenser under cooling mode. The refrigerant flow distribution of the two-circuit and three-circuit outdoor heat exchanger was keep uniform according to adjusting the extra electronic expansion valves installed in the outlet tube of each circuit under cooling mode. The refrigerant of evaporator was also set by adjusting the extra electronic expansion valves. Since the control of perfect even exit superheat of evaporator or subcooling of condenser in individual circuits sometimes was very difficult, the exit temperature difference less than 1K would be as uniform refrigerant flow distribution.

The figure 3 to figure 5 showed the effect of the refrigerant flow distribution of the two-circuit evaporator on the system performance with one-circuit, two-circuit and three-circuit condenser under cooling mode, separately. Thereinto, the refrigerant flow distribution of the condenser kept uniform as best as possible when adjusting that of evaporator.

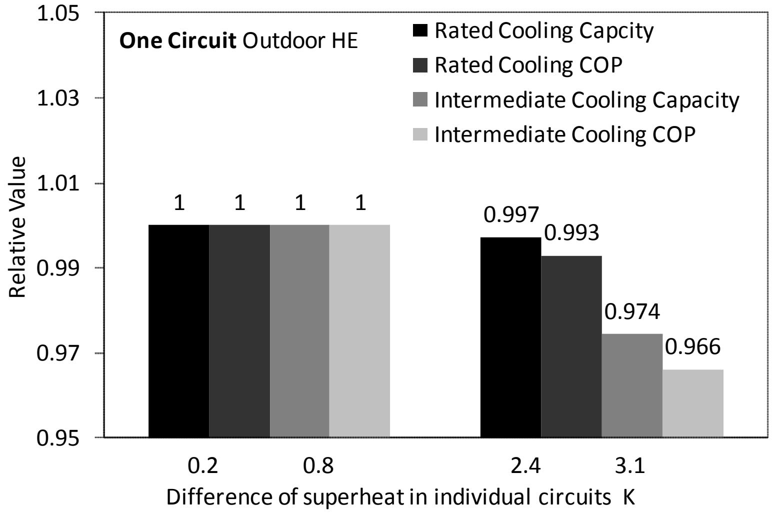

Figure 3: Variation with the refrigerant flow distribution of evaporator for one circuit outdoor heat exchanger

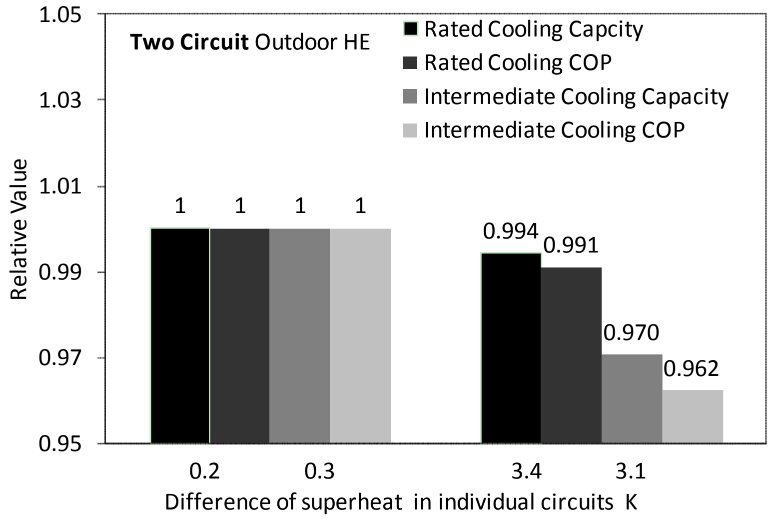

Figure 4: Variation with the refrigerant flow distribution of evaporator for two circuit outdoor heat exchanger

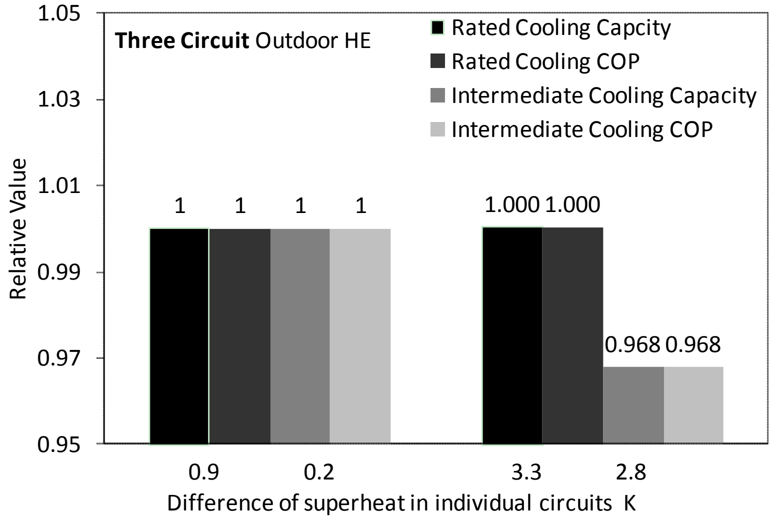

Figure 5: Variation with the refrigerant flow distribution of evaporator for three circuit outdoor heat exchanger

As seen in figure 3, figure 4 and figure 5, the cooling capacity and COP (Coefficient of performance) showed a decrease when the temperature difference of superheat increased in individual circuits. It was noted that the degradation of capacity and COP under intermediate cooling mode was much more than that under rated cooling mode at the similar evaporator temperature difference of superheat in individual circuits. Compared with the uniform refrigerant flow distribution of evaporator, when the evaporator had a exit superheat difference reached 3K, the capacity and COP under rated cooling mode both dropped no more than 1%, while the degradation of that under intermediate cooling mode achieved 2.6%~3.2% and 3.2%~3.4%, respectively for the three kinds of condensers.

The test results showed that the refrigerant flow mal-distribution may become more pronounced for lower mass flow rate under intermediate cooling mode. That was probably that lower refrigerant mass flow rate would result in much lower refrigerant side heat transfer coefficient in the evaporator, so a lower exit superheat difference would bring a larger difference of refrigerant mass flow rate in individual circuits, which affected evaporator performance detrimentally. The degradation of evaporator performance would cause the decrease of evaporator pressure, and the pressure ratio of the compressor would increase. Hence, the compressor consuming power was increased. The decrease of capacity and the increase of power both cause the more degradation of COP than that of capacity.

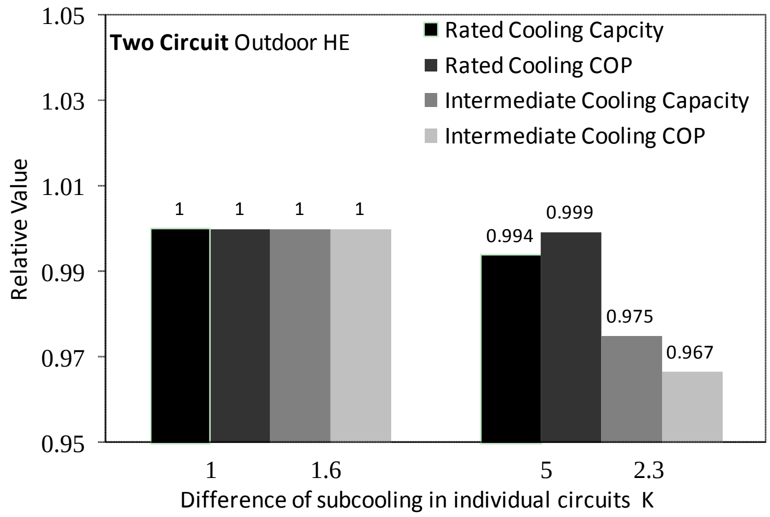

Figure 6: Variation with the refrigerant flow distribution of the two circuit outdoor condenser

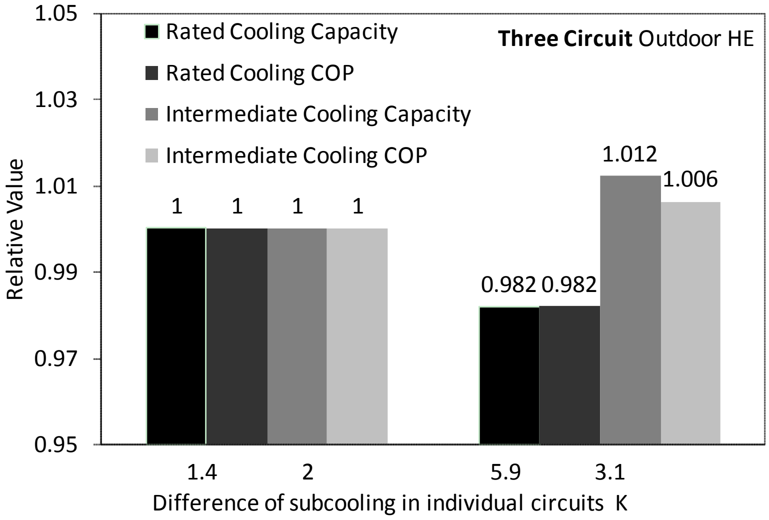

Figure 7: Variation with the refrigerant flow distribution of the three circuit outdoor condenser

The one-circuit outdoor heat exchanger had no problem about the refrigerant flow distribution, so only the refrigerant flow distribution characteristics of the two-circuit and three-circuit outdoor heat exchanger were studied. The figure 6 and figure 7 showed the effect of the refrigerant flow distribution of the two-circuit and three-circuit condenser on the system performance with the same two-circuit evaporator under cooling mode, respectively. The same way, the refrigerant flow distribution of the evaporator kept uniform as best as possible when adjusting that of condenser.

It can be also noticed that the system performance on the temperature difference of exit subcooling of condenser under rated cooling mode was much more sensitive than that under intermediate cooling mode from figure 6 and figure 7. The capacity and COP both decreased less than 1% when the temperature difference of exit subcooling of two-circuit condenser increased from 1K to 5K under rated cooling mode, however, the degradation of those achieved about 2.5% and 3.3% when the temperature difference increased from 1.6K to 2.3K under intermediate cooling mode. As for the three-circuit condenser, the capacity and COP both had obvious decrease when the temperature difference increased from 1.4K to 5.9K under rated cooling mode, while those showed a little increase when the temperature difference increased from 2K to 3.1K under intermediate cooling mode. Compared with the results of refrigerant flow distribution characteristics of evaporator, the system performance affected by that of evaporator was much more significantly.

Under the intermediate cooling mode, the performance dropped significantly for the two-circuit condenser but gained a little for the two-circuit, although the latter subcooling temperature difference variation was obviously higher. Explanation of this phenomenon was because the latter refrigerant flow distribution of evaporator was more uniform. In practice, the temperature difference of superheat of evaporator for the two-circuit condenser system was controlled at 3K, while that for the three-circuit condenser system was controlled at 0.2K. The test results also showed that the system performance affected by the temperature difference of superheat of evaporator was much more significantly than that by the temperature difference of subcooling of condenser.

Overall, when the temperature difference of exit superheat of evaporator or subcooling of condenser was less than 3K, the capacity and COP decrease less than 3% under cooling mode.

3.2 Effect on System Performance under Heating Mode

Since the system performance affected by the refrigerant flow distribution of evaporator was much more significantly than that of condenser, only the refrigerant flow distribution of the two-circuit and three-circuit outdoor heat exchanger as evaporator were tested under heating mode. Also, the refrigerant flow distribution of the condenser kept uniform as best as possible. In test, the temperature difference of subcooling of condenser for the condenser (two-circuit indoor heat exchanger) system was controlled at 0.3K and 0.4K for the three-circuit evaporator under rated and intermediate heating mode, respectively, while that for the three-circuit evaporator system was controlled at 1K and 0.7K, respectively.

The figure 8 and figure 9 showed the effect of the refrigerant flow distribution of the two-circuit and three-circuit evaporator on the system performance with the same two-circuit condenser under heating mode, respectively. It showed a rapid decrease in both capacity and COP when the temperature difference of exit superheat increased. The degradation of the heating capacity and COP for the two-circuit evaporator achieved about 2.5% and 3.6% when the temperature difference increased from 0.6K to 4.5K under rated heating mode, and that achieved about 1.5% and 1.3% when the temperature difference increased from 0.2K to 0.7K under intermediate heating mode. As for the three-circuit evaporator, the degradation of capacity and COP was a litter higher than those for the two-circuit evaporator under rated and intermediate heating mode even though the former temperature difference of superheat was higher in figure 9. The results indicated that evaporator with more circuits had a greater effect on the system capacity under rated heating mode than that with fewer circuits. It was also because that the refrigerant flow distribution would be more uneven for evaporator with more circuits due to lower refrigerant flow rate in individual circuit.

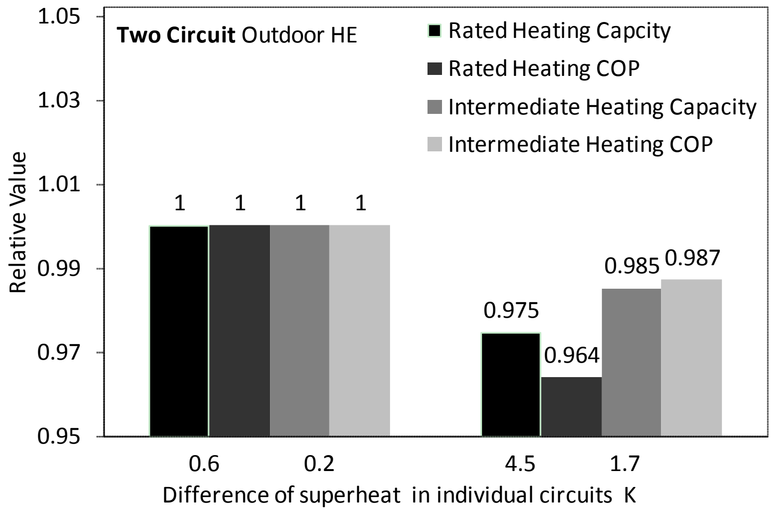

Figure 8: Variation with the refrigerant flow distribution of the two circuit outdoor evaporator

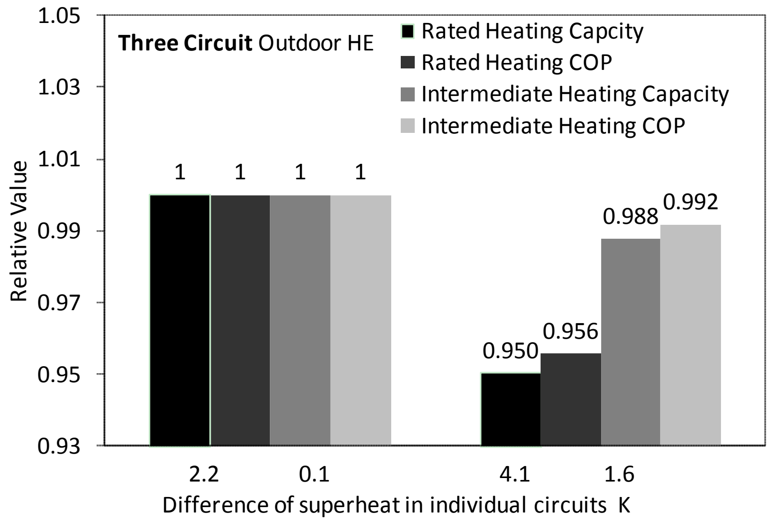

Figure 9: Variation with the refrigerant flow distribution of the three circuit outdoor evaporator

From the data showed in figure 8 and figure 9, the capacity and COP both decreased less than 2% when the temperature difference of exit superheat was less than 2K under intermediate heating mode, and that would decrease about 3%~5% when the temperature difference of exit superheat was more than 4K under rated heating mode. The results also suggested that the temperature difference of exit superheat less than 2K can be considered as uniform refrigerant flow distribution in evaporator.

3.3 Sensitivity Analysis

Table 2 presents the test results of system using the two-circuit indoor heat exchanger and three-circuit outdoor heat exchanger with the best refrigerant flow distribution. It was noticed that the opening of extra EEV was different for adjusting the best refrigerant flow distribution under different work conditions no matter for indoor heat exchanger or outdoor heat exchanger in table 2. Furthermore, the adjusting direction was nearly completely adverse for cooling mode and heating mode. Hence, how to adjust the refrigerant flow distribution was very important to improve the system overall performance.

Table 2: Test results

Conditions		Rated Cooling	Intermediate cooling	Rated heating	Intermediate heating
Extra EEV Opening (B)	Indoor HE	(500,440)	(500,380)	(500,500)	(500,500)
	Outdoor HE	(460,500,500)	(400,500,500)	(500,440,120)	(380,500,250)
Maximum TD of Superheat or Subcooling (K)	Indoor HE	0	0	1.1	0.7
	Outdoor HE	0.5	2	0.4	0.1

The following was the sensitivity analysis of system performance on extra EEV opening of the two-circuit indoor and three-circuit outdoor heat exchanger (HE). The relative value of system performance of all analysis was based on the best refrigerant flow distribution as presented in table 2.

3.3.1 Rated cooling condition

The figure 10 and figure 11 showed the variation of system rated cooling performance and the maximum exit temperature difference of indoor and outdoor heat exchanger with the extra EEV opening of outdoor heat exchanger at the same extra EEV opening of indoor heat exchanger. It can be noted that the maximum exit temperature difference of outdoor heat exchanger was increasing with the decrease of extra EEV opening of circuit 3, and the rated cooling capacity and COP were also decreasing. The rated cooling capacity and COP both dropped about 8% when the extra EEV opening of outdoor heat exchanger was the best opening under rated heating condition. At the moment, the maximum exit temperature difference of outdoor heat exchanger reached 7K, which indicated that the circuit 1 was not entirely condensed and the circuit 3 was subcooled completely, so that the heat transfer performance of outdoor heat exchanger dropped greatly. However, in relative wide variation of extra EEV opening, the rated cooling capacity and COP dropped not significantly although the maximum exit temperature difference of outdoor heat exchanger reached 6.7K.

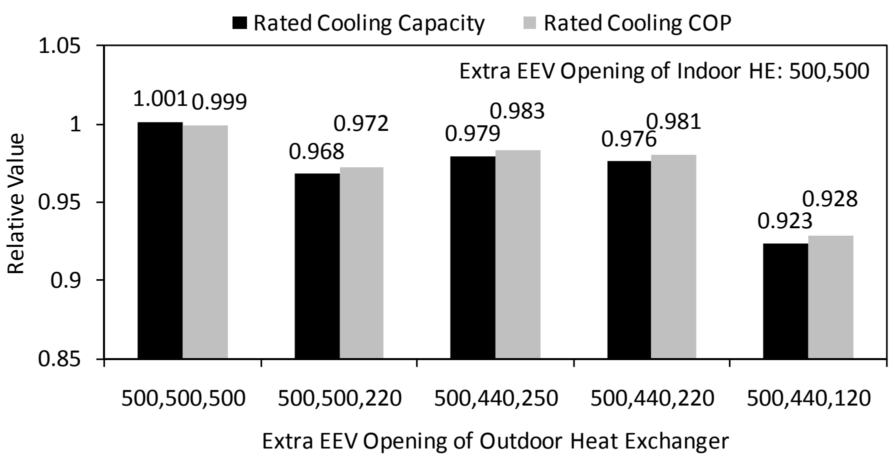

Figure 10: System performance with the extra EEV opening of outdoor heat exchanger

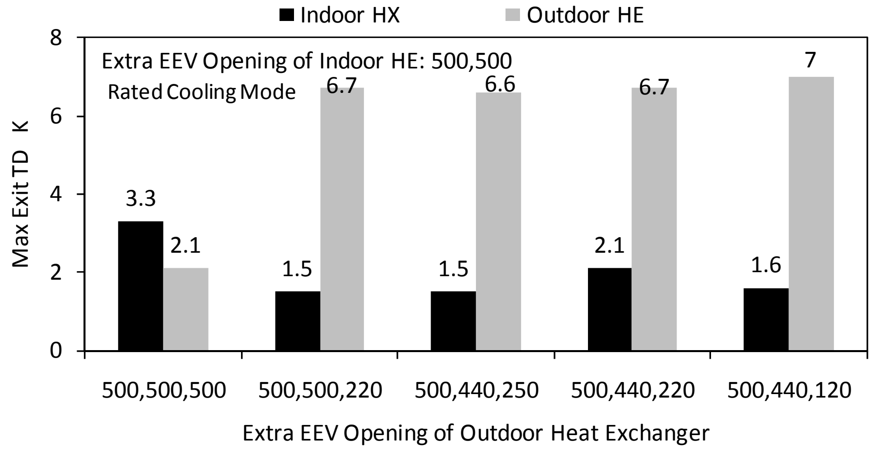

Figure 11: Maximum exit temperature difference with the extra EEV opening of outdoor heat exchanger

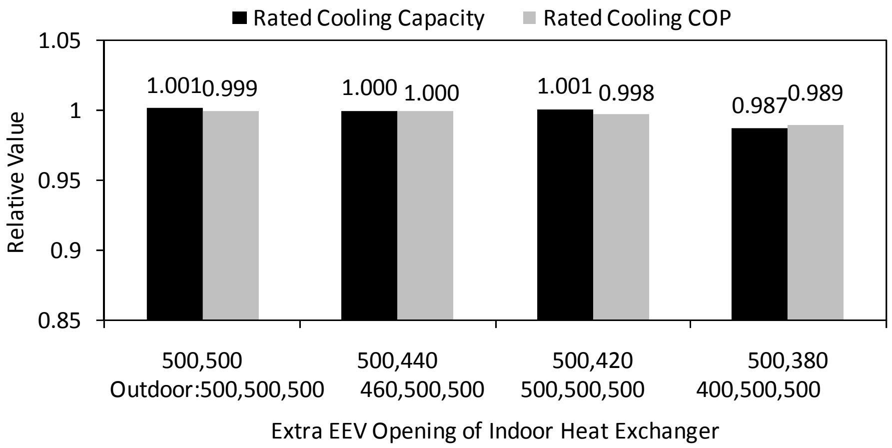

Figure 12: System performance with the extra EEV opening of indoor heat exchanger

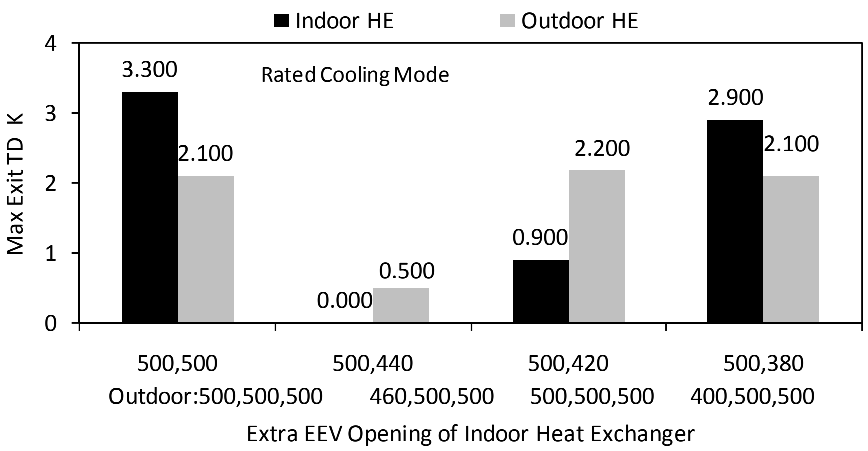

Figure 13: Maximum exit temperature difference with the extra EEV opening of indoor heat exchanger

When the extra EEV opening of outdoor heat exchanger was higher than 220B, the system performance had nearly no change. So the extra EEV opening of outdoor heat exchanger changed from (500,500,500) to (400,500,500) can be considered as no change for the system performance. The figure 12 and figure 13 can be as the variation of system rated cooling performance and the maximum exit temperature difference of indoor and outdoor heat exchanger with the extra EEV opening of indoor heat exchanger at the same extra EEV opening of outdoor heat exchanger. Also, the system rated cooling capacity and COP had almost no change in a wide area of the extra EEV opening of indoor heat exchanger thanks to the maximum exit temperature difference of indoor heat exchanger within 3K as showed in figure 13.

3.3.2 Intermediate cooling condition

The extra EEV opening of indoor heat exchanger had relatively higher effect on system performance than that of outdoor heat exchanger as presented in figure 13. The intermediate cooling capacity and COP increased 3.26% and 4.59%, respectively, when the extra EEV opening of indoor heat exchanger was adjusted from (500, 500) to (500, 380) with the same (500, 500, 500) opening of outdoor heat exchanger, however, the system performance was almost no change when the opening of outdoor heat exchanger was adjusted from (500, 500, 500) to (400, 500, 500) with the same (500, 380) opening of indoor heat exchanger.

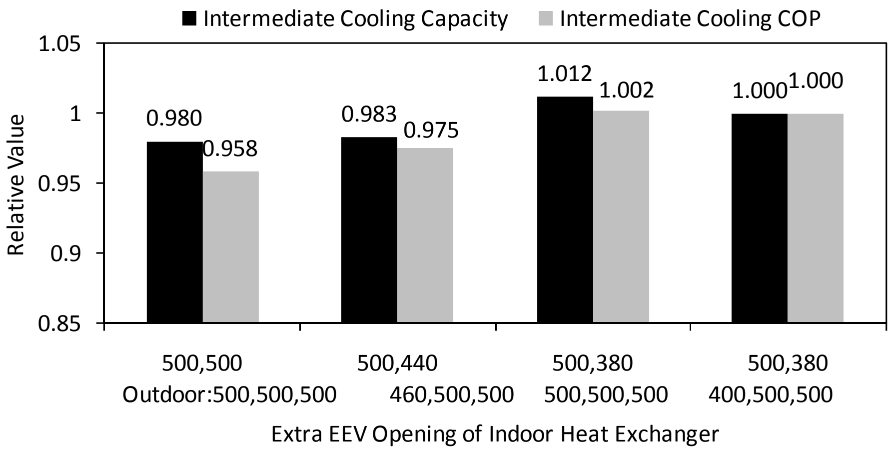

Figure 14: System performance with the extra EEV opening of indoor heat exchanger

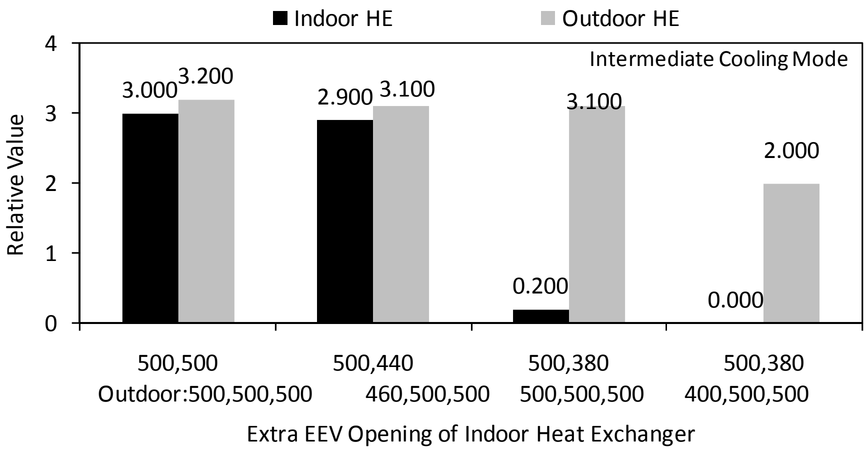

Figure 15: Maximum exit temperature difference with the extra EEV opening of indoor heat exchanger

3.3.3 Rated heating condition

The extra EEV opening of indoor heat exchanger had a great influence on the system rated heating performance as shown in figure 16, although the indoor heat exchanger was using as condenser. The maximum exit temperature difference of indoor heat exchanger reached 9.7K when the extra EEV opening of indoor heat exchanger was adjusted (500, 380) which was the best refrigerant distribution of intermediate condition. This significantly reduced the system performance.

When the best refrigerant flow distribution of indoor heat exchanger was adjusted, the refrigerant flow distribution of outdoor heat exchanger would greatly affect the system performance as shown in figure 16. The extra EEV opening especially for the circuit 3 was the most significant effect on the refrigerant flow distribution of outdoor heat exchanger as seen in figure 17, so that the system performance had been influenced. It can be seen that the increase of rated heating capacity and COP achieved 11.25% and 6.27%, respectively, when the extra EEV opening of the circuit 3 of the outdoor heat exchanger was adjusted from 500 to 120. The rated heating capacity and COP can increase 26% and 14%, respectively when the refrigerant flow distribution of indoor and outdoor heat exchanger was both adjusted the best state.

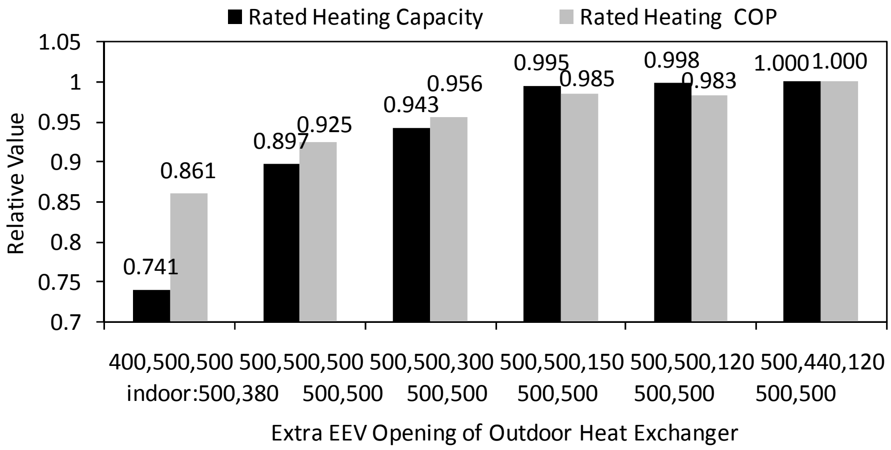

Figure 16: System performance with the extra EEV opening of outdoor heat exchanger

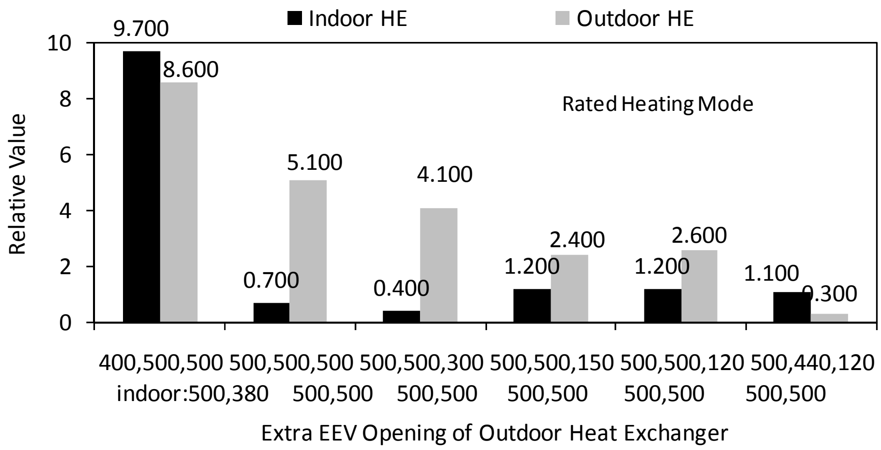

Figure 17: Maximum exit temperature difference with the extra EEV opening of outdoor heat exchanger

3.3.4 Intermediate heating condition

Compared with the rated heating condition, the extra EEV opening of indoor heat exchanger had a relative lower influence on the refrigerant flow distribution. The exit temperature difference of indoor heat exchanger reached 9.7K for rated heating condition, which was just 2.4K for intermediate condition when the extra EEV opening of indoor heat exchanger was adjusted (500, 380) as shown in figure 17 and figure 19. This suggested that the refrigerant flow distribution of indoor heat exchanger was the most sensitive under rated heating condition. The adjustment of refrigerant distribution characteristic of indoor heat exchanger should mainly be operated on the rated heating mode.

The refrigerant flow distribution of outdoor heat affected by the extra EEV opening of outdoor heat exchanger was not significantly as seen in figure 19. The maximum exit temperature difference of outdoor heat exchanger was no more than 3K when the extra EEV opening of outdoor heat exchanger was adjusted from the best state of intermediate cooling condition to the best state of intermediate heating condition, so the degradation of system performance was not as distinctive as that of rated heating condition.

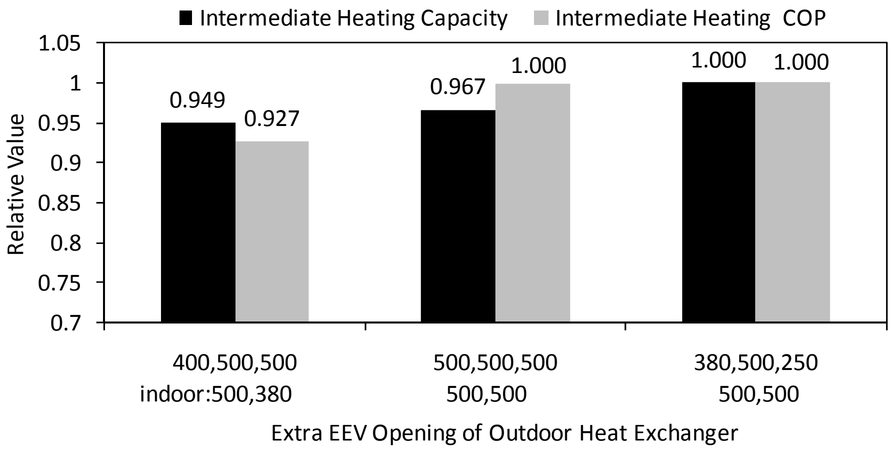

Figure 18: System performance with the extra EEV opening of outdoor heat exchanger

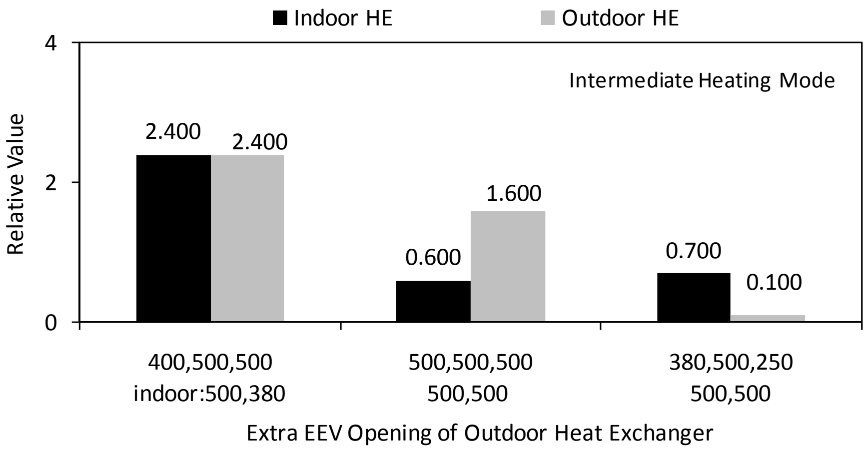

Figure 19: Maximum exit temperature difference with the extra EEV opening of outdoor heat exchanger

4. CONCLUSIONS

An experimental study was carried out to investigation the effect of refrigerant flow distribution on the system performance of a residential heat pump air conditioner. The outdoor heat exchanger included “n shape” 1 -circuit, 2-circuit and 3-circuit arrangements, and the indoor heat exchangers were only “n shape” 2–cirucuit arrangement. During the testes, the extra EEV were used on each circuit to adjust the refrigerant distribution. On the basis of previous discussions, the following conclusions were made:

The refrigerant flow distribution of both outdoor and indoor heat exchanger had greatly influenced the system performance change of the residential heat pump air conditioner. The degradation of rating heating capacity and COP can reached 26% and 14% due to refrigerant flow mal-distribution of indoor and outdoor heat exchanger.

The refrigerant flow distribution which was be expressed with the temperature difference value of each circuit exit superheat as evaporator with no significant effect on system performance had been obtained both under cooling mode and heating mode. The temperature difference value was nearly the same at various test conditions and can be considered as 2K corresponding the variation of capacity and COP both within 3%.

According to the sensitivity analysis, the system performance and refrigerant flow distribution under heating mode especially rated heating mode was the most significantly affected by the extra EEV opening of both outdoor and indoor heat exchanger. The adjustment of refrigerant distribution characteristic of both the outdoor and indoor heat exchanger should mainly be operated on the rated heating mode.

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