Simulation and experimental evaluation of the effects of oil circulation in an inverter air conditioning system using R-22 and R-407C

P. Sarntichartsak a,*, V. Monyakul b, S. Thepa a, A. Nathakaranakule a

a School of Energy and Materials, King Mongkut’s University of Technology Thonburi, 126 Pracha Utit Road, Bangmod, Thungkru, Bangkok 10140, Thailand
b National Science and Technology Development Agency (NSTDA), 111 Thailand Science Park Paholyothin Road, Klong 1, Klong Luang, Phathumthani 12120, Thailand

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Abstract

A steady-state mathematical modeling and experimental study were carried out to investigate effects of oil circulation in an inverter air conditioner using R-22 and R-407C. To highlight the prediction of oil circulating mass fraction, the simplified sub-model of oil discharge was modified. The predicted results were validated with experimental data. Two tested lubricants were mineral oil (MO) and polyol-ester lubricant (POE). The experiments varied the compressor frequency in the range of 30–50 Hz for each compressor oil level (0.8 l and 1 l). The results showed that the circulating oil flow rates decreased with the reducing compressor frequency and with a lower oil charge. The lubricant concentration affected the system performance at high frequencies. The charged oil quantity of 0.8 l provided more efficient performance than 1 l. The vapor velocity of R-407C is inadequate high enough to entrain the liquid MO at its lowest frequency. The immiscible mixture of R-407C/MO is not suitable used in the inverter air conditioning system. The proposed model obtains better results for the miscible mixture. The model prediction agreed with the measured values in the compressor frequency of 40–50 Hz.

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1. Introduction

In most air conditioner, the actual cooling load is not constant. Generally, the refrigeration systems are designed for the expected peak load. The variation of load suggests that there should be some capacity controlling techniques for use during continuous system operation. The inverter driven air conditioner with varying cooling capacity has been studied during the last two decades and the well-known technique is to control the rotational speed of the compressor with respect to its cooling load. Its advantages are energy conservation and comfort of the indoor environment over on/off compressor cycles.

R-22 is ozone depleting agent that is used in compressor and in 2020 Montreal Protocol will ban the use of it in 2020. Presently, R-407C and R-410A are considered as the major substitutes of R-22. R-407C is a non-ozone-depleting blend of R-32, R-125 and R-134a and their nominal mass fraction composition are 0.23, 0.25 and 0.52, respectively. Since their vapor pressure and performance characteristics are similar to those of R-22, the blend of these agents is the suitable replacement for R-22.

In practical vapor compression system, a small amount of lubricant oil is necessary for lubricating the sliding parts in compressors. However, part of the lubricant oil may migrate from the compressor to other parts of the system. The viscosity of lubricant oil should be three orders higher than that of the refrigerant and the surface tension should be approximately one order higher. Lubricant oil may have significant impacts on the compression system performance...
because it affects the transport properties and the associated heat transfer characteristics of the mixtures. The presence of the lubricant oil would considerably affect the transport properties of the mixture and may have a significant impact on the associated heat transfer characteristics of the refrigerant. Therefore, it may have a significant impact on the system performance. Especially, in variable capacity systems, the lubricant may be overcharged into compressor for the sufficient oil level in compressor crankcase due to inadequate lubricant return at speeds lower than the rated speed.

The circulation of lubricant in the compressor can be affected by the lubricant properties in liquid-phase and the refrigerant miscibility. In general, R-22 and MO are miscible over most of the expected ranges of operating conditions for normal air conditioner. Miscibility is generally believed to promote the return of lubricant to the compressor. R-407C is miscible with POE lubricant and immiscible with MO but the POE lubricant is more expensive than MO, the experiment was interested in using MO with R-407C.

According to previous works, Kruse and Schroeder [1] and McMullan and Murphy [2] studied the influence of oil on the performance of the R-12 heat pump system. Their results demonstrated that the circulating lubricant could degrade the heat exchangers performance. Popovic et al. [3] investigated the effects of miscibility and viscosity on performance of the R-134a system. He suggested that the performance could be improved by usage of the miscible lubricants with low viscosities. Lottin et al. [4,5] modeled the R-410A refrigeration system and simulated the oil effects on the performance. The simulation showed that oil had significant effects on the system performance when the oil mass fraction was above 0.5%. Moreover, performance of the heat exchanger reached its optimum when the oil concentration was 0.1%. Sekhar et al. [6] experimented the performance of a R-134a/R-600a/R-290 mixture on refrigeration system with mineral oil. They found that the new mixture performed better than R-12 and the miscibility between the retrofitted refrigerant and lubricant were good. In terms of compressor, Winandy
and Cuevas [7] conducted experimental analysis of the lubricant oil returning to compressor for several operating conditions in parallel scroll compressors. The compressors were equipped with two oil level measurements and a special connecting pipe. This configuration could present serious troubles linked to oil return, especially at the part load. Kim and Lancey [8] modeled the oil distribution in the rotary compressor and the model could predict the oil flow rate at various operating conditions. Recently, Navarro et al. [9] tested five compressors working with POE oil and found that the oil concentration leaving the compressor was independent of operating conditions.

From the mentioned literature review, there are not many models that simulate the oil circulation within the inverter air conditioning system. Therefore, the aims of this work were:

1. To model the effects of circulating lubricant in a variable capacity system using R-22 and R-407C as refrigerant.
2. To investigate the effects of quantity of mineral oil and polyol-ester lubricant in the variable capacity rotary compressor on the compressor performance.
3. To investigate the impact of using mineral oil with R-407C in an inverter driven air conditioner.

2. Experimental setup and procedure

2.1. Experimental apparatus

A schematic diagram of the experimental setup for the variable capacity system is shown in Fig. 1. It consisted of an indoor and outdoor chamber. The test unit was a split type inverter air conditioner with a rated cooling capacity of 5.27 kW using R-22 as the refrigerant. The condensing unit used a variable speed rotary compressor. An inverter was used to control the speed. The volumetric displacement was 77.7 cm$^3$ rev$^{-1}$ and the rotational speed ($N$) was 1425 rev m$^{-1}$ at nominal frequency. The crankshaft diameter was 25 mm. The oil pick-up plate was installed within an oil gallery of 15 mm diameter. On the shaft wall, five radial oil feeding holes of 3 mm diameter were drilled as shown in Fig. 2. An oil separator was installed at discharged line. The geometric parameters of the heat exchangers are shown in Table 1. The working fluid is throttled by three capillary tubes. The inner tube diameter is 1.1 mm. The tube length was optimized to 914 mm. All the connected pipes were thermally insulated, the inside tube diameter of liquid line, suction line and discharge line were 8.9 mm, 13 mm and 8.9 mm, respectively. The sampling tube was inserted into a liquid line and it was connected with a sample bottle of about 50 ml. For R-407C test, refrigerant weight was adjusted and the filter-dryer was changed to be compatible with retrofitted fluid.
2.2. Measurements

The temperature and pressure of refrigerant were measured at several locations (as shown in Fig. 1) using T-type thermocouples and pressure gauges with accuracies of ±0.1 °C and ±0.2% of full scale, respectively. Refrigerant temperatures within the coil were measured by 24 thermocouples mounted on the surface of the copper return bend and they were well insulated from the ambient temperature. A flow meter with an accuracy of ±0.2% was installed in the liquid line.

Air temperatures were measured using RTDs (Pt100) with an accuracy of ±0.05 °C. The face air velocity was measured with a hot-wire anemometer with an accuracy of ±1%.

The kilowatt-hour power meter was used to measure the energy consumption of the system. To measure the input power, voltage and electrical current, the clip-on power meter with an accuracy of ±0.01% of full scale was used.

Three data loggers were used to collect data from testing apparatus. The experimental data were recorded continuously for 60 min with 10 s intervals. The uncertainties of the cooling capacity and COP estimated by analysis were approximately 0.8% and 1%, respectively. The kilowatt-hour power meter was used to measure the energy consumption of the system. To measure the input power, voltage and electrical current, the clip-on power meter with an accuracy of ±0.01% of full scale was used.

The test condition and procedure

1. The lubricant was filled into compressor of 0.8 l or 1 l (these volumes are recommended by manufacturers). The difference of oil level was 20 mm as shown in Fig. 2. In this study, two compressors were provided with MO and POE lubricant. The grades of the tested lubricants are ISO-46 and ISO-32, respectively.
2. The system was evacuated for 1 h.
3. The tested air condition was carried out corresponding to the ASHRAE test condition “A”. The inlet air state at the indoor and outdoor units were maintained of 27 °C dry bulb 19.5 °C wet bulb, and 35 °C dry bulb 24 °C wet bulb, respectively.
4. The vapor phase of R-22 was charged into the system inlet of the compressor. The amount of R-22 charged to the system, based on manufacturing recommendations, was around 1.4 kg. For retrofitted fluid tests, the amount of R-407C of 1.5 kg must be charged into a liquid line.
5. The compressor frequency is varied over 30, 35, 40, 45 and 50 Hz.
6. The influences of oil circulation were observed with and without an oil separator.
7. The tests were carried out in steady-state conditions. The oil concentration was determined by sampling the liquid region downstream of the condenser, just prior to entering the capillary tube. This method is based on ASHRAE standard 41.4-1996 [10].

Note that the system was clean inside tube. Each test was carried out three times under the same operating conditions.

3. System modeling

The simulation program is based upon steady-state mathematical models of the components in the refrigeration system including a variable speed compressor, two finned tube heat exchangers and capillary tubes. The thermodynamic and transportational properties of the refrigerants were evaluated by using REFPROP computer program. The thermophysical properties of lubricant oils and mixer of lubricant oils with refrigerants were estimated by using set of mathematical relations shown by Conde [11]. Note that Cho and Tae [12] presented that the local oil concentrations did not affect the flow pattern of refrigerant during phase change.

3.1. Modeling of the compressor

3.1.1. Variable speed compressor

Threlkeld [13] developed the approximate expression for compression work assuming a polytropic process as

\[
W = \frac{m_r}{\eta_m} \left( \frac{k}{k-1} \right) P_i v_i (1 - PR^{\frac{1}{k}}) 
\]

(1)

The pressure ratio (PR) can be calculated from a well-known correlation given by Threlkeld [13] for ideal gas under a polytropic process. In this study, the differences in compressor efficiency due to oil were found to be negligible. To calculate the mass flow rate of refrigerant \(m_r\) imposed by the compressor, the following equation is used as

\[
m_r = \eta_s NV \frac{N}{v_m} 
\]

(2)

\[
N = \frac{120f(1-s)}{P} 
\]

(3)
3.1.2. Simplification of the oil distribution model

Previous work, modeling of the rotary compressor includes the oil distribution sub-model was proposed by Kim and Lancey [8]. In this study, the complex calculation of oil supply system was simplified. In Fig. 2, the oil pick-up plate was installed inside the crankshaft. An oil gallery was provided in the crankshaft. On the wall of the shaft, the radial oil feeding holes were drilled for the oil which was contained in the oil gallery to flow into the chambers inside the roller. As the crankshaft rotated, the lubricant oil in the oil sump was pulled into the oil gallery inside the shaft and delivered into various lubrication elements. Commonly, an amount of oil return into an oil sump and the remaining oil flowed to other components in the system. The model considered the oil gallery and the oil pulled plate as a centrifugal pump. The oil flow rate \( q_i \) can be calculated from the operating point of the system resistance curve and the performance curve of shaft pump as following equations:

\[
q_i = q_{\text{max}} \sqrt{1 - (H/H_{\text{ho}})} \quad (4)
\]

\[
H_{\text{o}} = \zeta_{\text{sh}} \frac{\omega^2}{2g} \left( \frac{D_{\text{oil}}}{2} \right)^2 \quad (5)
\]

\[
q_{\text{max}} = c_{v,\text{sh}} A_r \sqrt{2gh_E} \quad (6)
\]

The rotational speed of the rotary compressor was regulated by the inverter so the model varied the inverter’s electrical frequency in order to adjust the rotational speed. Generally, the lower limit of the electrical frequency is 30 Hz to avoid problems for the compressor lubricating by splash. Therefore the range 30–50 Hz is considered. The model assumed that the circulated oil can return back to the compressor at any frequency.

Previous works applied the system resistance curve that the static head was neglected. In this study, the static suction head and the static discharge head were included. When reducing the oil level within compressor, the total static head was elevated, thus, the oil flow rate was reduced. The head coefficient \( (c_{v,\text{sh}}) \) and the discharge flow coefficient \( (\zeta_{\text{sh}}) \) were assumed as 0.15 and 0.5, respectively. The discharged oil into refrigeration cycle \( (q_{\text{out}}) \) was assumed to be approximately 0.5 of the entire oil flow rate \( (q_i) \).

3.2. Modeling of the condenser

3.2.1. Performance of finned tube condenser

In this study, the heat exchanger was divided into a number of small and controllable volumes thus the total capacity was calculated by summing the heat transfer rate of each region. The model could calculate the heat transfer capacity of the coil for complex circuit arrangements. The heating capacity \( (Q_c) \) was calculated from the following equation:

\[
Q_c = \eta m_a C_p a (T_{e, i} - T_{a, i}) \quad (7)
\]

Due to air heating by dry-fin condenser the humidity ratio was constant \( (w_{c, in} = w_{c, out}) \) while the dry bulb temperature \( (T_a) \) was raised. The NTU-effectiveness method was used to analyse the heat exchangers. The effectiveness \( (\varepsilon) \) of the phase charge region could be determined from Eq. (8) and the correlation that was presented by Wang et al. [14] for single-phase region was

\[
\varepsilon = 1 - \exp(-\text{NTU}) \quad (8)
\]

Consequently, the overall heat transfer coefficient \( (U_t) \) can be calculated from correlation shown by Hewitt et al. [15]. It was derived from the assumptions that the fouling resistance and the thermal resistance at the junction of fins and tubes were negligible.

3.2.2. Air-side heat transfer coefficient

The condenser was a fin-and-tube heat exchanger. It comprised the copper tubes assembled with the L-footed type of aluminum fins. The total surface area \( (A_c) \) was a summation of tube and fin surface area. The minimum flow area \( (S_{\text{min}}) \) was the minimum free flow cross-section area between tubes in a plane which was perpendicular to the direction of flow. The total frontal area \( (S_o) \) was a product between the height of the coil \( (L_{\text{coil}}) \) and the length of the coil \( (L_{\text{coil}}) \). The maximum air velocity \( (V_{\text{max}}) \) was air velocity entering the heat exchanger \( (V_{\text{a}}) \) divided by the ratio of \( S_{\text{min}}/S_o \). The Reynolds number of air \( (R_e) \) passing through the minimum cross-section was calculated from maximum air velocity. The air-side heat transfer coefficient of staggered array of tubes \( (h_{\text{sp}}) \) for \( 2 \times 10^3 < R_e < 4 \times 10^4, 0.13 < s/l_f < 0.57 \) and \( 1.15 < p_1/p_2 < 1.72 \), was calculated from the Nusselt number in Eq. (9) [15]. The factor of fluid properties variation \( (f_j) \) could be taken as 1 for air heating and factor of number of tube rows \( (f_2) \) may be taken as 0.92, 0.84 and 0.76 for 3, 2 and 1 rows, respectively:

\[
Nu_a = 0.242 R_e^{0.658} \left( \frac{s}{R_l} \right)^{0.297} \left( \frac{p_1}{p_2} \right)^{-0.091} P_a^{1/3} f_j f_2 \quad (9)
\]

For corrugated fins, the heat transfer coefficient \( (HTC) \) should be multiplied by 1.2 to account for the increase in turbulence. In this case, a continuous-plate fin was used. The imaginary rectangular or hexangular fin was approximated as a circular fin. The equivalent fin diameter \( (D_{e}) \) was used as the circular fin diameter \( (D_f) \) for the calculation of the fin efficiency. The HTC for the total surface taking into account fin efficiency \( (h_{\text{e,f}}) \) could be calculated from an equation as shown by Hewitt et al. [15].

3.2.3. Refrigerant-side heat transfer coefficients

The refrigerant flow in the condenser was classified into vapor phase, two-phase and liquid phase regions. The refrigerant HTC \( (h_{e,i}) \) in the single-phase region was calculated using the Gnielinski [16] correlation multiplied with the enhancement factor as shown by Copetti et al. [17]. The condensation of HTC was calculated from the Sami et al. correlation [18] and Goto et al. correlation [19] for
R-22 and R-407C, respectively. Note that the heat transfer in the returned bend was assumed to be negligible.

3.2.4. Refrigerant pressure drop

There were two types of pressure drop in the copper tube, which were that in the straight tube section and in the returned bend section. The single-phase pressure drop can be determined by the equation and the enhancement friction factor that shown by Copetti et al. [17]. The two-phase pressure drop was calculated by the homogeneous model that given by Choi et al. [20] for both R-22 and R-407C. For the returned bend, the pressure drop correlation of Chisholm [21] and Geary [22] were used for single-phase flow and phase change flow, respectively.

3.3. Modeling of the evaporator

3.3.1. Performance of finned tube evaporator

The effectiveness could be determined from correlation similar to condenser model. The cooling capacity \( Q_e \) could be evaluated from Eq. (10) as follows:

\[
Q_e = \dot{m}_w (i_{a,i} - i_{a,sat,r,i})
\]

The number of transfer units (NTU) was calculated from the following equation as shown by Kim and Bullard [23]. For wet fin, the overall heat transfer coefficient \( U_{r,w} \) was defined by the following equation:

\[
\frac{1}{U_{r,w} A_e} = \frac{b_1(D_i - D_h)}{K_i A_i} + \frac{b_4}{h_{i,a_i} A_i} + \frac{1}{h_{a,w} \left( \frac{A_w}{b_2} + \frac{\eta_{f,w} A_f}{b_1} \right)}
\]

3.3.2. Air-side heat transfer coefficient

The method for developing the evaporator model was the same as that used for the condenser model. However, the cooling coil is commonly operated with the wetted surface, so the condenser model must be modified. The air-side HTC of the wet fin \( h_{a,w} \) could be calculated from the following equation. The coefficients \( b \) involving enthalpy–temperature ratio that were illustrated by Threlkeld [13]:

\[
h_{a,w} = \frac{1}{C_p h_{a,f} + \frac{\delta_w}{K_w}}
\]

The wet fin efficiency \( \eta_{f,w} \) was calculated from the correlation that was given by McQuiston and Parker [24]. For dry fin \( \eta_p \), the differential value of \( w_a - w_{wall} \) was defined as zero. Moreover, Threlkeld [13] gave Eq. (13) for predicting the change of air enthalpy \( i_a \) with respect to the change of humidity ratio \( w_a \), when moist air flows across the totally wet-fin coil. In this study, the apparatus dew point temperature was assumed as the evaporating temperature \( T_e \) and Lewis number \( (Le) \) was assumed as 0.9. A water film thickness was equal to 0.1016 mm [13]. The condensing rate of water \( (m_w) \) was evaluated by mass balance concept:

\[
\frac{di_a}{dw_a} = Le \left( \frac{i_a - i_{a,w}}{w_a - w_{a,w}} \right) + (i_g - 2501 Le)
\]

3.3.3. Refrigerant-side heat transfer coefficients

The refrigerant flow in the evaporator was divided into superheated flow regions and two-phase flow region. The HTC of evaporation inside tube \( (h_{r,i}) \) was calculated from the Sami et al. correlation [18] for R-22 and Gungor and Winterton correlation [25] for R-407C. The heat transfer enhancement was obtained from the experimental data by Passos et al. [26].

For superheated flow region, the HTC was calculated in a way similar to the condenser model because liquid oil flowed from the evaporator to the compressor. The oil was soluble with the non-evaporate quantity of refrigerant. The enthalpy change due to the change in the solubility [27]. Therefore, the solubility sub-model was included in this model.

3.3.4. Refrigerant pressure drop

Chen et al. [28] developed the empirical correlation based on the homogeneous model. These equations were calculated with evaporation of R-22 and R-407C. Note that the pressure-drop penalty factor was determined from the tested result by Passos et al. [26]. The pressure drop of the returned bend and the single-phase flow were calculated corresponding to the condenser model.

3.4. Capillary tube model

Chung [29] and Gu et al. [30] proposed the numerical procedure for simulation of the adiabatic flow in a capillary tube for R-22 and R-407C. In Fig. 3, the homogeneous model was divided into the sub-cooled flow region and the two-phase flow region using the difference in the flash point. The length of the sub-cooled region \( (L_{sub}) \) could be determined from the well-known correlation by Chung [29]. In the liquid–vapor phase region, the zeotropic mixture was different from that of pure refrigerant due to effects of temperature glide. Assuming the mini-pressure drop through the phase changed flow region. The outlet quality \( (x_{out}) \) of a section could be determined from the correlation given by Gu et al. [30]. A section length \( (\Delta L) \) could be calculated from Eq. (14). The calculation was performed until summation of the divided lengths equaled to the actual length of the capillary tube:

\[
\begin{align*}
\text{Condenser} & \quad \text{Liquid} & \quad \text{Two Phase} & \quad \text{Evaporator} \\
A_i & \quad P_{lhp} & \quad A_{f1} & \quad \text{Flash Point} & \quad P_{flash} \\
A_{w1} & \quad L_{sub} & \quad \Delta L & \quad P_{lhp} & \quad A_k \\
1 & \quad L_{sub} & \quad P_{flash} & \quad L & \quad A_k
\end{align*}
\]

Fig. 3. Schematic diagram of a capillary tube.
\[ \Delta L = \frac{A_{\text{cap}}(P_{r,1} - P_{r,0}) - m_r(V_{r,1} - V_{r,0})}{f_{\text{in}}m_r(V_{r,1} - V_{r,0})/D_{h}} \]  

(14)

4. Simulation algorithm

Fig. 4 shows the flow chart of the performance simulation program. The simulation consists of two iteration loops. All inputs were taken from specification of the components, inlet air conditions, compressor frequency and quantity of the filled oil. The calculation started by guessing the refrigerant mass flow rate, absolute temperature and pressure at compressor inlet. Consequently, the evaporating temperature and the superheating temperature were derived. Using the compressor model evaluated the escaped oil flow rate and the circulated oil concentration. After compressor operation started, the oil level was reduced. In the first calculation, the oil discharged rate was predicted by the reference level at the initial condition. The lost oil charge was estimated by the circulated oil concentration to multiply the refrigerant charge. Consequently, the new oil level was obtained and the new oil discharged rate was recalculated. This process was continuously carried out until the oil level was at the convergence.

The cooling capacity and the outlet air conditions were calculated from the evaporator model. The numerical approach was used to find the condensing pressure and the sub-cooled temperature by the capillary tube model. The heating capacity, the discharged temperature and the outlet air temperature were calculated from the condenser model. The compressor model was used to find the new value as if the mass flow rate from inner iteration loop was not equal to the initial guess, then the new value must be adjusted, and the inner loop must be started again. After the convergence of the inner iteration loop, the outer iteration loop was started for the calculation of the refrigerant conditions at the input of the compressor. In the case that the pressure and temperature of refrigerant vapor were not equal to the initial estimated values, then the evaporation pressure and superheated temperature must be adjusted, and the inner loop must be repeated again. The iterative calculation was continuously performed until inner and outer iteration loops were at the convergence.

The computer program was written corresponding to above procedure. Note that to operate the system with the optimum mass charge was the assumption of this model. The refrigerant mass charge could be evaluated from the mean density to multiply the inner volume within the whole heat exchanger. Moreover, the mass charge of lubricant oil could be estimated by the oil mass fraction and the quantity of oil remained in the compressor crank-case at approximately 70% at the normal frequency of 50 Hz.

5. Results and discussion

5.1. Experimental results

Fig. 5(a) shows that the mass flow rate of R-22 and the circulated oil concentration \( C_g \) of MO decreased with a reduction of the rotational speed of the compressor which was due to a change in the input electrical frequency. The proportion of the reducing liquid oil flow rate was higher than the vapor refrigerant flow rate. The circulated concentration with 1 l filled lubricant was higher than with 0.8 l charging due to the lower total static head. In this test, the oil separator was not 100% efficient, therefore, an amount of oil was still discharged to the condenser. The refrigerant flow rate was slightly diminished whilst increasing the oil concentration due to an increase in the viscosity of the mixture.

The system performances are shown in Fig. 6(a). When the compressor frequency was diminished, the cooling capacity decreased while the COP increased due to a reduction of the input power. Both performances decreased with an increase in the oil concentration. The oil fraction affected the COP more than the cooling capacity, due to
the input power being slightly increased. The influence of the lubricant may be neglected when the circulated mass fraction was less than approximately 0.6%. The best performances occurred with a charging of 0.8 l lubricant oil. Note that to prevent the failing of a compressor, the oil quantity must be carefully maintained at higher than 0.4 l by perpetual recharging after drainage.

Fig. 5 (b) shows the mass flow rate of R-407C and the circulated mass fraction of MO and POE lubricants which decreased with a reduction in the compressor frequency. The mass flow rate of the retrofitted refrigerant with MO was lower than that with POE lubricant because MO lubricant had higher liquid viscosity. Moreover, the influence of viscosity increased the pressure drop within the evaporator and condenser so the capacities dropped. Consequently, the mixture density at the compressor inlet diminished with the reduction in the refrigerant flow rate. A higher density of the POE oil resulted in the higher oil mass fraction and oil flow rate of POE lubricant than that of MO. The experiment regulated the rotational speed of the compressor to adjust the mass flow rate and the reduction in the mass flow rate diminished the vapor velocity. Furthermore the oil viscosity was mainly influenced by the operating temperature and it was reduced with an increase of the evaporating pressure. Therefore, the adequate vapor velocity to move the liquid oil was diminished. However, the mixture of R-407C with MO yielded a sufficient amount of the returned oil at a high frequency. When the vapor velocity of R-407C/MO at the return line was reduced from 6.96 m s\(^{-1}\) with a frequency of 50 Hz to 2.42 m s\(^{-1}\) with
30 Hz, the oil level reduced continuously. It could be concluded that this vapor velocity of R-407C was inadequately to move the MO back to the compressor.

The experimental work investigated the effect of an immiscible mixture as a working fluid on the system performance as shown in Fig. 6(b). The evaporating HTC of R-407C/POE lubricant mixture was higher than R-407C/MO due to its lower viscosity. The cooling capacity of the R-407C/POE system was subsequently higher, the lower vapor density at the suction line was obtained so the refrigerant flow rate was lower as shown in Fig. 5(b). The input power of the R-407C/MO system was higher due to the higher flow rate and the pressure ratio. The influence of the lubricant may be neglected when the circulated mass fraction was lower than approximately 0.4%. The best performances occurred with a charging of 0.8 l corresponding to the R-22 system. However, the long term replacement of the hydrofluorocarbon (HFC) and MO in the retrofitted system should be considered [6]. For this apparatus, the conventional air conditioning system using R-407C/MO had reliability of the performance with 480 h operation.

5.2. Model validation

The model was validated using experimental data. The comparison of the circulated oil fraction in the model and the experimented data was shown in Fig. 7 and the results agreed within ±10% accuracy. The model tended to over-predict at low frequencies. This was because the sub-model of compressor under-predict the total static head of lubricant.

Fig. 8 shows the refrigerant flow rate that the model tended to slightly under-predict at a low frequency, while it tended to slightly over-predict at higher frequencies, the discrepancies were within ±5% accuracy. Note that the refrigerant flow rate was a summation of the oil flow rate and the refrigerant discharge rate. The mixture of refrigerant and lubricant as the homogeneous fluid was assumed in this model, it corresponded to the miscible mixture. Hence, the mass flow rate was affected with the immiscible mixture. The discrepancy in the result was owing to the flow characteristics as pulse flowing at low frequency.

Fig. 9 shows the agreement of the different tested results with the simulated results of the COP. The results agreed within ±10%. For the R-407C/MO, the agreement was worst at lower frequencies where the model tended to over-predict the cooling COP. Due to the parameter such as polytropic index was adjusted as a constant value according to the normal operation.

Fig. 10 shows the comparison of the measured cooling capacity with the corresponding simulation at a higher frequency. The prediction tended to over-predict the capacity.

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Fig. 7. Comparison of experimental and simulated circulated oil concentration.

Fig. 8. Comparison of experimental and simulated refrigerant flow rate.

Fig. 9. Comparison of experimental and simulated COP.
at lower frequencies. For the actual R-407C/MO system, the lubricant was separated flowing along the tube. The oil layer affected the HTC by reducing the evaporating heat transfer, while the model considered the liquid refrigerant and oil as homogeneous fluid. Therefore, this system had a higher discrepancy.

6. Conclusions

This paper show the experimental study and the mathematical modeling of the inverter air conditioner. The effects of lubricant oil was investigated. The following are the conclusions that were summarized from the experimental and simulation results:

1. The discharged oil rate depended on the charged oil level in the compressor. Therefore, the quantity of charged lubricant affected the system performance. The oil quantity of 0.8 l was more effective than 1 l. To prevent the failing of compressor, however, the lost oil quantity of refrigerant drainage necessary was to be recharged, especially, for the R407C system.

2. The vapor velocity of R-407C was sufficient to entrain the non-volatile liquid MO at a compressor frequency of 35–50 Hz, while the poor return of the lubricant occurred at the lowest frequency. This study supported the decision to the use of MO with R-407C in the small inverter air conditioner. However, other considerations such as thermal stability, lubricity were not investigated. Further studies, the reliability of the variable capacity system performance using immiscible mixture should be addressed in the long term.

3. Unlike previous studies, this proposed model was capable of predicting the escaped oil flow rate. The simulated model extended its applicability to the existing retrofitted existing systems. The model prediction agreed with the measured values in a range of 40–50 Hz.

4. The predicted model obtained better results for the mixture of R-22/MO and R-407C/POE. The model could be used as the analytical tool to predict the operation prior to a retrofit of an existing system.

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