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EVALUATION OF THE THERMAL PERFORMANCE OF FORCED CONDENSERS FOR BUS AIR-CONDITIONING SYSTEMS

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Abstract. *Air-conditioning systems are widely used by the automotive industry to ensure the comfort of the occupants of the vehicle. However, such systems are responsible for a significant increase in fuel consumption, which is even more drastic in buses, generally used for public transportation. One of the ways to mitigate this problem is to improve the thermodynamic performance of the refrigeration system, considering different possibilities for the refrigeration cycle. For this reason, the present work has the objective of comparing the thermodynamic performance of two condensers assembly configurations typically used in air-conditioning systems of buses. The first configuration considers the two condensers connected in series, while the second considers the condensers connected in parallel. To perform the comparative analysis, a mathematical model was developed based on the first principles laws. The mathematical model considers the geometric parameters of the condenser and operating conditions, such as the evaporator temperature, the ambient temperature, the speed of the compressor and the air flow supplied by the fans. The mathematical model was validated and presented a good agreement with the experimental measurements. The mathematical model was used to compare the two condensers and showed that the serial configuration enhances the condenser heat transfer in 6.5%.*

Keywords: *air-conditioning, buses, condenser configuration, thermal analysis*

1. INTRODUCTION

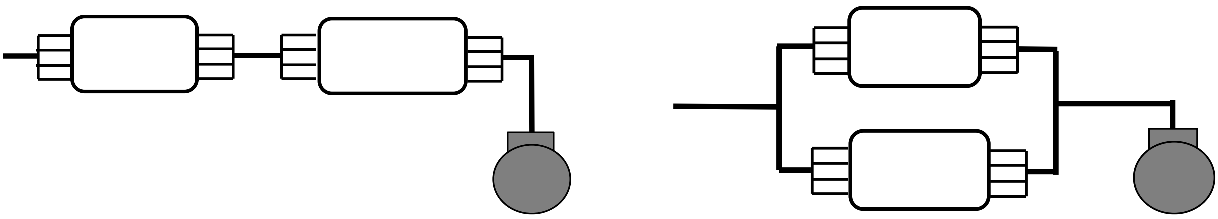
In comparison to cars automobiles, buses are a lower cost and energy efficient option to commuters. This alternative not only reduces the pollutant emissions per occupants, but also drastically decreases the number of vehicles in traffic (EU, 2009). To provide thermal comfort to the passengers, buses can be equipped with air-conditioning systems. However, the use of the air-conditioning requires an extra amount of fuel during its operation (Shah, 2009). Due to the large number of passengers and high frequency of door opening, this amount of fuel is even more drastic for this application. One alternative to mitigate this problem is to improve the thermodynamic performance of the refrigeration system, considering different possibilities for the refrigeration cycle and its components. The condenser is one of the fundamental components for the operation of an air-conditioning system based on the mechanical vapor compression principle. The purpose of the condenser is to accept the high temperature and pressure gas from the compressor and reject its heat to the environment, so that the refrigerant will condense and become liquid (Hundy et al., 2008). Although several studies regarding the air-conditioning systems of car passengers were developed in the recent years (Jabardo et al. 2002; Qi et al. 2010; Ng et al. 2014; Da Silva and Melo, 2016; Da Silva and Cordova, 2017; Gillet et al. 2018), only a small number of studies available in the literature are focused on buses (Hegar et al. 2013). Therefore, the present work has the objective of comparing the thermodynamic performance of two condensers assembly configurations typically used in air-conditioning systems of buses. The first configuration considers the two cooling system condensers connected in series, while the second one considers the condensers connected in parallel. To carry out the comparative analysis, a first principles laws mathematical model was developed and validated with experimental data. The mathematical model considers the geometric parameters of the condensers under evaluation and operating conditions, such as the evaporator temperature, the ambient temperature, the speed of the compressor and the air flow supplied by the fans.

2. AIR-CONDITIONING SYSTEM AND EXPERIMENTAL TESTS

The schematic representation of the compressor and the two condensing sections configurations investigated in this study are presented in Fig. 1. Each configuration is equipped with two condensers. Figure 1a shows the serial configuration in which the two condensers are connected in series, while Fig.1b shows the parallel configuration. As can be seen, the fluid refrigerant flow is divided in four streams at the condenser inlet and unified again at the outlet position. The condensers are tube-finned heat exchangers assembled with copper tubes and aluminum fins. The external diameter of the tubes is 9.5 mm and the fin density is 3.5 fins/cm. The tubes are in a staggered arrangement and the distance between its centers is 25 mm. The length, height and width of each condenser are 1360 mm, 206 mm and 155 mm, respectively. Four axial fans operating in parallel produce the airflow on the condensers external surface. The external diameter of the fans is 330 mm. In addition, the compressor is a reciprocating model with 550 cm^3 of displacement per revolution and the fluid refrigerant is R-134a. This components are typically used in interurban buses, as shown in Fig. 2. In this applications, the condenser section may be located on the top of the vehicle roof, in which the ambient airflow is admitted by the frontal and lateral grids of the condensing section and rejected at the top.

In order to carry out experimental measurements, pressure transducers and thermocouples were installed in the air-conditioning system. Two pressure transducers were installed in the inlet and outlet positions of the compressor, while fourteen T-type thermocouples where installed in the air flow and on the external surface of the tubes along the condenser. The uncertainty of the pressure and temperature transducers were ±0.1 bar and ±0.5ºC, respectively. Steady-state measurements were carried out with both condenser configurations at VMG Aires (2019) facilities.

Furthermore, the airflow rate of the condensation sub-system was measured with a wind tunnel at Polo Laboratory facilities. In order to identify the operation point of the fans, the measurements were carried out using the entire condensing section of the air-conditioning system, taking into account the fans, heat exchangers, grids and the assembly structure. The airflow measurement uncertainty of the wind tunnel was evaluated as ±10%.



(a) Serial configuration (b) Parallel configuration

Figure 1. Configurations of the condensers under analysis

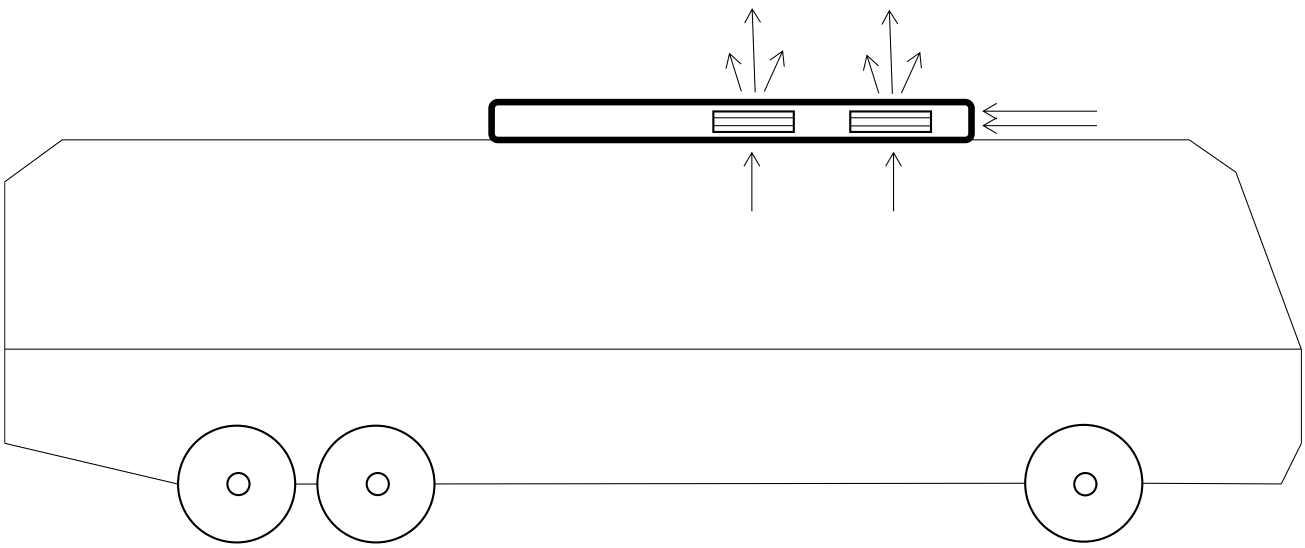


Figure 2. Schematic representation of the condensing section located on the roof of the bus

3. MATHEMATICAL MODEL

The mathematical model described in this work is based on the condenser thermal-hydraulic evaluation approach developed by Hermes et al. (2002). This formulation consists in an iterative procedure to evaluate the compressor discharge pressure that satisfy the energy balance on the refrigerant and air sides of the condenser. The refrigerant pressure drop inside the condenser is also considered. The main assumptions of the mathematical model are: (i) steady state operating condition, (ii) uniform air distribution on the condenser surface (iii) semi-empirical correlation for the compressor performance, (iii) fixed thermodynamic state at the compressor inlet and (iv) a constant subcooling at the exit of the condenser.

3.1 Condenser model

The condenser model was divided in three regions that correspond to the superheated region (*sup*), two-phase region (*tp*) and subcooled region (*sub*). The following equations were used to evaluate the condenser heat transfer rate (Q_{cond}).

$$Q_{cond} = Q_{cond,\text{sup}} + Q_{cond,tp} + Q_{cond,sub} \tag{1}$$

$$Q_{cond,\text{sup}} = \dot{m}c_{p,v}\left(T_{comp,e} - T_{cond,i}\right) \tag{2}$$

$$Q_{cond,tp} = \dot{m}\left(h_{cond,v} - h_{cond,l}\right) \tag{3}$$

$$Q_{cond,sub} = \dot{m}c_{p,l}\left(T_{cond,se} - T_3\right) \tag{4}$$

where $\dot{m}$ is the refrigerant mass flow rate, $T_{\text{comp,e}}$ is the refrigerant temperature at the compressor exit, $T_{cond,i}$ is the refrigerant temperature at the condenser inlet, $h_{\text{cond,v}}$ and $h_{cond,l}$ are the saturated vapor and liquid enthalpies of the refrigerant, $T_{cond,se}$ is the saturated refrigerant temperature at the end of the two-phase section and T_3 is the refrigerant temperature at the condenser outlet. On the refrigerant side, the heat transfer coefficients of the superheated and subcooled regions were obtained with the Dittus and Boelter (1930) correlation, whereas the correlation proposed by Shah (1979) was used for the two-phase region. Moreover, the air-side heat transfer coefficient was evaluated using the correlation proposed by AWF (1995), while the outlet temperature of the air in each region was calculated by the ε-NTU method. The pressure drop in the refrigerant side were calculated by the correlation proposed by Ould Didi et al. (2002) model in the two phase region, while the friction factors for the single phase regions were evaluated using the Moody factor (Moody, 1984).

3.2 Compressor model

The air-conditioning system was equipped with an externally driven reciprocating compressor produced by Valeo (2019). The manufacturer performance curves for cooling capacity and power consumption were used to adjust the empirical correlations proposed by Li (2012). Equation (6) was used to evaluate the refrigerant mass flow rate, while Eq. (7) were used to evaluate the compressor power consumption (W_{comp}).

$$\eta_v = c_1 + c_2\left(\frac{P_d}{P_a}\right)^{\frac{1}{n}} \tag{5}$$

$$\dot{m} = \eta_v \rho_a V_w N \tag{6}$$

$$W_{comp} = \frac{P_a \dot{m}}{\rho_a}\frac{k}{k-1}\left[\left(\frac{P_d}{P_a}\right)^{\frac{n-1}{n}} - 1\right]\left(c_3 + \frac{c_4}{P_a} + \frac{c_5}{P_d}\right) + c_6 \tag{7}$$

where P_d is the compressor discharge pressure, P_a is the compressor suction pressure, n is the polytropic coefficient, V_w is the compressor displacement, N is the compressor speed and ρ_a is the refrigerant density in the compressor inlet. In addition, c_1-c_6 are empirical coefficients.

4. RESULTS

Figure 3 shows the four fans characteristic curve and the operating line measured in a wind tunnel. The intersection of these curves represents the airflow rate supplied by the four fans to the condensers, considering the pressure drop related to the grids, condensers and the condensing section structure. The air flow rate measured at the intersection point is 5360 m^3/h.

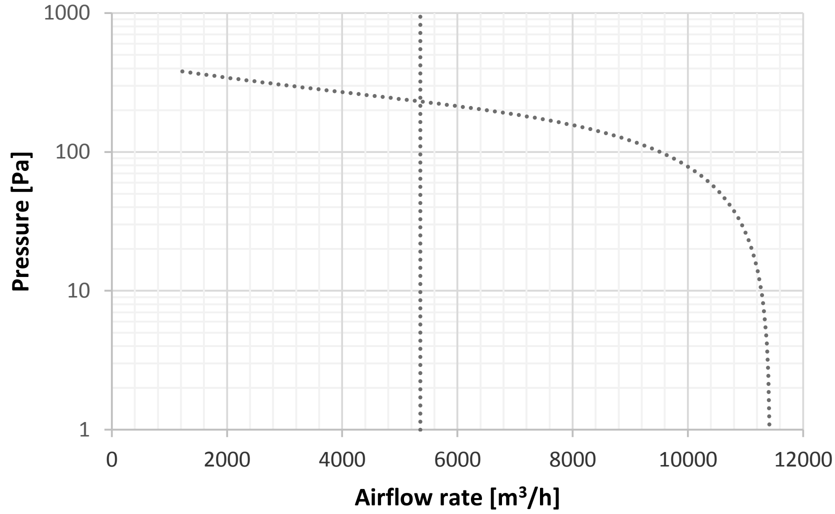


Figure 3. Operating point of the fans

Figure 4 and Figure 5 show a comparison between simulated results and the reference values, obtained on the compressor manufacturer performance curves. Figure 4 compares the compressor refrigerant mass flow rate while Fig. 5 compares the power consumption for different operating conditions. A good agreement is observed between the results, with errors smaller than 3% for both variables.

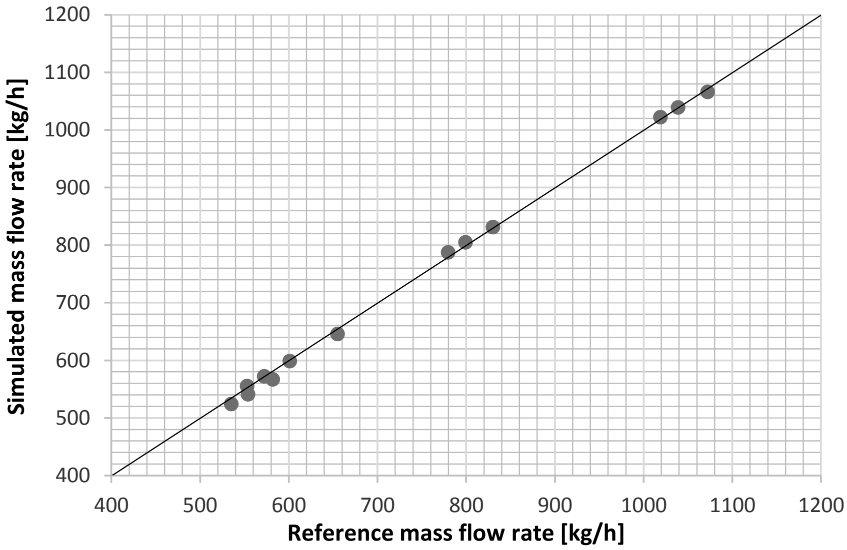


Figure 4. Compressor mass flow rate comparison

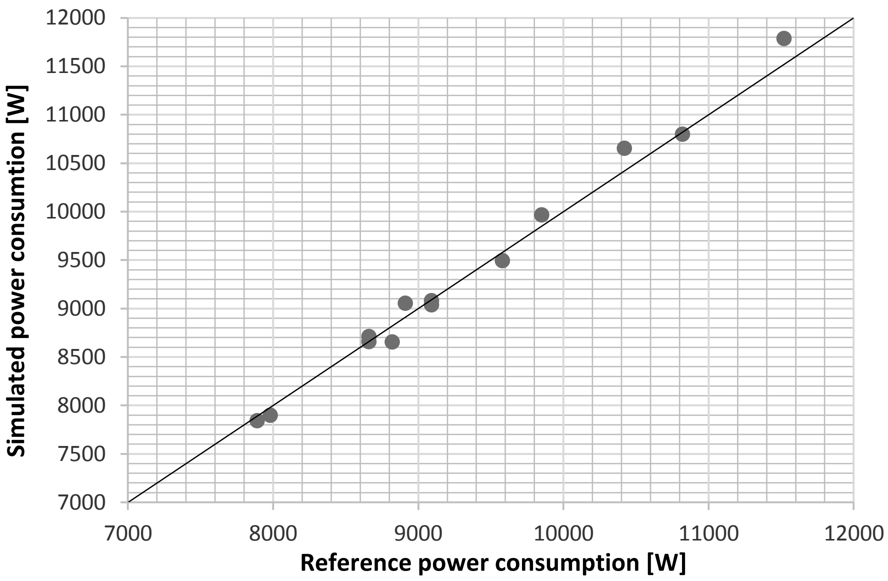


Figure 5. Compressor power consumption comparison

The mathematical model of the system, comprised by the compressor and the condensers, was also validated with experimental results. Table 1 shows a comparison between the experimental measurements and simulation predictions for serial and parallel condensers configurations. As can be seen, the mathematical model was able to predict the compressor discharge pressure and temperatures with errors smaller than 0.4 bar and 6.5°C, respectively.

Table 1. Comparison between experimental and simulated results

Serial configuration (T_{amb} = 31.0ºC)				Parallel configuration (T_{amb} = 27.5ºC)			
	Experimental	Simulation	Error		Experimental	Simulation	Error
P_d [bar]	15.3	15.7	0.4 bar	P_d [bar]	12.7	12.9	0.2 bar
T_d [°C]	87.4	93.6	6.2°C	T_d [°C]	80.1	84.0	3.9°C

After the validation process, the mathematical model was used to compare the parallel and serial condensers configuration and identify which one presents the best the thermodynamic performance. The simulations were carried out considering a fixed evaporator temperature of 2.5°C, a superheat of 10.0°C at the compressor suction line, a compressor speed of 1590 rpm and a 3.0°C of subcooling in the condenser outlet. The air flow rate supplied by the fans was fixed in 5360 m^3/h.

Figure 6 shows the effect of the ambient temperature in the discharge pressure for serial and parallel condenser configurations. Environmental temperatures between 27.0°C and 40.0°C, that represent bus air conditioning applications, were investigated. As can be seen, the parallel configuration presents higher discharge pressures over the entire range of temperatures. In addition, it is observed that the difference between the serial and parallel configurations increases with the ambient temperature from 4% at 27.0°C up to 8% at 40.0°C.

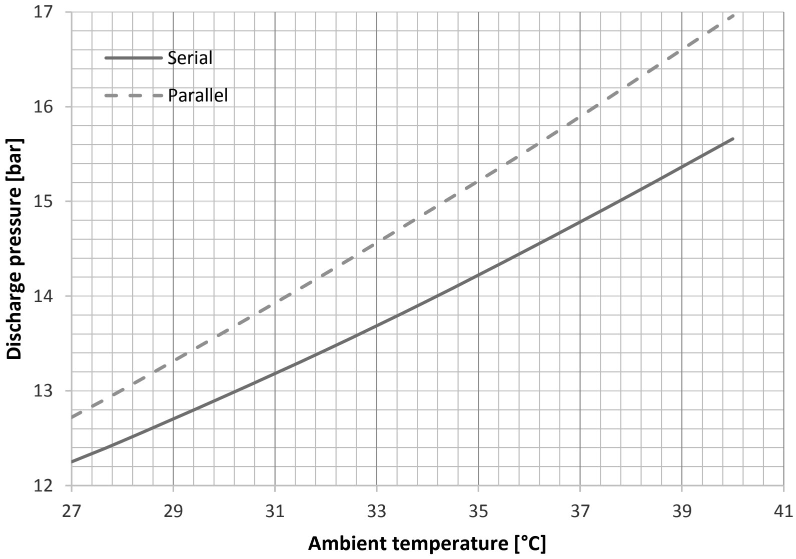


Figure 6. Discharge pressure comparison

The effect of the ambient temperature in the refrigerant mass flow rate is presented in Figure 7. The higher discharge pressure observed in the parallel configuration in Fig. 6 affects the compressor performance, which in turn, results in a smaller refrigerant mass flow for this configuration. The results also show that the mass flow rate reduction is higher for the parallel configuration in comparison to the serial.

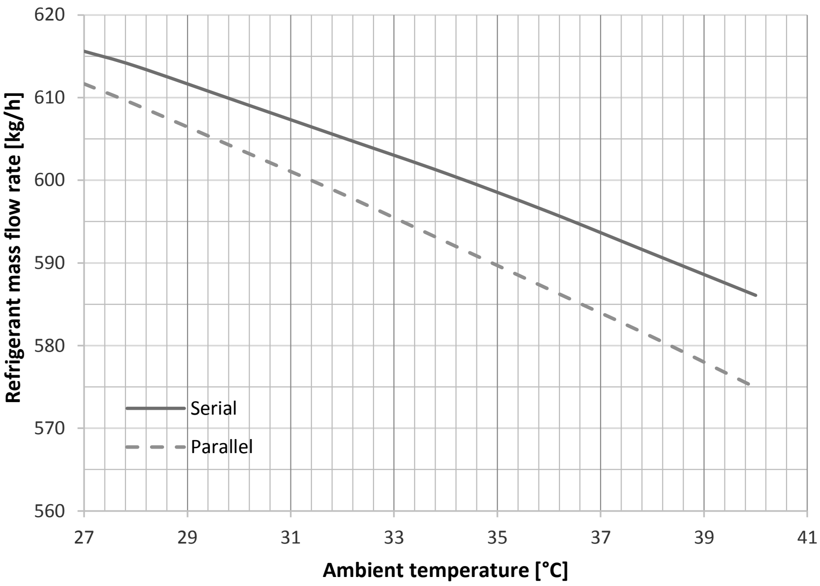


Figure 7. Refrigerant mass flow rate comparison

Figure 8 compares the refrigerant pressure drop of both configurations. The pressure drop represents the difference between the inlet and outlet condenser pressures, and it is mainly related to the friction between the fluid refrigerant and the tubes internal surfaces. The comparison show that the parallel configuration results in an average pressure drop ten times smaller than the observed in the serial configuration. This significant difference is justified by the division of the refrigerant mass flow rate in two streams before the condensing section (Fig. 1b) combined with the smaller compressor mass flow rate observed in Fig. 7.

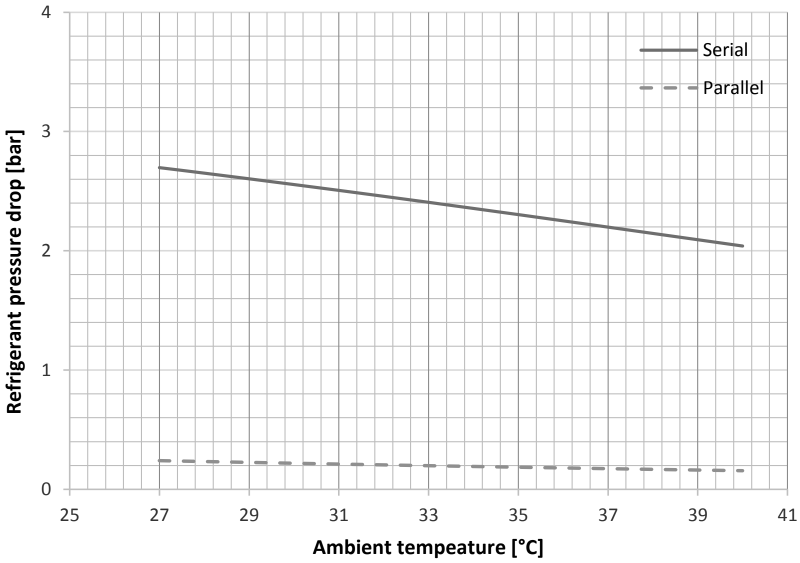


Figure 8. Refrigerant pressure drop comparison

Figure 9 shows the overall coefficient of heat transfer for both condenser configurations and different ambient temperatures. As can be seen, the serial configuration presents a higher overall coefficient of heat transfer. An average difference of 7.5% is observed over the entire range of investigated temperatures. As the condenser surface and the air flow rate are identical for both configurations, this difference is mainly explained by the higher refrigerant mass flow rate of the serial configuration. The high mass flow rates results in higher fluid velocities, which in turn produces high heat transfer coefficients. The higher heat transfer coefficients observed in the serial configuration also justify how this configuration is able to operate with lower discharging pressures, as depicted in Fig. 6.

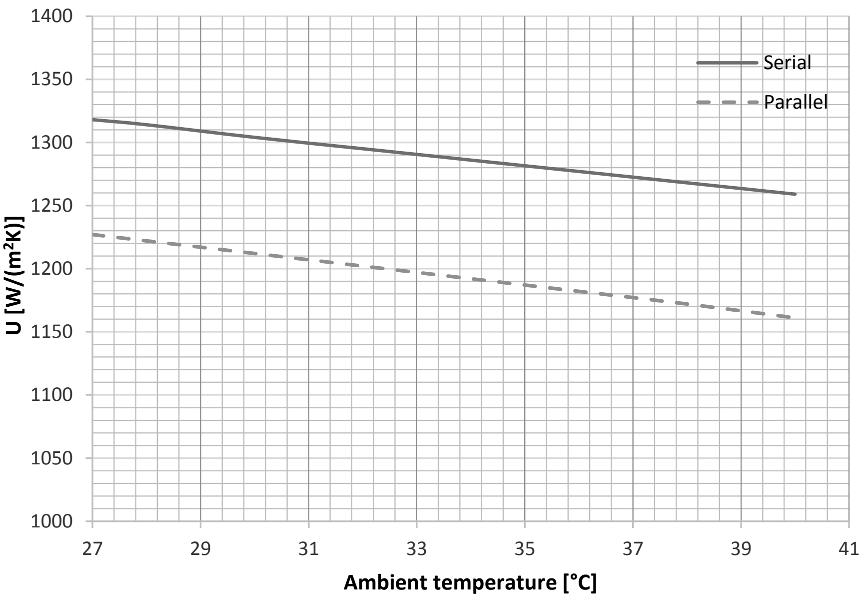


Figure 9. Overall coefficient of heat transfer comparison

The mathematical model was also used to evaluate the condenser heat transfer rates for both configurations. An ambient temperature of 36.0ºC was considered for this analysis. Figure 10 compares the heat transfer rates for the vapor, two-phase and liquid regions of the condenser. It shows that the serial configuration results in a 6.5% higher heat transfer rate. As can be seen, the main difference was observed in the two phase region, due to a 7.5% larger overall heat transfer coefficient. This results also shows that the higher refrigerant pressure drop of the serial configuration is offset by the larger heat transfer rate due to its larger refrigerant mass flow rate (Fig. 7).

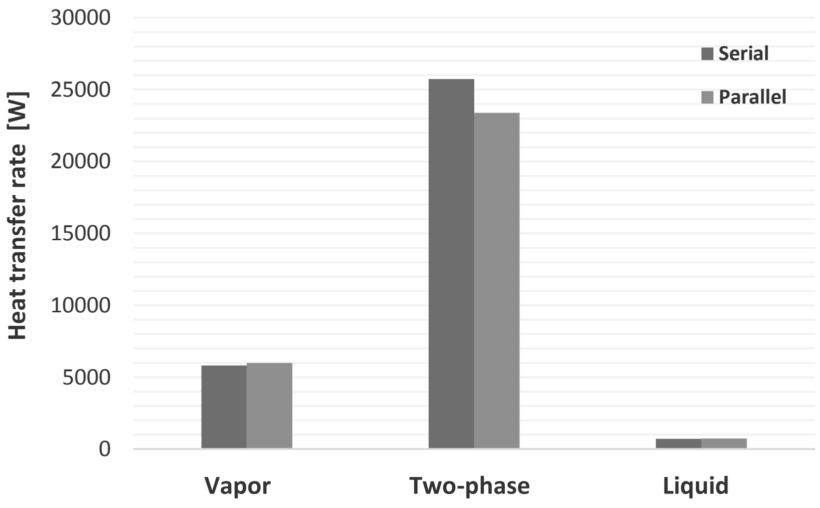


Figure 10. Comparison between the heat transfer rates

Finally, Figure 11 compares the coefficient of performance (COP) of the condensers under evaluation. As expected, the COP of the system decreases with the ambient temperature. This result is mainly justified by the increase in the compressor discharge pressure with the increase of the ambient temperature presented in Fig. 6. In addition, it was o observed that, over the entire range of ambient temperatures, the serial configuration presents a higher COP. It is observed that the COP difference between the serial and parallel configurations increases with the ambient temperature from 4.4% at 27°C up to 9.4% at 40°C. These results confirm the serial configuration as the higher energy efficiency option for this application.

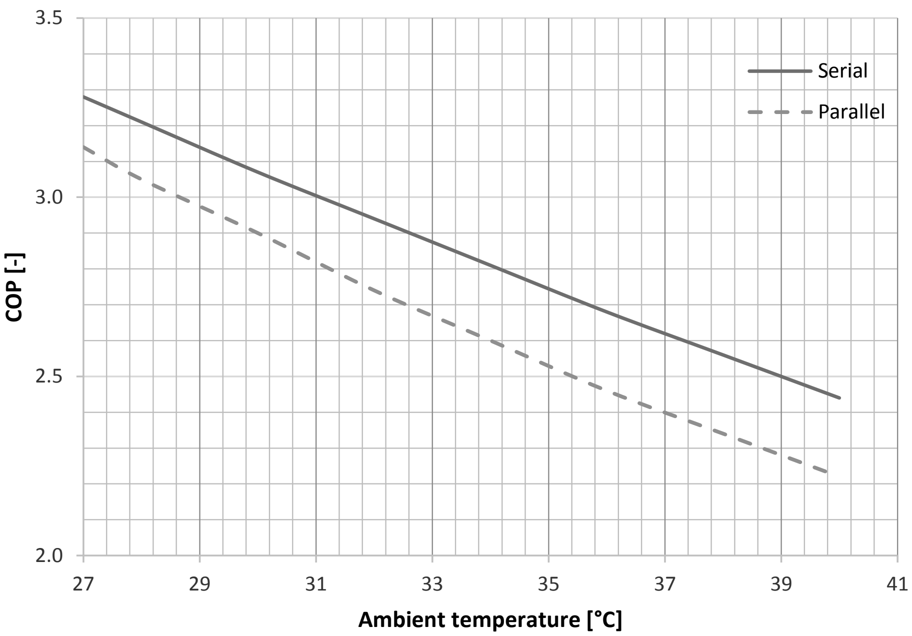


Figure 11. Coefficient of performance comparison

5. CONCLUSIONS

Condensers play an important role in the efficiency of the air-conditioning system as they have a strong effect on the operating pressures of the system, which in turn influence the energy consumption. In this study, the thermal performance of two condensers configurations for buses air-conditioning applications were compared. The first configuration considered two condensers assembled in series, while the second considered them in parallel. Both configurations has the same heat transfer area and receive the same air flow rate. A mathematical model was developed based on the first principle laws and considers the geometric parameters of the condenser and operating conditions, such as the evaporator temperature, the ambient temperature, the speed of the compressor and the air flow supplied by the fans. The compressor model results were compared with manufacturer performance curves and presented a good agreement, with errors smaller than 3% for power consumption and mass flow rate. Experiments were carried out for the bus air-conditioning system equipped with both condensers configurations. The experimental data were used to validate the air-conditioning system mathematical model and presented maximum errors of 0.2 bar and 6.5ºC for the discharge pressure and temperature, respectively. In addition, the air flow rate of the fans were measured in a wind tunnel considering the pressure drop related to the grids, condensers and the condensing section structure. The results show that, under the same operating conditions, the serial configuration is able to operate at lower discharge pressures, which in turn results in higher mass flow rates. Although the parallel configuration presented a lower refrigerant pressured drop, its overall heat transfer coefficient was 7.5% smaller than the results observed in the serial configuration. Moreover, the mathematical model results showed that the serial configuration results in a 6.5% higher heat transfer rate, justified by a higher overall heat transfer coefficient. These results show that the higher refrigerant pressure drop of the serial configuration is compensated by its larger heat transfer rate. Finally, COP differences up to 9.4% were evaluated, indicating the serial configuration as the higher energy efficiency option for this application.

6. ACKNOWLEDGEMENTS

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