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APPLICATION OF PERIPHERAL FINS IN A VARIABLE-SPEED AIR CONDITIONER

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Abstract. *In air-conditioning applications, the used of hydrocarbons refrigerants as Propane (R-290) has growth extensively in the last years, due to more restricted regulations regarding the power consumption and the environmental. Hence, this paper proposes a numerical evaluation of a condenser that makes use of peripheral fins on a split-type air-conditioning system. The heat exchanger tube length was part of the evaluation, while three other types of fins were also simulated for comparison purposes: plain, wavy and louvered fins. In order to assess the system performance, the SEER rating computed in accordance with the AHRI 210/240 was included. Additionally, the system cooling capacity, the condenser thermal conductance and the system refrigerant charge were part of the analysis. As a manner to enhance the numerical evaluation, the Oil Circulation Ratio (OCR) of the system was estimated and the presence of the lubricating oil POE ISO 22 mixed with R-290 was taken into account on the thermophysical properties and correlations. Results have shown that for the simulated range, heat exchangers with peripheral fins provided higher thermal conductance (UA), SEER and cooling capacities than other types of fins. When the available heat transfer area is constrained in the peripheral fin condenser, cooling capacities similar to the other fins were achieved, with a strong decrease of the refrigerant charge.*

Keywords: *peripheral fin, air conditioning, SEER, variable-speed.*

1. INTRODUCTION

With the increasing of quality of life, especially in developing countries, the use of air-conditioning (AC) equipment has grown exponentially, as well as the required power generation (Issac and Van Vuuren, 2009). As a manner to promote the rational use of refrigerating equipment, the application of compressor inverters allows the cooling capacity modulation via compressor speed. As a result, indoor temperature fluctuations are reduced, as well as the AC power consumption (Li *et al*., 2014).

As a response to environmental concerns, the use of natural refrigerant fluids has been applied extensively most due to their thermodynamic characteristics that are suitable for typical refrigerating equipment. Accordingly, studies and applications that consider the fluid Propane (R-290) as the low-GWP (Global Warming Potential) refrigerant can be found in the open literature (Devotta and Padalkar, 2005; Padalkar *et al*., 2014). On the other hand, safety issues related to R-290 flammability makes the refrigerant charge mitigation a critical aspect when it comes to AC applications.

Additionally, the presence of the lubricating oil along the AC components plays a harmful role in the system performance (McMullan *et al*., 1988; Lottin *et al*. 2013; Sarntichartsak *et al*., 2006), and due to this reason, this aspect should be considered at the design of such system, especially when high Oil Circulation Ratio (OCR) is found. Furthermore, due to the regulation published by the European Union which schedules the application of low-GWP refrigerants (EU regulation No 517, 2014), assertive heat transfer enhancement techniques for heat exchangers focused on the AC final performance will become an essential need for the refrigeration industry.

Thus, this study proposes the analysis of the performance of a split-type AC system where the condenser applies a novel type of fin, called peripheral fin (Pussoli, 2010), as shown in Fig. 1. As evidenced by Pussoli (2010), this type of fin provided high heat transfer per volume ratio. However, a significant air pressure drop was reported in experimental tests. For a fixed cooling capacity, the use of such fin in a AC system can contribute to the reduction of the condenser area and, consequently, volume. Therefore, due to this reason, a decrease of refrigerant charge can be achieved using this heat transfer enhancement technique.

The influence of the condenser tube length in the system performance is also evaluated. Also, for comparison purposes, three other types of fins were included in the assessment: plain, wavy and louvered fins. In this study, a variable-speed AC which works with R-290 was modeled and the presence of the POE ISO 22 was incorporated at the thermodynamic properties computation, as well as for the heat transfer coefficient and friction factor. Beside the system cooling capacity, the condenser thermal conductance, the system refrigerant charge and the SEER rating defined by AHRI 210/240 (2008) are part of the numerical analysis.

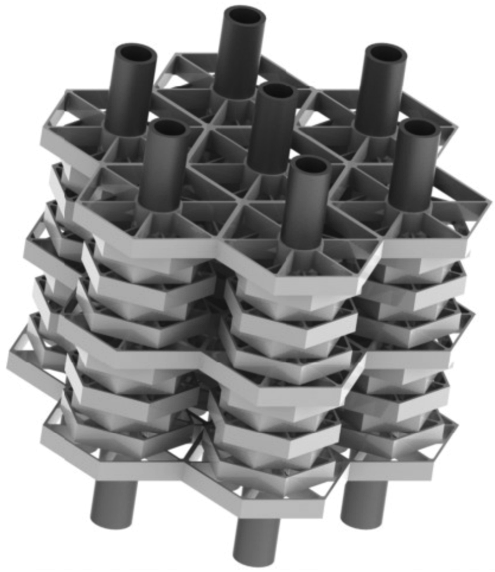

Figure 1. Schematic representation of a tube array with peripheral fins (Pussoli, 2010).

2. MATHEMATICAL MODEL

In order to evaluate the influence of the evaporator dimensional parameters, each AC component was modeled and implemented via the ESS (Engineering Equation Solver) software. The description of mathematical modeling and its validation were described by previous studies (Ribeiro and Barbosa, 2016, Ribeiro and Barbosa, 2018). For the sake of conciseness only essential information will be repeated here.

Thermodynamic properties of the R-290/POE ISO 22 mixture were calculated using the departure function method (Edmister and Lee, 1984; Elliot and Lira, 1990) combined with the Peng-Robinson (1976) equation of state. The oil mass flow rate was estimated based on the system operating conditions, and the presence of the lubricant oil on heat transfer and pressure drop was taken into account in the modeling. The total system mass flow rate $\dot{m}$ and the scroll compressor power input $\dot{W}_{comp}$ are obtained as

$$\dot{m} = \frac{\eta_v V_{comp} N_{comp}}{v_1} \tag{1}$$

and

$$\dot{W}_{comp} = \frac{\dot{m}\left(h_{2,s} - h_1\right)}{\eta_g} \tag{2}$$

where compressor volumetric and global efficiencies (η_v and η_g) were obtained through the study described by Pereira (2012). It is assumed that the heat losses through the compressor shell are neglected in this study, since the compressor is thermally and acoustically insulated by inside the AC condensing unit. Thus, the compressor discharge enthalpy $h_{comp,o}$ is obtained via a simple energy balance along the compressor shell is applied as follows

$$h_{comp,o} = h_{comp,i} + \frac{\dot{W}_{comp}}{\dot{m}} \tag{3}$$

The peripheral fin modeling was performed based on the porous media theory. For the computation of the heat transfer coefficient from the air-side, the correlation proposed by Handley and Heggs (1968) was used. For the evaluation of the air pressure drop, the Montillet et al. (2007) correlation was employed. The heat exchangers are modeled based on thermodynamic zones, where each zone corresponds to thermodynamic state (i.e. liquid-vapor or sub-cooled). Applying an energy balance on the working fluid side to find the sub-cooled heat transfer $\dot{Q}_{c,sc}$, and using the heat exchanger effectiveness for $UA_{c,sc}$ (Kays and London, 1984), the condenser length related to the sub-cooled zone is computed as

$$L_{c,sc} = UA_{c,sc}\left[\frac{1}{\eta_f h_{air}\pi D_e(A_{sc}/A_c)}\right]+UA_{c,sc}\left[\frac{1}{h_{sc}\pi D_i}\right]+UA_{c,sc}\left[\frac{ln(D_e/D_i)}{2\pi k_{co}}\right] \quad (4)$$

where the first, second and third terms are referenced to the heat transfer at the air side, mixture side and conduction through the copper tube, respectively. In order to obtain h_{air}, correlations of Wang et al. (2000), Wang et al. (1999a) and Wang et al. (1999b), for plain, wavy and louver fins, respectively, were used. For the mixture heat transfer coefficient h_{sc}, the correlation of Gnielinski (1976) were applied. Since, $L_c = L_{c,tp} + L_{c,sc}$, the condenser thermal conductance $UA_{c,tp}$ and the heat transfer rate $\dot{Q}_{c,tp}$, both related to the condensing zone, are represented as

$$\frac{1}{UA_{c,tp}} = \frac{1}{\eta_f\, h_{air}\pi D_e(A_{sc}/A_c)\, L_{tp}} + \frac{1}{\eta_f h_{tp}\pi D_i L_{tp}} + \frac{ln(r_e - r_i)}{2\pi k_{co} L_{tp}} \quad (5)$$

and

$$\dot{Q}_{c,tp} = UA_{c,tp}\, \Delta T_{c,lm} \quad (6)$$

To obtain of the condensing heat transfer coefficient h_{tp}, the correlation of Dobson and Chato (1998) was applied, as well as the Bassi and Bansal (2003) expression which takes into account the lubricating oil effect. During the iterative procedure, the working fluid pressure is considered converged when the heat transfer rate criterion is achieved as follows

$$\left|\dot{Q}_{c,mix} - \dot{Q}_{c,sc} + \dot{Q}_{c,tp}\right| > 1 \quad (7)$$

where $\dot{Q}_{c,mix}$ is the condenser heat transfer rate regarding the mixture side. The evaporator is characterized by presence of only the two-phase zone, so it can be concluded that $L_e = L_{e,tp}$. Furthermore, the numerical procedure used to compute $\dot{Q}_{e,tp}$ is similar of the aforementioned equations. The evaporating heat transfer coefficient was obtained as proposed by Wattelet (1994) and Schlager et al. (1990). The refrigerating system capacity $\dot{Q}_e$ is calculated as follows

$$\dot{Q}_e = \dot{Q}_{e,tp} + \dot{Q}_{lat} + \dot{W}_e \quad (8)$$

where $\dot{Q}_{lat}$ and $\dot{W}_e$ are the evaporator latent heat and fan power input, respectively. The latent heat is driven by the difference of absolute humidity w and was estimated based on the overall mass conductance as follows

$$\dot{Q}_{lat} = UA_m\left(w_{air,i} - w_{air,e}\right)h_{air,lv} \quad (9)$$

and

$$UA_m = \frac{UA_e}{Le^{2/3} c_{p,air}} \quad (10)$$

where the Lewis number Le is considered as a unitary value. The correlation of Müller-Steinhagen and Heck (1986) combined with the expression of Zürcher et al. (1998) was applied in order to estimate the two-phase flow pressure drop.

At this study, it was considered a fixed degree of superheating and sub-cooling when compared to pure R-290. Therefore, it has been assumed that the outlet conditions of both heat exchangers are considered known as

$$P_e = P_{sat}\left(T_{e,o} - \Delta T_{sh}\right) \quad (11)$$

$$P_c = P_{sat}\left(T_{c,o} + \Delta T_{sc}\right) \quad (12)$$

where the term P_{sat} stands for saturation pressure of pure R-290. Terms $T_{e,o}$ and $T_{c,o}$ denote the evaporator and condenser outlet temperature, respectively. The apparent level of superheating ΔT_{sh} and sub-cooling ΔT_{sc} arbitrated in this study was defined in 2 K. This approach has already been used by several previous studies (Negrão and Hermes,

2011) and it avoids significant computational efforts with complex modelling of expansion devices. The discharge line, the evaporator inlet line and the suction line were modelled as simple heat exchangers and due to this reason, the thermal conductance of a connecting line can be represented as

$$\frac{1}{UA_l} = \frac{1}{h_{air,l}\pi D_l\, L_l} + \frac{1}{h_{mix,l}\pi D_{i,l} L_l} + \frac{ln(r_{e,l} - r_{i,l})}{2\pi k_{co} L_l} \tag{13}$$

where air heat transfer coefficient $h_{air,l}$ was computed as proposed by Churchill and Chu (1975), considering a natural convection heat transfer mechanism. With the exception of the evaporator inlet line which used the same evaporator heat transfer expressions, all other connecting lines applied the Gnielinski (1976) correlation to obtain the mixture heat transfer coefficient $h_{mix,l}$. The mixture charge along the AC system was computed separately for each component and connecting line, considering the internal volume and density. For components where two-phase flow exists, the approach suggested by Ma *et al*. (2009) was adopted. The void fraction model of Premoli *et al*. (1971) was applied for annular flow, whereas the Taitel and Barnea (1990) modeling was considered for intermittent flow. Also, the two-phase flow pattern map for horizontal tubes of Kattan et al. (1995) was used for the determination of the mixture charge.

The modeling was applied to the AC system manufactured by Whirlpool Latin America, model Consul CBV-09-CBY09. The evaporator dimensions, fans and components connecting lines applied in this study were referred to this refrigerating system. The variable-capacity scroll compressor applied here was assembled by Embraco – Whirlpool Compressor Unit and has its speed ranging from 50 to 150 Hz. The fan curves of each heat exchanger were incorporated in the numerical model, in order to emulate the hydraulic matching between the heat exchanger and its fan. The descriptions of the main dimensions of the AC components, as well as the fan representative coefficients, are summarized in Ribeiro and Barbosa (2016). The SEER computation was performed according to the AHRI 210/240 (2008) standard, where five different environmental conditions and compressor speed were applied. Table 1 shows the AC conditions considered in this study for the SEER calculation. In table 1, condition A2 was stipulated as the rating condition for the AC performance.

Table 1. Conditions for SEER calculation.

Condition	Indoor – Dry / Wet Bulb Temperature	Outdoor - Temperature	Speed
A2	26.67/19.44 °C	35.00 °C	150 Hz
B2	26.67/19.44 °C	27.78 °C	150 Hz
Ev	26.67/19.44 °C	30.55 °C	83.33 Hz
B1	26.67/19.44 °C	27.78 °C	50 Hz
F1	26.67/19.44 °C	19.44 °C	50 Hz

3. RESULTS

Fig. 2 and 3 display the SEER rating and AC cooling capacity for different condenser tube lengths, with different types of fins. For these simulation cases, the number of row in any direction is kept constant, as well as the fin density. Thus, the variation of the condenser frontal area only depends on the tube length. The AC cooling capacity was computed for the rating condition (A2).

As can be seen, the increase of tube length promoted higher SEER parameter for all types of fins. This fact is easily explained by the increase of the total heat transfer area which is resulted from longer tube lengths, despite the decrease of the air average velocity across the evaporator. As the net effect, the area increasing had more influence on the condenser thermal conductance than the air heat transfer coefficient. Also, the lack of condenser area, as consequence of lower tube lengths, increases the importance of heat transfer enhancement techniques. Thus, the use of louver fin is reasonable for the lower range of tube length. On the other hand, for a higher tube length range, plain fins promoted higher SEER rating and AC cooling capacity than wavy and louver fins. The large available heat transfer area turned heat transfer enhancement techniques, such as the louver fin, less effective.

Furthermore, as shown in Fig. 1 and 2, the condenser with peripheral fins resulted in higher SEER rating and cooling capacity $\dot{Q}_e$, when compared to other fins. The high available heat transfer area in conjunction with high air heat transfer coefficient makes the use of peripheral fins a promising solution for AC performance enhancement. It is important to mention that the AC condenser are hydraulic coupled with a fan in the numerical analysis. Therefore, the increase of air pressure drop promoted by peripheral fins did not result in an extensive degradation of the supplied air flow rate.

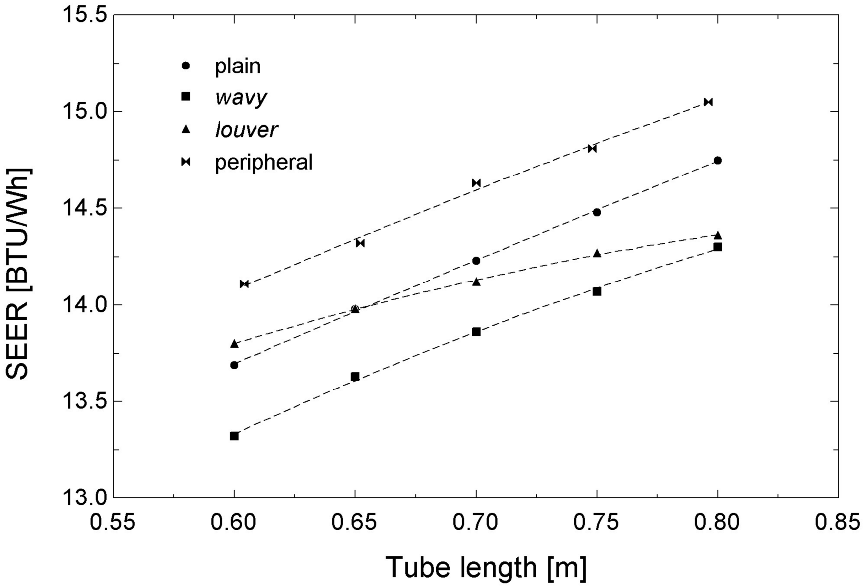

Figure 2. SEER rating as function of the condenser tube length.

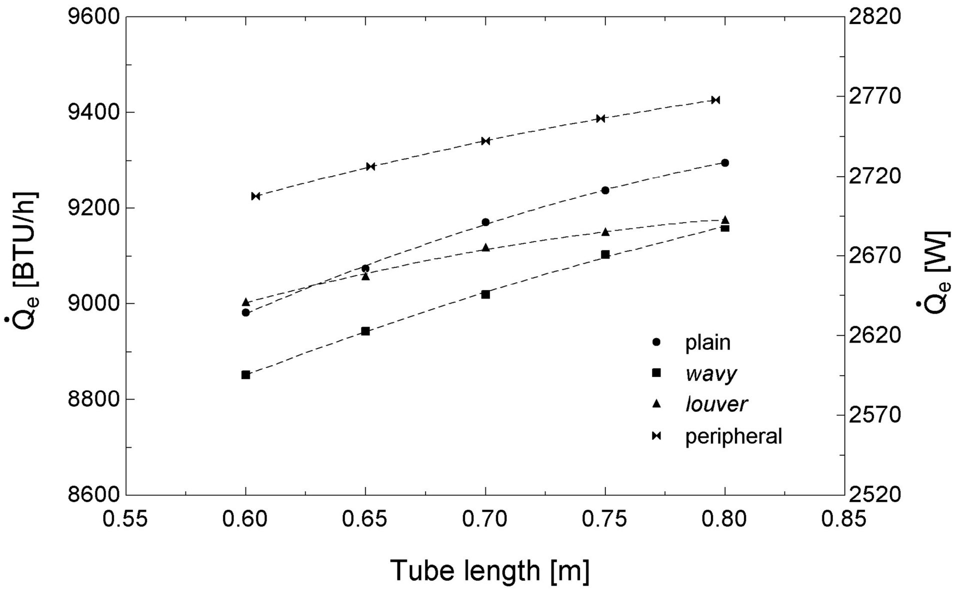

Figure 3. Cooling capacity as function of the condenser length.

Fig. 4 shows the condenser thermal conductance as function of the tube length and fin type. As mentioned before, the condenser with peripheral fin provided the highest UA_c, for the tube length range. These results justify the AC performance enhancement promoted by the use of peripheral fins. Moreover, this conclusion provides an opportunity to reduce the volume of the peripheral fin condenser, in order to mitigate the total charge of R-290/POE ISO 22.

As mentioned before, peripheral fin condenser achieved the highest condenser thermal conductance UA_c, for all types of fins. Therefore, the transversal number of tubes was decreased for the peripheral fin condenser, as a manner to match the cooling capacity obtained by the other condenser configurations. This modified peripheral fin condenser displayed in Fig. 5 has lower available heat transfer area, and consequently, internal volume. As a result, this modified configuration promoted lower mixture charges than other condensers, as shown in Fig. 5. For natural and flammable refrigerants like R-290, the use of a peripheral fin condenser may be a promising solution to diminish refrigerant charge and meet safety requirements concerning residential AC applications.

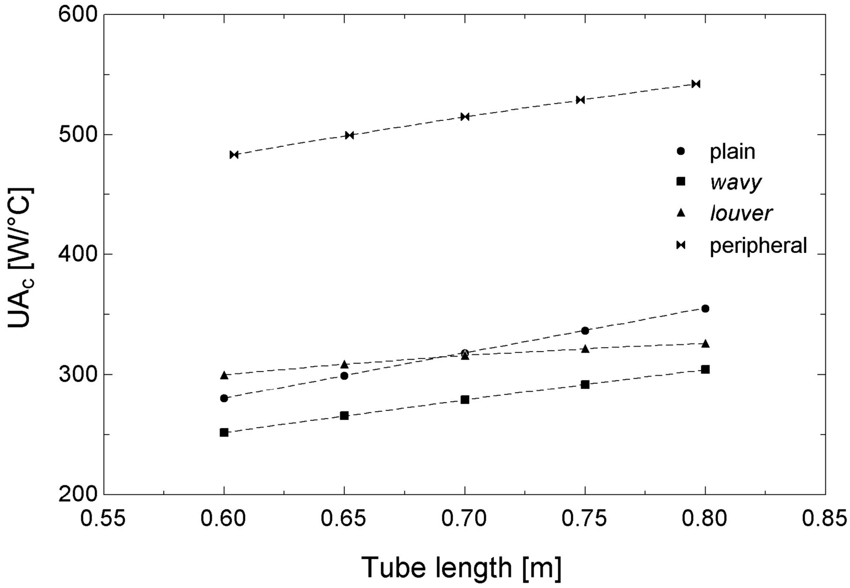

Figure 4. Condenser thermal conductance as function of the condenser length.

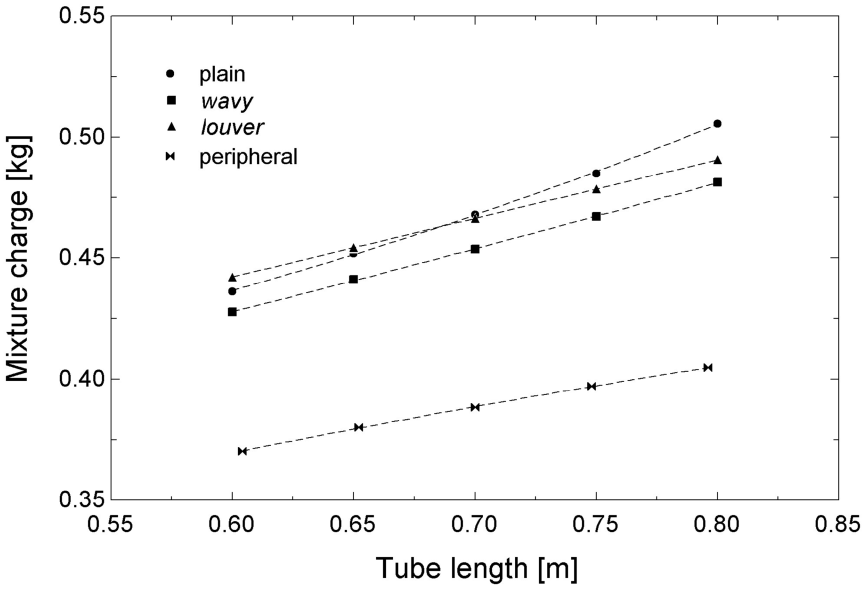

Figure 5. Comparison of refrigerant charge for different finned condensers.

4. CONCLUSIONS

This study presented an evaluation of the AC system performance for where the condenser applies peripheral fins as the solution for the heat transfer enhancement. The length of the tube was varied as a manner to numerically assess their impact on the system cooling capacity, SEER and refrigerant charge. For the evaluation, three more types of fins were simulated in this study: plain, wavy and louver.

The AC model considers all the system components, including their connecting lines. The oil circulation ration of POE ISO 22 was estimated based on the AC system conditions and thermodynamic properties of the mixture R-290/POE ISO 22 were computed via the departure function approach. Moreover, the mixture effect was taken into consideration for the computation of the heat transfer coefficient and friction factor.

Results have shown that the increase of total condenser area through the increase of tube length provided higher values of SEER rating. In the simulated range, the peripheral fin condenser promoted higher SEER values.

With the intention to keep a lower cooling capacity, the internal volume of the peripheral fin condenser was reduced, as well as the heat transfer area. Considering this new condenser configuration, the total refrigerant charge was strongly decreased, which is desirable when flammable refrigerants as R-290 are used in the residential AC industry.

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