Development of Heat Pump System for the Greenhouse Heating

Dr. Song, Hyun-Kap
Dept. of Agr. Machinery Eng.
Chungbuk National University
Cheongju 360-763, Korea

Dr. Ryou, Young-Sun
Agr. Mechanization Institute
Suwon, Korea

Park, Jong-Kil
Dept. of Agr. Machinery Eng.
Graduate school
Chungbuk National University
Cheongju 360-763, Korea

ABSTRACT

It is desirable to use the renewable energy for the greenhouse heating in winter season, it makes possible not only to save fossil fuel and conserve green farm environment but also to promote the quality of agricultural products and reduce the agricultural production cost.

In this study the heat pump system was developed to use the natural energy as thermal energy resources for the thermal environment control of the greenhouse.

Key word : Heat Pump, COP, AVACTHE, Ambient Temperature

1. INTRODUCTION

In order to use the natural energy resources as much as possible for the greenhouse heating the heat pump was designed on the basis of the weather conditions in the winter season of Korea. But it's not easy to compose the heat pump system operating with high COP at the ambient temperature below -5℃.

In this study to solve this problem, the AVACTHE(Automatic Variable Area Capillary Type Heat Exchanger) was devised and installed in the refrigerant circuit of the heat pump system, and the AVACTHE effect on the performance of heat pump system was tested as a function of the ambient temperature and the greenhouse heating effect of the heat pump installed with AVACTHE has been analyzed experimentally.

2. THEORETICAL ANALYSIS OF THE HEAT PUMP SYSTEM

2.1 P-H Diagram for the Heat Pump Design

The Pressure-Enthalpy diagram of the heat pump is shown in Fig. 1.

As shown in Fig. 1, the dot line is cycle of heat pump without AVACTHE and solid line is cycle of heat pump with AVACTHE installed between condenser outlet and evaporator inlet. The degree of superheat(①'~①) was appeared before the compressor inlet and the degree of supercool(③'~④) before the expansion valve inlet.

2.2 Power Requirement of Heat Pump Compressor

The compressor power requirement of heat pump in the heating mode could be determined as follow:

$$W_{R_H} = \frac{W_{th}}{\eta_i \cdot \eta_m} = \frac{\dot{G}_{OH}(h_2 - h_1)}{\eta_i \cdot \eta_m}$$

$$= \frac{h_2 - h_1}{h_2 - h_5} \cdot \frac{\dot{Q}_{OH}}{\eta_i \cdot \eta_m} \qquad \text{-----------------(1)}$$

$$W_{th} = \frac{h_2 - h_1}{h_2 - h_5} \cdot \dot{Q}_{OH}$$

$$\dot{G}_{OH} = \frac{\pi D^2}{4 V_1} \cdot L \cdot z \cdot n \cdot \eta_v \qquad \text{-----------------(2)}$$

Substituting equation (2) into equation (1)

$$W_{R_H} = (h_2 - h_1) \cdot \frac{\pi D^2}{4 V_1} \cdot L \cdot z \cdot n \cdot \frac{\eta_v}{\eta_i \cdot \eta_m} \qquad \text{---------(3)}$$

Where, η_i : compression efficiency

η_m : mechanical efficiency of compressor

η_v : volume efficiency

D : piston diameter of compressor (㎜)

$\dot{G}_{OH}$: refrigerant mass flow rate in the heating mode (kg/hr)

h_1 : enthalpy of refrigerant in the initial state of compression process (㎉/kg)

h_2 : enthalpy of refrigerant at the end of compression process (㎉/kg)

h_5 : enthalpy of refrigerant at the end of expansion process (㎉/kg)

L : piston stroke (㎜)

n : revolution velocity (N/s)

$\dot{Q}_{OH}$: greenhouse heating load (㎉/hr)

V_1 : specific volume of refrigerant (㎥/kg)

W_{R_H} : the real power requirement of heat pump compressor in the heating mode (㎾)

z : number of piston

2.3 Heat Transfer Area of Condenser

The condenser heat transfer area is determined by the following equations:

$$\dot{Q}_{CO} = A_t \cdot U \cdot \Delta T_{LM}$$

$$A_t = \frac{\dot{Q}_{CO}}{U \cdot \Delta T_{LM}} \qquad \text{------------------------(4)}$$

Terms of ΔT_{LM} and U in the equation (4) can be written as follow

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\dfrac{\Delta T_1}{\Delta T_2}\right)} \qquad \text{----------------(5)}$$

$$\Delta T_1 = (T_{h.in} - T_{L.out})$$

$$\Delta T_2 = (T_{h.out} - T_{L.in})$$

$$\frac{1}{U} = \frac{1}{\bar{\alpha}_{air}} + \frac{D_r}{2\lambda_t}\ln\frac{D_r}{D_i} + \frac{1}{\alpha_m}\frac{D_r}{D_i} \qquad \text{---------(6)}$$

And $\bar{\alpha}_{air}$ and α_m terms in this equation (6) can be written as follow:

$$\bar{\alpha}_{air} = \left(\frac{\eta_f \cdot A_f + A_W}{A_t}\right)\left[0.242\left(\frac{v_{\max} \cdot \rho_a \cdot D_r}{\mu_a}\right)^{0.688}\left(\frac{s}{h}\right)^{0.297} P_{ra}^{\frac{1}{3}}\right]\frac{\alpha_a}{D_r} \qquad \text{--(7)}$$

$$\alpha_m = \frac{\alpha_{in} + \alpha_{out}}{2} = \frac{\alpha_{LO}}{2}\left[\left\{1 + X_{in}\left(\frac{\rho_L}{\rho_g} - 1\right)\right\}^{\frac{1}{2}} + \left\{1 - X_{out}\left(\frac{\rho_L}{\rho_g} - 1\right)\right\}^{\frac{1}{2}}\right] \qquad \text{--(8)}$$

$$\alpha_{LO} = 0.021\frac{\lambda_L}{D_i} \cdot Re_{LO}^{0.8} \cdot Pr_l^{0.43}$$

Where, $\bar{\alpha}_{air}$: CHTC of air (W/㎡K)

α_m : average CHTC of refrigerant in the condenser (W/㎡K)

α_{in} : CHTC of refrigerant at condenser inlet (W/㎡K)

α_{out} : CHTC of refrigerant at condenser outlet (W/㎡K)

α_{LO} : convection heat transfer coefficient(CHTC) of saturated refrigerant in liquid phase (W/㎡K)

λ_t : thermal conductivity of tube wall (W/mK)

η_f : efficiency of high fins

μ_a : viscosity (kg · s/㎡)

ρ_g : density of refrigerant in vapor phase (kg/㎥)

ρ_L : density of refrigerant in liquid phase (kg/㎥)

A : total surface area of the finned tube bank (㎡)

A_f : surface area of the fins (㎡)

A_t : total surface area of condenser tubes without fins (㎡)

A_W : surface area of the tube between fins (㎡)

D_i : inside diameter of tube (㎜)

D_r : outside diameter of tube (mm)

Pr_a : Prandtle number of air

Pr_l : Prandtle number of refrigerant in liquid phase

$\dot{Q}_{CO}$: radiated heat from condenser(= $\dot{Q}_{OH}$: greenhouse heating load) (kcal/hr)

ΔT_{LM} : logarithm mean temperature difference (K)

$T_{h.in}$: refrigerant temperature at condenser inlet (K)

$T_{h.out}$: refrigerant temperature at condenser outlet (K)

$T_{L.in}$: air temperature at condenser inlet (K)

$T_{L.out}$: air temperature at condenser outlet (K)

S : space between fin and fin (mm)

U : overall heat transfer coefficient of condenser (W/m²K)

$v_{\max}$: velocity of the air passing through the minimum cross sector (m/s)

x_{in} : quality of refrigerant at condenser inlet (%)

x_{out} : quality of refrigerant at condenser outlet (%)

2.4 Heat Transfer Area of AVACTHE

Heat transfer area of capillary tube in the AVACTHE can be determined by the following equations:

$$U \cdot N \cdot A_{to} \cdot \Delta T_{LM} = \dot{G}_{O.si}(0.15\rho_L+0.85\rho_g)\times(0.15c_{P_L}+0.85c_{P_g}) \cdot \Delta T_{LM}$$

$$\therefore A_{to}=\frac{\dot{G}_{O.si}(0.15\rho_L+0.85\rho_g)\times(0.15c_{P_L}+0.85c_{P_g})}{U \cdot N} \quad \text{--(9)}$$

U and $\dot{G}_{O.si}$ terms in this equation can be written as follow:

$$\frac{1}{U} = R_A+R_B \cdot \left(\frac{D_1}{D_2}\right)+\frac{D_1}{2K_W} \cdot \ln\left(\frac{D_1}{D_2}\right)+\frac{1}{\alpha_A}+\frac{1}{\alpha_B} \cdot \left(\frac{D_1}{D_2}\right) \quad \text{-(10)}$$

$$\dot{G}_{O.si} = \frac{1}{3600\rho_{si}} \cdot \frac{P_{(ps)} \cdot 0.7355_{(kW/ps)} \cdot 860_{(kcal/kW\,hr)} \cdot COP}{h_2-h_4} \quad \text{-(11)}$$

$$R_A=0.00035 \ (m^2K/W) \qquad R_B=0.000175 \ (m^2K/W)$$

$$\alpha_A=\frac{Nu_{si} \cdot Kw_{si}}{De_{si}} \qquad \alpha_B=\frac{Nu_{si} \cdot Kw_{si}}{De_{si}}$$

Where, α_A : convective film heat transfer coefficient on the outer fouling layer (W/m²K)

α_B : convective film heat transfer coefficient on the inner fouling layer (W/m²K)

ρ_g : density of refrigerant in gas phase (kg/m³)

ρ_L : density of refrigerant in liquid phase (kg/m³)

A_w : cross section area of tube outside (m²)

COP : Coefficient of Performance of heat pump

c_{P_g} : specific heat of refrigerant in gas phase (kcal/kg °K)

c_{P_L} : specific heat of refrigerant in liquid phase (kcal/kg °K)

D_1 : outside diameter of capillary tube (mm)

D_2 : inside diameter of capillary tube (mm)

De_{si} : equivalent diameter of th shell (mm)

De_{ti} : equivalent diameter of capillary tube bundle (mm)

$\dot{G}_O$: mass flow rate of refrigerant (kg/sec)

$\dot{G}_{O,si}$: volume flow rate of refrigerant in the shell

h_2 : enthalpy of refrigerant compressed by compressor (kcal/kg)

h_4 : enthalpy of refrigerant at condenser outlet (kcal/kg)

Kw : thermal conductivity of the capillary tube wall (W/m²K)

Kw_{si} : thermal conductivity of the shell side

Kw_{ti} : thermal conductivity of the tube side

N : number of capillary tube

Nu_{si} : Nusselt number of refrigerant in the shell

Nu_{ti} : Nusselt number of refrigerant in the tube

P : compressor power (ps)

R_A : fouling resistance of outer fouling layer on the tube

R_B : fouling resistance of inner fouling layer on the tube

ΔT_{LM} : logarithm mean temperature difference (K)

T_A : bulk temperature of fluid A flowing in the shell (K)

T_B : bulk temperature of fluid B flowing in the shell (K)

U : overall heat transfer coefficient of shell and tubes (W/m²K)

3. COMPOSITION OF HEAT PUMP SYSTEM EXPERIMENTAL EQUIPMENT

3.1 Heat Pump System and Experimental Equipment

In order to develop the greenhouse heating system using natural energy resources, the heat pump with function of air to water was composed for the greenhouse heating, and the test facility was equipped in the heat pump circuit(Ref. Fig. 2-(a), (b)).

As the ground water temperature could be kept a range of 7~10℃ even in the winter season, the water to water and the water to air heat pump systems using the water as low temperature heat resources are studied and realized in the these days.

However it is not always possible to find out the abundant ground water resources around the greenhouse area, therefore the air to air and air to water heat pump system using the ambient air as low temperature heat resources have to be studied for greenhouse heating system in the winter season. But it's not easy to compose the heat pump system operating with high COP at the ambient air temperature below -5℃.

In this study to solve this problem, the AVACTHE was installed in the circuit of heat pump system as shown in Fig. 2, and the details of tested the heat pump system was shown in table 1.

Table 1. Detail of the tested Greenhouse-Heat pump system

Items / System	Size	Mode	Working Fluid
Heat pump	3 ps	Heating Mode (Air to Water)	R22

3.2 Experimental Method

In order to test the thermal characteristics and COP of the heat pump system(air to water) the experimental variables and measuring items were arranged as shown in table 2.

Table 2. Combination of experimental variables and measuring items

system / variable / measuring items		Heat Pump System Air to Water Heat Pump (Open Loop)
· Ambient temperature (℃)		-12 ~ 15
· Condenser inlet temperature of heat transfer fluid (℃)		7 ~ 15
· Condenser outlet temperature of heat transfer fluid (℃)		30 ~ 85
· Water or air flow rate in the heat pump circuit		100 ~ 300 (l/h)
· Refrigerant changing rate (kg/m')		240 ~ 400
· Suction pressure in the heat pump circuit (kg/㎠)		1.5 ~ 4.5
· Discharge pressure in the heat pump circuit (kg/m')		11.5 ~ 25.0
· Electric power consumption (W)		1800 ~ 3500
· Variation of heat transfer area of AVACTHE (㎠)	1-Exchanger	111.8
	2-Exchanger	123.0
	3-Exchanger	134.2
· Data acquisition interval (min)		1

This experimentation was conducted from January 1st to April 30th, 1996. As the ambient air temperature was varied from -12℃ to 15℃ during the experimental period, the ambient air temperature aground the evaporator and the temperature of water supplied to the condenser were controlled easily by the natural weather conditions.

4. RESULTS AND DISCUSSION

4.1 AVACTHE Effect on the Condenser Inlet Temperature

The Fig. 3-(a) presents the AVACTHE effect on the condenser inlet temperature, it shows that the condenser inlet temperature was increased in proportion to the

AVACTHE area, this result suggests that the heat pump COP can be improved by AVACTHE installation in the refrigerant circuit.

The Fig. 3-(b) presents the ambient temperature effect on the condenser inlet temperature in case of the bypass and the AVACTHE installation. In case of both bypass and AVACTHE, the condenser inlet temperature was increased in proportion to the ambient temperature. The temperature of condenser in case of AVACTHE was 15~16% higher than in case of the bypass(Without AVACTHE).

4.2 Ambient Temperature Effect on the Heat Exchange in AVACTHE Tubes

The ambient temperature effect on the heat exchange in the AVACTHE tubes is presented in Fig. 4, it shows that the inlet and outlet temperature difference of tubes in AVACTHE is decreased inversely proportional to the ambient temperature, and the inlet and outlet temperature difference of AVACTHE at the ambient temperature below 0℃ is larger than that at the ambient temperature above 0℃, this results indicate that the AVACTHE exerts greater influence at the ambient temperature below 0℃ than that at the ambient temperature above 0℃.

4.3 Construction of Standard Charging Charts in the Heating Mode

However the heat pump system was best designed and constructed, if the refrigerant was not accurately charged in the heat pump circuit, the system could not operate with best performance. Therefore it is important to construct the optimum refrigerant charging charts and to use these charts for the best performance of the system.

The heat pump performance is affected not only by the ambient temperature but also the suction and discharge pressure that are changed in response to the different refrigerant charging levels.

These circular, triangular and rectangular points in Fig, 5-(a), (b) are experimental values obtained for various heated water temperature in this study, and solid and dot lines are values taken from "typical heating data chart". As shown in Fig. 5-(a), (b), the experimental values were well agreed with the values of "typical heating data chart". Thus, this chart composed of suction and discharge pressure as a function of ambient temperature for various heated water temperature could be used as the standard chart for the heat pump charge.

4.4 R_{22} Charge Effect on the Compressor Power Consumption and on the Condenser Energy Gain

4.4.1 R22 charge effect on the compressor power consumption

The Fig. 6 presents the R_{22} charge effect on the power consumption of heat pump compressor for various levels of water flow rate at the condenser inlet. In the heating mode of heat pump, the power consumption of compressor was increased in proportion to R_{22} charge mass per the volume of heat pump pipe line.

In all cases of condenser inlet water flow rate of 150 l/h, 200 l/h and 300 l/h, the maximum power consumption was shown at the R_{22} charge mass of 380 kg/㎥. And then the power consumption was increased inversely proportional to the water flow rate of condenser inlet.

Thus it is necessary to accurately analyze the relationship between the compressor power consumption and the energy gain of condenser as a function of R_{22} charge mass per pipe volume.

4.4.2 R22 charge effect on the thermal energy gain of condenser

The R_{22} charge effect on the thermal energy gain of condenser for various level of water flow rate at the condenser inlet was presented in Fig. 7.

As shown in Fig. 7, the energy gain was increased in proportion to the R_{22} charge and the increasing rate of energy gain below 290 kg/㎥ of the R_{22} charge was larger than that above 290 kg/㎥ of the R_{22} charge. It is notable that below 290 kg/㎥ of R_{22} charge, the thermal energy gain was not affected by the water flow rate in the condenser but above the R_{22} charge of 290 kg/㎥ the energy gain was increased in proportion to the water flow rate.

The maximum energy gain in the condenser was achieved at 380 kg/㎥ of the R_{22} charge that the maximum power consumption of compressor was occurred. Thus the optimum charge level can be determined by the COP analysis but not only by the condenser energy gain and the power consumption of compressor.

4.3.3 R22 charge effect on the COP

R_{22} charge effect on the COP for various water flow rate at the condenser inlet was presented in Fig. 8.

As shown in Fig. 8, the COP was increased in proportion to R_{22} charge over a range of 170 kg/㎥ to 400 kg/㎥ and water flow rate of 150 l/h to 300 l/h, but above the 350 kg/㎥ of R_{22} charge, the COP was inversely proportional to the R_{22} charge. In other words, the maximum value of COP was occurred at 350 kg/㎥ of R_{22} charge.

It is notable that the maximum power consumption of compressor and the maximum thermal energy gain of condenser were occurred at 380 kg/㎥ of R_{22} charge, but the maximum COP analyzed with the compressor power consumption and the energy gain was occurred at 350 kg/㎥ of R_{22} charge.

4.5 AVACTHE Effect on the Heat Pump Performance

4.5.1 AVACTHE effect on the power consumption of compressor

The power consumption of compressor as a function of the ambient temperature was shown for various AVACTHE area in Fig. 9.

It showed that the power consumption was increased in proportion to the ambient temperature and the power consumption in case of bypass(without AVACTHE) was larger than that in other two cases of AVACTHE.

The difference between the power consumption in case of bypass and that in case of AVACTHE at the ambient temperature below 0℃ was about 6.6% and the difference at the ambient temperature above 0℃ was about 10.8%.

4.5.2 AVACTHE effect on the thermal energy gain of condenser

The AVACTHE effect on the thermal energy obtained from the condenser as a function of the ambient temperature was shown in Fig. 10.

As shown in Fig. 10, at the ambient temperature below 0℃ the thermal energy gain of condenser was negligible difference between the bypass and the AVACTHE, but at the ambient temperature above 0℃ the energy gain in case of bypass was larger than that in case of AVACTHE.

4.5.3 AVACTHE effect on the COP

The COP variation depending on the AVACTHE area was presented as a function of the ambient temperature in Fig. 11.

As shown in Fig. 11, the COP of heat pump installed with AVACTHE was larger than that without AVACTHE at the ambient temperature below -5℃, but at the ambient temperature above -5℃ was inverse.

As a result it was very important for the AVACTHE to be installed in the circuit of refrigerant not only for the smooth operation of heat pump but also for the COP increase at the ambient temperature below -5℃.

5. CONCLUSION

In these days the greenhouse installed area has been increased rapidly in Korea, therefore the fossil fuel consumption also increased for several years in proportion to the greenhouse heating area.

Now it is very important to use the renewable energy instead of fossil fuel for the greenhouse heating. In this study the heat pump system was developed to use the renewable energy for the greenhouse heating, but it was not easy to compose the heat pump system operating with high coefficient of performance(COP).

The COP of heat pump was affected not only by the ambient temperature but also by the refrigerant charging levels. To solve the COP decrease at the ambient temperature below -5℃, the AVACTHE was devised and installed in the refrigerant circuit and the standard charging charts were constructed to control the refrigerant charge levels, and the thermal characteristics of the heat pump system was analyzed experimentally.

These results could be summarized as follows:

1. Optimum suction pressure of the heat pump was 1.6~2.0 kg/㎠ at the ambient temperature of -10~-5℃, and optimum discharge pressure was 15~22 kg/㎠ at the ambient temperature of -8~8℃.

2. The AVACTHE makes possible no only to operate the heat pump smoothly but also promote the COP to 4.2 at the ambient temperature below -5℃.

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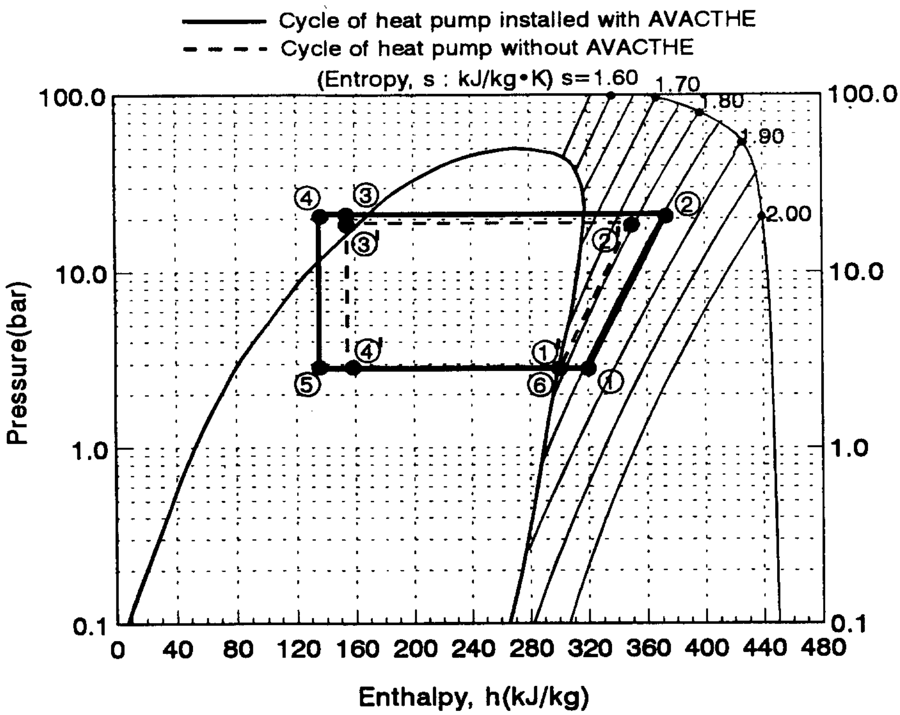


Fig. 1 P-h diagram of heat pump

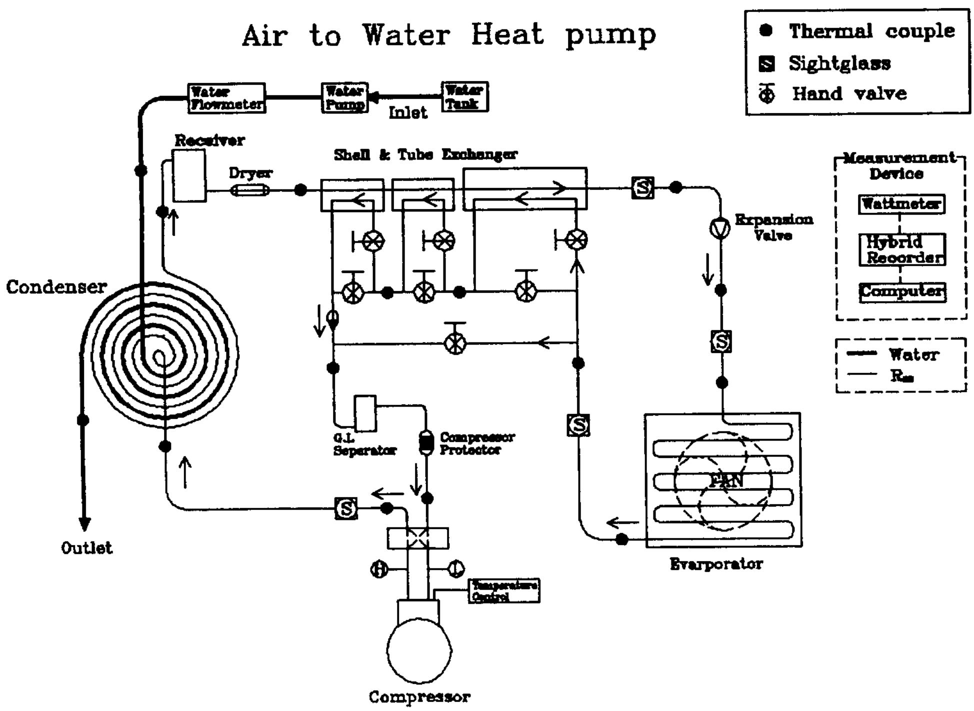


Fig. 2-(a) Heat pump system and experimental apparatus

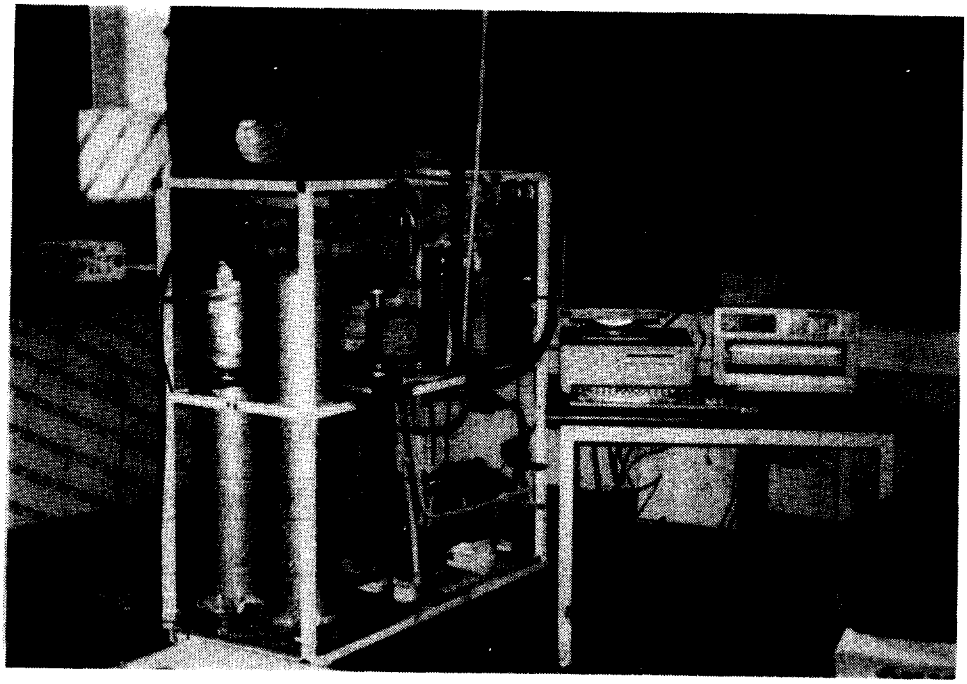


Fig. 2-(b) Photo of heat pump system

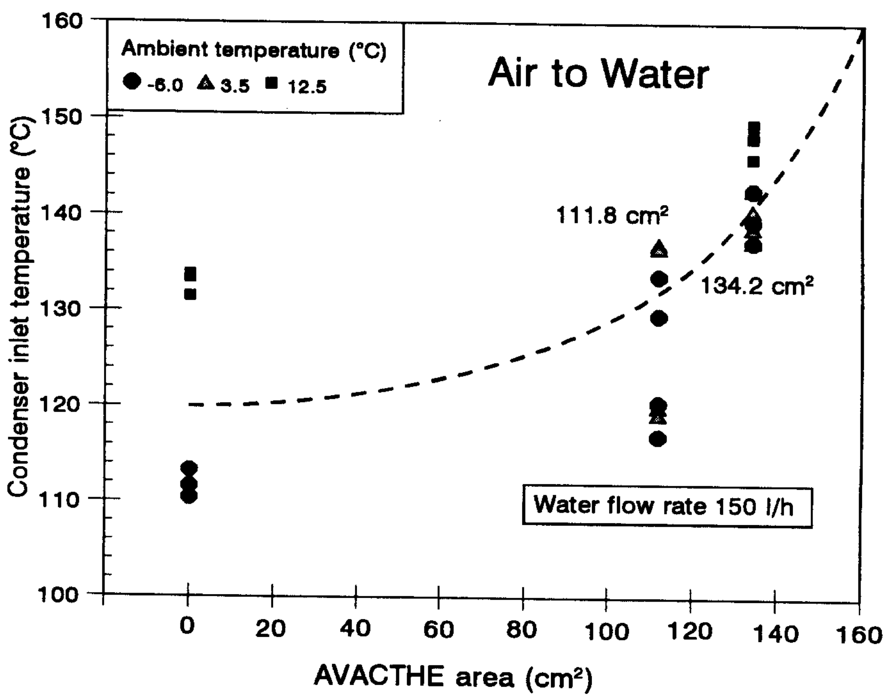


Fig. 3-(a) AVACTHE effect on the condenser inlet temperature

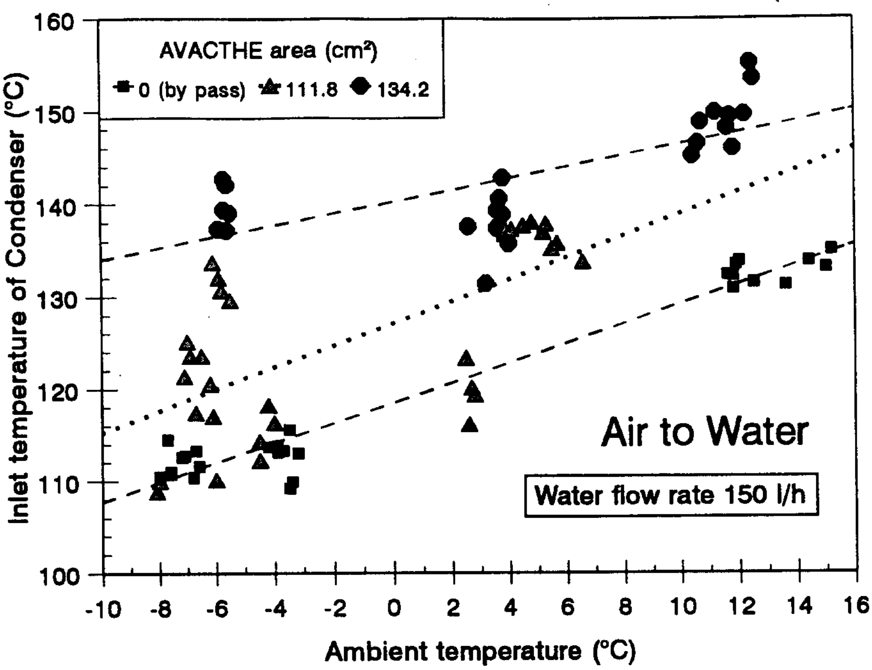


Fig. 3-(b) Ambient temperature effect on the condenser inlet temperature

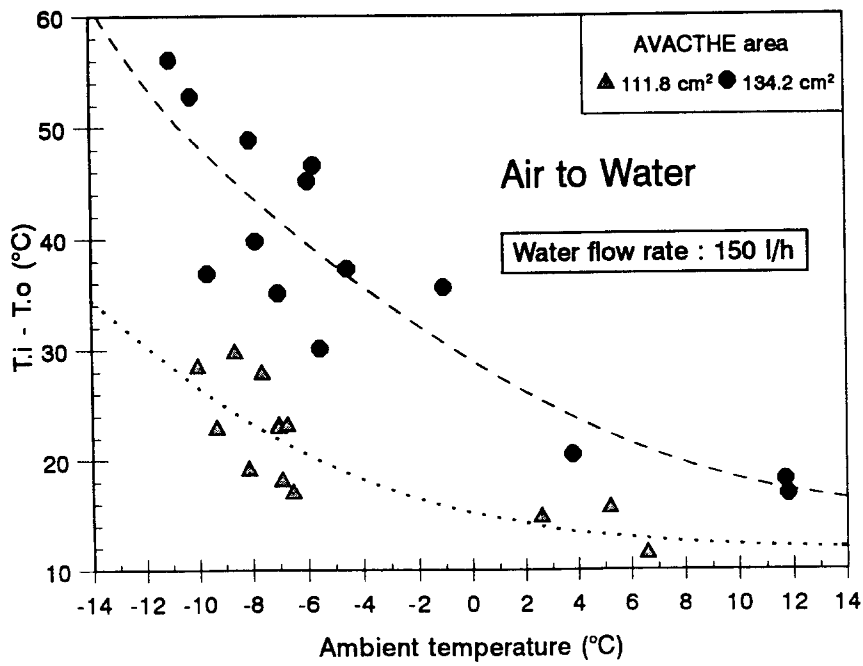
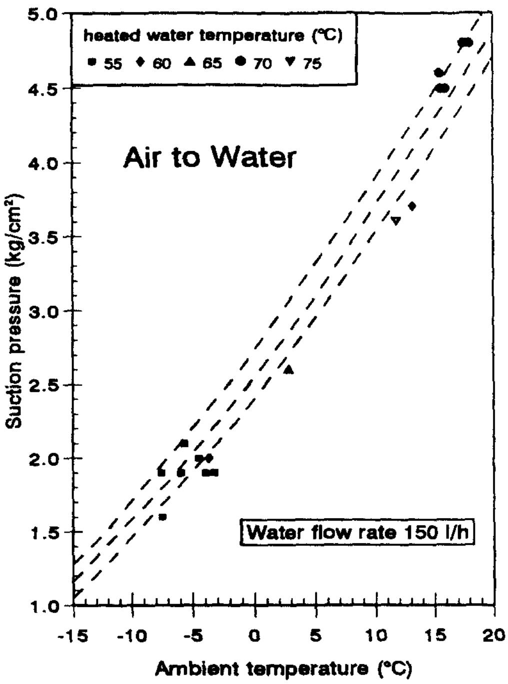
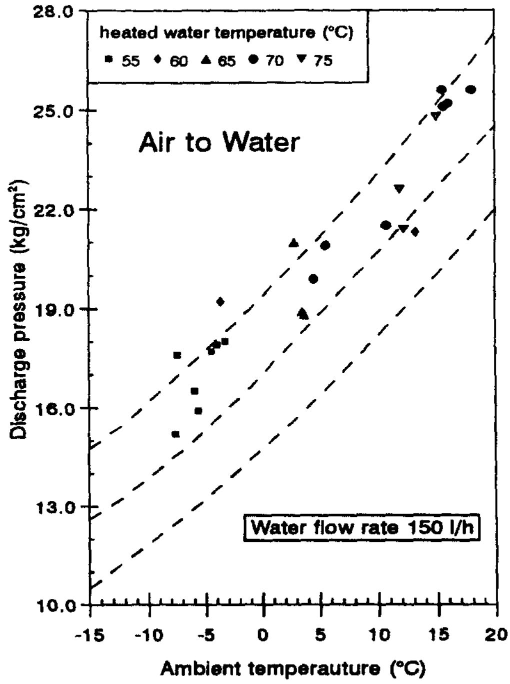


Fig. 4 Ambient temperature effect on the inlet and outlet temperature difference of AVACTHE tubes



(a) Suction pressure-Ambient temperature



(b) Discharge pressure-Ambient temperature

Fig. 5 Refrigerant standard charging chart in the heating mode

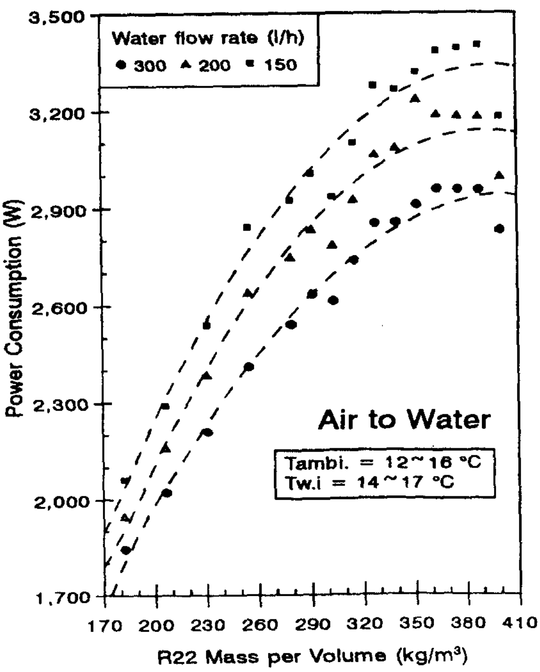


Fig. 6 Relationship between R22 charge mass and compressor power consumption

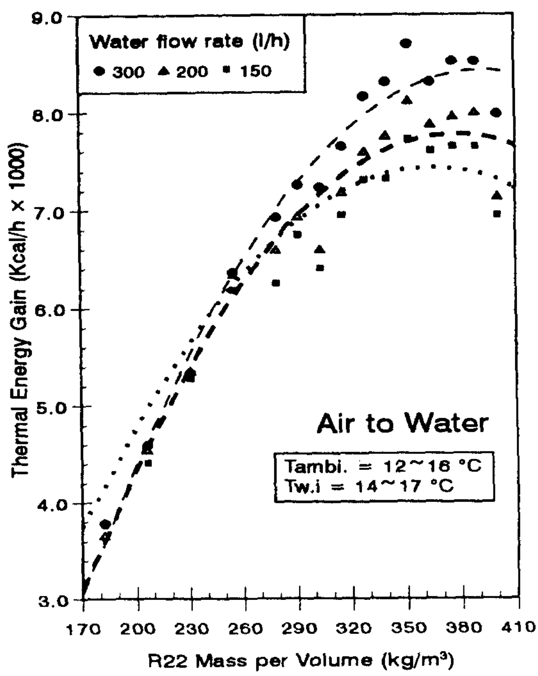


Fig. 7 R22 charge effect on the thermal energy gain of condenser in accodrdance with water flow rate

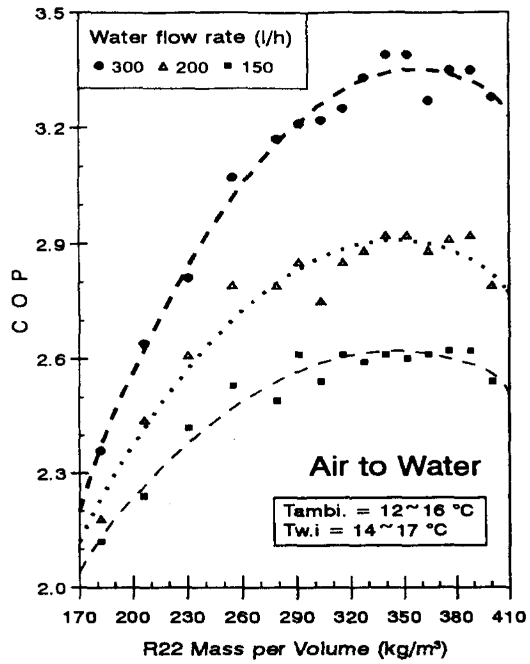


Fig. 8 R22 charge effect on the COP for various water flow rate in condenser

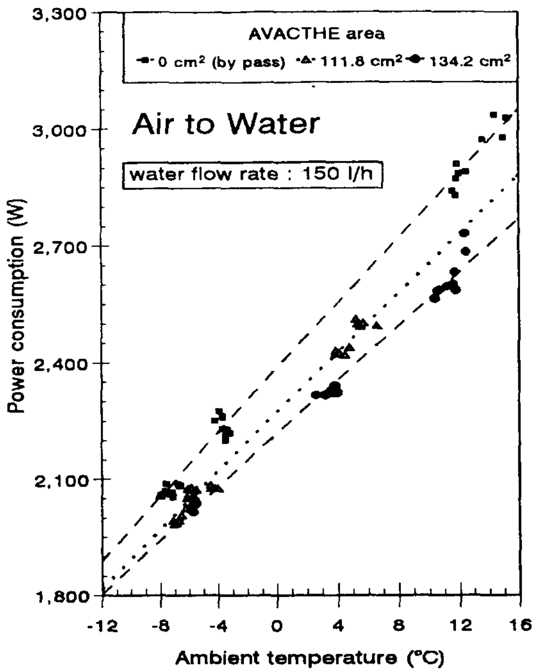


Fig. 9 The AVACTHE effect on the compressor power consumption as a function of the ambient temperature

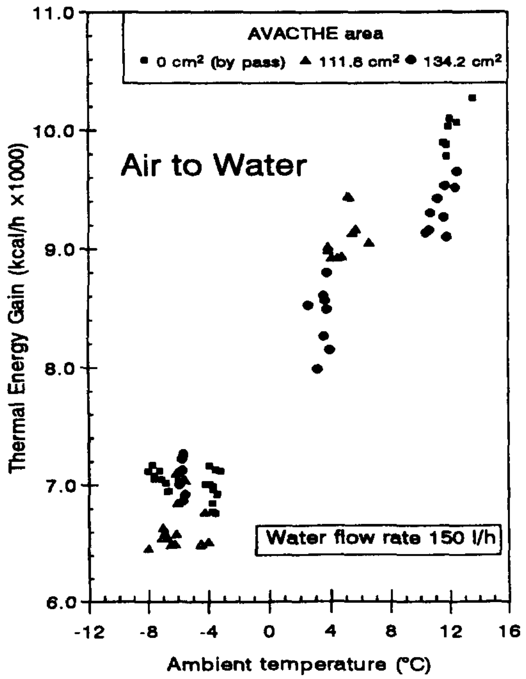


Fig. 10 The AVACTHE effect on the thermal obtain energy obtained from condenser as a function of the ambient temperature

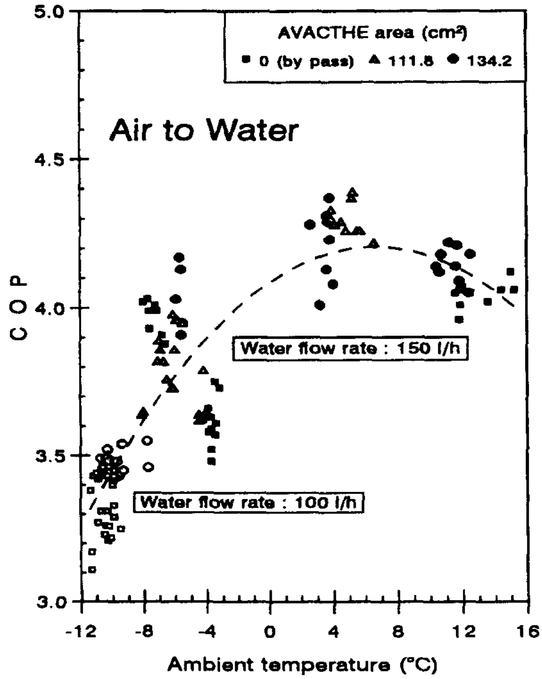


Fig. 11 The AVACTHE effect on the COP as function of the ambient temperature