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MATHEMATICAL MODELING OF AN AUTOMOTIVE AIR-CONDITIONING SYSTEM UNDER STEADY-STATE CONDITIONS

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Abstract. *Brazil is among the 20 world's largest passenger cars manufacturers and has produced 1.7 million units in 2016. One important subsystem of these vehicles is the automotive air-conditioning that is designed to improve the security and the welfare of the occupants. As the air-conditioning systems is the largest auxiliary load to vehicle engine, its operation results in an increase of the fuel consumption and greenhouse gas emissions. For these reasons, one of the obstacles of the automotive industry is to provide thermal comfort in a sustainable and energy efficient way. Due to cost and time restrictions, the mathematical modeling approach has become essential to predict the performance of automotive air-conditioning systems in the car industry. Considering this scenario, the aim of this paper is to present a mathematical model to evaluate the energy performance of automotive air-conditioning systems. The mathematical model was developed based on the conservation laws and empirical correlations. It accounts for variables such as: the environmental conditions, the compressor speed, the expansion device geometry and the heat exchangers geometry and air flow rates. The results obtained with the mathematical model can either be used to evaluate the energy performance of existing systems or support the design of new concepts.*

Keywords: *Automotive Air-Conditioning, MAC system, Mathematical Model, Energy Performance.*

1. INTRODUCTION

Currently, Brazil is among the 20 world's largest passenger cars manufacturers and has produced 1.7 million units in 2016 (OICA, 2016). Many of these vehicles are equipped with factory-installed mobile air-conditioning (MAC) systems that not only improve the safety but also the comfort of the occupants. On the other hand, the MAC system is the largest subsystem in a vehicle, which in turn results in an increase up to 7.5% on the fuel consumption and direct and indirect greenhouse gas emissions and (Shah, 2009).

The literature review presented by Da Silva and Melo (2016) showed that the design of an automotive air-conditioning system considers several parameters such as the vehicle space restrictions, thermal comfort, durability, insulation, noise, vibration among others. Moreover, one of the challenges of the automotive industry is to provide thermal comfort in a sustainable and energy efficient way. Due to cost and time restrictions, the automotive industry has been using the mathematical modeling approach to predict the performance of automotive air-conditioning systems in the car industry. In addition, there is a growing body of literature that recognizes the importance of mathematical modeling approach applied to MAC systems. For instance, Lee et al. (2000) proposed a mathematical model for an automotive air-conditioning and showed that it can be used to size the proper condenser for a specific system. Jabardo et al. (2002) proposed a steady state mathematical model for a MAC system equipped with a variable capacity compressor. The numerical predictions were compared with experimental results with maximum deviations of ±20%. In addition, Tian et al. (2005) proposed a dynamic model for a MAC system and observed that the fast transients observed in the compressor and the expansion devices can be neglected. Ng et al. (2014) used neural networks theory to predict the air-conditioning system behavior and proposed control strategies able to provide thermal comfort with low energy consumption.

Based on the previous facts, the purpose of this paper is to develop a steady-state mathematical model for an automotive air-conditioning system taking into account variables such as the environmental conditions, the compressor speed, the expansion device geometry and the heat exchangers geometry and air flow rates. The model will be designed to predict the main parameters of the system as well as the energy performance under different operating conditions.

2. MODELED AIR-CONDITIONING SYSTEM DESCRIPTION

Figure 1 shows a typical thermostatic expansion valve receiver-dryer (TXV-RD) system used by the automotive industry and its corresponding pressure-enthalpy diagram. The cycle starts with the compression of the fluid refrigerant in the vapor phase that exits the evaporator at position 1. During the compression process, the temperature of the refrigerant vapor is increased to a value above the temperature of the surroundings. The super-heated refrigerant that exits the compressor at position 2 flows to the condenser and transfers heat to the surroundings in order to become liquid. The high pressure liquid is accumulated in the receiver-dryer, which allows only liquid refrigerant to flow to the thermostatic expansion valve, as indicated by position 3. Next, the liquid refrigerant expands through the thermostatic valve, which regulates the mass flow rate according to the refrigerant superheat measured at the evaporator exit. Finally, the two-phase low pressure refrigerant that exits the thermostatic expansion valve at position 4 flows through the evaporator, exchanges heat with the external air flow and returns to the initial thermodynamic state of the cycle. While the air flows through the external side of the evaporator its temperature and humidity are reduced. The cold dry air, which exits the evaporator, is then circulated to the passenger compartment.

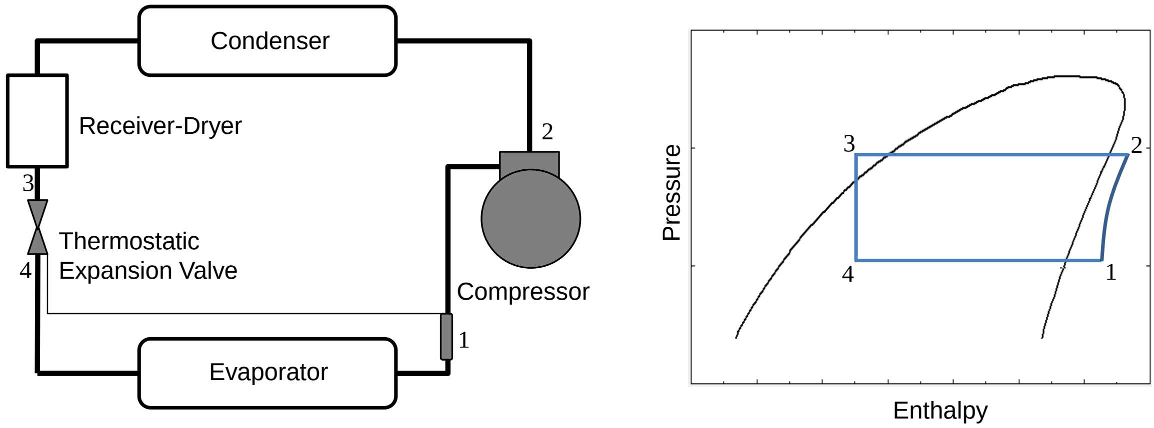

Figure 1. Schematic representation of a TXV-RD system and its corresponding pressure-enthalpy diagram

3. MATHEMATICAL MODEL DESCRIPTION

The mathematical model presented on this work is based on the formulation of Jabardo et al. (2002) which employs the conservation laws and empirical correlations of the main components of the automotive air-conditioning system. The fluid refrigerant is the R134a and the following main assumptions were considered: (i) steady state conditions, (ii) average heat transfer coefficients on the heat exchangers, (iii) semi-empirical correlations for the compressor and the thermostatic valve, (iii) polytropic compression process, (iv) expansion device orifice area is realted to the pressure difference between the minimum and maximum valve openings, (v) the receiver is modelled as a volume between the thermostatic valve and the evaporator, (vi) the latent heat of the air was not considered on the evaporator model.

3.1 Compressor

The compressor used on this work is a Delphi 7CVC165 externally driven reciprocating compressor. It was modeled using its performance curves for cooling capacity and power consumption. The following equations were used to evaluate the volumetric efficiency (η_v), the isentropic efficiency (η_s), the compressor mass flow rate ($\dot{m}_{comp}$) and the compressor power consumption ($\dot{W}_{comp}$)

$$\eta_v = \left(1 - \frac{C_v}{V_w}\left(\left(\frac{P_d}{P_a}\right)^{\frac{1}{n}} - 1\right)\right)\psi_1(N,V_w) \tag{1}$$

$$\eta_s = \psi_2\left(N\right) \tag{2}$$

$$\dot{m}_{comp} = \eta_v \rho_a V_w N \tag{3}$$

$$\dot{W}_{comp} = \eta_s \dot{W}_s \tag{4}$$

where C_v is the clearance volume, V_w is the swept volume, P_d is the compressor discharge pressure, P_a is the compressor suction pressure, n is the polytropic coefficient, N is the compressor speed, ρ_a is the refrigerant density in the compressor inlet and $\dot{W}_s$ is the compressor isentropic power consumption. In addition, ψ_1 and ψ_2 are power functions of N and V_w.

3.2 Expansion valve

The expansion device is a thermostatic expansion valve that adjusts the orifice area according to the evaporator pressure and the superheating at the evaporator exit position. Equations (5) and (6) were used to evaluate the maximum thermostatic valve mass flow rate ($\dot{m}_{\text{valve,o}}$) and the mass flow rate when the valve is partially opened ($\dot{m}_{\text{valve}}$)

$$\dot{m}_{valve,o} = k_v \sqrt{\rho_l \left(P_{cond,e} - P_{evap,i} \right)} \tag{5}$$

$$\dot{m}_{valve} = \tau m_{valve,o} \tag{6}$$

where k_v is valve coefficient, ρ_l is the refrigerant density at the valve inlet, $P_{cond,e}$ is the pressure at the condenser exit, $P_{evap,i}$ is the pressure at the evaporator inlet, τ is the valve orifice opening area factor.

3.3 Condenser model

The condenser is a microchannel heat exchanger described by Mamani (1997). To carry out the analysis the condenser was divided in three regions that correspond to the superheated (*sup*), two-phase (*tp*) and subcooled (*sub*) sections. The following equations were used to evaluate the condenser heat transfer rate (Q_{cond}).

$$Q_{cond} = Q_{cond,\sup} + Q_{cond,tp} + Q_{cond,sub} \tag{7}$$

$$Q_{cond,\sup} = \dot{m} c_{p,v} \left(T_{comp,e} - T_{cond,i} \right) \tag{8}$$

$$Q_{cond,tp} = \dot{m} \left(h_{cond,v} - h_{cond,l} \right) \tag{9}$$

$$Q_{cond,sub} = \dot{m} c_{p,l} \left(T_{cond,se} - T_3 \right) \tag{10}$$

where, $\dot{m}$ is the mass flow rate, $T_{comp,e}$ is the refrigerant temperature at the compressor exit, $T_{cond,i}$ is the refrigerant temperature at the condenser inlet, $h_{cond,v}$ 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 valve inlet. 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. In turn, 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.

3.4 Evaporator model

The evaporator investigated was the fin and tube heat exchanger described by Mamani (1997). In order to evaluate que evaporator heat transfer rate (Q_{evap}), the thermal analysis was performed by dividing the evaporator in two sections, correspondent to the two-phase and superheated, as showed by equations (11), (12) and (13)

$$Q_{evap} = Q_{evap,tp} + Q_{evap,\sup} \tag{11}$$

$$Q_{cond,tp} = \dot{m}\left(h_4 - h_{evap,v}\right) \tag{12}$$

$$Q_{evap,\sup} = \dot{m}c_{p,l}\left(T_{evap,e} - T_1\right) \tag{13}$$

where $\dot{m}$ is the mass flow rate, h_4 is the enthalpy at the valve exit and $h_{evap,v}$ is the saturated vapor enthalpy of the refrigerant, $T_{evap,e}$ is the saturated refrigerant temperature at the end of the two-phase section and T_1 is the refrigerant temperature at the evaporator exit position. On the refrigerant side, the heat transfer coefficients of the superheated region was obtained with the Dittus and Boelter (1930) correlation, whereas the correlation proposed by Klimenko (1988) was used for the two-phase region. The air-side heat transfer coefficient was evaluated by a correlation proposed by AWF (1995) for fin and tube heat exchanger and the outlet temperature of the air in each region was calculated by the ε-NTU method.

3.5 Solution procedure

The solution procedure is shown in Figure 1. As can be seen, there are three iterative loops required to simultaneously solve the set of algebraic equations. The input variables are the air temperatures and the air flow rates on the evaporator and the condenser, and the refrigerant subcooling temperature in the condenser exit. The solution of this procedure identifies the system operating parameters related to the input variables. The procedure initiate with guessing values for the evaporator pressure (P_e), condenser pressure (P_c) and the superheating at the evaporator outlet (*Sup*). As can be seen, the first loop corrects the superheating, the second corrects P_e based on the difference of the mass flow rate between the compressor and the expansion device, while the last loop corrects P_c using the error on the subcooling degree (*Sub*). The Newton-Raphson technique was used to evaluate the new values of the guessed variables based on the errors.

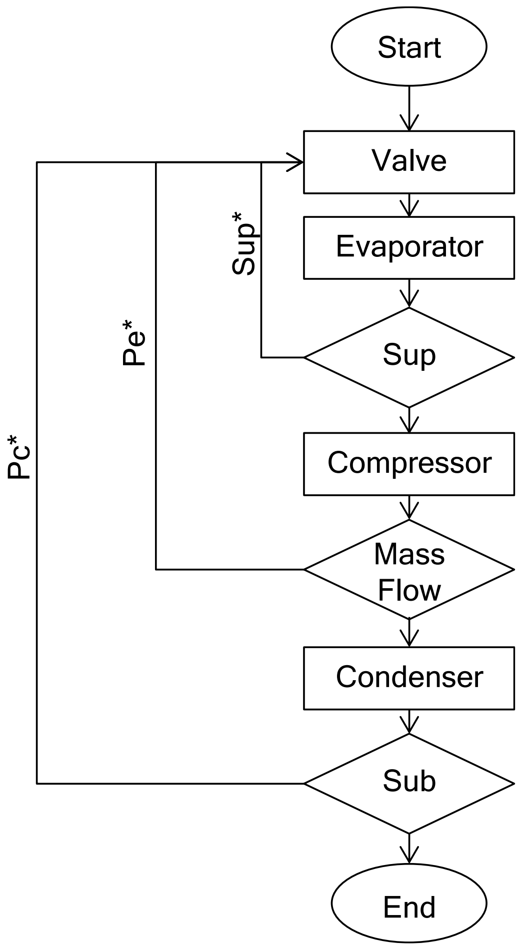

Figure 2. Solution procedure of the mathematical model

4. RESULTS AND DISCUSSIONS

The mathematical model was used to predict the system performance under different operating conditions. In order to carry out a sensibility analysis, the air flow rate of the evaporator was changed between 400 to 600 kg/h. The increase of this parameter causes a rise on the evaporator thermal load, pushing the system to a different operating condition. During the simulations, the inlet condenser air velocity and inlet temperature were fixed in 3 m/s and 30°C respectively, the evaporator inlet air temperature was 20°C and the compressor speed was 3300 rpm.

Figures 3 and 4 show the effect of the evaporator air flow rate on the condenser and evaporator pressures, respectively. As can be seen, the change on the evaporator air flow rate increases the evaporator and condenser pressures. The evaporator pressure behaviour is due to the increase in the air side convective heat transfer coefficient promoted by the change on the air flow rate, which in turn raises the heat absorbed by the fluid refrigerant, resulting in an increase of the saturation pressure and temperature. Although the condenser air temperature and air flow rates are fixed, the condenser pressure is also affected because the compressor and the expansion device need to identify a new condition in which the new condenser heat transfer rate is satisfied Stoecker (1985). As can be seen in Fig. 4, the increase of the evaporator air flow rate results in an increase of the condenser pressure.

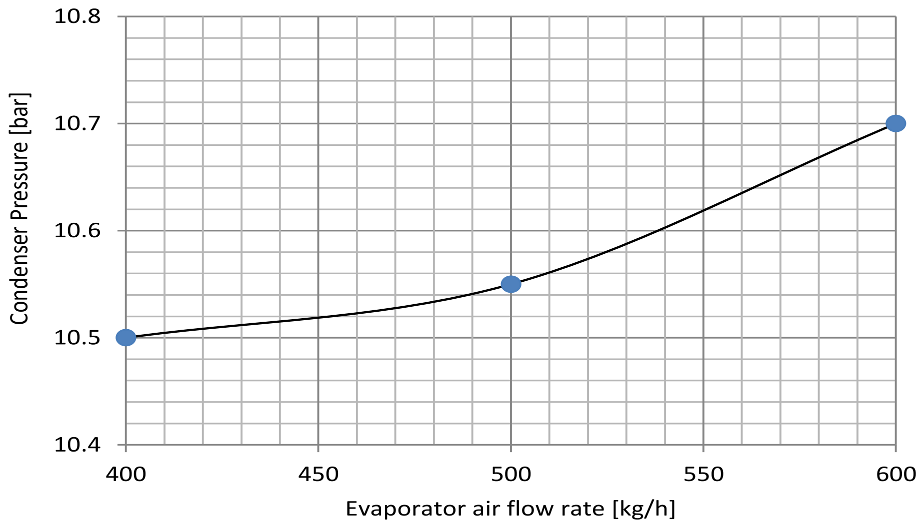

Figure 3. Effect of evaporator air flow rate on the condenser pressure

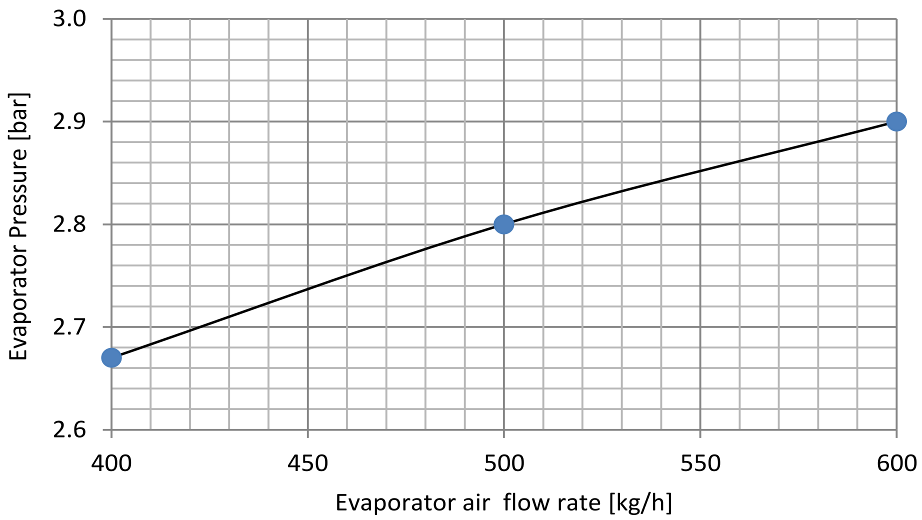

Figure 4. Effect of evaporator air flow rate on the evaporator pressure

Figure 5 represents the intersection of equations 3 and 5, in which the compressor and expansion valve mass flow rates are identical, as expected for the steady-state operating condition. It is observed that the increase in the evaporator air flow rate forces the air-conditioning system to operate at higher refrigerant mass flow rates. Despite the condenser pressure rise, the compressor mass flow rate increases due to the combined effect of the evaporator pressure and the refrigerant density at the compressor inlet. The increase of the expansion device mass flow rate is mainly due to the change in the orifice area of the thermostatic valve (see equation 5).

The cooling capacity of the system as a function of the evaporator air flow rate is illustrated in Figure 6. This behavior is justified by: (i) the increase on the air side convective heat transfer coefficient associated with a higher air velocity through the evaporator and (ii) the increase on the refrigerant mass flow rate depicted in Figure 5. In this case, as the thermal load increase due to the change in the evaporator air flow rate, the expansion valve opens to allow more refrigerant flow into the evaporator.

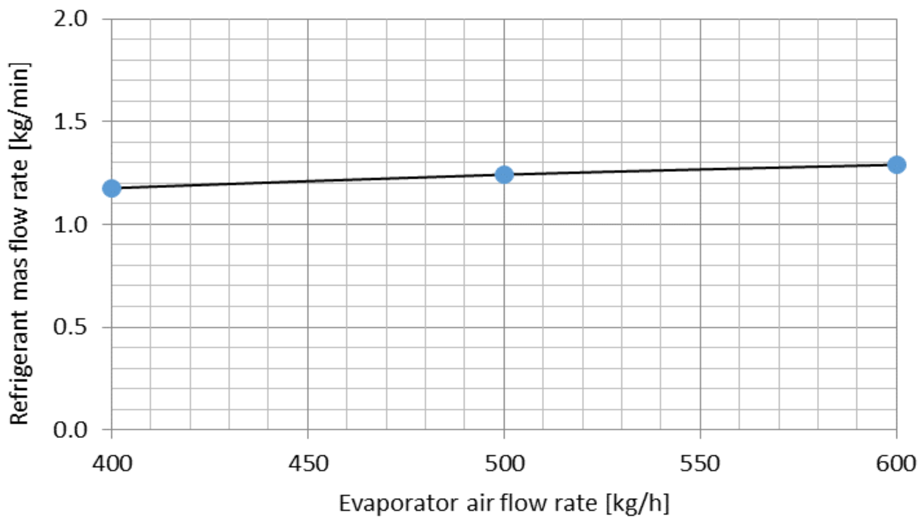

Figure 5. Effect of evaporator air flow rate on the refrigerant mass flow rate

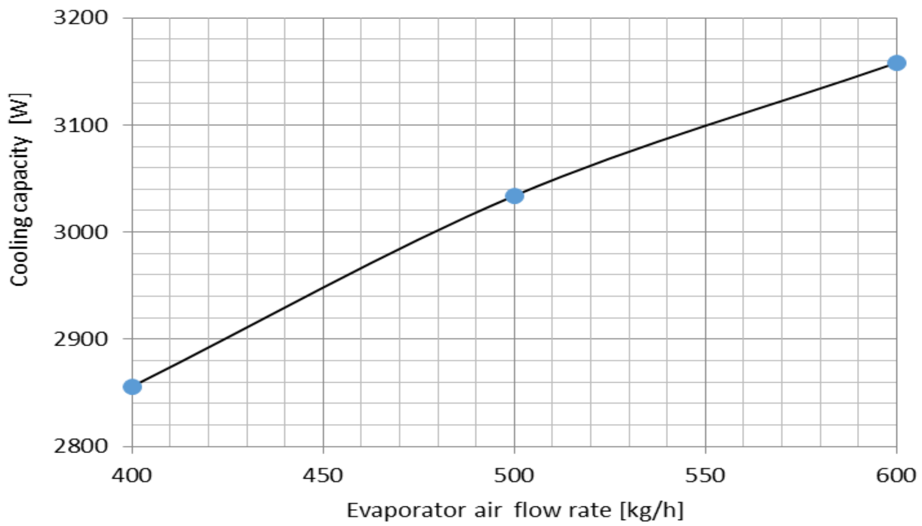

Figure 6. Effect of evaporator air flow rate on the cooling capacity

Although it has been observed an increase in the cooling capacity with the evaporator air flow rate, Figure 7 shows that the additional cooling capacity does not necessarily result in a reduction of the evaporator outlet air temperature. This behavior is explained by the higher air mass flow rate flowing through the coil, combined with the increase in the evaporator pressure (Figure 4), which in turn results in a higher coil temperature.

Figure 8 shows a nearly linear increase of the compressor power consumption with the evaporator air flow rate. The rise of the power consumption is mainly associated with the increase of the refrigerant mass flow rate, as the isentropic efficiency of the compressor was not significantly affect. The global effect of the evaporator air flow rate on the air-conditioning system energy efficiency is depicted in Figure 9, which shows the coefficient of performance (COP). As can be seen, the system COP slightly increases with the evaporator air flow rate, showing that the increase of the compressor power consumption is compensated by the enhancement of the cooling capacity inside the investigated region.

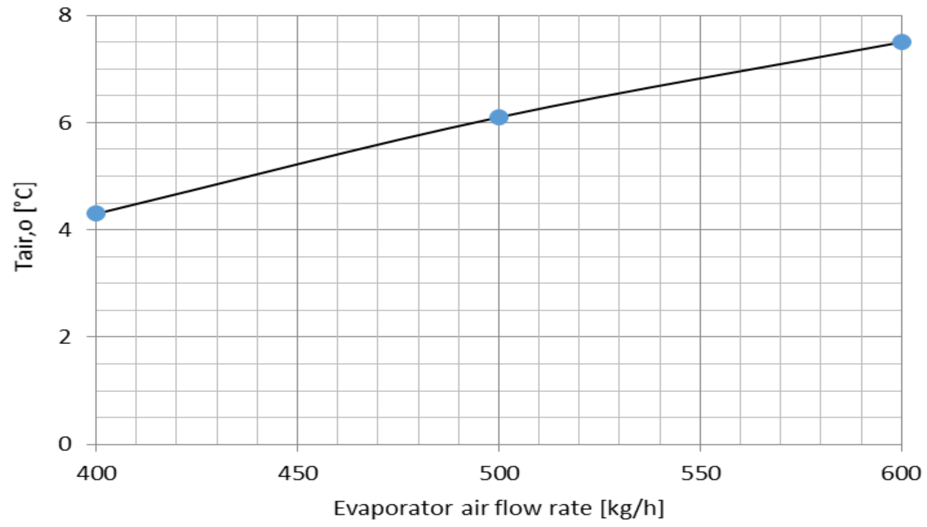

Figure 7. Effect of evaporator air flow rate on the evaporator outlet air temperature

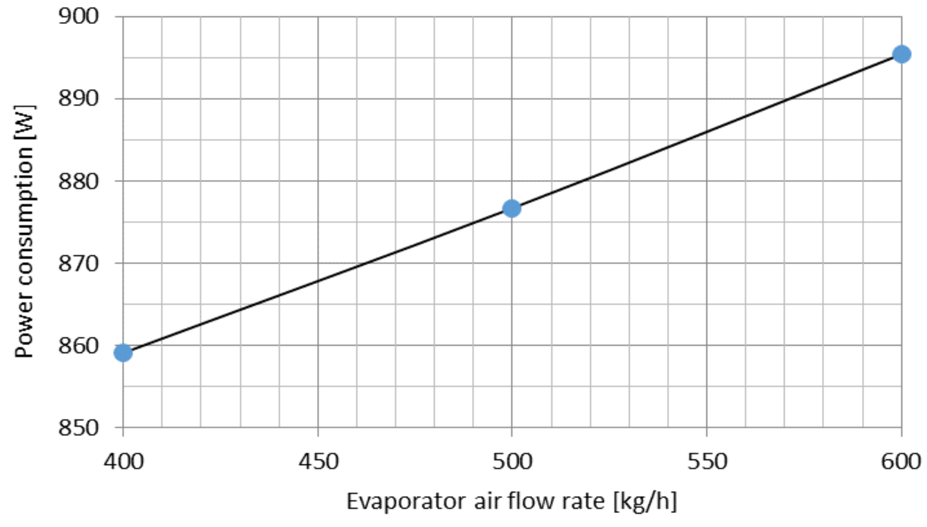

Figure 8. Effect of evaporator air flow rate on the compressor power consumption

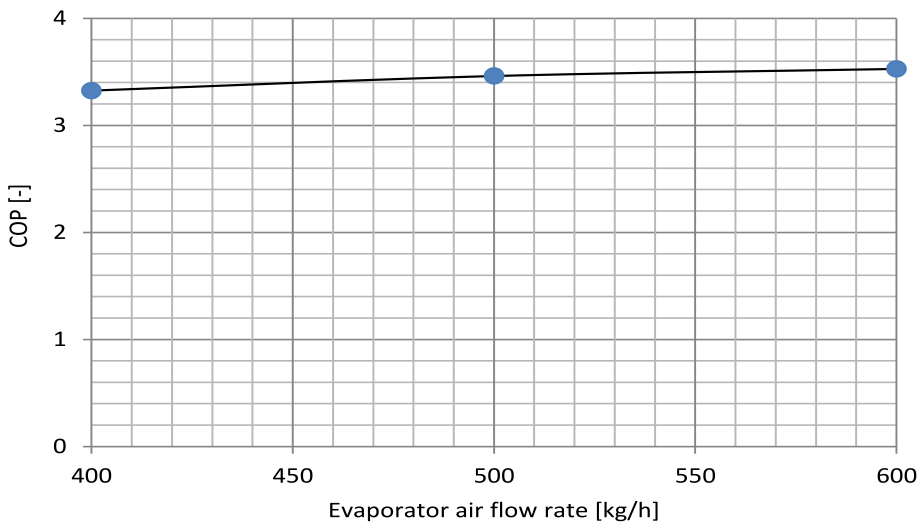

Figure 9. Effect of evaporator air flow rate on the coefficient of performance

In addition to the sensibility analysis, the mathematical model was also employed to investigate the effect of the refrigerant charge on the air-conditioning system performance. This analysis was carried out comparing the evaporator outlet air temperature and the compressor power consumption for different values of refrigerant subcooling at the condenser outlet, which in turn, are related to the system refrigerant charge (SAE, 2008).

The simulations were performed considering the air flow rate of the evaporator equal to 500 kg/h, the inlet condenser air velocity and inlet temperature of 3 m/s and 30°C respectively, the evaporator inlet air temperature of 20°C and the compressor speed of 3000 rpm. Figure 10 shows that as the refrigerant subcooling degree at the condenser exit is increased, the evaporator outlet air temperature initially decrease until it reaches a minimum around 2.8ºC. After this point, the increase in the subcooling, by charging refrigerant to the system, results in an undesired increase of the evaporator outlet air temperature. Moreover Figure 11 shows that the increase of the subcooling degree results in a continuous increase of the compressor power consumption. Therefore, in this analysis, the increase of the power consumption is only compensated by the reduction of the evaporator outlet air temperature until the subcooling reaches 1ºC, which indicates the optimal refrigerant charge for this operating condition.

Although the simulated results obtained in this paper agree with the expected behavior of air-conditioning systems, the authors point out that this model must be improved to consider the latent heat load of the air in the evaporator, the pressure drops on the refrigerant and the air sides and also validated with experimental results.

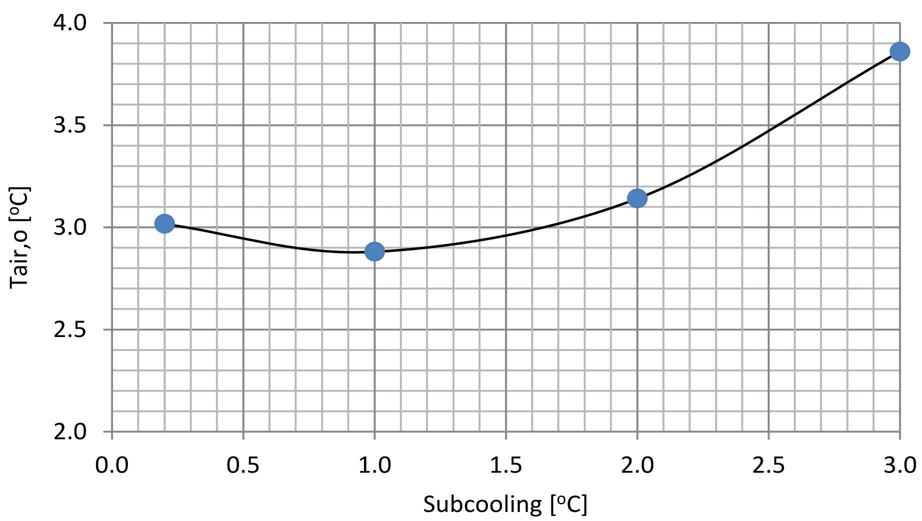

Figure 10. Effect of the subcooling on the evaporator outlet air temperature

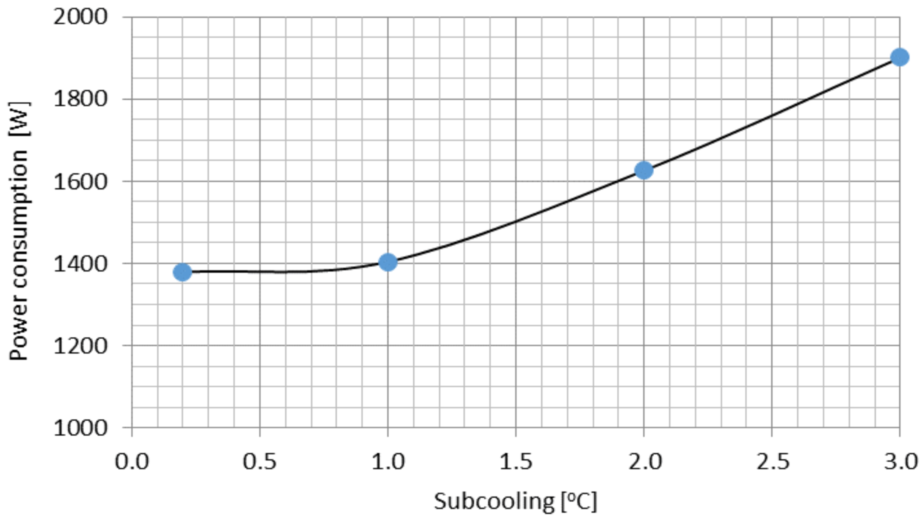

Figure 11. Effect of the subcooling on the compressor power consumption

5. CONCLUSIONS

This paper presents a mathematical model for a TXV-RD automotive air-conditioning system. The model was developed based on the conservation laws and empirical correlations. An iterative procedure, based on the Newton-Raphson technique, was implemented to solve the model, which consists in the simultaneously solution of a set of algebraic equations. A sensibility analysis was carried out by changing the evaporator air flow rate between 400 to 600 kg/h, which alters the evaporator thermal load. The effect of evaporator air flow rate was investigated on the main parameters of the automotive air-conditioning system such as the condenser and evaporators pressures, the refrigerant mass flow rate, the cooling capacity, the compressor power consumption, the evaporator outlet air temperature and the COP. Moreover, the model was used to analyze the effect of the refrigerant charge on the air-conditioning system, when it was identified an optimal operating condition relating the subcooling and the evaporator outlet air temperature. The authors also point out that, in order to improve the accuracy of the presented mathematical model, it is required to consider the latent heat load of the air in the evaporator, the pressure drops on the refrigerant and the air sides and also validate the model with experimental results.

6. ACKNOWLEDGEMENTS

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