Modelling of an R-290/POE ISO 22 Variable Speed Air Conditioner System under SEER Conditions

Guilherme B. Ribeiro
Aerospace Science and Technology Department, Brazil, gbribeiro@ieav.cta.br

Jader Riso Barbosa Jr.
Federal University of Santa Catarina, Brazil, jrb@polo.ufsc.br

Follow this and additional works at: http://docs.lib.purdue.edu/iracc

http://docs.lib.purdue.edu/iracc/1561

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
Modelling of an R-290/POE ISO 22 Variable Speed Air Conditioner System under SEER Conditions

Guilherme B. Ribeiro¹*, Jader R. BARBOSA JR.²

¹Aerospace Science and Technology Department, Nuclear Energy Division, São José dos Campos, São Paulo, Brazil
gbribeiro@ieav.cta.br

²Federal University of Santa Catarina, Mechanical Engineering Department, Florianópolis, Santa Catarina, Brazil
jrb@polo.ufsc.br

* Corresponding Author

ABSTRACT

Air-conditioning applications using propane (R-290) have several environmental and thermodynamic advantages over more commonly used refrigerants, such as R-410A and R-22. This paper presents the development of a mathematical model for variable capacity air conditioning systems that use R-290/POE ISO 22 as refrigerant/lubricant. The thermodynamic performance of the refrigeration system is evaluated in terms of the SEER (Seasonal Energy Efficiency Ratio). The thermodynamic properties of the refrigerant/lubricant mixture were obtained from a departure-function approach using the Peng-Robinson equation of state. The effect of the oil on the condenser and evaporator heat transfer coefficients and pressure drops was also taken into account. Sub-models were developed for each component of the air conditioning system, including the connecting lines and the scroll compressor.

1. INTRODUCTION

In air conditioning (AC) systems, a significant fraction of the compressor lubricating oil flows through the system components during steady-state operation. Several studies point out that the oil circulation ratio (OCR) depends principally on the compressor speed (Navarro et al., 2005) and significantly reduces the system performance (McMullan et al., 1988; Lottin et al., 2003; Winandy and Cuevas, 2003; Sarntichartsak et al., 2006; Galván et al., 2011).

Empirical approaches as reviewed by Thome (1995) are widely applied in the prediction of oil-refrigerant thermophysical properties. Marcelino Neto and Barbosa (2013) advanced a departure function approach to calculate the thermodynamic properties of R-600a/AB ISO 5. A cycle analysis of a household refrigeration system was also performed.

The present study consists of modelling an split-type AC system equipped with a scroll compressor. The thermodynamic properties of the R-290/POE ISO 22 mixture were calculated from a departure-function approach using the Peng and Robinson (1976) equation of state. Additionally, the oil effects on the heat transfer coefficients were also taken into account. The performance of the AC system was evaluated as a function of the OCR, and conditions established in the AHRI 210/240 (2008) standard, i.e., the ambient air temperature and humidity and the compressor speed, were used to calculate the SEER (Seasonal Energy Efficiency Ratio).


2. MATHEMATICAL MODEL

2.1 Mixture Thermophysical Properties

The thermodynamic properties were calculated via the departure function approach (Elliot and Lira, 1990). Thus, the variation of a generic thermodynamic property is obtained as:

\[ \phi_{\text{fin}} - \phi_{\text{ini}} = (\phi_{\text{fin}} - \phi_{\text{fin}}^{\text{id}}) + (\phi_{\text{fin}}^{\text{id}} - \phi_{\text{ini}}^{\text{id}}) - (\phi_{\text{ini}} - \phi_{\text{ini}}^{\text{id}}) \]  

(1)

where the superscript \( \text{id} \) stands for the ideal gas model and the first and third terms are the \( \phi \) departure functions at the final and initial states, respectively. The Peng and Robinson (1976) enthalpy and entropy departure functions were obtained from Edmister and Lee (1984). The International Institute of Refrigeration (IIR) reference state was adopted. Specific properties of a two-phase liquid-vapor mixture are calculated by:

\[ \phi = X\phi_v + (1 - X)\phi_l \]  

(2)

where \( X \) is the vapor mass quality of the R-290/POE ISO 22, and \( \phi_v \) and \( \phi_l \) are the thermodynamic properties at the saturated vapor and liquid states, respectively. According to ASHRAE (2010), the oil vapour pressure can be considered negligible when compared with pressures commonly encountered in refrigeration systems. Thus, it is assumed that the refrigerant mass fraction in the vapor phase is equal to unity. Figure 1 shows a sketch of the vapor-liquid equilibrium diagram, where the dew point line was ignored due to the small vapor pressure of the lubricating oil. Applying the lever rule, the vapor quality \( X \) is obtained as:

\[ X = \frac{AB}{BC} = \frac{z_{\text{ref}} - x_{\text{ref}}}{1 - x_{\text{ref}}} \]  

(3)

where \( z_{\text{ref}} \) is the overall refrigerant mass fraction and \( x_{\text{ref}} \) is the refrigerant mass fraction in the liquid phase (solubility). Considering \( z_{\text{ref}} = 1 - OCR \), the mixture vapor quality is calculated directly as function of OCR and \( x_{\text{ref}} \) as follows:

\[ X = 1 - \frac{OCR}{1 - x_{\text{ref}}} \]  

(4)

Figure 1. Vapor-liquid equilibrium of an oil-refrigerant mixture.

Figure 2. Vapor-liquid equilibrium of R-290/POE ISO 22 mixture.
The mass fraction in the liquid phase was calculated as function of bubble point and temperature via the Peng and Robinson (1976) equation of state with the classical quadratic mixing rule. The binary interaction parameter was determined using solubility curves shown in Fig. 2 and available from the oil supplier.

The mixture dynamic viscosity, thermal conductivity and specific heat capacity were determined from the empirical correlations of Kedzierski and Kaul (1993), Filippov (1968) and Jensen and Jackman (1984), respectively. The lubricating oil critical pressure and temperature, acentric factor and specific heat capacity were calculated via the group contribution methods of Constantinou and Gani (1994). The remaining oil thermophysical properties were obtained directly from the manufacturer. The properties were evaluated as function of temperature and fitted using a fourth-order polynomial.

2.2 Scroll Compressor

The refrigerant mass flow rate $\dot{m}$ and compressor power input $W_{\text{comp}}$ were calculated as suggested by Gosney (1982) as follows:

$$\dot{m} = \frac{\eta_v V_{\text{comp}} N_{\text{comp}}}{v_1}$$

$$W_{\text{comp}} = \frac{\dot{m}(h_2 - h_1)}{\eta_g}$$

where $v_1$ is the mixture specific volume at the compressor inlet. The scroll compressor volumetric and overall efficiencies ($\eta_v$ and $\eta_g$) were obtained from the lumped model of Pereira (Pereira and Deschamps, 2010; Diniz et al., 2012; Pereira, 2012), which contains specific models for the gas leakage and heat transfer in the scroll compression chambers. The compressor efficiencies were calculated as function of compressor speed, $N_{\text{comp}}$, suction temperature and suction and discharge pressures. The mixture discharge temperature is obtained from the discharge enthalpy, $h_2$, from a simple energy balance in the compressor as follows:

$$h_2 = h_1 + \frac{W_{\text{comp}}}{\dot{m}}$$

Heat losses through the compressor shell were neglected since compressors are generally thermally and acoustically insulated inside AC condensing units.

2.3 Condenser and Evaporator

The louvered fin condenser was modelled in zones corresponding to thermodynamic state of the flowing fluid (i.e. liquid-vapor or sub-cooled mixture). Since the geometry of the condenser is fixed, the heat transfer area for each zone depends on the heat transfer characteristics. Considering a simple energy balance on the mixture side to find the sub-cooling heat transfer rate, $\dot{Q}_{c,\text{sc}}$, and using the heat exchanger effectiveness to determine the overall conductance $UA_{c,\text{sc}}$ (Kays and London, 1984), the sub-cooled region length is given by:

$$L_{c,\text{sc}} = UA_{c,\text{sc}} \left[ \frac{1}{\eta_f h_{\text{air}} \pi D_e (A_{\text{sc}}/A_c)} + UA_{c,\text{sc}} \left[ \frac{1}{\eta_{g_{\text{sc}}} \pi D_t} \right] + UA_{c,\text{sc}} \left[ \frac{\ln(D_e/D_t)}{2\pi k_{\text{sc}}} \right] \right]$$

where the terms on the right correspond to the heat transfer at the air side, mixture side and conduction through the tube wall, respectively. The Wang et al. (1999) correlation for louver fins was used to estimate $h_{\text{air}}$, and the correlation of Gnielinski (1976) was used to compute $h_{\text{sc}}$. Since, $L_c = L_{c,\text{tp}} + L_{c,\text{sc}}$ (fixed geometry), the two-phase region thermal conductance and heat transfer rate are given by:

$$\frac{1}{UA_{c,\text{tp}}} = \frac{1}{\eta_f h_{\text{air}} \pi D_e (A_{\text{sc}}/A_c) L_{\text{tp}}} + \frac{1}{\eta_{f_{\text{tp}}} \pi D_e L_{\text{tp}}} + \frac{\ln(\tau_e - \tau_l)}{2\pi k_{\text{co}} L_{\text{tp}}}$$

16th International Refrigeration and Air Conditioning Conference at Purdue, July 11-14, 2016
The total condenser heat transfer rate is the sum of the heat transfer rates in the sub-cooled and two-phase zones, \( \dot{Q}_{c,sc} \) and \( \dot{Q}_{c,tp} \). The two-phase heat transfer coefficient, \( h_{tp} \), was calculated through the Dobson and Chato (1998) correlation combined with the Bassi and Bansal (2003) relationship to account for lubricating oil effects. The condenser model was implemented so that the following relationship was used as a convergence criterion for the condensing pressure:

\[
|\dot{Q}_{c,mix} - \dot{Q}_{c,sc} - \dot{Q}_{c,tp}| > 1 \text{ W}
\]

where the heat transfer rate \( \dot{Q}_{c,mix} \) is obtained by the energy balance on the mixture side, applying the difference of enthalpy at condenser inlet and outlet. The calculation procedure to determine the heat transfer rate in the evaporator is similar, but the evaporator is characterized by only having a two-phase zone, as the lubricating oil does not evaporate. The internal heat transfer coefficient (mixture side) was calculated using the correlations of Wattelet (1994) and Schlager et al. (1990). The heat transfer rate in the evaporator (cooling load), \( \dot{Q}_e \), is a sum of three terms as follows:

\[
\dot{Q}_e = \dot{Q}_{sen} + \dot{Q}_{lat} + W_e
\]

where \( \dot{Q}_{sen} \) is a sensible heat transfer rate calculated based on the product of an overall thermal conductance, \( U_Ae \), and a log-mean temperature difference for the evaporator. \( \dot{Q}_{lat} \) and \( W_e \) are the evaporator air-side latent heat transfer rate and fan power input, respectively. The latent heat was estimated based on the mass transfer coefficient of the evaporator and humidity ratio difference between the inlet and outlet of the evaporator:

\[
\dot{Q}_{lat} = h_{air,m}(w_{air,i} - w_{air,e})h_{air,lv}
\]

\[
h_{air,m} = \frac{h_{air}}{Le^{2/3}c_p,air}
\]

where a unit Lewis number, \( Le \), has been assumed. The two-phase pressure drops in the heat exchangers was calculated using a combination of the correlations by Müller-Steinhagen and Heck (1986) and Zürcher et al. (1998) to take mixture effects into account.

2.4 Expansion Device

The solution algorithm assumed fixed values for the apparent superheating and sub-cooling degrees (taking the pure R-290 as a reference) at the outlet of the evaporator and condenser, respectively. Therefore, it has been assumed that the outlet conditions of both heat exchangers are considered known as

\[
T_5 = T_{sat}(P_e) + \Delta T_{sh}
\]

\[
T_3 = T_{sat}(P_i) - \Delta T_{sc}
\]

where the term \( T_{sat} \) stands for saturation temperature of pure R-290. The apparent superheating, \( \Delta T_{sh} \), and sub-cooling, \( \Delta T_{sc} \), degrees used in this study was defined in 2 K. Similar approaches for pure refrigerants have already been used in previous studies (Gonçalves et al., 2008; Hermes et al., 2009; Negrão and Hermes, 2011) to avoid significant computational efforts with complex modelling of expansion devices.

2.5 Connecting Lines

The compressor suction and discharge lines and the evaporator inlet line were modelled as simple heat exchangers (Kays and London, 1984). The generic thermal conductance of a connecting line can be written as:
In Eq. (17), the external heat transfer coefficient $h_{\text{air,1}}$ was obtained from the Churchill and Chu (1975) correlation for natural convection. The single-phase mixture heat transfer coefficient $h_{\text{mix,1}}$ was calculated using the Gnielinski (1976) correlation, except for the evaporator inlet line, where the correlations of Wattelet (1994) and Schlager et al. (1990) were applied. The last item of eq. (17) is related to the heat transfer across the insulating material which was installed along the evaporator connecting line.

### 2.6 Fans

The air flow rate across each heat exchanger is obtained iteratively, based on the fan characteristic curve and the air pressure drop. The fan efficiency and the power input are calculated in the same manner, and the latter is later incorporated into the system total power input. The characteristic curves provide the hydraulic performance and efficiency behavior of the single-speed fans and were represented by simple algebraic equations whose coefficients were supplied by the manufacturer. The volumetric air flow rates of the evaporator and condenser fans are given by

$$V_e = A_1 + B_1 \cdot \Delta P_{\text{air,e}} + C_1 \cdot (\Delta P_{\text{air,e}})^2$$

$$V_c = D_1 + E_1 \cdot \Delta P_{\text{air,c}} + F_1 \cdot (\Delta P_{\text{air,c}})^2$$

where $\Delta P_{\text{air}}$ is the air pressure drop across the heat exchangers. The electrical efficiencies of the two fans were written in terms of the volumetric air flow rate as follows

$$\eta_e = A_2 + B_2 \cdot V_e + C_2 \cdot (V_e)^2$$

$$\eta_c = D_2 + E_2 \cdot V_c + F_2 \cdot (V_c)^2$$

With the fan electrical efficiency, the electrical power consumptions are directly obtained from

$$W_e = \frac{\Delta P_{\text{air,e}} V_e}{\eta_e}$$

$$W_c = \frac{\Delta P_{\text{air,c}} V_c}{\eta_c}$$

### 3. OPERATING CONDITIONS

In order to determine the geometric characteristics of heat exchangers and connecting lines, a split-system model Consul CBV09-CBY09 (manufactured by Whirlpool) was simulated. The scroll compressor was simulated with a speed ranging from 50 to 150 Hz. As mentioned above, the SEER was quantified based on the five conditions of AHRI 210/240 (2008). Furthermore, the cooling capacity and total power input of the air conditioner at the cooling capacity rating condition (i.e., 27/35°C internal and external ambient temperatures, for a frequency of 150 Hz) was evaluated separately. The total AC power input, $W_{\text{sys}}$, was considered as the sum of the power input related to the compressor $W_{\text{comp}}$ and heat exchangers fans.

### 4. RESULTS

Values of OCR ranging from 1 to 7% were simulated, which are well above the levels usually found in small capacity air conditioners. As can be seen in Fig. 3, a strong degradation of the SEER is observed with increasing OCR. A change in concentration from 1 to 3% is responsible for a SEER decrease from 14 to 11.5 BTU/h/W. Furthermore, the SEER degradation progressively increases for higher levels of OCR.
A similar effect is verified in Fig. 4, which shows the system cooling capacity \( \dot{Q}_c \) as a function of the OCR. It is important to emphasize that these data were extracted from the cooling capacity rating condition proposed on AHRI 210/240 (2008). Similarly to Fig. 3, a strong reduction of \( \dot{Q}_c \) is observed as the OCR increases. Despite the increase in the mixture mass flow rate, the increase in the OCR reduces the specific refrigerating effect, which has a greater impact on the system cooling capacity.

![Figure 3. Performance factor SEER as a function of the oil circulation ratio.](image)

![Figure 4. Cooling capacity as a function of the oil circulation ratio.](image)

Fig. 5 shows the influence of the OCR on the AC system power input, \( \dot{W}_{dis} \). As can be seen, the total power input increases rapidly with the increasing OCR. In addition to the increase of the mixture mass flow rate, augmenting the OCR causes the specific compression work (enthalpy difference) to increase. More specifically, an increase of 80 W is obtained in the AC power input when the OCR shifts from 1 to 3%. This is in general agreement with the mixture pressure-enthalpy diagrams and cycle analysis Marcelino and Barbosa (2013). Overall, the present results demonstrate the importance of an efficient oil pumping system in the compressor, allowing for good lubrication of the moving parts without generating a significant oil circulation through the other components of the refrigeration system.
5. CONCLUSIONS

A method to calculate the thermodynamic properties of refrigerant-oil mixtures based on the theory of departure functions combined with the Peng and Robinson (1976) equation of state was used to evaluate the influence of POE ISO 22 on the overall performance of an air-conditioner system operating with R-290. The system uses a scroll compressor and the expansion valve was modelled as suggested by Gonçalves et al. (2008). Heat exchangers and all connecting lines were modelled according to the heat exchanger effectiveness approach.

The effect of the OCR on the system performance was such that, although the mixture mass flow rate increases with the OCR, the associated reduction of the refrigerant effect resulted in lower cooling capacities for higher values of the OCR. Moreover, the increase in the specific compression work also contributes to the degradation of the AC system power input and, consequently, to the SEER performance factor. The results indicated the need for a compressor lubricating system that delivers the smallest possible amount of oil in the discharge line.

NOMENCLATURE

\begin{align*}
A & \quad \text{Area} \quad (\text{m}^2) \quad \dot{Q} & \quad \text{Heat transfer rate} \quad (\text{W}) \\
c_p & \quad \text{Specific heat} \quad (\text{J/kg K}) \quad r & \quad \text{Radius} \quad (\text{m}) \\
D & \quad \text{Diameter} \quad (\text{m}) \quad T & \quad \text{Temperature} \quad (\text{K}) \\
h & \quad \text{Enthalpy} \quad (\text{J/kg}) \quad U_A & \quad \text{Thermal conductance} \quad (\text{W/K}) \\
h & \quad \text{Heat transfer coefficient} \quad (\text{W/m}^2 \text{ K}) \quad v & \quad \text{Specific volume} \quad (\text{m}^3/\text{kg}) \\
k & \quad \text{Thermal conductivity} \quad (\text{W/m K}) \quad V & \quad \text{Compressor displacement} \quad (\text{m}^3) \\
L & \quad \text{Length} \quad (\text{m}) \quad \dot{V} & \quad \text{Volumetric flow rate} \quad (\text{m}^3/\text{s}) \\
Le & \quad \text{Lewis number} \quad (--) \quad w & \quad \text{Humidity ratio} \quad (\text{kg}_w/\text{kg}_{\text{air}}) \\
m & \quad \text{Mass flow rate} \quad (\text{kg/s}) \quad \dot{W} & \quad \text{Work} \quad (\text{W}) \\
N & \quad \text{Compressor speed} \quad (\text{Hz}) \quad x & \quad \text{Mass fraction in the liquid phase} \quad (--) \\
OCR & \quad \text{Oil Circulation Ratio} \quad (--) \quad X & \quad \text{Mixture quality} \quad (--) \\
P & \quad \text{Pressure} \quad (\text{Pa}) \quad z & \quad \text{Overall mass fraction} \quad (--) \\
\Delta & \quad \text{Difference} \quad \eta & \quad \text{Efficiency} \\
\pi & \quad \text{Pi number} \quad \phi & \quad \text{Generic variable}
\end{align*}

Figure 5. System power input as a function of the oil circulation ratio.
REFERENCES


ACKNOWLEDGEMENT

The authors would like to thank Whirlpool and Embraco-Whirlpool Compressor unit for the technical support.