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The transient state modeling and optimization of refrigerant charge amount for a household refrigerator

가정용 냉장고의 동적 해석 모델을 통한

냉매 량 최적화

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Abstract

The transient state modeling and optimization of refrigerant charge amount for a household refrigerator

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Energy consumption regulation of household refrigerator has been continuously strengthened worldwide. In Korea, energy efficiency rating labeling was enforced from 1992. According to data from the Korea Energy Management Corporation (KEMCO), energy consumption of refrigerators has been reduced 59% over 15 years. Because of that, most refrigerator manufacturers are seeking ways to improve thermal and electrical performance of their products. In general, researching and development method of refrigerator is trial and error procedure, which is time consuming and costly. Hence, numerical analysis represent an essential tool to improve time and cost. Therefore, a study for the numerical simulation of the refrigerator has been conducted actively.

In this study, experiments were conducted together with the numerical simulation to find the refrigerant charging amount of minimizing the power consumption of household refrigerator, which has capillary tube with heat exchanger. The experimental data were obtained in refrigerant charge range from 80 g to 125 g at ambient temperature of 25℃. According to the experimental data, the power consumption of household refrigerators has been minimized at the refrigerant amount 104g. The numerical simulation was computed under the same condition of the experiments by finite difference method. The numerical data showed that power consumption was minimized at refrigerant amount 98 g. Numerical simulation result was compared with experimental data, and it was found that optimal refrigerant charge amount was well predicted by the model.

Keywords: Key Words : Household refrigerator, Optimization of refrigerant charge amount, Transient state modelling, Non-adiabatic capillary tube.
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Nomenclatures

L	length [m]
d	diameter [m]
r	radius [m]
A	area [m^2]
V	volume [m^3]
x	vapor quantity
θ	angle of inclination with horizontal [deg]
v	velocity [m/s]
n	rps [rev/s]
p	pressure [N/m^2]
$\dot{m}$	mass flow rate [kg/s]
τ	shear stress [N/m^2]
v	specific volume [m^3/kg]
u	internal energy [kJ/kg]
C_p	specific heat at constant pressure [kJ/kg℃]
k	heat conductivity [W/mK]
h	heat transfer coefficient [W/m^2K]

U	overall heat transfer coefficient [W/m^2K]
Re	Reynolds number
Pr	Prandtl number
f	friction factor
NTU	number of heat transfer units

Greek

ρ	density [kg/m^3]
μ	dynamic viscosity [Pa.s]

Subscript

eff	effectiveness
in	inlet
out	outlet
i	inner
o	outer

l	liquid
v	vapor
lo	liquid only
e	evaporated
w	wall
c	cross-sectional/ capillary
s	suction
min	minimum

Chapter 1. Introduction

1.1 Background of the study

As the refrigerator makes human life more comfortable and convenient, the demand of refrigerator is continuously increased. However the refrigerator has to operate continuously for storing food. Thus energy consumption regulation of household refrigerator has been continuously strengthened worldwide. In Korea, energy efficiency rating labeling was enforced from 1992. The power consumption of refrigerator has been main factor for choosing purchasing product by customer. Because of that, most refrigerator manufacturers are seeking ways to improve thermal and electrical performance of their products. In general, researching and development method of refrigerator is trial and error procedure, which is time consuming and costly. Hence, numerical analysis represent an essential tool to improve time and cost. Therefore, a study for the numerical simulation of the refrigerator has been conducted actively.

The majority of research is to improve the efficiency of the refrigerator. In general, the refrigerator is operating intermittently for maintain temperature of food storage compartment. If temperature of food storage compartment is lower than designated temperature, the compressor is running. For the opposite case, the compressor is stopped. Thus the performance of refrigerator is always changed by operation condition. Commonly, the model of numerical simulation is divided steady state and transient. Each method has advantages and disadvantages. In this study, numerical simulation was calculated with transient state for accurate analysis. Also internal heat exchanger of the capillary tube is another key concern. The capillary tube is increase effective cooling capacity of the refrigerator by exchanging heat with suction line. Thus, internal heat exchanger of the capillary tube was concerned at numerical and experimental model.

1.2 Literature review

There is preceded research about numerical simulation of household refrigerator. The studies of numerical simulation are divided steady state and transient. Generally steady model was favored as less computing time. After that, the transient model and combined model were used to increase accuracy.

Goncalves and Melo (2004) studied the steady-state behavior of top mounted freezer type refrigerator by measuring and modeling the performance characteristic of each of its compartments. However, electricity consumption of refrigerator was not matched well.

More improved model was presented by Hermes and Melo (2006). The model was used to simulated same type of refrigerator with Goncalves and Melo (2004). However, their model was including of compressor on-off control. As a result, difference between numerical simulation and experiential data was within 10% band.

Bruno et al (2010) studied the transient-state behavior of top mounted freezer type refrigerator by measuring and modeling with quasi-steady approach. Their model was calculated using hybrid semi-empirical model (steady state refrigeration loop and transient refrigerated compartments). As a result, difference about power consumption between numerical simulation and experiential data was within 2% band.

1.3 Objectives and scope of the study

The major objective is optimization of refrigerant charging amount of household refrigerator using numerical and experimental method. This numerical analysis was conducted under the transient state condition with non-adiabatic capillary tube. The experiment also was conducted for finding optimal charging quantity of the refrigerant to minimize power consumption under the same with condition of numerical analysis.

In chapter two, the components modeling are conducted and the numerical analysis of overall refrigerator is conducted. Although the many research established simulation of refrigerator, the analysis for refrigerator with steady state modeling is not sufficient. Therefore the analysis is meaningful under transient state condition about household refrigerator. Also, the internal heat exchanger of capillary-tube was included to the numerical analysis model.

In chapter three, the experiment is conducted with various charging quantity of the refrigerant to household refrigerator. The objective is optimization of charging amount of refrigerant in household refrigerator. The experiments are conducted with designated refrigerant charging amount at each case. And the other condition of refrigerator is the same.

In chapter four, concluding remarks are given alone with the summary of this study

Chapter 2. Transient state modelling of the household refrigerator

2.1 Introduction

A refrigerator is a thermal system which removes heat from stored foods usually by latent heat of the refrigerant. The vapor compression cycle is the most widely used household refrigerator in general. The household refrigerator is consisting of several components. General components are compressor, condenser, capillary tube with heat exchanger, evaporator with accumulator and cabinet. A schematic of household refrigerator is shown in Fig. 2.1. In the compression process, the refrigerant is compressed and then condensed to liquid refrigerant in the condenser. In the expansion process, the condensed refrigerant passes through the capillary tube, and turn into a low pressure state. Then the low pressure refrigerant evaporates by absorbing heat from stored foods.

A characteristic of household refrigerator is that the compressor experience on-off operation for temperature control. If the temperature in food storage compartment of the refrigerator is greater than the designated turn-on temperature, the compressor is operating. If the temperature in food storage compartment of refrigerator is less than the designated turn-off temperature, the compressor is stopped. Because of these characteristic, it is difficult to analysis as a steady state model. This study tries to analyze the household refrigerator by the transient state modelling

2.2 Specification of the studied household refrigerator

Generally there are many type of refrigerator. Typically, the refrigerator is classified TMF (top mounted freezer), LMF (low mounted freezer) and SBS (side by side). In this study, the side by side refrigerator was investigated. The studied refrigerator is configured with freezer and fridge. The exterior and interior of refrigerator model is shown Fig. 2.2. The total volumes of the refrigerator are 830 liter. The refrigerator uses inverter compressor. And it use isobutene as refrigerant. The detail specifications of refrigerator model are represented in Table 2.1.

2.3 Component modelling

2.3.1 Compressor modelling

Since the compressor is one of the important components in the refrigerator, accurate compressor modelling is required. However the transient state modelling requires a great deal of computing time, compressor in the transient state modelling is calculated with simple concept. In this study, isentropic efficiency is used to analyze the behavior of the compressor.

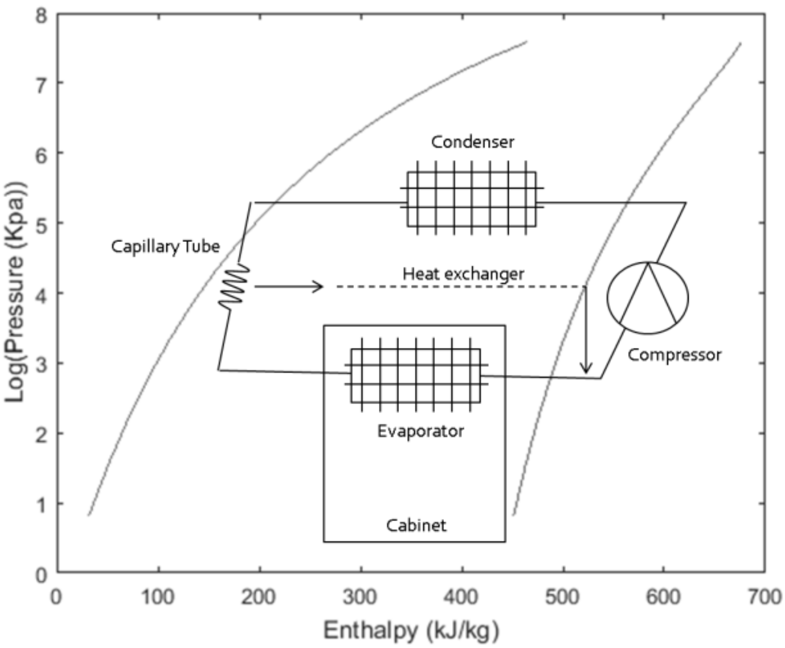

Figure 2.1 Schematic of household refrigerator

Figure 2.2 Exterior and interior of refrigerator model

Table 2.1 Specification of refrigerator model

Item		Specification
Model Code		RH83H
Prices		2,000$
Type		Side by side
Size (w x h x d)		912x1774x920 mm
Volume	Freezer	307 liter
	Fridge	523 liter
	Total	830 liter
Refrigerant		Isobutene 96g
Compressor	Stroke volume	15 cc
	Type	Inverter
	RPM	1450~3600
Condenser	Type	Circular fin and tube
Evaporator	Type	Rectangular fin and tube
	Accumulator	132cc
Capillary	Type	Non-adiabatic
	Heat Exchanger	Lateral, With Suction Line

The volumetric efficiency is the basis for predicting volumetric flow rate of refrigerant in the compressor. It is defined as the ratio of actual volume flow rate to the displacement flow rate of the compressor. In this model, clearance volumetric efficiency was used. Clearance volumetric efficiency depends on the re-expansion of refrigerant trapped in the clearance volume and is defined by

$$V_{eff} = 1 - V_c \times \left(\frac{\rho_{out}}{\rho_{in}} - 1\right) \quad (2.1)$$

where V_s is the clearance volume ratio, ρ_{in} density of refrigerant entering compressor, and ρ_{out} density of refrigerant after compressor. The mass flow rate of the refrigerator is determined by Eq. (2.2)

$$\dot{m} = V_{eff} \times V_d \times \mathrm{n} \quad (2.2)$$

where V_d is the displacement volume, n rotation per sec of the compressor.

2.3.2 Heat exchanger modelling

The evaporator and condenser were air cooled heat exchanger. The heat exchanger was treated fin and tube type with plain fin. The secondary fluid flows into the heat exchanger by the electric fan.

The governing equations are shown in Eq. (2.3), Eq. (2.4), and Eq. (2.5).

Continuous equation:

$$\frac{d\rho}{dt} = -\frac{d\rho \mathrm{v}}{dx} \tag{2.3}$$

Momentum equation:

$$\frac{d\rho \mathrm{v}}{dt} = -\frac{d\rho \mathrm{v}^2}{dx} - \frac{dp}{dx} - \frac{A_w}{A_c}\frac{\tau}{L} - \rho g sin\theta \tag{2.4}$$

Energy equation:

$$\frac{d\rho u}{dt} = -\frac{d\rho \mathrm{v}(h + \frac{\mathrm{v}^2}{2})}{dx} - \frac{A_w}{A_c}\frac{h}{L}(T_w - T) \tag{2.5}$$

To reduce the calculation time, above equations are modified one-dimension form. Eq. (2.3) and Eq. (2.4) represent mass conservation and momentum conservation for the refrigerant. Eq. (2.5) represents energy equation for refrigerant. The heat exchanger was divided numerous pieces. And then each grid of heat exchanger follows governing equations.

To calculate the energy equation, the overall heat transfer coefficient was calculated by thermal resistance method. The overall heat transfer coefficient is defined by Eq. (2.6)

$$\frac{1}{UA} = \frac{1}{hA_r} + \frac{ln\left(\frac{r_o}{r_i}\right)}{2\pi kL} + \frac{1}{hA_a} \tag{2.6}$$

For the uncertainty of inlet and outlet temperatures of the secondary flow, effectiveness-NTU method was used and is defined by Eq. (2.7)

$$NTU = \frac{UA}{C_{min}} = -ln(1 - \varepsilon) \tag{2.7}$$

where ε is the effectiveness of heat exchanger. The effectiveness was chosen by flow type of the heat exchanger.

When the refrigerant flows alone the heat exchanger, there must be a phase change. To calculate heat exchangers accurately, it is necessary to analyze local heat transfer coefficient. So correlations of heat transfer coefficient are selected suitable for single and two phases.

For single phase regions in the evaporator and condenser, Dittue and Boelter correlation (1930) was selected.

$$h = 0.23\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4\text{ or }0.3} \tag{2.8}$$

where the exponential is 0.4 for heating and 0.3 for cooling.

For the condensing heat coefficient, Chen correlation (1987) was selected.

$$Nu = 0.018\left(\frac{\mu_v}{\mu_l}\right)^{0.078}\left(\frac{\rho_l}{\rho_v}\right)^{0.6} Re_l^{0.2}\left[Re_{lo} - Re_{tp}\right]^{0.7} Pr_l^{0.65} \tag{2.9}$$

For the evaporating heat coefficient, Gungor and Winterton (1987) correlation was used.

$$E = 1 + 3000Bo^{0.8}\left(\frac{x}{1-x}\right)^{0.75}\left(\frac{\rho_l}{\rho_v}\right)^{0.41} \tag{2.10}$$

In the correlation, E is acceleration factor by phase change.

For the air-side heat transfer coefficient of the evaporator, Wang (1999) correlation was used

$$j = 0.086 {Re_D}^{P3} N^{P4} \left(\frac{F_p}{D_c}\right)^{P5} \left(\frac{F_p}{D_h}\right)^{P6} \tag{2.11}$$

In the correlation, j was the Coburn factor.

For the air-side heat transfer coefficient of the condenser, Mon (2003) correlation was used.

$$Nu = 0.356 Re^{0.6} Pr^{\frac{1}{3}} \left(\frac{A}{A_t}\right)^{-0.15} F^{0.173} \left(\frac{S_t}{S_l}\right)^{-0.475} \tag{2.12}$$

2.3.3 Capillary tube modelling

The expansion devise makes lower pressure by throttling. There several types of expansion devices. In this study, a capillary-tube was used as an expansion device. General capillary-tube exchanges heat with the suction line.

The governing equations are shown in Eq. (2.3), Eq. (2.4), and Eq. (2.5). The capillary tube was divided numerous pieces. And then each grid of capillary tube follows governing equations.

To calculate the energy equation, the overall heat transfer coefficient was calculated by thermal resistance method. The cross section of the non-adiabatic capillary tube is shown in Fig. 2.2. This process includes convective heat transfer from the refrigerant inside the capillary tube to the capillary tube wall, conduction from the tube wall to the solder joint, conduction from the solder joint to the suction line wall and from the suction line wall to the refrigerant inside the suction line. Therefore, the overall thermal conductance, UA, can be given by

$$\frac{1}{UA} = \frac{1}{h_c r_c \pi dz} + \frac{\ln\left(\frac{r_{c,o}}{r_{c,i}}\right)}{2\pi k_{c,w} dz} + \frac{\delta}{k_j w dz} + \frac{\ln\left(\frac{r_{s,o}}{r_{s,i}}\right)}{2\pi k_{s,w} dz} + \frac{1}{h_s r_s \pi dz} \tag{2.13}$$

Eq. (2.13) was suggested by Bansal (1998) and represents the overall heat transfer coefficient in non-adiabatic capillary tube with lateral configuration.

The refrigerant phase is changed across the capillary tube by pressure drop and heat from suction line. To calculate capillary-tube accurately, it is necessary to analyze local heat transfer coefficient. So correlations of heat transfer coefficient are selected suitable for single and two phases.

For single phase regions in the capillary-tube, Grielinski (1976) equation was used.

$$h_{sp} = \frac{k\left(\frac{f}{8}\right)(Re - 100)Pr}{\left(1 + 1.27\left(\frac{f}{8}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)\right)D} \tag{2.14}$$

For the two phase heat transfer coefficient, Mezavila (1996) assumption was used

$$h_{tp} = \infty \tag{2.15}$$

For single phase regions in the suction line, Eq. (2.3) was used. And for two phase regions in the suction line, Eq. (2.10) was selected.

Since a pressure drop occurs through the capillary-tube, to calculate capillary-tube accurately, it is necessary to analyze local friction factor.

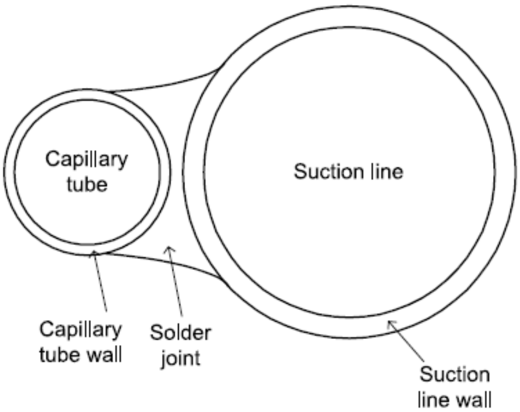

Figure 2.3 Cross section of the capillary-tube

For the single phase friction factor, Churchill (1977) equation was used

$$f_{sp} = 8[(\frac{8}{Re})^{12} + \frac{1}{(A+B)^{1.5}}]^{\frac{1}{12}} \tag{2.16}$$

For the two phase friction factor, Lin (1997) equation was used.

$$f_{tp} = \emptyset^2 f_{sp}\left(\frac{v_{sp}}{v_{tp}}\right) \tag{2.17}$$

2.3.4 Accumulator modelling

The accumulator was placed downstream of the evaporator for preventing liquid flow to the compressor. The mass and energy equation was adapted in the accumulator. Because the accumulator is relatively short, the friction loss can be neglected. A schematic of accumulator is shown in Fig 2.4. The accumulator separate liquid refrigerant ($\dot{m}_l$) and vapor refrigerant ($\dot{m}_v$) by the gravity. In accumulator inlet, liquid and vapor refrigerant are mixed. Through the accumulator, liquid refrigerant ($\dot{m}_l$) is removed and evaporated refrigerant ($\dot{m}_e$) added. Finally vapor refrigerant ($\dot{m}_v$) and evaporated refrigerant ($\dot{m}_e$) are exhausted from outlet of the accumulator.

2.3.5 Cabinet modelling

The cabinet is preventing heat penetration by insulation. Representatively polyurethane is used insulation of the refrigerator.

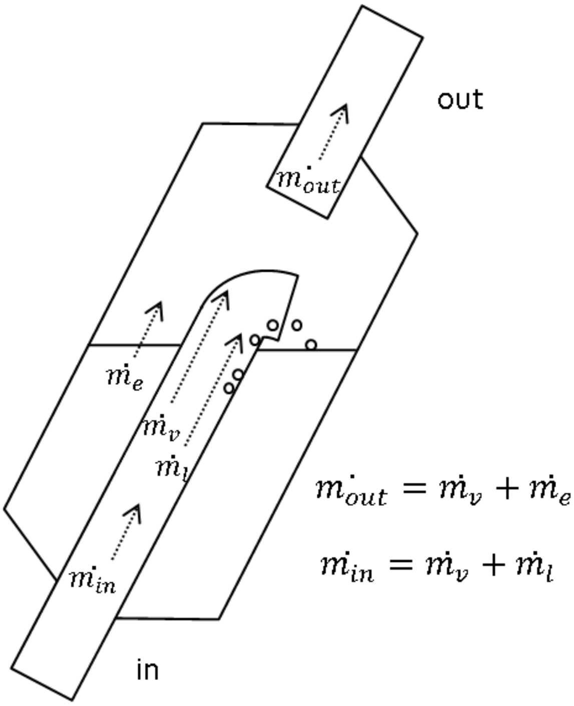

Figure 2.4 Schematic of accumulator

When the cabinet is a system, mass balance is consist of discharge and suction air caused by density change. Then energy balance is divided cabinet and inner air. Each system should be flow energy equation. The cabinet energy balance is consisting of heat flux from ambient, heat flux from inner air and internal energy deviation of the cabinet by time. The inner air energy balance is consisting of heat flux from cabinet, heat flux from evaporator and internal energy deviation of inner air by time. Because the wall between freezer and fridge influence inner temperature, it was included in the transient model.

2.4 Simulation Procedure

The target of simulation is general side by side refrigerator which is configured with fridge and freezer. All components except compressor are divided several pieces which follow mass, momentum and energy equation except compressor. The transient model starts with equilibrium state at ambient temperature. If all components are converged under given conditions, next time step is started. The operation condition of compressor is decided by inner temperature of refrigerator. If the inner temperature is below the target temperature, the compressor is stopped. And simultaneously evaporating and condensing pressure is become equal and is the same with relevant pressure of inner temperature. If the inner temperature is above the target temperature, the compressor is run. And simultaneously, evaporating and condensing pressure are separated according to given condition. Therefore, all performance data follow periodic inner temperature of the refrigerator. The numerical simulation is conducted at each refrigerant charging amount. A flow chart of transient model is showed in Fig. 2.5. The convergence conditions of model are mass flow rate by capillary tube, charging amount, heat flux of capillary tube and degree of superheat at compressor inlet. The capillary tube has counter-flow heat exchanger with suction line. Direct calculation of heat exchange between capillary and suction is difficult. Therefore, the heat exchange between capillary and suction is calculated using heat flux. After each calculation heat flux is adjusted until converged. It is need additional convergence calculation loop about heat flux by internal heat exchanger. By the way, searching DSH (degree of superheat) and heat flux of capillary are closely related, conditions are merged together.

2.5 Simulation results of the studied household refrigerator

The numerical analysis proceeds with various refrigerant charging amounts. As shown in the Fig. 2.6, the electric consumption is minimized at 95 g of refrigerant. However, according to least square polynomial, the optimal charging quantity of refrigerant is changed to 98 g. In general, optimization method of the least square polynomial is used in practice. The numerical analysis was calculated at 60-second intervals for 24 hours. The characteristics of refrigeration cycle over time are shown in Fig. 2.7. As results of numerical simulation, charge amount of each component was changed by time. Refrigerant distribution in each component at refrigerant charge of 110 g is shown in Fig. 2.8. The refrigerant charge of accumulator was changed by operating condition. In the beginning of compressor operation, the refrigerant charge amount is decrease to supply refrigerant to the condenser. And then excessive refrigerant is return to the accumulator.

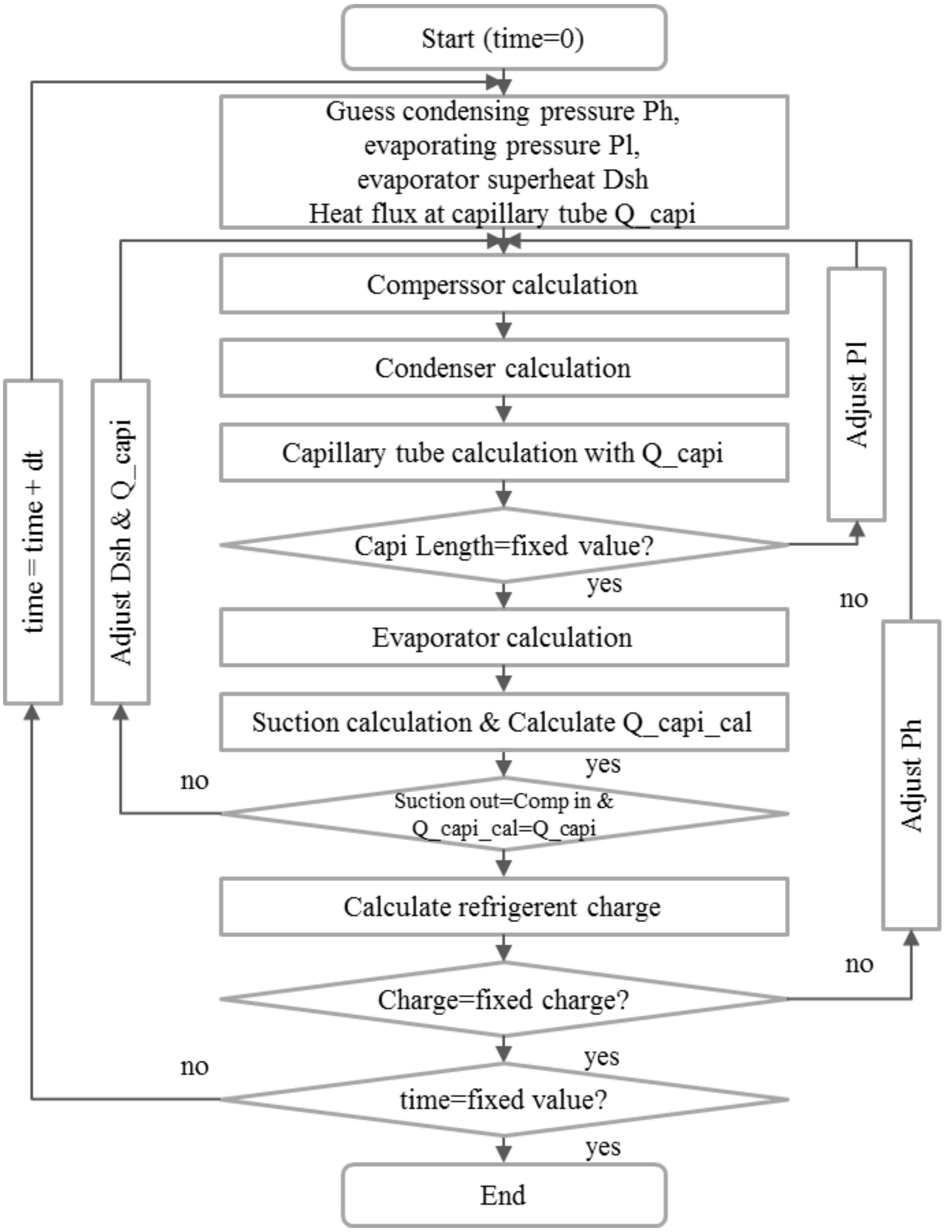

Figure 2.5 Flow chart of transient state modelling

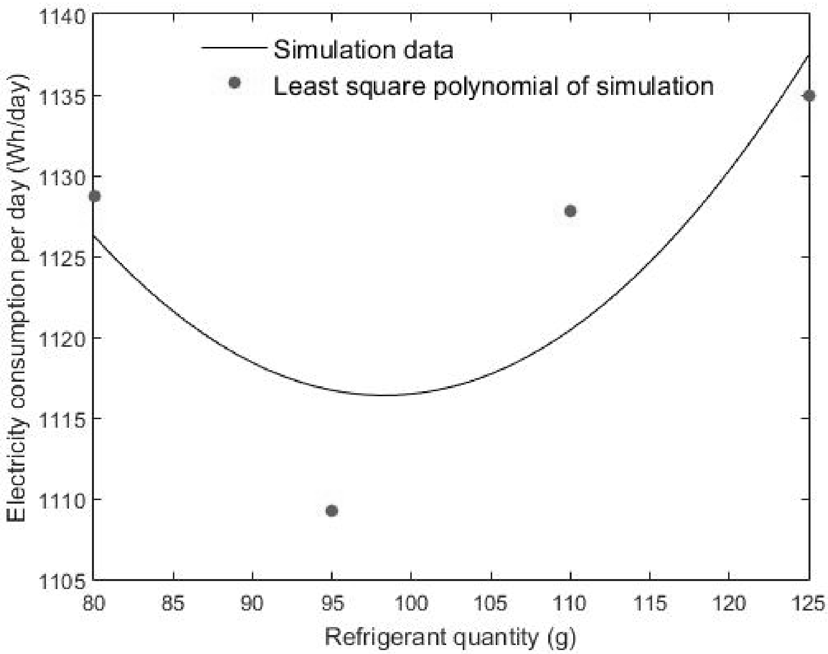

Figure 2.6 Numerical analysis results of household refrigerator

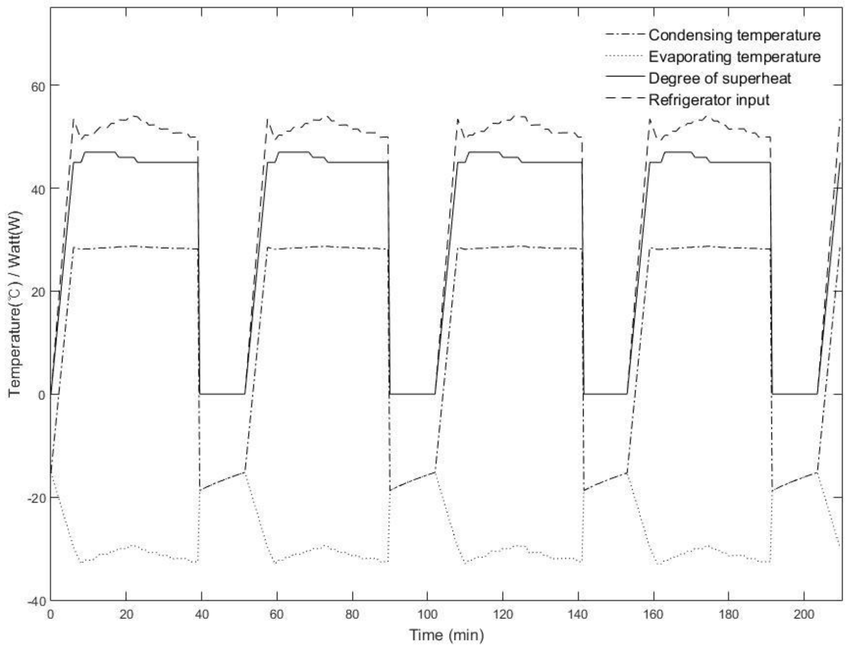

Figure 2.7 Characteristics of refrigeration cycle over time at refrigerant charge of 110 g

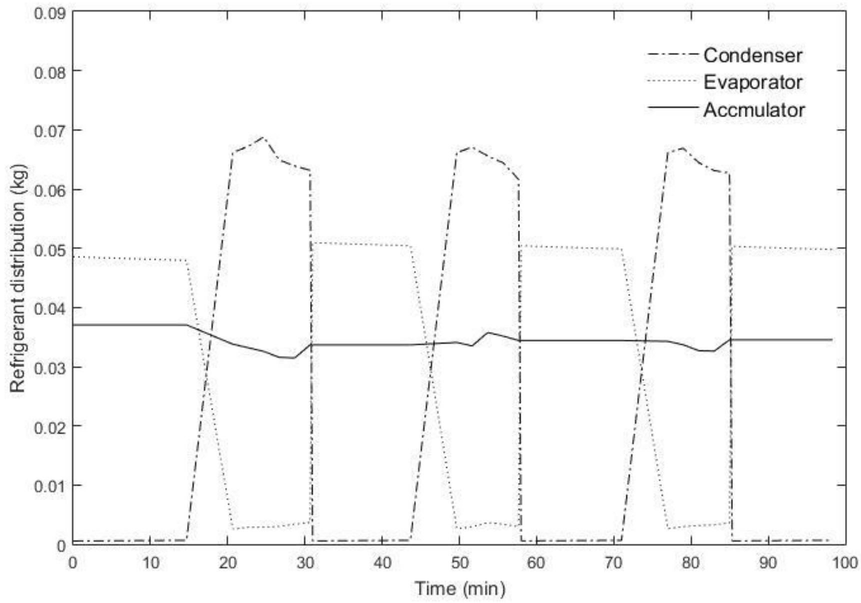

Figure 2.8 Refrigerant distribution in each component at refrigerant charge of 110 g

Chapter 3. Experimental study on the household refrigerator

3.1 Introduction

In the previous chapter, the numerical data showed that electricity consumption was minimized at refrigerant charge of 98 g. To verify the optimal charging quantity of refrigerant in practice with various conditions, the experimental studies were required. Therefore the experiment was conducted with the experimental set-up in refrigerant charge range from 80 g to 125 g. In this paper, the experiment conducted at ambient temperature of 25℃, which is test temperature of energy consumption evaluation at Korea.

3.2 Experimental setup and measurement

3.2.1 Experimental setup

The experimental setup is well represented in schematic of Fig. 3.1. Basically the refrigerator is installed in constant temperature and humidity chamber. The specification of constant temperature and humidity chamber is described in Table 3.1. And then the refrigerator is operated under set temperatures of freezer and fridge. Because the evaporator is installed in freezer, fridge is cooled by divaricated air at freezer.

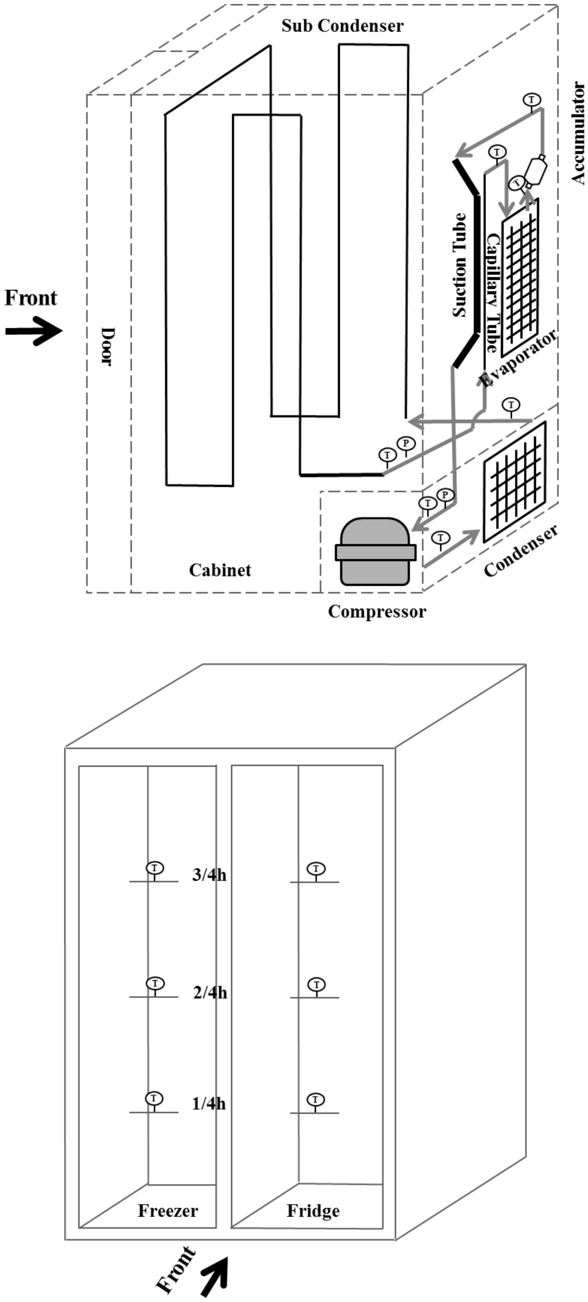

Figure 3.1 Schematic of experimental set up

Table 3.1 Specifications of the constant temperature and humidity chamber

Classification		Specification
Model		EBL-6H20W0P2CJ-22
Temperature	Range	-30~80℃
	Constancy	±0.3℃
	uniformity	±0.75℃
Humidity	Range	10~95%
	Constancy	±2.5%
	uniformity	±5.0%
Size		W3020 x H2450 x D4070

Figure 3.2 Picture of the constant temperature and humidity chamber

3.2.2 Measurement

The specifications of measuring devices are represented in Table 3.2. The measured data about temperature, pressure, power are logged at interval of 30 seconds. The measured data of temperature are divided into refrigeration cycle and storage compartment of refrigerator. Temperatures are measured by T-type thermocouples, which are twisted. All components of refrigeration cycle are measured temperature at inlet and outlet of those. And each storage compartment of refrigerator is measured temperature at 3 points. The condensing and evaporating pressures are measured at outlet of condenser and suction line. The measuring points of thermocouples and pressure gages are shown in the Fig. 3.1. The electric data including power consumption and input are measured at power line of refrigerator. All measured data are sending to data acquisition system. The desktop computer is used for data acquisition system with receiving module.

Table 3.2 Specifications of the measuring instruments

Absolute pressure transducer	
Model	Sensotech, Z
Full scale	500 psi
Accuracy	± 0.25% of full scale
Multi-channel recorder	
Model	Yokogawa, DA 100 Expendable type
Max. input channel	60 channels (up to 300 channels)
A/D integration time	20 ms (50 Hz), 16.7 (60 Hz)
Compensation accuracy for reference junction	± 0.25℃ or 0.4%
Maximum resolution	1 mV, 0.1℃
Measurement accuracy	± 0.05% of reading +0.7℃ ± 0.05% of reading +2 digits

Table 3.2 Specifications of the measuring instruments (continued)

Power meter	
Model	Yokogawa, WT1600
Voltage range	1.5/3/6/10/30/60/100/150/300/600/1000V
Current	50A input element: 1/2/5/10/20/50A
Frequency	DC, 0.5 Hz to 1MHz Accuracy:±(0.05% of rdg+1dgt)
Basic accuracy	±(0.1% of rdg+0.05% of rng)
A/D converter	16-bit Resolution

3.3 Test conditions

As the purpose of this paper is to find optimal charging quantity of the refrigerant at minimum electric consumption, the charging quantity of the refrigerant is important. The charging quantity is varied from 80 g to 125 g with each gap of 15 g. The household refrigerator is exchanging heat with the ambient air. and the condition of ambient air has to be maintained a steady state. The constant temperature and humidity chamber performs a role for this state as the ambient condition is set to 25℃ 70%. The every condition is also kept steady state. As the household refrigerator is operated by temperature of compartment, performance data are periodic. Thus performance data of each case are selected after stable condition. The detailed operating conditions are represented in Table 3.3.

Table 3.3 Test conditions of household refrigerator

Parameter	Value
Refrigerant	R-600a
Refrigerant charge (g)	80, 95, 110, 125
freezer temperature (℃)	-19
fridge temperature (℃)	2
Compressor speed (RPM)	1650
Ambient condition	25℃/70%

3.4 Experimental results

The experiment was conducted with changing the charging quantity of the refrigerant. and the results is shown in the following figure and table. As shown in the Fig. 3.3, the electric consumption is minimized at 110 g of refrigerant. However, according to least square polynomial, the optimal charging quantity of refrigerant is changed to 104 g. In general, optimization method of the least square polynomial is used in practice. Trend of the operation ratio of compressor is similar with trend of the electric consumption. In case of deficiency of refrigerant, the operation ratio of compressor is decreased by adding refrigerant. However, In case of excess of refrigerant, the operation ratio of compressor is increased by adding refrigerant. This trend is occurred by capacity of accumulator. The comparison of numerical simulation and experimental result is shown in Fig. 3.4. Compare with the experimental results, optimal refrigerant charge amount is underestimated about 6 g. The numerical model has being a good agreement with experimental results. However, numerical result has not matched well with polynomial. That is caused by non-adiabatic capillary tube. According to heat flux of capillary tube, cycle performance is extremely changed.

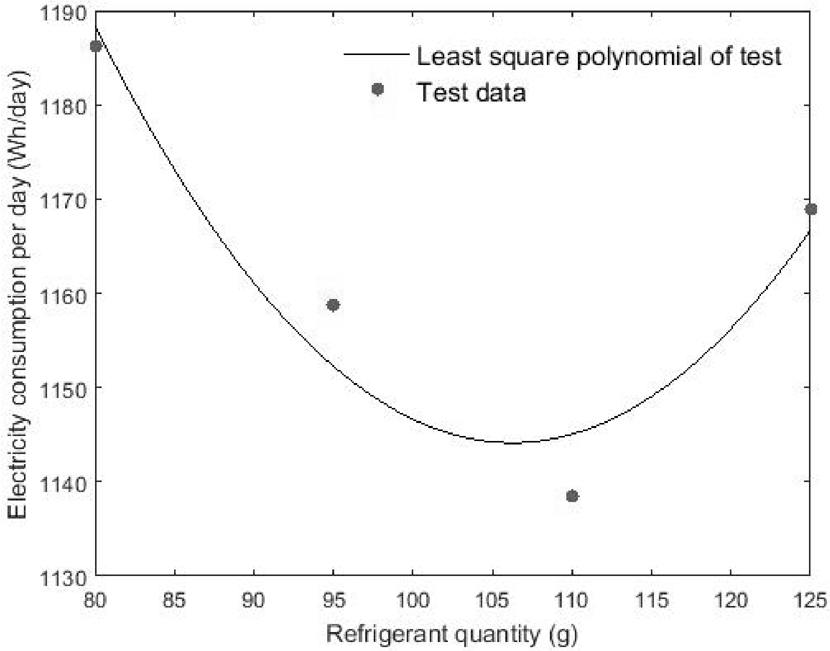

Figure 3.3 Experimental results of household refrigerator

Table 3.4 Experimental results of household refrigerator

Refrigerant charge amount	80g	95g	110g	125g
Freezer temperature	-15.8℃	-16.3℃	-16.5℃	-16.5℃
Fridge temperature	1.7℃	1.2℃	1.0℃	1.2℃
Electricity consumption (Wh/day)	1186.2	1158.5	1138.5	1168.9
Operation ratio of compressor (%)	79.1	73.5	71.3	72.0

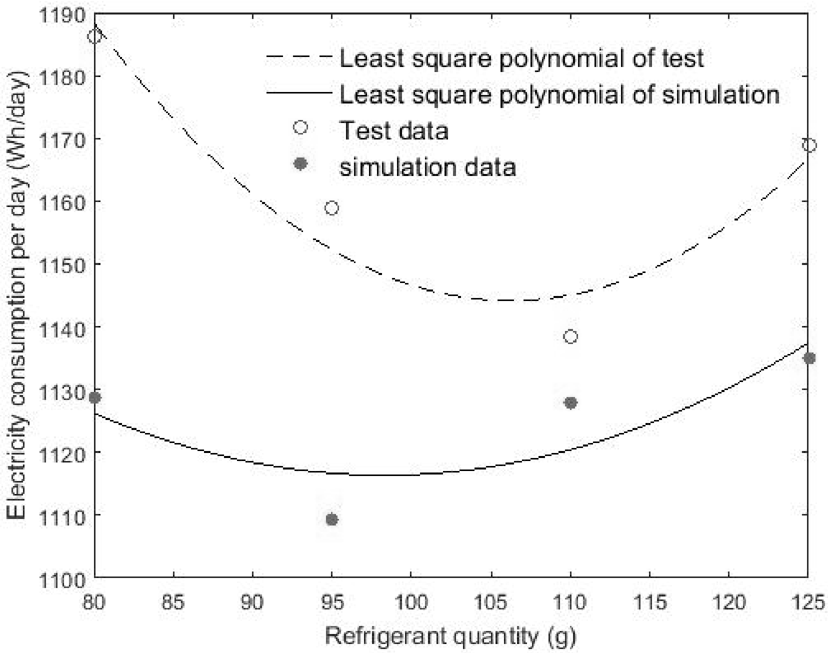

Figure 3.4 Comparison of numerical simulation and experiment

Chapter 4. Conclusion

In this research, the numerical study and experiment were conducted to find the optimal refrigerant charging amount for minimum power consumption of the household refrigerator.

The numerical study was to establish the components modeling and comprehensive analysis of household refrigerator under the transient state. For a reflection of practicality, the modeling of compressor is including solution of oil and refrigerant by fitting values of separate experiment. The founded model was based on thermodynamic theories and the concept of fluid dynamics.

The experimental study was conducted with side by side refrigerator at the constant temperature and humidity chamber. It was verified that the optimal refrigerant charging amount of minimum power consumption is exist.

From this study it was to know the following.

(1) To estimate performance of refrigerator, the numerical simulation was carried out with transient model.

(2) Optimal refrigerant charge amount was obtained for minimum electricity consumption.

(3) Optimal refrigerant charge amount was matched well with numerical model.

References

Goncalves, J. M. & Melo, C., 2004, Experimental and numerical steady-state analysis of a top-mount refrigerator, 10th International Refrigeration Conference at Purdue, West Lafayette, USA, R-078.

Hermes, C. J. & Melo, C., 2006, A dynamic simulation model for fan-and-damper controlled refrigerators, 11th International Refrigeration Conference at Purdue, West Lafayette, USA, R-036

Zhelezny, P. V. Zhelezny, V. P., Procenko, D. A. & Ancherbak, S. N., 2007, An experimental investigation and modelling of the thermodynamic properties of isobutane–compressor oil solutions: Some aspects of experimental methodology. International Journal of Refrigeration, 30(3), 433-445.

Mezavila, M. M. & Melo, C., 1996, Capheat: An Homogeneous Model to Simulate Refrigerant Flow Through Non Adiabatic Capillary Tubes. Proceedings of the IIR-Purdue Refrigeration Conference. West Lafayette, USA, pp. 95–100, 1996

Borges, B. N., Hermes, C. J., Gonçalves, J. M., & Melo, C. (2011). Transient simulation of household refrigerators: a semi-empirical quasi-steady approach. Applied Energy, 88(3), 748-754.

Xu, B., & Bansal, P. K. (2002). Non-adiabatic capillary tube flow: a homogeneous model and process description. Applied Thermal Engineering, 22(16), 1801-1819.

초 록

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이 연구에서, 열교환을 하는 모세관을 갖는 가정용 냉장고의 소비전력을 최소화하는 냉매 충전량을 찾기 위해 수치 해석 및 실험을 진행하였습니다. 실험은 주위온도 25 ℃ 에서 냉매 충전량을 80g에서 125g까지 15g 간격으로 변경하며 진행하였습니다. 실험결과에 따르면, 가정용 냉장고의 소비 전력은 냉매 충전량 104g에서 최소화되었습니다. 수치 해석은 유한 차분 법을 통해 실험과 동일한 조건으로 계산하였습니다. 수치해석 결과는 전력 소모가 냉매 충전량 98g에서 최소화되었습니다. 수치해석 결과와 실험 데이터를 비교해보면, 수치해석 모델이 최적의 냉매 충전량을 잘 예측하는 것을 확인할 수 있었습니다.

주요어: 가정용 냉장고, 냉매 충전량 최적화, 동적 해석 모델, 열교환을 하는 모세관.

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