The Performance Analysis of the Air-to-Air Heat Pump Package Air Conditioner

Zhou Zicheng
Guangdong Rowa Air Conditioner Co. Ltd. Guangdong 529200

ABSTRACT

In this paper, computer simulation technique is employed to analyze the performance of a KFR-73LW/AY air-to-air heat pump package air conditioner . From the performance obtained at different air flow rates and different heat exchanging areas , the results are presented with great satisfaction and they can be used for system optimization.

Key words: Package air conditioner Heat pump

1. INTRODUCTION

Usually, many experimental work will be done to obtain the satisfying performance of air-to-air heat pump package air conditioner during designing stage. Parameters are tested to match the refrigeration system under different conditions such as compressors of different sizes, fans of different air flow rates or different area heat exchangers. It takes large amounts of both manual work and experimental time. However, using computer simulation technique, we are able to get good results with great ease and effectiveness.

In this paper, a KFR-73LW/AY air-to air heat pump package air conditioner is taken as an example to modify fin pitch, airflow rate of condenser and evaporator. With the good heat pump performance accomplished by the simulation, not only does it demonstrate the method we described, but also gives us a direction to search for the optimal heat pump system parameters.

2. MATHEMATICAL MODEL

2.1 COMPRESSOR MODEL

There are two ways to simulate the compressor performance. One is to calculate the heating capacity and the refrigerant mass flow rate by using the refrigerant thermodynamic properties and refrigeration cycle. While the others is to use the ten coefficient polynomials. In this paper, the latter method is used. The ten coefficients data are provided by the compressor company.

The power conception (w) and the refrigerant mass flow rate (lbm/s) of compressor at different evaporating temperatures and different condensing temperatures can be calculated by the following polynomial equations:

$$P = P_1 + P_2 \times (S) + P_3 \times (D) + P_4 \times (S^2) + P_5 \times (S \times D) + P_6 \times (D^2) + P_7 \times (S^3) + P_8 \times (D \times S^2) + P_9 \times (S \times D^2) + P_{10} \times (D^3) \quad \ldots(1)$$

$$M = M_1 + M_2 \times (S) + M_3 \times (D) + M_4 \times (S^2) + M_5 \times (S \times D) + M_6 \times (D^2) + M_7 \times (S^3) + M_8 \times (D \times S^2) + M_9 \times (S \times D^2) + M_{10} \times (D^3) \quad \ldots(2)$$

Where: P = power conception (w);
M = refrigerant mass flow rate (lbm/s);
S = suction temperature (^{o} F);
D =discharge temperature (^{o}F);
$P_1 \sim P_{10}$ = coefficient in eq.(1);
$M_1 \sim M_{10}$ = coefficient in eq.(2).

P_1=6.028E2; P_2=2.661E0; P_3=6.671E0; P_4=-6.912E-2; P_5=-4.730E-2 ; P_6=2.883E-2 ; P_7=7.253E-4 ; P_8= 2.878E-4; P_9=9.786E-5; P_{10}=4.919E-4 ; M_1=2.526E2; M_2=4.192E0; M_3=-2.372E0; M_4=4.184E-2; M_5=-3.538E-3; M_6=2.515E-2; M_7=-2.58E-5; M_8=-2.402E-5; M_9=2.533E-5; M_{10}=-1.009E-4.

2.2 MODELS OF CONDENSER AND EVAPORATOR

$$Q = m_1 c_1 (t_1' - t_1'') = m_2 c_2 (t_2'' - t_2') \quad \ldots(3)$$

$$\frac{t_1'' - t_2''}{t_1' - t_2'} = \exp\{-\frac{KA}{m_2 c_2}[1 + \frac{m_2 c_2}{m_1 c_1}]\} \quad \ldots(4)$$

$$NTU = \frac{KA}{m_2 c_2} \quad \ldots(5)$$

Where:

Q= heat transfer rate;

m_1,m_2= two mass flow rate in the heat exchanger;

c_1,c_2= specific heat of two streams;

t_1’,t_1’’ and t_2’,t_2’’=enter temperature and exit temperature of two streams;

K= heat transfer coefficient;

A= area.

3.EVALUATION OF THE SIMULATION MODEL

The simulation model is evaluated by the experimental test at the different ranges of capacity and different rating conditions. The results are shown in table 1.

Table 1 Simulation and test data

type	Capacity w		EER/COP	
	simulation	measured	simulation	measured
KFR-50LW/AY Cooling model, rotary compressor	4883.34	4865.1	2.5415	2.45
KT3FR-60LW/AY Cooling model, T3 condition, Reciprocating compressor	6142.47	6048	2.071	1.96
KFR-73LW/AY Heating Model ,Scroll Compress	7915.06	7805	2.917	2.97

4.MATCH PARAMETERS

Several groups which consist of different air flow rate and different fin pitch are shown in table 2.

Table 2 Match parameters

group	Indoor heat exchanger		outdoor heat exchanger	
	Air flow rate m^3/h	Fin pitch mm	Air flow rate m^3/h	Fin pitch mm
A	1100	1.5	2700	1.8
B		1.6		1.8
C		1.7		1.8
D	1100	1.5	2700	2.0
E		1.6		2.0
F		1.7		2.0
G	1100	1.5	2800	1.8
H		1.6		1.8
I		1.7		1.8
J	1100	1.5	2800	2.0
K		1.6		2.0
L		1.7		2.0
M	1200	1.5	2700	1.8
N		1.6		1.8
O		1.7		1.8
P	1200	1.5	2700	2.0
Q		1.6		2.0
R		1.7		2.0
S	1200	1.5	2800	1.8
T		1.6		1.8
U		1.7		1.8
V	1200	1.5	2800	2.0
W		1.6		2.0
X		1.7		2.0

5.PERFORMANCE ANALIZE

5.1 HEATING CAPACITY, POWER CONCEPTION AND COP OF HEAT PUMP

Fig. 1 shows the heating capacity , power conception and COP of the air-to-air heat pump package air conditioner at the rating conditions and different parameters. It can be concluded that, when the air flow rate increase and the fin pitch decrease, the larger the heating capacity and COP become , the smaller the power conception is .

When the indoor air flow rate changes from 1100 m^3/h to 1200 m^3/h, the fin pitch changes from 1.5 mm to 1.7 mm, and the outdoor air flow rate changes from 2700 m^3/h to 2800 m^3/h, the fin pitch changes from 1.8mm to 2.0mm, the heating capacity changes from 7891.83 w to 7945.99 w , the power conception changes from 2593.46 w to 2745.99 w, and the COP changes from 2.88 to 3.058. Obviously, the power conception and the COP change greater and the heating capacity changes smaller .

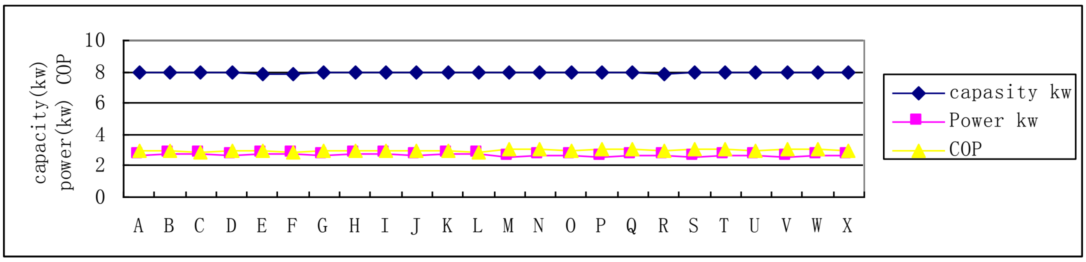

Figure 1 The heating capacity (kw), power conception (kw) and COP

5.2 HEAT EXCHANGER PERFORMANCE OF CONDENSER

Fig.2 shows the heat transfer coefficients of condenser refrigerant side in the vapor region, two phase region and sub cooled region. It can be seen that the heat transfer coefficients will be larger when the airflow rate increase and the fin pitch decrease. It can also be seen that the heat transfer coefficient changes smaller in the vapor region and the sub cooled region, the two phase regions where the main effects lie on.

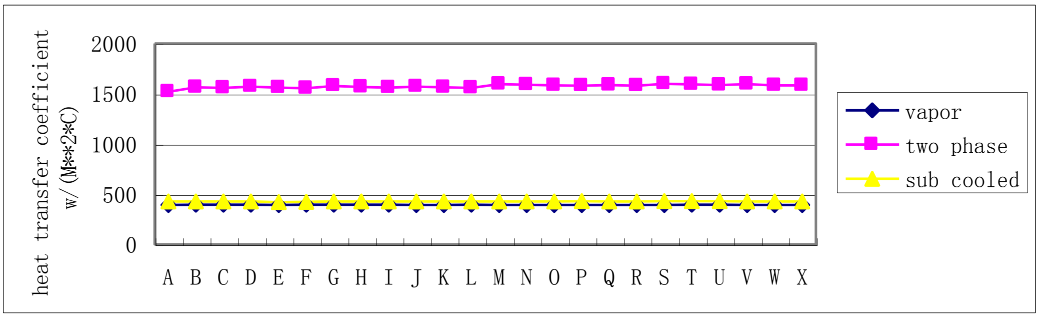

Figure 2 Heat transfer coefficients at the refrigerant side

Fig.3 shows the heat transfer coefficient of the airside of condenser. It can be seen that the air side heat transfer coefficient is much smaller than the refrigerant side, and it will become larger when the air flow rate increase and fin pitch decrease.

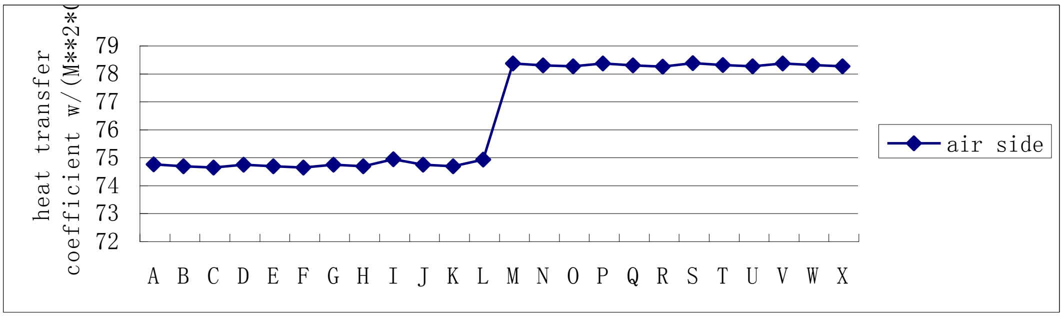

Figure 3 Air side heat transfer coefficient

5.3 HEAT EXCHANGER PERFORMANCE OF EVAPORATOR

Fig. 4 shows the heat transfer coefficients of evaporator refrigerant side in the vapor over heated region and the two phase region. It can be seen that the heat transfer coefficient is mainly dependent on the airflow rate.

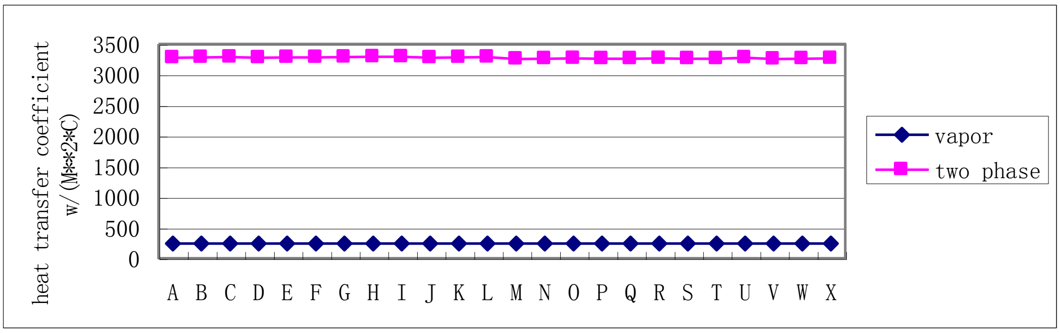

Figure 4 heat transfer coefficients at the refrigerant side

Comparing with the condenser and evaporator, the heat transfer coefficient in the evaporator two phase region is about two times as that in the condenser. Therefore, the heat exchanger area of evaporator is usually smaller than that of the condenser. Fig.5 shows the dry coil region and the wet coil region of airside heat transfer coefficients. They also depend on the airflow rate.

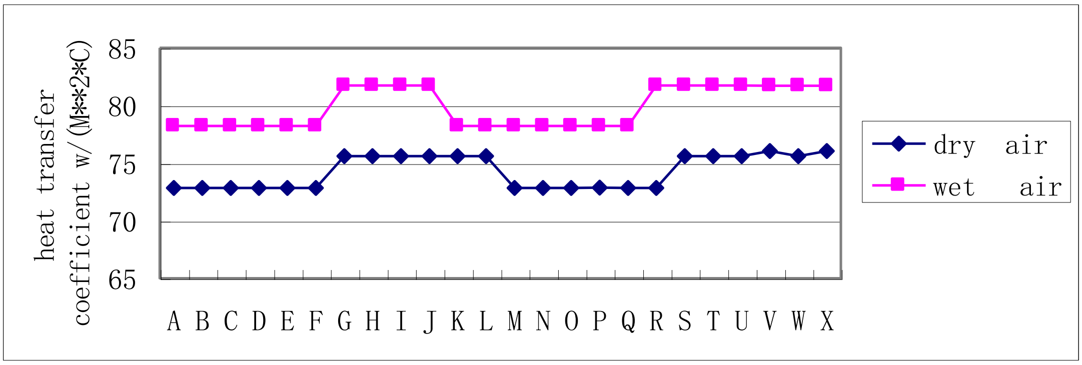

Fig. 5 Air side heat transfer coefficients

It is similar to the condenser that the air side heat transfer coefficients either in the dry coil region or in the wet coil region are much smaller than in the two phase region. It about 1/45 to 1/40 of the two phase region.

6.CONCLUSIONS:

6.1 It is evident that the computer simulation is a powerful method used for heat pump air conditioner optimal design.

6.2 The power conception and COP of air-to-air heat pump package air conditioner vary more obviously than those of the heating capacity when the air flow rate and the fin pitch change.

6.3 The heat transfer coefficient at the refrigerant side is much greater than that at the air side, and the heat transfer coefficient in the evaporator is greater than that in the condenser.

6.4 It is important that the refrigerant flow rate must be divided into the same parts to each refrigerant circuit during the heat pump design procedure, because the heat transfer coefficient in these two phase regions is much greater than that in the other regions.

7. REFERENCES:

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