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Experimental and Theoretical Studies on the Transient Characteristics During Speed Up of Inverter Heat Pump

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EXPERIMENTAL AND THEORETICAL STUDIES ON THE TRANSIENT CHARACTERISTICS DURING SPEED UP OF INVERTER HEAT PUMP

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ABSTRACT

A series of tests were performed to verify the transient characteristics of heat pump in heating and cooling mode when operating speed was varied over the 30 to 102Hz. One of the major issues that has not been addressed so far is transient characteristics during speed modulation. The model for cycle simulation has been developed to predict the cycle performance under conditions of increasing drive frequency and the results of the theoretical study were compared with the results of the experimental study. The simulated results were in good agreement with the experimental result within 10%. The transient cycle migration of the liquid state refrigerant causes significant dynamic change in system. Thus, the migration of refrigerant was the most important factor whenever do experimental results analysis or develop simulation model.

NOMENCLATURE

A	: heat transfer area, m^2
d	: tube diameter, mm
H	: enthalpy, kJ/kg
h	: heat transfer coefficient, $W/m^2\,K$
m	: mass, kg
$\dot{m}$	: mass flow rate of refrigerant, kg/h
Q	: heat capacity, W
P	: pressure, MPa
Γ	: conversion factor(=1/427kcal/kgm),
t	: time
T	: temperature, ℃
U	: overall heat transfer coefficient, W/m^2K
V	: volume, m^3
v	: specific volume, m^3/kg
x	: length, m / quality
Z	: flow direction
ρ	: density(kg/m^3)

INTRODUCTION

A system with a variable speed compressor is becoming more beneficial because it can respond properly to outdoor loads in order to complete the necessary capacity. It is called a "variable-speed or modulating heat pump" which operates on cooling or heating mode by changing the speed of the compressor using an inverter circuit. In most previous works on unsteady state cycle modeling, the temperature variations caused by the irregular migration of refrigerant during start up or shutdown are experimented or predicted[1]~[3]. For the system using a variable speed compressor, however, it is necessary to develop a simulation model that predicts the transient state of the cycle during the change of compressor driving frequency. The simulation model can reflect on the design of the system by analyzing the properties and behaviors of refrigerant according to time. Even though there have been many studies on variable-speed heat pumps, studies on unsteady state behaviors are conducted by only a few researchers [4],[5]. In consequence, it is essential to clarify the characteristics during speed changing by experiments and simulation models for various conditions. Therefore, the purpose of this study is to develop a systematic design process and a design program of variable-speed heat pump cycle through analyzing the transient characteristics of the cycle during a compressor speed change.

EXPERIMENTAL SET UP AND PROCEDURE

A cycle can be optimized through improving efficiency of each major component such as designing heat exchanger pass, improving efficiency of distributor, modifying length and diameter of capillary tube, changing

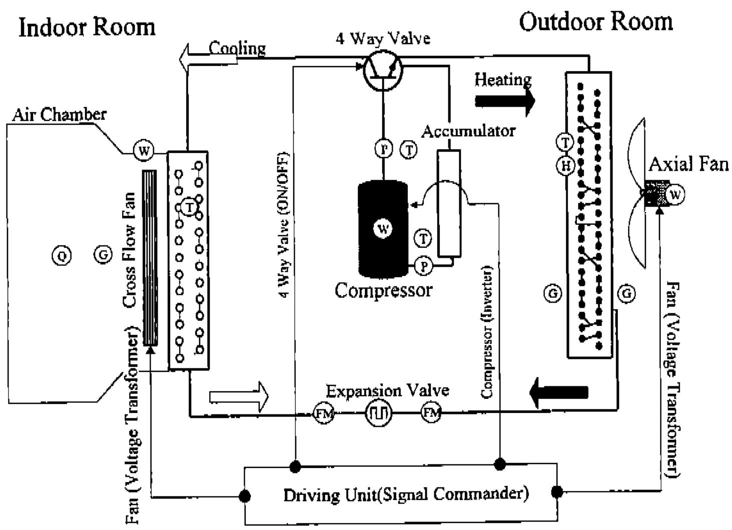

Fig. 1 Schematic diagram of the system unit set up

Table 1 Specification of heat pump system.

Component	Specification	Unit
	Capacity : 3570(C) / 4300(H)	kcal/h
Cycle	Power :1455(C)/1700(H)	W
	Refrigeration Oil : 4GS	
	Charging Amount : 850	g
	Dehumidification : 2.3	l/h
	Type : High Side Scroll	
Compressor	Stroke Volume : 15.6	m^3/rev.
	Frequency : 30 ~ 110	Hz
	Power : 1 ϕ -200V-60Hz	
	Air Flow Rate : 10.0 / 10.5	m^3/min.
Indoor	Fan Type : Cross Flow	
Unit	Size : 880x302x183	mm
	Type : 2Row-2Path-12Step	
	Air Flow Rate :26.4/28.3	m^3/min.
Outdoor	Fan Type :Axial	
Unit	Size : 800x530x270	mm
	Type : 2Row-4Path-24Step	
	Type : Capillary Tube	
Expansion	Diameter: 1.7 ~ 1.9	mm
Device	Length: 1000 ~ 1100	mm

In Fig.1, (P) is a pressure measuring point, (T) is a temperature measuring point, (W) electric consumption, (G) air flow rate, (H) enthalpy, (Q) capacity, and (FM) is a mass flow rate measuring point. To explain briefly about the test procedures, after two minutes from steady, desired drive frequency was set and the unsteady state data were stored in a file every 5-second. The measuring points are consisted of 24 temperature, 2 pressure and 1 mass flow rate signals, and also the capacity, compressor electric input, and airflow rate can be measured. Because all the instruments are open to the noise from the inverter driver, the counter measurement for reducing the noise is needed. Grounding or filtering using an electrolysis condenser can solve the problem. In this experiment, the mass flow meter and compressor inlet and outlet were wound with grounding copper wire, and thermocouples and pressure gauge lines were filtered with103~104 pF capacity electrolysis condensers attached to each channel of instrument. The specification of the system of this experiment is shown in Table 1.

EXPERIMENTAL RESULT

The suction and discharge pressure variations as a function of time during speed up are shown in Fig.2. When the compressor speed increases, the suction pressure drop influences greater to the system than discharge pressure. If the suction pressure decreases suddenly, liquid refrigerant flows into the inlet and evaporates in the compressor causing sudden increase of pressure. From the visualization tests, sudden flooding could be observed in the compressor and vaporized refrigerant flowed into the inlet in a few seconds but than again refrigerant liquid flowed in. The two sharp spikes shown during speed up from 30 to 70Hz are the results of that. In normal case, there is only one slow increase in pressure, but if there is big change such as 70 to 102Hz, a sharp peak can be seen and a transient state occurs at discharge pressure . In the case of speed up from 70 to 102Hz, evaporation occurs and the refrigerant is completely boiled off in the suction side of the compressor causing the sudden rise of the discharge pressure and then the inflow of liquid state refrigerant causes the sudden drop.

The Fig.3 shows the temperature variations of the compressor inlet and outlet as a function of time during speed up. The discharge temperature increases slowly as drive frequency increases, and it takes more time to reach the steady state, as increasing frequency width is greater. In order to reach the steady state during speed up from 70 to 102Hz, it takes more than twice as long as the case of 90 to 102Hz, and the suction temperature rises and falls with a great range because of the boiling in the compressor as mentioned. In the case of 90 to 102Hz, the inflow of liquid state refrigerant like other cases does not occur, but the temperature increases due to the immediate boiling up and then falls back. That is because there is a shortage of spare refrigerant for corresponding to the increase of mass

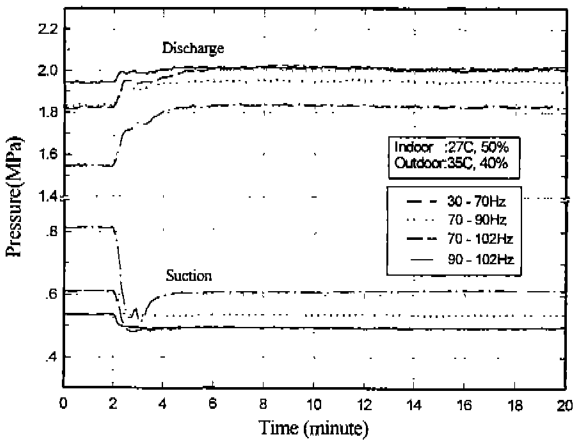

Fig.2 Measured refrigerant pressure under raise up frequency conditions in cooling mode.

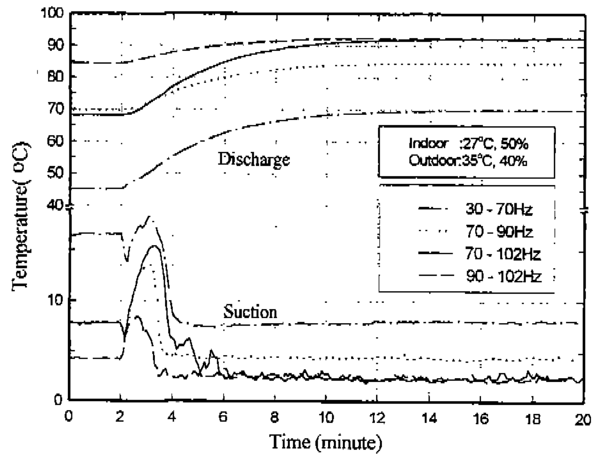

Fig.3 Measured refrigerant temperature at compressor suction and discharge under raise up frequency conditions in cooling mode.

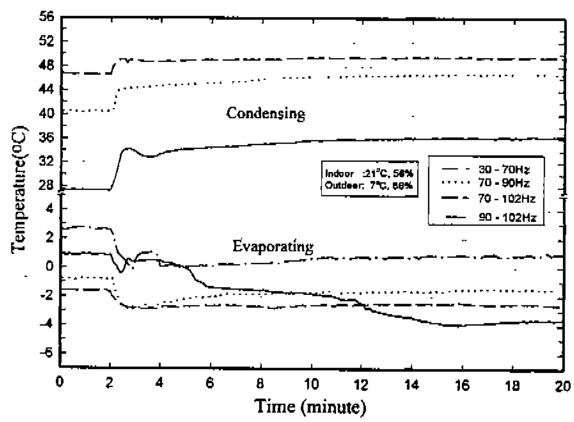

Fig.4 Measured refrigerant temperature at indoor and outdoor heat exchanger under raise up frequency conditions in heating mode.

flow rate since the refrigerant mass flow rate is great relatively high frequency.

Fig.4 shows the evaporating and condensing temperature variations as a function of time during speed up. The condensing temperature is increase after change, but the evaporating temperature, which is the average of 4 passes, drops continuously in the case of increasing frequency from 30 to 102Hz. This result shows that the evaporating temperature decreases due to the frosting of the evaporator. The possibility for frosting becomes greater for all increasing conditions except for relatively low frequency region such as 30 to 70Hz case.

Generally, the possibility of frosting increases when the relative humidity is over 70% and the air temperature is below 5 ℃. For an R-22 refrigerant, if the air temperature is below 5 ℃ the evaporating temperature is less than 0 ℃ satisfying Eq.(1)

$$T_p < T_{dew} \text{ and } T_p < 0\ ℃ \qquad (1)$$
$$T_{dew} = f(T_{db}, RH)$$

In this experiment, the relative humidity 86% of the heating standard conditions(KS standards) was used and the evaporating temperature was below -1 ℃ which fell under the frosting conditions. In the evaporator with 4 passes, non-uniformly frosting occurs because the refrigerant mass flow rates in each pass are different due to poor distribution.

SIMULATION MODEL

A numerical analysis model is developed to predict the transient state characteristics of the system during compressor speed changes using the analysis modeling of each major component as follows.

Compressor

The energy and mass conservation equations and vapor state equations are applied for compression process. The temperature variation of the refrigerant is driven from the energy and mass conservation equations and pressure variation is obtained from the actual vapor state equation. The equation for refrigerant mass flow rate according to the compressor speed is also additionally needed for an unsteady state modeling. The temperature variation equation driven from the 1st law of thermodynamic is differentiated on time and can be expressed by mass and volume rates as following Eq.2. The equation for pressure changes on time can be obtained from the state equation differentiated on time as following Eq.3.

$$\frac{dT}{dt}=\frac{\dot{Q}+\sum(H_i-H)\frac{dm_i}{dt}-\left(\frac{dV}{dt}-v\frac{dm}{dt}\right)\left\{\left(\frac{\partial H}{\partial v}\right)_T-\left(\frac{\partial P}{\partial v}\right)_T\cdot v\right\}}{m\left[\left(\frac{\partial H}{\partial T}\right)_v-\left(\frac{\partial P}{\partial T}\right)_v\cdot v\right]} \tag{2}$$

$$\frac{dP}{dt}=P\left\{\frac{1}{m}\frac{dm}{dt}+\frac{1}{T}\frac{dT}{dt}-\frac{1}{V}\frac{dV}{dt}\right\} \tag{3}$$

Heat exchangers

Modifying the lumped parameter model proposed by MacArthur (1984,1989)[6], the cycle model by Domanski & Didion[7], a simulation model for heat exchangers is developed. The following unsteady discrete equation in terms of enthalpy differences can be achieved after multiplying H_i' with the expanded unsteady 1st order continuity and energy equation with neglected momentum terms:

$$(H_i'-H_i^o)\frac{\rho_i^o V_i}{\Delta t}=\dot{m}_j'(H_i'-H_j')+\dot{m}_{j-1}'(H_{j-1}'-H_i')+(UA)_i(T_{air,i}'-T_{ref,i}') \tag{4}$$

The heat resistance of air is over 70% of the total resistance (*1/UA*), the properties of air are great factors on model calculation. McQuiston (1978), Nakayama (1983), Gray and Webb (1986), and Nagaka (1990) presented general equations on finned tube exchangers with flat fins and enhanced fins. However, because the shapes of fins are various and the airflow passing the fin is very complicate, the estimation of the exact heat transfer of air is very difficult. Since the fin, Φ 7-LG, used in this experiment has also an unique shape and arrangement, the correlation with the experimental heat transfer coefficient is used rather than the theoretical correlation. Fig.5 shows the experimental heat transfer coefficient of air for the indoor unit.

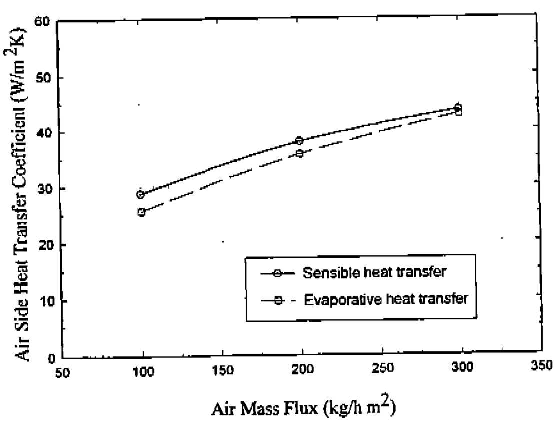

Fig. 5 Experimental heat transfer coefficient in the indoor unit versus mass flux for 2 Row,19Fpi

Expansion device

The expansion device model used in this study is based on the idea of the pressure drop due to the friction and the mass flow rate control according to the characteristics of Fanno flow proposed by Domanski (1982)[7]. Though the refrigerant at the capillary tube entrance is sub-cooled and mostly passing the saturated liquid line, it is necessary to consider two-phase flow because there could be no more sub-cooled liquid at the instant when the mass flow rate changes with the changed compressor driving frequency. If the saturated temperature at the condenser outlet pressure is lower than the entrance temperature, the calculation for two-phase flow should be made, and if it is higher the calculation at the entrance should be made for sub-cooling region. The capillary tube simulation model for cooling and heating mode is a simple linear model without the consideration of thermodynamically non equilibrium effect assuming that the flow is one-dimensional and adiabatic and the friction has a great effect on the pressure drop.

4-way valve

Because there is hot and cold refrigerant in a 4-way valve, there occurs a pressure drop because of the conduction, convection, and radiation heat transfer and its complex flow structure. The temperature of suction gas increases due to the convection heat transfer between two fluids while the discharge side temperature decreases. The difference between cooling and heating mode can be neglected because the effect of convection and radiation from surrounding is very small. Therefore, 1^{st} order simple linear correlation are used.

Cycle modeling

In this cycle model, the prediction of transient state characteristics during the change of operating frequency is the main purpose. Subprograms were made to show the system change in infinitesimal time using governing equations including unsteady terms for each major component. One of the typical solving methods for an unsteady system is to solve simultaneously the whole governing equations enumerated with dependent variables on infinitesimal time. Because these are one-dimensional differential equations on space or time, the 4^{th} order Runge-Kutta method can be applied. This method has very good convergence, but has a disadvantage of somewhat low accuracy. Therefore, after solving the unsteady state governing equations for each major component separately on infinitesimal time and balancing with the results, the calculation for the next time step was taken orderly in this model. The initial conditions from the steady state refrigerant distribution for each frequency were used. Because the initial conditions influence greatly to the simulation results, the simulation was performed using measured refrigerant property data referring to the related references and experiment in order to obtain precise initial conditions. The initial conditions used in this simulation are shown in Table 2. Assuming no slip of motor except high and low frequencies, the system was simulated using saturation temperature from evaporating and condensing pressure. The surface temperature of the compressor varied with operating frequency, but because the fluctuation was not so great during steady state operation, the steady state values were used as the initial values.

Table 2 Initial conditions at typical frequencies.

Input Name	Start Frequency	Rotating Speed	Evaporating Pressure	Condensing Pressure	Compressor Surface Temperature
Unit	(Hz)	(rpm)	(MPa)	(MPa)	(℃)
Cooling	30	1800	0.82	1.55	44
	90	5400	0.53	1.91	82
Heating	30	1800	0.55	1.23	38
	90	5400	0.44	2.04	93

SIMULATION RESULT AND DISCUSSION

The variations of capacity, refrigerant mass flow rate, pressures and condensing and evaporating temperature were simulated using a theoretical model when the drive frequency in a steady state was changed to a desired frequency. The theoretical results were compared with the experimental results for 70Hz. Fig.10 shows the compressor suction and discharge pressure variations during sudden increase of drive frequency from 30Hz to 50, 70, and 102Hz in cooling mode. In the case of 70Hz, though the simulation values are a little smaller than the experimental ones, the overall variation trend is very similar. When the drive frequency is changed from 30 to 102Hz,unlike when moved to 50Hz, the drop of evaporating temperature can be seen. That is due to the inflow of a large quantity of liquid state refrigerant into the evaporator from the increase of the drive frequency, and the sudden pressure rise due to the evaporation of refrigerant in the compressor cannot be predicted. The evaporating temperature variations during sudden increase of drive frequency from 30Hz to 50, 70, and 102Hz in cooling mode are shown in Fig.11. Due to the frequency increase, the evaporating pressure drops abruptly causing the drop of its temperature. After the evaporating temperature drops, it increases at once since the shortage of refrigerant in the evaporator occurs. When refrigerant circulates again in the evaporator, the evaporating temperature decreases slowly to the steady state. The precision of these variations of temperature depends on prediction of the void fraction of two-phase region in the evaporator, and it effects the calculation of cooling capacity. The prediction during increasing system frequency was fairly good, but the temperature and pressure of the condenser and the compressor discharge pressure were low. The reason of the low values of the condenser is due to the under

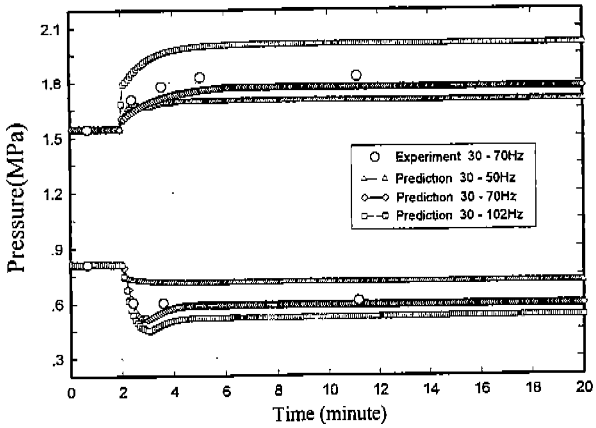

Fig. 10 Calculation results of suction and discharge pressure for various system frequencies according to time under standard cooling condition. (30Hz Start / 50, 70, 102Hz Goal)

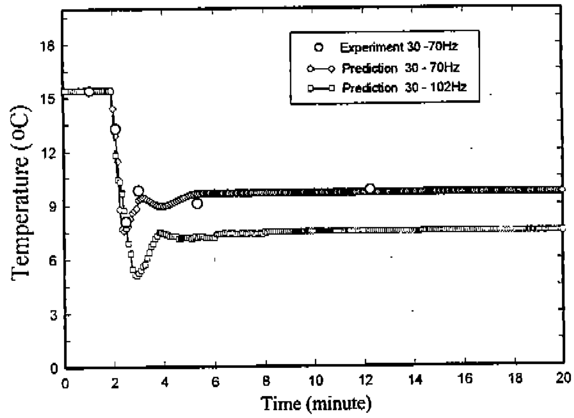

Fig. 11 Calculation results of evaporating temperature for various system frequencies according to time under standard cooling condition.(30Hz Start / 70, 102Hz Goal)

prediction of the compressor discharge rather than inaccurate heat exchanger modeling. Therefore, the application of more exact compressor modeling is necessary.

CONCLUSIONS

The main cause for the transient phenomena was due to the excessive refrigerant into compressor from increased refrigerant mass flow rate and suction pressure drop. The temperature varied a lot due to the shortage of refrigerant in evaporator and capillary entrance. The transient for heating mode was relatively shorter than that for cooling because of smaller suction pressure drop, but in 102Hz high speed conditions, it was necessary to maintain the evaporating temperature to above -1 ℃ in order to prevent frosting in evaporator surface. The transient state model for each major component of the system was developed, and the results of the simulation were in good agreement with the experimental ones within 10% for transient conditions during speed up. From the results it could be drawn that the transient cycle migration of the liquid state refrigerant caused significant dynamic change in system. Therefore, the migration of refrigerant was the most important factor in accuracy for an experimental result analysis or development of transient model. Since only standard conditions were considered in this study, transient characteristics of system under various temperature conditions as well as overload conditions should be defined through further studies.

REFERENCES

1.Murphy, W. E. and Goldschmidt, V. W., 1985, "Cyclic Characteristics of a Typical Residential Air Conditioner-Modeling of Start-Up Transients," *ASHRAE Transactions*, No.2A, pp. 427-444.

2. Miller, W. A., 1988, "Laboratory efficiency comparisons of modulating heat pump components using adjustable speed drives," ASHRAE Transactions, Vol.94, pp. 874-891.

3.Tanaka, N. and Ikeuchi, M., 1982, "Experimental Study on the Dynamic Characteristics of a Heat Pump," *ASHRAE Transactions*, No.2, pp. 323-331.

4.Yasuda, T.,Yanagisawa, T., Izushi, M., 1994, "A dynamic model of a vapor compression refrigeration cycle," JAR *Transactions,* No.3, pp263-275. (in Japanese)

5. Matsuoka, F., Nagatomo, H., 1988, "Dynamic response and electrical control for the air conditioner," JAR *Transactions,* No.5, pp43-54. (in Japanese)

6. MacArthur, J. W. and Grald, E.W., 1989, "Unsteady Compressible Two Phase Flow Model for Predicting Cyclic Heat Pump Performance and a Comparison With Experimental Data," Int. J. of Refrigeration, Vol.12, No.1 , pp. 29-41.

7. Domanski, P. A. and Didion, D. A., 1983, "Computer Modeling of The Vapor Compression Cycle with Constant Flow Area Expansion Device," Report No. NBS Building Science Series 155, *NBS*.