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A numerical study on oil retention and migration characteristics in the heat pump system †

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Abstract

In HVAC system, the oil circulation is inevitable because the compressor requires the oil for lubrication and sealing. A small portion of the oil circulates with the refrigerant flow through the system components while most of the oil stays or goes back to the compressor. Because oil retention in refrigeration systems can affect system performance and compressor reliability, proper oil management is necessary in order to improve the compressor reliability and increase the overall efficiency of the system. This paper describes a numerical analysis of oil distribution in each component of the commercial air conditioning system including the suction line, discharge line and heat exchanger. In this study, system modeling was conducted for a compressor, discharge line, condenser, expansion valve, evaporator and suction line. Oil separation characteristics of the compressor were taken from the information provided by manufacturer. The working fluid in the system was a mixture of a R-410A refrigerant and PVE oil. When the oil mass fraction (OMF) was assumed, oil mass distribution in each component was obtained under various conditions. The total oil hold-up was also investigated, and the suction line contained the largest oil hold-up per unit length of all components.

Keywords: Oil mass fraction (OMF); Retention; Migration; Oil hold-up; Void fraction; Heat pump

1. Introduction

A conventional heat pump system consists of compressor, condenser, evaporator and expansion device. Only compressor among them has the moving part, therefore, it requires lubricating oil to prevent wear. Accordingly, it cannot help avoiding for the oil to be discharged from compressor with the refrigerant. Most of the discharged oil can return to the compressor by a separator, however, the rest of the oil is accumulated or migrates in a certain part of the heat pump system such as discharge line, heat exchangers, or suction line. It can lead to bring many unfavorable effects such as trapping more oil in each component, decreasing the heat transfer performance, and increasing the pressure drop. Furthermore, amount of the oil in the compressor can decrease depending on the oil separator's characteristics. It can also lead a severe mechanical failure over the moving parts of the compressor. Therefore, oil must be returned to the compressor and proper oil level should be maintained. Oil returning characteristics are affected by physical properties of fluids, flow pattern, flow conditions and surrounding conditions. ASHRAE presented considerable designing factors for refrigeration system and it contains proper oil-refrigerant relation and mentioned oil returning issues [1]. Sumida *et al.* has tested R-410A to observe flow patterns in the liquid line and evaluate oil return characteristics. He suggested that oil can be accumulated in the liquid refrigerant line since the oil moving velocity was smaller than the liquid refrigerant velocity [2]. Riedle *et al.* were among the first re-

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searchers to systematically characterize the flow of oil-refrigerant mixtures. Their analytical model based on minimum gas velocities introduced the concepts of void fraction, oil entrainment, and liquid film thickness for oil-refrigerant mixtures [3]. Schlager *et al.* conducted experiments in order to determine the quantity of oil in smooth and micro-fin tubes during evaporation and condensation of refrigerant-oil mixtures. They showed that the parameters that affect oil retention were mass flux, oil mass fraction, mixture viscosity, evaporator exit conditions, and evaporation pressure [4]. Biancardi *et al.* conducted experimental and analytical studies to determine the lubricant circulation characteristics of HFC/POE in a residential heat pump system [5]. Additionally, oil return characteristics in vertical upward flow were experimentally and theoretically investigated by Mehendale et al. The critical mass flow rate for preventing oil film reversal in a vertical pipe for vapor refrigerant with R-22, R-407C, and R-410A with MO and POE was pointed and was compared with the results by Jacobs et al. [6-7]. Burton *et al.* have studied vapor-liquid equilibrium for R-32 and R-410a with POE oil and compared some pre-existing correlations [8]. Fukuta et al. suggested measurement method for oil-refrigerant mixture concentration ratio using refractive index [9]. Lottin et al. have performed study on the effect of lubricating oil to refrigeration system by experimentally and analytically with R-410a and POE oil. They confirmed that the oil retention in the heat exchanger degrades its efficiency significantly and causes more pressure drop. The whole system performance thereby goes worse the oil retention amount increases [10]. Finally, Yana Mota et al. visualized the flow of R-404a and POE oil mixture in capillary tube and revealed that oil influences the vaporization and flow regime [11].

In this study, simulation method is suggested to describe refrigeration system and then, physical model is established determining oil retention amount in each component. Oil retention is obtained for each component under various operating conditions.

2. Modeling of the heat pump system and oil retention

2.1 Modeling of the heat pump system

The heat pump system mainly consists of compressor, condenser, expansion valve and evaporator. The simplified schematic diagram of the heat pump system

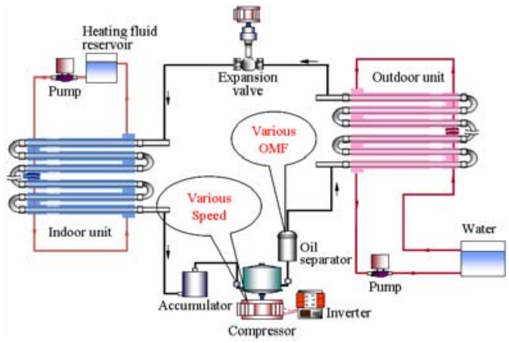

Fig. 1. Schematic of heat pump system.

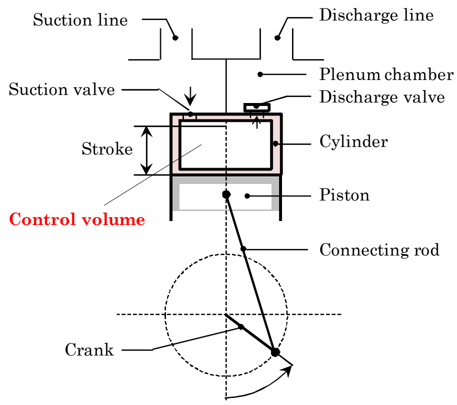

Fig. 2. Schematic diagram of a compressor.

is depicted in Fig. 1. The compressor volumetric efficiency is used to decide mass flow rate and compressor discharge enthalpy is calculated by isentropic efficiency. Corresponding Eqs. are shown as follows.

$$\eta_v = 1 - m\left(\frac{v_{suc}}{v_{dis}} - 1\right) \tag{1}$$

$$\dot{V}r = \frac{\pi}{4} B^2 S \times \frac{rpm}{60} \tag{2}$$

$$\dot{m}_{comp} = \frac{\dot{V}r \cdot \eta_v}{v_{suc}} \tag{3}$$

$$h_{dis} = h_{suc} + \frac{h_{dis,isen} - h_{suc}}{\eta_{isen}} \tag{4}$$

where η_v, η_{isen}, v_{suc}, and v_{dis} denote the volumetric efficiency, isentropic efficiency, suction volume and discharging volume, respectively. B and S are the bore and stroke length of the compression chamber as shown in Fig. 2, and finally h means the enthalpy of oil-refrigerant mixture in the suction and discharging line. An orifice model is adopted as expansion device.

The general orifice equation is shown below to calculate the mass flow rate flowing through the expansion valve.

$$\dot{m}_{\exp} = C_d A \sqrt{2\rho_i (P_i - P_o)} \quad (5)$$

where C_d, ρ_i, P_i, and P_o denote the orifice coefficient, density of the inlet flow and pressure of the inlet and outlet flow, respectively. The refrigerant passing through the heat exchanger made of concentric dual tubes is assumed to flow counter currently. Whole sections of discharge line, condenser, evaporator, and suction line are divided as 1000 grids, and energy conservation equation is applied to each subsection. Furthermore, continuity equation and momentum equation are also used to calculate mass flow rate and pressure drop as shown in Eqs. (6)-(8).

$$\frac{\partial(\rho A)}{\partial t} + \frac{\partial \dot{m}}{\partial z} = 0 \quad (6)$$

$$\frac{\partial(\rho A h)}{\partial t} + \frac{\partial(\rho \dot{m} h)}{\partial z} = U\tilde{P}(T_o - T_r) \quad (7)$$

$$(\rho A c_p)_o \frac{\partial T_o}{\partial t} + (\dot{m} c_p)_o \frac{\partial T_o}{\partial z} = U\tilde{P}(T_r - T_o) \quad (8)$$

For all system components, mass flow rate of the refrigerant-oil mixture is balanced iteratively either at compressor or expansion device at a given pressure. All correlations used for the heat exchanger modeling is given in Table 1.

2.2 Modeling of oil retention

2.2.1 Oil retention in suction and discharge line

The suction line and discharge lines are assumed to be vertical annular tubes. In the suction and discharge lines of which length are 50 m because the heat pump system is designed for commercial air conditioning, refrigerant flows in the gas phase and oil is assumed to flow in annular liquid film. The driving force of oil flow is shear interfacial force caused by refrigerant flow as shown in Fig. 3. Then, oil velocity profile is obtained as Eq. (9) applying Navier-Stokes equation to each flow based on the above assumption.

$$u_{oil}(y) = \left(\frac{1}{\rho_{oil}\nu_{oil}}\frac{dP}{dx} - \frac{g}{\nu_{oil}}\right)\left(\frac{y^2}{2} - \frac{a}{2}y\right) + \frac{V_{interface} \cdot y}{a} \quad (9)$$

Integrating the velocity, oil volume flow rate can be obtained as Eq. (10).

$$\begin{aligned} \dot{Q}_{oil} &= 2\pi R \int_0^a u_{oil}(y)dy \\ &= \left(\frac{g}{\nu_{oil}} - \frac{1}{\rho_{oil}\nu_{oil}}\frac{dP}{dx}\right)\frac{a^3 \pi R}{6} + V_{interface} a \pi R \\ &= \frac{\dot{m}_{oil}}{\rho_{oil}} \end{aligned} \quad (10)$$

At the interface between oil and refrigerant, shear stress effecting oil flow is expressed as Eq. (11).

Table 1. Correlations used for the heat exchanger modeling.

	Experimental correlations	Details
Traviss	$\frac{hD}{k_l} = \frac{0.15\mathrm{Pr}_l \mathrm{Re}_l^{0.9}}{F_2}[\frac{1}{X_{tt}} + \frac{2.85}{X_{tt}^{0.476}}]$ Re : Reynolds number ($\mathrm{Re} = GD/\mu$)	Condensation (two phase)
Gungor & Winterton	$h_{tp} = Eh_l$ $E = 1 + 3000Bo^{0.86} + 1.12(\frac{x}{1-x})^{0.75}(\frac{\rho_L}{\rho_v})^{0.41}$	Evaporation (two phase)
Dittus & Boelter	$h_l = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}\frac{k_l}{d}$ Re : $\frac{G(1-x)d}{\mu_l}$ (Liquid - only Reynolds number) Pr : $\frac{\mu_l c_{p,l}}{k_l}$ (Prandtl number) G : mass flux(kg/m^2s)	Single Phase Refrigerant
Gnielinski	$h_G = \frac{(\varepsilon/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7\sqrt{(\varepsilon/8)}(\mathrm{Pr}^{2/3}-1)}[1+(\frac{d}{L})^{2/3}]K$ $\varepsilon = (1.82\log\mathrm{Re} - 1.64)^{-2}$, $K = (\mathrm{Pr}/\mathrm{Pr}_w)^{0.11}$	Secondary Fluid (water)

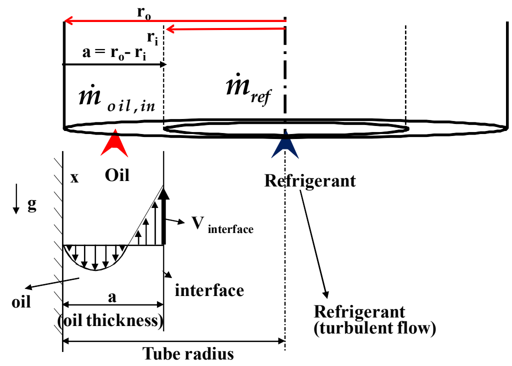

Fig. 3. Cross section of the discharge and the suction line.

$$\tau_{oil_a} = \mu_{oil}\left[\left(\frac{1}{\rho_{oil}\nu_{oil}}\frac{dP}{dx} - \frac{g}{\nu_{oil}}\right)\frac{a}{2} + \frac{V_{interfaace}}{a} \right] \qquad (11)$$

On the other side, shear stress of refrigerant side can be expressed like Eq. (12).

$$\tau_{ref_a} = 0.0225\rho_{ref}V_{ref}^2\left(\frac{2\nu_{ref}}{V_{ref}D_i}\right)^{0.25} \qquad (12)$$

By Eqs. (10)- (12), we can calculate the annular oil film thickness ‘ a ’ as well as the oil retention amount in each grid.

2.2.2 Oil retention in heat exchanger

In the heat exchanger, three parts of refrigerant flow such as liquid phase, two phase and gas phase have to be considered because oil film can be solved in liquid phase refrigerant by its maximum solubility. In the cross sectional area of the tube, the area ratio of the vapor phase refrigerant is called void fraction. In this study, Premoli void fraction model which includes various factors below is used to calculate oil retention amount [12].

At first, OMF (oil mass fraction) and local OMF are defined as following Eqs.

$$OMF = \frac{\dot{m}_{oil}}{\dot{m}_{oil} + \dot{m}_{ref}} \qquad (13)$$

$$OMF_{local} = \frac{OMF}{1 - x_{mix}} \qquad (14)$$

$$x_{mix} = \frac{\dot{m}_{ref,gas}}{\dot{m}_{ref,gas} + \dot{m}_{ref,liq} + \dot{m}_{oil}} \qquad (15)$$

Then, a void fraction as shown in Fig. 4 can be ex-

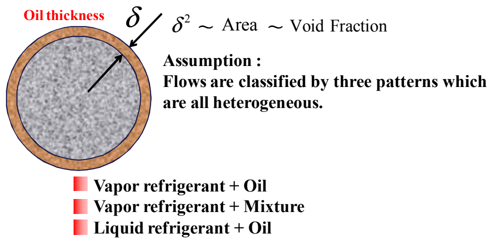

Fig. 4. Cross section of the tube and flow conditions (Inner area can be regarded as the void region).

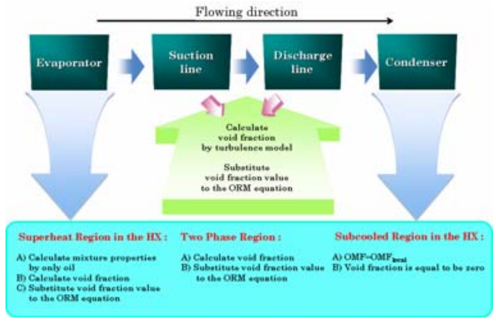

Fig. 5. Summary of process to obtain the oil retention mass in each component.

pressed like Eq. (16) which was obtained by experimental correlation.

$$\alpha = \frac{1}{1 + S \cdot \dfrac{1 - x_{mix}}{x_{mix}} \cdot \dfrac{\rho_{ref,vap}}{\rho_{mix,liq}}} \qquad (16)$$

where

$$S = 1 + B_1\left[\frac{y}{1 + yB_2} - yB_2\right]^{1/2}$$

$$B_1 = 1.578 \cdot \mathrm{Re}_{mix}^{-0.19} \cdot \left(\frac{\rho_{mix,liq}}{\rho_{ref,vap}}\right)^{0.22}$$

$$B_2 = 0.0273 \cdot We_{mix} \cdot \mathrm{Re}_{mix}^{-0.51} \cdot \left(\frac{\rho_{mix,liq}}{\rho_{ref,vap}}\right)^{-0.08}$$

$$\mathrm{Re}_{mix} = \frac{G_{tot} \cdot D_h}{\mu_{mix,liq}}, \; We_{mix} = \frac{G_{tot} \cdot D_h}{\sigma_{mix,liq} \cdot \rho_{mix,liq}}$$

If a void fraction in each grid is obtained, oil retention mass can also be calculated by Eq. (17).

$$\text{Oil retention mass} = OMF_{Local} \cdot V_{segment} \cdot (1 - \alpha) \cdot \rho_{mix,liq}$$

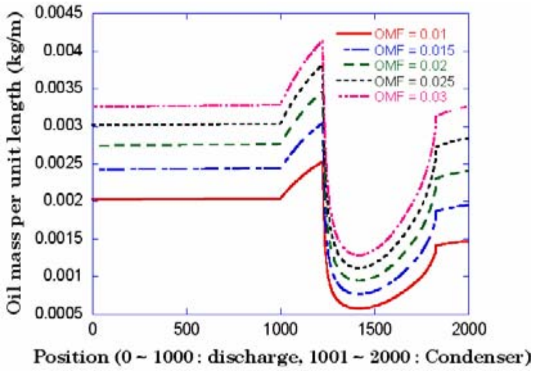

Fig. 6. Oil mass per unit length for various OMF conditions in discharge line and condenser.

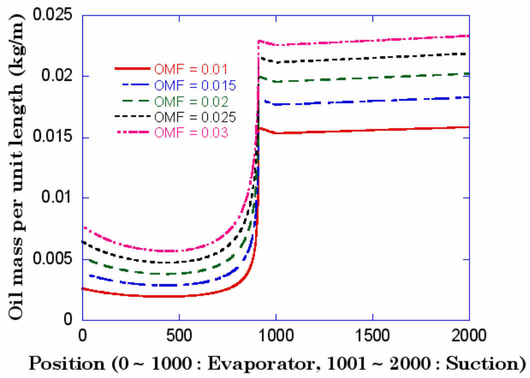

Fig. 7. Oil mass per unit length for various OMF conditions in evaporator and suction line.

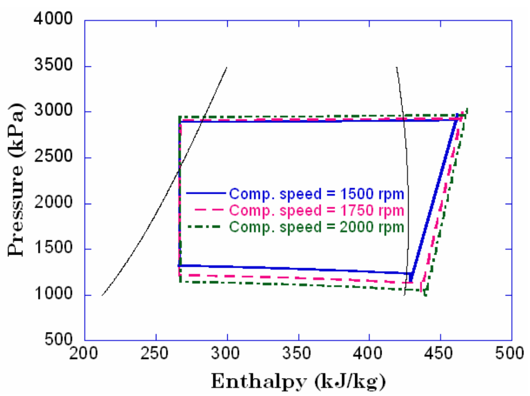

Fig. 8. Pressure-enthalpy curve under various compressor speed conditions.

$$\text{Oil retention mass} = OMF_{Local} \cdot V_{segment} \cdot (1-\alpha) \cdot \rho_{mix,liq} \tag{17}$$

Finally, the method to obtain oil retention mass in each component is summarized in Fig. 5.

3. Simulation results

The simulation was conducted under various oil mass fractions. In the graphs, the transverse axis presents not only the number of grids in each component but the moving direction of the refrigerant flow from low number to high number. Additionally, the oil retention amount and each component oil retention ratio of total oil hold up were also investigated.

3.1 Oil retention distribution under various oil mass fraction conditions

The Fig. 6 shows oil mass distribution in the discharge line and condenser under various OMF conditions, while the Fig. 7 shows it in the evaporator and suction line. At same condition, the oil retention in discharge line and suction line increases very slightly because the refrigerant velocity decreases due to the gravity and the pressure drop although the heat transfer does not exist. However, the oil retention in the suction line is about 7 times larger than in the discharge line because the refrigerant velocity in the suction line is slower and the viscosity of the oil-refrigerant mixture in the suction line is about 30 times larger due to low evaporating temperature [13].

In the mean time, the velocity and the temperature of the superheated refrigerant in the condenser inlet decrease sharply due to the heat transfer and pressure drop until the refrigerant meets two phase region. Thus, the oil retention in the condenser inlet increases rapidly. Then, the oil retention decreases in the two phase region when the oil is mixed with the liquid phase refrigerant. However, the oil retention increases again when the liquid phase refrigerant occupy in cross section of the tube more than the gas phase refrigerant. Besides, the oil retention in the subcooled region increases a little because the velocity and the temperature of the refrigerant decrease.

Meanwhile, the oil retention in the two phase region of the evaporator inlet decreases because the liquid phase refrigerant decreases and the shear stress between the gas phase refrigerant and the oil film increases. Nevertheless, the oil retention increases radically in the high mass quality region because the oil-refrigerant mixture viscosity increases rapidly although the velocity of the refrigerant becomes high. Then, the oil retention in the superheated region of the evaporator decreases very slightly because the property of the mixture does not change rapidly and the velocity of the refrigerant becomes faster in this region.

Finally, OMF effects on the oil retention in the discharge line, the superheated region and subcooled region of the condenser more than in two phase region as shown in Fig. 6. Furthermore, it also affects the oil retention in the suction line more than in two phase region of the evaporator as shown in Fig. 7.

3.2 Oil retention distribution under various compressor speed conditions

Fig. 8 shows the system performance change when the compressor speed increases from 1500 rpm to 2000 rpm. In this case, temperature change in the evaporator and suction line is larger than in the condenser and discharge line, which leads to change the oil mass retention in the evaporator and suction line more effectively. As shown in Fig. 9, the oil retention in the discharge line can be reduced as the compressor speed increases because the shear stress of the refrigerant becomes larger, which can influence the superheated region in the condenser. However, the oil retention cannot be affected by compressor speed in the two phase region and subcooled region because the condensing temperature and pressure were little influenced by the compressor speed. Fig. 10 shows the oil mass distribution in the evaporator and suction line under various compressor speeds. According to Fig. 8, the oil retention in both the evaporator and suction line can be predicted to be larger by its lower viscosity according to the lower evaporating temperature and pressure when the compressor speed increases. However, contrary to the expectation, the oil mass retention in the suction line is reduced as seen in Fig. 10, which means the faster refrigerant velocity in the evaporator by increasing compressor speed can generate more shear stress, and drag more oil flow in spite of lower viscosity. Finally, it can be one of the effective methods for returning the oil to increase compressor speed because the most oil exists in the suction line.

3.3 Analysis of the simulation results

Tables 2 and 3 show the oil retentions per unit length according to the each component under various OMF and compressor speed conditions, respectively. When the OMF value is 1%, the portion of

Table 2. Summary of oil retention results in R410A/VG68 under various OMF conditions.

Part	1.0 wt.% (g/m)	1.5 wt.% (g/m)	2.0 wt.% (g/m)	1.0 wt.% (%)	1.5 wt.% (%)	2.0 wt.% (%)
Suc.	15.6	18.0	19.9	69.6	66.9	65.1
Evap.	3.6	4.8	6.0	16.1	17.9	19.8
Dis.	2.0	2.4	2.7	8.9	9.1	8.9
Cond.	1.2	1.6	1.9	5.4	6.1	6.2
Total	22.4	26.8	30.5	100	100	100

Table 3. Summary of oil retention results in R410A/VG68 under various compressor speed conditions.

Component	1500 rpm (g/m)	1750 rpm (g/m)	2000 rpm (g/m)	1500 rpm (%)	1750 rpm (%)	2000 rpm (%)
Suc.	19.9	18.5	18.3	65.1	55.2	52.4
Evap.	6.0	10.5	12.3	19.8	31.5	32.3
Dis.	2.7	2.6	2.5	8.9	7.8	7.1
Cond.	1.9	1.9	1.8	6.2	5.5	5.2
Total	30.5	33.5	34.9	100	100	100

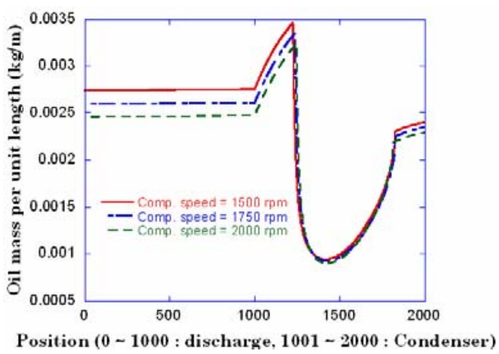

Fig. 9. Oil mass per unit length under various compressor speed conditions in discharge line and condenser.

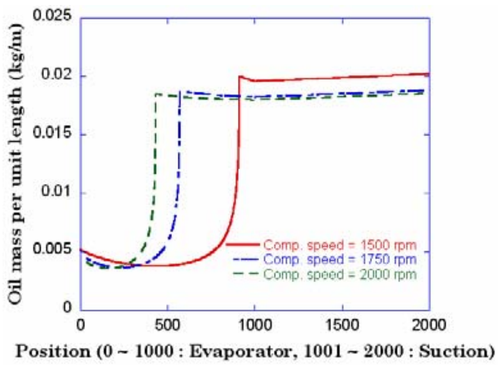

Fig. 10. Oil mass per unit length under various compressor speed conditions in evaporator and suction line.

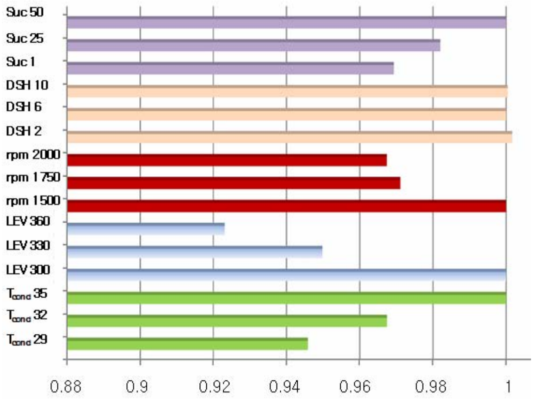

Fig. 11. Total oil hold-up ratio in the system compared to the reference condition (OMF = 2%, $T_{water,\ cond.\ inlet}$ = 35°C, Opening = 300, Compressor speed =1500 rpm).

oil retention in the suction line is 69.6% as shown in Table 2, which is the largest portion of all components compared with 16.1% in the evaporator, 8.9% in the discharge line and 5.4% in the condenser. Furthermore, we can see the oil unsolved in the refrigerant is difficult to be migrated in the superheat region. Thus, it can be an effective method to reduce the oil mass retention in the suction line and superheat region in the evaporator. From Table 3, the oil mass retention in the suction can be reduced by increasing the compressor speed. Additionally, the total oil hold-up in the whole system component was also investigated under various compressor speeds, suction lengths, degrees of superheat, openings of the expansion valve and condensing temperatures as well as under various OMF conditions. Fig. 11 presents the total oil hold-up except the compressor compared to the reference condition. As the ratio of total oil hold-up becomes smaller, the oil returns to the compressor very well. In Fig. 10, the most effective oil return method is to increase the compressor speed and to decrease the expansion valve opening.

4. Conclusions

The oil retention in the discharge line, suction line, condenser, and evaporator of the commercial air conditioning system was simulated on this study. Additionally, when the OMF increases 0.5% at a time from 1% to 3%, the oil retention in the discharge line, the suction line and the single phase region of the heat exchanger are more influenced by OMF rather than in the two phase region of the heat exchanger. Furthermore, the efficient method of returning the oil is to increase the compressor speed and to decrease the expansion valve openings. These results can be used to design each component and to find the optimized oil return method through the verification with the experimental study.

Acknowledgment

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