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International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2010

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ABSTRACT

VRF system has generally large size outdoor unit which can cover big capacity. Large size outdoor heat exchanger requires many refrigerant paths, because of pressure drop during evaporation and condensation. The increase of refrigerant path number may cause mal-distribution of refrigerant. Especially, heat exchangers which have vertically long geometry and uneven air velocity profile have very serious refrigerant distribution problem. The appropriate differential refrigerant flowrate control can lead to the increase of heat exchanger performance. The distribution problems are more important for high efficient evaporator. In this paper, how to control refrigerant flowrate using small diameter tubes after distributor will be introduced. And we will review the effect of the appropriate refrigerant flowrate control of outdoor heat exchanger. Our result showed that the decrease of the standard deviation of evaporator outlet temperatures of 1°C can lead to the heating COP increase of nearly 10%.

1. INTRODUCTION

VRF (Variable Refrigerant Flow) System is large scale and high performance multi air conditioning system for buildings. It was introduced in Japan more than 20 years ago and has become popular in many countries. VRF systems are used in approximately 50% of medium-sized commercial buildings (up to 6,500m^2) and one-third of large commercial buildings (more than 6,500m^2) (Goetzler, 2007). VRF system has been evolved to increase the system performance, which is lower energy consumption, ease of installation and maintenance, and robustness. To make energy saving VRF system, high performance compressor and heat exchanger were introduced (Kim, 2008). And high capacity sub-cooler unit was applied to give a high freedom of installation limitation (Yoo, 2000).

Outdoor heat exchanger assembly for VRF heat pump system is comprised of manifold, fin-tube heat exchanger, distributor, and small diameter tubes. Figure 1 shows the schematics of heat exchanger assembly of outdoor unit. Outdoor unit of VRF heat pump system operates as not only a condenser for cooling operation but also an evaporator for heating operation. When it operates as a condenser, the inlet of refrigerant is the manifold. And refrigerant flows through heat exchanger and it exits through small diameter tubes installed at each path. Finally, it leaves heat exchanger assembly through the distributor combining small diameter tubes. At the manifold, the state of refrigerant is gas phase. Vertically long manifold, a cylindrical pipe, generally shows bad distribution characteristics due to the effect of gravity, but mal-distribution is not serious due to very low density of gas phase.

When outdoor heat exchanger operates as an evaporator for heating operation, on the other hand, refrigerant enters to heat exchanger through small diameter tubes after refrigerant distributor as a two phase flow. After evaporating in the fin-tube heat exchanger, it leaves the heat exchanger assembly through manifold merging superheated refrigerant gas. One of the differences between condenser and evaporator is the state of refrigerant at the inlet and outlet. The inlet and outlet refrigerant condition of condenser is single phase: gas at the inlet and liquid at the outlet. But the refrigerant condition of evaporator outlet is superheated gas phase and the inlet of evaporator is two phase: gas and liquid. Two phase flow has very unstable flow characteristics. Its flow pattern varies unstably due to many factors: flowrate, void fraction, vapor-liquid velocity ratio, and geometry of flow path (Collier, 1981). These unstable effects of two phase flow disturb the appropriate refrigerant distribution for evaporator in a distributor. Mal-distribution of refrigerant induces lower efficiency of heat exchanger. So, the design of distributor is one of the important factors for evaporator design.

Through the distributor, refrigerant is distributed to each path of heat exchanger. When the profile of the inlet air flow to heat exchanger is uniform and each path structure of heat exchanger is same, same mass flowrate supply of refrigerant may be the best solution to get maximum efficiency of heat exchanger. But in the real situation, air velocity profile and path structure are different with each part. So, differential refrigerant mass flowrate control need mainly comes from the uneven air velocity profile and different path structure. Unless refrigerant flowrate is controlled appropriately on purpose, some paths may show excessive superheat and the other paths may show two phase state at the exit of the path. This result will lead to the inefficient use of heat exchanger. To prevent the inefficiency, refrigerant mass flowrate should be controlled. This paper deals with one of the method to control the refrigerant mass flowrate to increase the heat exchanger efficiency.

Refrigerant mass flowrate can be controlled by small diameter tube like capillary tube. Park *et al* (2008) showed the effect of capillary tube geometry on mass flowrate and suggested correlation considering capillary coil shape. Zhang *et al* (2001) suggested correlation for two phase pressure drop in small diameter tubes using 119 data. Their correlation was developed by modifying the Friedel correlation.

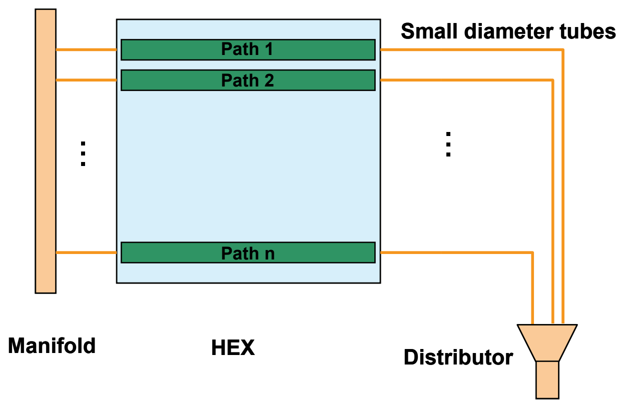

Figure 1 Schematics of heat exchanger assembly of VRF outdoor

2. DESIGN PROCEDURE FOR DIFFERENTIAL FLOWRATE CONTROL

There are many factors that induce unbalance between heat exchanger paths; uneven air velocity profile, different path structure, chassis structure, parts locations installed for heat pump cycle behind heat exchanger. Some of them show mixed effect and some of them shows independent effect. We select two of them, air velocity profile and path structure, as parameters for designing differential refrigerant mass flowrate. The other factors may influence to air velocity profile. So the air velocity profile implies other factors.

Design procedure for the differential refrigerant mass flowrate control is shown in Figure 2. First, the air velocity profile was estimated and measured. The large outdoor heat exchanger has uneven air velocity profile. Figure 3 shows the air velocity profile. This uneven air velocity profile mainly comes from the geometry of the outdoor chassis and the location of fans. Many large VRF systems have the chassis geometry shown Figure 4. Fan located at the top of the chassis induces air flow from the side to the top. In this situation, air velocity of the upper part is large and velocity of the lower part, on the other hand, is low. We designed path structure as simple as it could be to verify the effect of the differential mass flowrate control. The simple path structure is shown in Figure 5.

Second, path simulation to get heat transfer rate, mass flowrate, and pressure drop for each path was performed. The simulation was performed under the corresponding air velocity shown in Figure 3. Evap-Cond program of NIST was used for the simulation and some conditions were fixed to simplify the simulation. The fixed conditions are listed in Table 1. Our target of design is to adjust outlet temperature of refrigerant of each path and minimize the standard deviation of outlet temperatures. So outlet temperature was fixed to be same for all path outlets during the simulation. This simulation results was compared to the experimental results and analyzed the accuracy.

Third, small diameter tube specification was calculated to distribute refrigerant to each path differentially based on the previous path simulation results. The major premise is that the sum of the pressure drop at small diameter ($\Delta P_{capillary}$) and heat exchanger path (ΔP_{path}) is same with each path. At the second step, needed mass flowrate and pressure drop corresponding to air velocity for each path was calculated. Then needed pressure drop for equalizing total pressure drop ($\Delta P_{capillary} = \Delta P_{total} - \Delta P_{path}$) between each path can be calculated. With the needed pressure drop and mass flowrate, the specification of each small diameter tube was designed with the equation of modified Friedel's correlation (Zhang *et al*, 2001). The used correlation is shown below.

$$\frac{dP}{dz} = \left.\frac{dP}{dz}\right|_{lo} \Phi_{lo}^2 \tag{1}$$

$$\left.\frac{dP}{dz}\right|_{lo} = \frac{2 f_{lo} G^2 v_l}{D} \tag{2}$$

$$f_{lo} = 0.079 * \mathrm{Re}^{-\frac{1}{4}} \quad \text{for Turbulent flow}$$

$$f_{lo} = \frac{16}{\mathrm{Re}} \quad \text{for Laminar flow}$$

$$\Phi_{lo}^2 = (1-x)^2 + 2.87x^2\left(\frac{P}{P_c}\right)^{-1} + 1.68x^{0.8}(1-x)^{0.25}\left(\frac{P}{P_c}\right)^{-1.64} \tag{3}$$

Finally, after applying the designed specification to the system, we obtained experimental data and analyzed the results. Due to many assumptions for simple and fast simulation, modification of small diameter tube specification should be done after analyzing the experimental results.

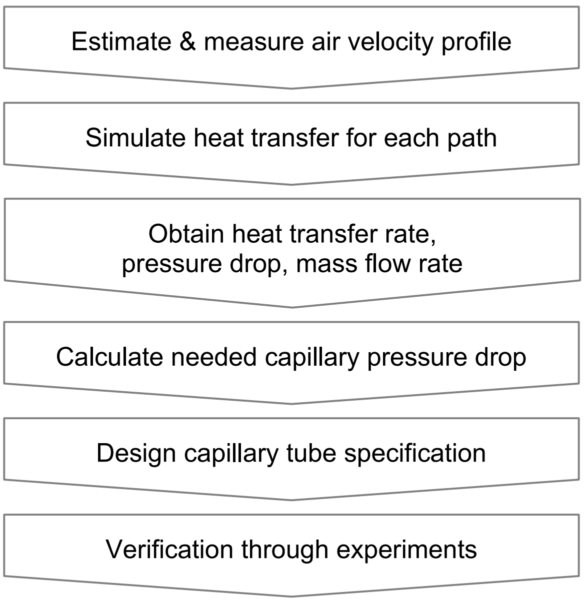

Figure 2 Diagram for design process

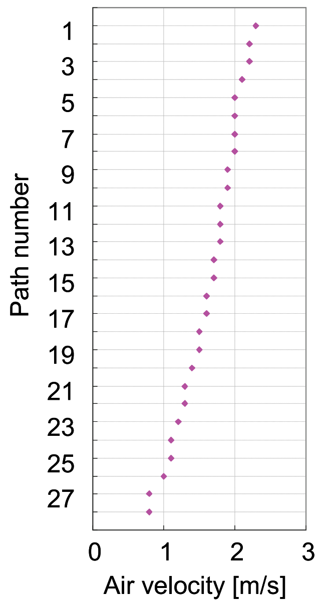

Figure 3 Air velocity profile of VRF outdoor unit

Table 1 Fixed condition for path simulation

Refrigerant	Inlet temperature	1℃
	Inlet quality	0.2
	Mass flowrate	Iteration until outlet temperature become same
Air	Inlet temperature	7℃
	Relative humidity	87%
	Velocity	Measured velocity (Figure 3)

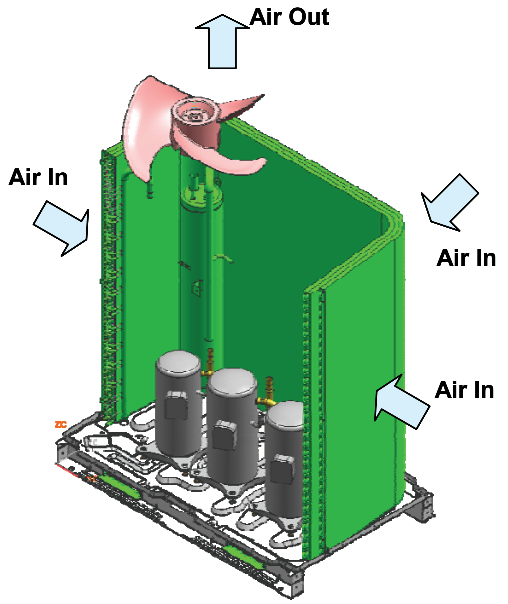

Figure 4 Outdoor chassis structure of large VRF system

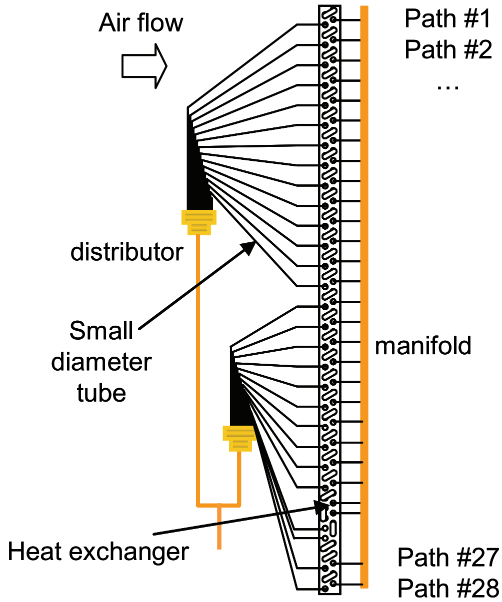

Figure 5 Path structure

3. EXPERIMENTS AND RESULTS

VRF heat pump system of 20Hp capacity was used for verifying the effect of differential refrigerant mass flowrate control. Simple path structure of outdoor heat exchanger (Figure 5) was adopted for the easy identification of the effect. Path number was 28 and two distributors were used for refrigerant distribution. Refrigerant was distributed to heat exchanger paths through small diameter tubes which have different specification with each other. Using the different small diameter tube specification, mass flowrate of refrigerant of each path were controlled. Totally 13 cases of heating condition were tested to verify the effect of the standard deviation of outlet temperature of paths. Outlet temperatures were varied by changing the small diameter tube specification. Figure 6 shows the effect of standard deviation of outlet temperature on COP of 20Hp heat pump system. The experiments were conducted by changing small diameter specification only for heating condition. The effect on system COP is nearly linear. The graph shows that the decrease of the standard deviation of evaporator outlet temperature of 1°C can lead to the heating COP increase of nearly 10%.

Figure 7 shows the temperature distribution result for 16Hp heat pump system which was adjusted finally. The average outlet temperature is 1.6 ^{o}C and standard deviation is 1.14 ^{o}C. Design target of the average outlet temperature was 2 ^{o}C but the result showed 1.6 ^{o}C which is slightly lower than target. We could get the maximum system COP at the lowest standard deviation of outdoor evaporator outlet temperature.

We also conducted more experiments by varying compressor operation frequency for the same system. And the system COP change and temperature distributions due to the capacity variation were analyzed. Figure 8 shows the behavior of temperature distribution varying system capacity. The standard deviation of evaporator outlet temperature is 1.14 ^{o}C, 1.27 ^{o}C, and 1.55 ^{o}C for 16Hp, 18Hp, and 20Hp respectively. Higher capacity at the same system shows higher standard deviation of outdoor evaporator outlet temperature and lower system COP. When refrigerant mass flowrate increases, the outlet temperature behavior becomes unstable at the same heat exchanger. And the evaporation temperature and system COP decreases, as the system capacity increases.

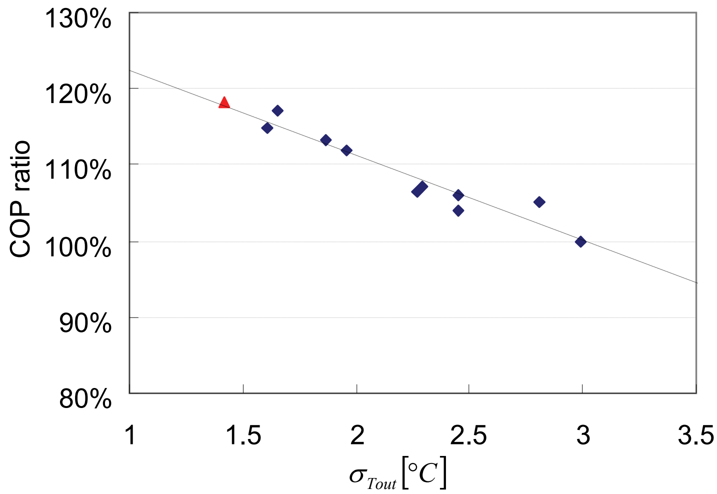

Figure 6 Effect of standard deviation of outlet temperature on system COP

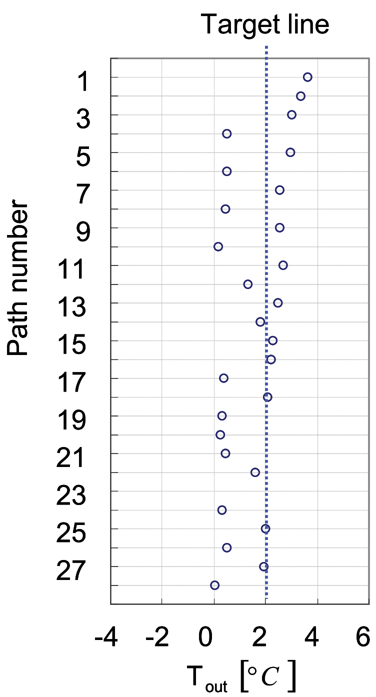

Figure 7 Temperature distribution of 16Hp system

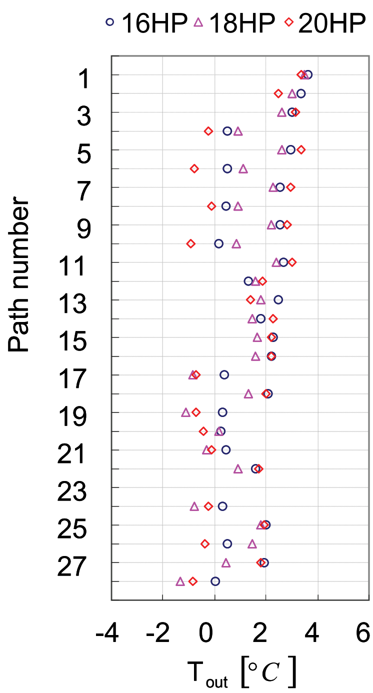

Figure 8 Temperature distribution w.r.t. system capacity

4. CONCLUSIONS

In this paper, the effect of the standard deviation of outdoor evaporator outlet temperature on system COP of heating condition was evaluated and analyzed. We could verify the relation between system COP and the standard deviation of outdoor evaporator outlet temperature. Main implications of this paper are like below.

- The decrease of 1°C of the standard deviation of outdoor evaporator outlet temperatures can lead to the increase of nearly 10% of heating COP.
- Larger capacity shows higher standard deviation of outdoor evaporator outlet temperature and lower system COP for the same system.
- Design considering the standard deviation of evaporator outlet temperatures is very important to get maximum heat exchanger performance

NOMENCLATURE

D	Diameter	(m)
dP/dz	Pressure gradient	(Pa/m)
f	Friction factor	(dimensionless)
G	Mass flux	(kg/m^2s)
P	Pressure	(Pa)
Pc	Critical pressure	(Pa)
Re	Reynolds number	(dimensionless)
v	Specific volume	(m^3/kg)
x	Vapor quality	(dimensionless)
σ	Standard deviation	(^oC)
Φ	Two phase multiplier	(dimensionless)

Subscripts

l	liquid phase
lo	all liquid pahse

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