25th ABCM International Congress of Mechanical Engineering
October 20-25, 2019, Uberlândia, MG, Brazil

COB-2019-1196
MOVING BOUNDARY MODEL FOR A GAS COOLER OF A CO_2 HEAT PUMP

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Abstract. *Recently, the use of natural fluids such as CO_2 in heat pumps for residential water heating has proved to be an ecologically alternative, especially due to the high global warming potential and ozone depletion layer of traditional fluids such as CFCs and HCFCs. This work aims to develop the mathematical model for the cooler of a solar heat pump operating with CO_2 using the moving boundary modeling. The model was built using the Fortran® programming language based on the equations of conservation of mass, energy and momentum and considering unidimensional, transient and with pressure drop flow. The model provides important variables for the CO_2 heat pump such as the water outlet temperature, the COP of the cycle, the CO_2 and water convection heat transfer coefficients and others variables. The model validation was realized by comparing the theoretical output data with experimental data from the literature. It observes a good agreement between these results, with a maximum deviation of 4.8% for the outlet temperature and 0.5% for the high pressures of the cycle. The comparative study allowed to conclude that the mathematical model presents good agreement results becoming an important tool to predict the dynamic behavior of the heat exchanger.*

Keywords: *dynamic model, moving boundary model, CO_2, gas cooler.*

1. INTRODUCTION

Recently, the modeling of systems is one of the areas of greatest growth in the field of scientific research, mainly in engineering, since it represents an important factor for the understanding of the dynamic behavior of the systems. Mathematical models can save time and capital: through them, it is possible to predict the results of any modifications in mechanical systems, preventing new inadequate equipment from being acquired. In addition, according to Koury et al. (2012), modeling can be a useful tool in optimizing the control of the cooling capacity of a system, and also in studies on the substitution of refrigerants that attack the environment.

The concern about preserving the environment has become institutionalized, and today it is one of the world's priorities. Since 1972, international agreements have been made with the aim of reducing the emission of greenhouse gases. In this way, the selection of refrigerants must take into account the environmental impact that may be caused by them. The CO_2, the fluid selected in this study, is a natural fluid with low global warming potential (GWP) when compared to synthetic fluids, and has a potential for ozone depletion (ODP) equal to zero (Srinivasan et al., 2010).

Sustainable energy sources have also been the subject of scientific research in order to reduce environmental impact. The use of electric showers, for example, for heating water consume large amounts of electricity. In developed countries, especially in Europe, in many households the heating of bath water is given through the heat pump systems. The great differential of this technology is reduced energy consumption, since heat pumps only transfer heat from a cold source to a hot source by pumping a refrigerant (Ralney, 2013).

This paper presents the development and formulation of a mathematical model to simulate the behavior of a heat pump for residential water heating, using CO_2 as refrigerant. The model is validated by comparing the numerical predictions with experimental results obtained from the literature.

2. METHODOLOGY

2.1 Prototype description

Figure 1 shows the prototype CO_2 heat pump of the Cooling and Heating Group (GREA) of the Federal University of Minas Gerais. The equipment was built with the help of the Maxtemper company and includes the gas cooler under study in this work. The heat pump has a tank which stores the hot water coming out of the cooler. The copper heat exchanger is positioned around this water tank. The reservoir and the heat exchanger are thermally insulated with polyethylene.

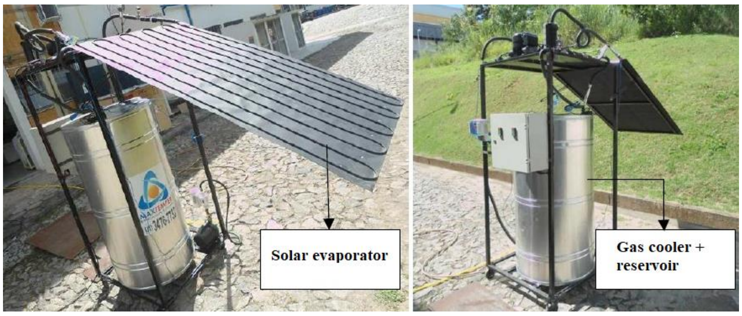


Figure 1. CO_2 heat pump from GREA – UFMG.
Available from: Oliveira (2016).

2.2 Mathematical model

The gas cooler mathematical model was developed to simulate the dynamic behavior of this heat exchanger operating with CO_2. The model needed to receive data from two other models that simulate the conditions for the compression and expansion of CO_2 before and after the gas cooler.

The compressor in this work is modeled in a simplified way. Equation 1 (Braga, 2019) is used to calculated the cycle low pressure (P_{f1}) in Pa, which at time zero is equal to the compressor output pressure (P_{f2}). From approximately t=120s the low pressure becomes permanent (Faria, 2013) and the compressor model becomes stable.

$$P_{f1} = 0.0023t^4 - 1.7356t^3 + 480.1t^2 - 56,800t + 6x10^6 \qquad (1)$$

The mass flow rate imposed by the compressor ($\dot{m}_{comp}$) is determined by Equation 2, where rot, cil, ρ_{f1}, η_v correspond to rotational speed, displacement, density of the fluid at the compressor inlet and volumetric efficiency respectively. The variation in mass flow between the inlet and the outlet of the compressor is not considered. Volumetric efficiency is considered to be 0.7.

$$\dot{m}_{comp} = rot \cdot cil \cdot \rho_{f1} \cdot \eta_v \qquad (2)$$

To simulate the behavior of the expansion device the Equation 3 presented by Martin (2007) is used:

$$\dot{m}_v = a_v C_d \sqrt{(P_{fs} - P_{f1})\rho_{fs}} \qquad (3)$$

where $\dot{m}_v$, a_v, C_d, P_{fs} e ρ_{fs} represent the mass flow rate of the fluid in the device input, area of the cross section of the orifice, discharge coefficient of the expansion device, the pressure and the density of the fluid at the outlet of the gas cooler respectively.

From the input mass flow rate of the expansion device and the mass flow rate of the heat exchanger ($\dot{m}_{fs}$), it is possible to calculate the error between the output mass flow rate of the cooler and the input mass flow rate of the device expansion (e_v):

$$e_v = \frac{\left|\dot{m}_v - \dot{m}_{fs}\right|}{\dot{m}_v} \tag{4}$$

If the difference between the two values is not within the acceptable limit ($e_v < 0.01$) the compressor output pressure must be changed and the calculations are repeated until achieving convergence.

2.3 Gas cooler model resolution method

In order to model the gas cooler the following assumptions are used: heat loss through the thermal insulation is negligible; no axial heat conduction; the axial thermal resistance along the inner tube wall is negligible; the compression and expansion process are adiabatic; helical pressure drop is negligible; inner tube symmetrically centered on the outer tube of the heat exchanger; the properties of the refrigerant are considered in average temperature; CO_2 initial quality in the gas cooler is 50%.

The developed mathematical model is transient, unidimensional and is based on the conservation equations of mass, energy and momentum. The CO_2 and water convection heat transfer coefficients are obtained from heat transfer correlations of the literature. The geometry of the gas cooler, the thermodynamic properties of the fluids (defined from the equations described in Span and Wagner (1996)) and the initial conditions at time zero are input data from the model. Figure 2 presents the gas cooler model flowchart.

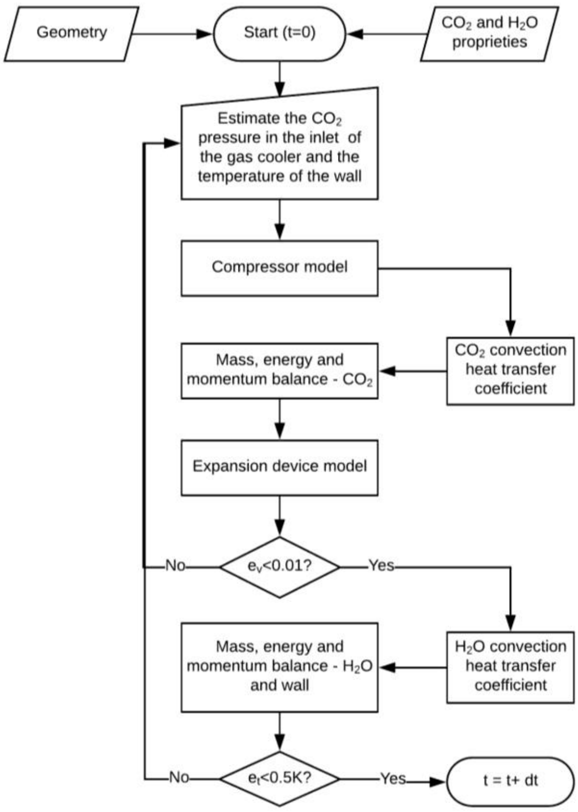


Figure 2. Gas cooler model flowchart.
Available from: Braga (2019).

The heat pump gas cooler consists of two concentric tubes where CO_2 flows in the inner tube and water flows in the outer tube in countercurrent. Initially (t=0), the temperature of CO_2 was considered 25°C and the pressure 6434 kPa (CO_2 saturation pressure at 25°C). Then a value for the CO_2 pressure in the inlet of the heat exchanger (P_{f2}) and a temperature of the water in the inner tube wall (T_p) was arbitrated. With this pressure, and with the pressure and temperature of CO_2 at the compressor inlet, the flow rate and enthalpy of CO_2 were calculated at the compressor outlet. After that, the mass flow rate of CO_2 in the output of this volume control was compared to the mass flowrate of CO_2 calculated from the expansion device model. If there was no convergence, a new CO_2 pressure was taken and the routine calculations were repeated. The model starts the balance equations of water and the wall, with the convergence of the mass flow rate of CO_2. The wall temperature calculated in the external balance (T_{pc}) is compared to the wall temperature estimated in the initial phase of the model (T_p). When the difference between them is less than or equal to 0.5K, the model converges and moves on to the next time step where the variables are recalculated and stored for data processing.

3. RESULTS AND DISCUSSION

3.1 Experimental validation

For the experimental validation of the model, simulations were performed for the same operating conditions of the Paulino (2019) tests, and the model data with the experimental data were confronted after reaching the steady regime. The solar heat pump was experimentally tested in shade and sun. Operation points were validated under both conditions and no significant differences were observed in the analysis of the deviations.

The discharge pressures of the cycle and the water outlet temperature were compared. Figure 3 shows the behavior of the inlet pressure (P_{f2}) and outlet pressure (P_{f3}) of the cooler as a function of the discharge coefficient of the heat pump needle valve (C_d). Figure 4 shows the values of the water outlet temperature (T_{as}) as a function of the opening of the needle valve, by increasing C_d.

It is possible to observe in Fig. 3 a decrease of the cycle high pressures at higher values of Cd. The mean deviation between the cycle high pressures found in the model and in the experimental tests was 0.5%. The mean deviation found in the comparison between theoretical and experimental T_{as} was 4.8%. At points whose cycle high pressures are close to the critical CO_2 pressure, there was a greater deviation between the confronted values of T_{as}. The deviations between the theoretical and empirical data are due to the uncertainties of the measuring instruments used in the experimental tests.

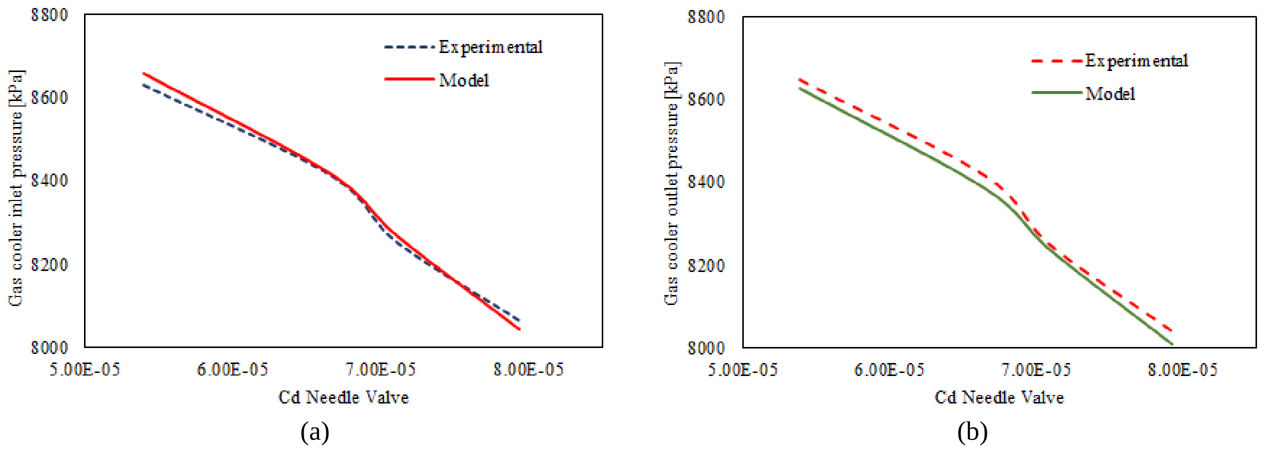


Figure 3. (a) Theoretical and experimental inlet pressure in the cooler due to the increase of C_d. (b) Theoretical and experimental outlet pressure in the cooler due to the increase of C_d.

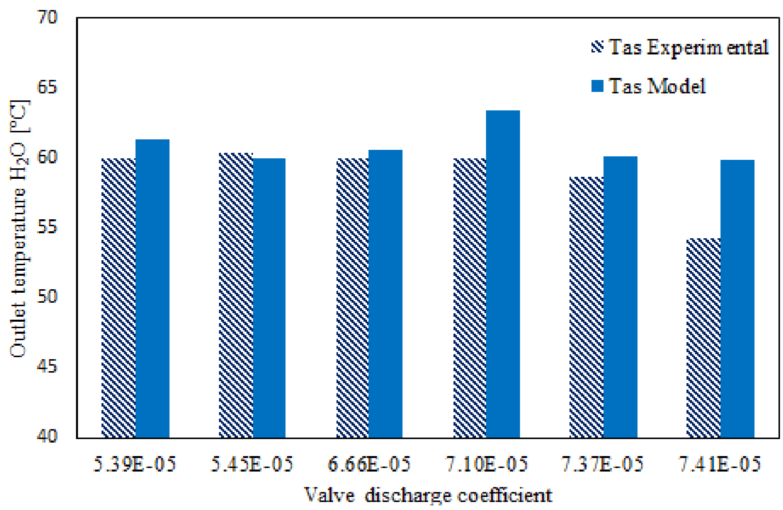
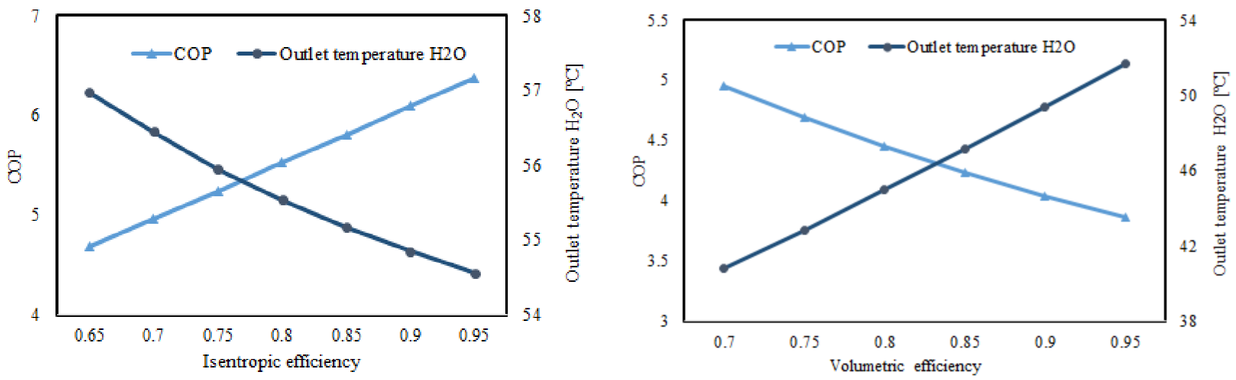


Figure 4. Theoretical and experimental water temperature due to the increase of Cd.

3.2 Influence of the isentropic and volumetric efficiency of the compressor

Figure 5 shows the behavior of the heat pump COP and the water outlet temperature when the isentropic or volumetric efficiency of the compressor is changed.

Figure 5(a) shows that the increase in the isentropic efficiency (η_s) of the compressor allows the increase of the system COP, since the work of the compressor ($\dot{W}$) decreases. There is also a drop in the rate of heat released by CO_2 in the cooler ($\dot{Q}$), but this drop is small in relation to $\dot{W}$. Due to the decrease of the heat exchanged in the gas cooler, T_{as} decreases. By Fig. 5(b) it is possible to notice that the system COP decreases and Tas increases with the increase in volumetric efficiency (η_v). Both $\dot{W}$ and $\dot{Q}$ increase, however, the increase of $\dot{W}$ is higher, resulting in lower COP and higher T_{as}.



(a) (b)

Figure 5. (a) COP and water outlet temperature depending on the isentropic efficiency of the compressor. (b) COP values and the water outlet temperature depending on the volumetric efficiency of the compressor.

3.3 Influence of overheating on compressor suction

Simulations were performed for degrees of superheat in the suction of the compressor (ΔT_{sup}) varying from zero to 9 °C. Figure 6 and Fig. 7 show the pressure versus CO_2 enthalpy plots of the different simulations and the COP plot as a function of the degree of overheating, respectively.

By means of Fig. 6, a decrease of the cycle high pressure, due to the increase of ΔT_{sup}, is noticed. For higher degrees of overheating, ρ_{f1} decreases, causing decrease in $\dot{m}_{f2}$. When $\dot{m}_{f2}$ decreases, the flow at the inlet of the needle valve ($\dot{m}_v$) should decrease, and the only output (ie variable) term from Eq. (3), which determines the value of $\dot{m}_v$, is the high pressure at the cooler output (P_{fs}).

Analyzing Fig. 7 it is possible to notice that the COP increases due to the increase of ΔT_{sup}. However, the increase in COP is attenuated when ΔT_{sup} reaches values higher than 5 °C.

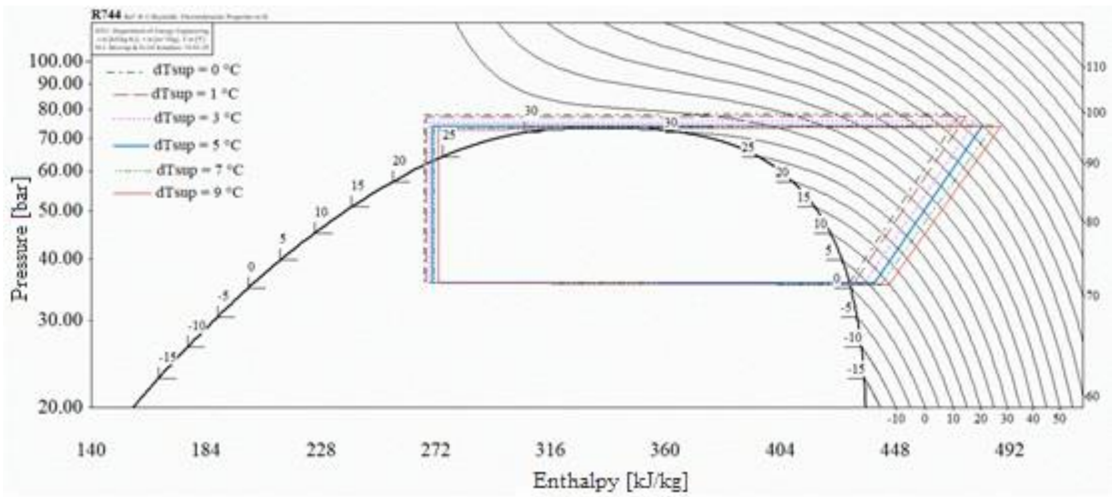


Figure 6. Pressure versus enthalpy diagram for different degrees of overheating in the suction of the compressor.

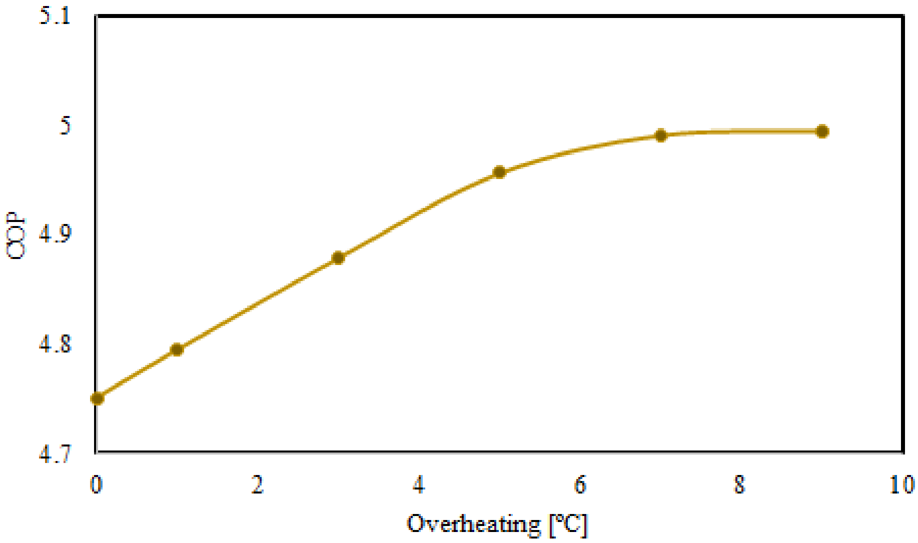


Figure 7. COP versus CO_2 overheating in the suction of the compressor.

3.4 Influence of the suction pressure step of the compressor

The suction pressure of the compressor, ie the low pressure of the cycle, was changed in ±100 kPa and ±300 kPa after the model reached a steady state, in 250s. The results of this step can be seen in Fig. 8.

The cycle low pressure when subjected to negative step, makes the gas cooler inlet pressure lower. The opposite occurs when the step is positive: the inlet pressure of the cooler increases when the low pressure is high. The output data of the gas cooler model maintained the same values for all low pressure steps. That is, in view of new low pressure values, the model responded by changing the high pressure points of the system, and kept the other output data unchanged.

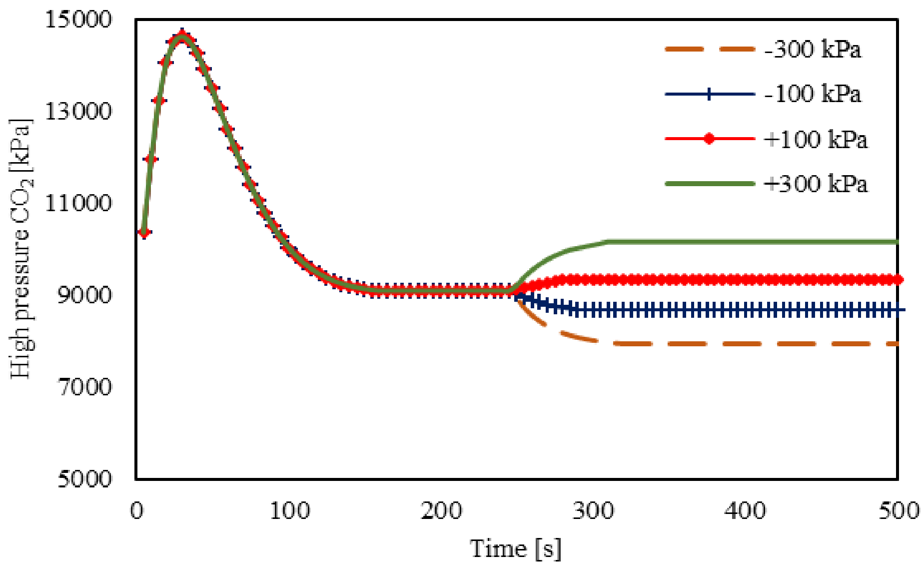


Figure 8. Gas cooler inlet pressure over time, with changes at low pressure at t = 250 s.

3.5 Simulation time

In order to measure the response time of the model simulations, two low cycle pressure steps were programmed, one at t = 250s, where the low pressure suffered a drop of 100 kPa and the other at t = 350s, where pressure down to the previous state. Figure 9 and Fig. 10 show the time required to simulate each time step and the total simulation time, respectively.

It is possible to notice that the simulation time of the model in the first time step is higher than the time of the other steps. This is due to the cooler input pressure estimate, which is not always the closest to the input pressure calculated in the model. In the time steps of 250 and 350s, as the model suffers by a pressure step, the simulation time becomes greater. When the model stabilizes, its simulation time per time step is approximately 2s.

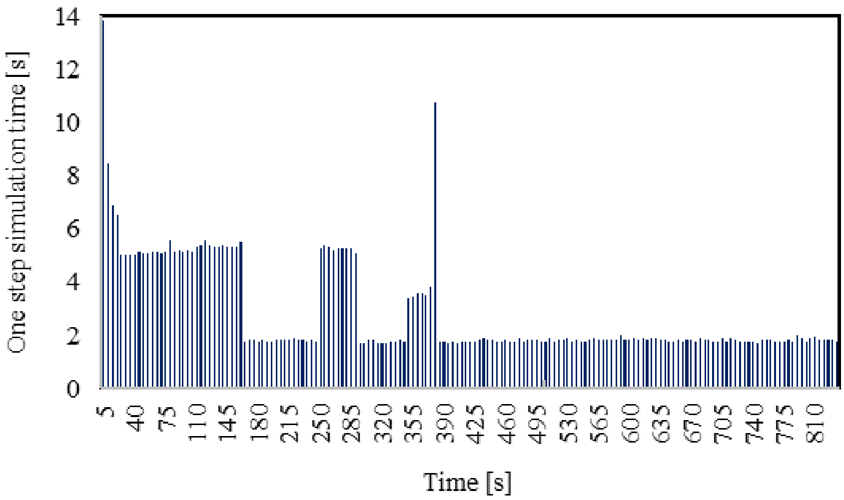


Figure 9. Simulation time of the model per time step.

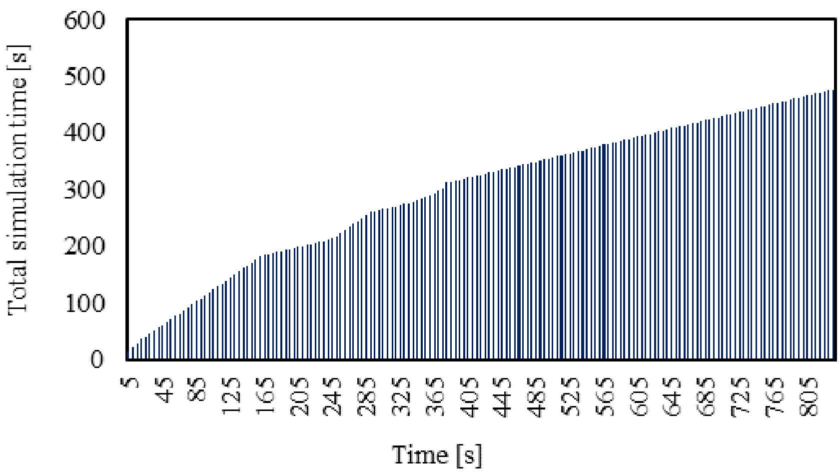


Figure 10. Total simulation time of the model.

4. CONCLUSIONS

In this work a dynamic model for a gas cooler of a CO_2 heat pump was developed using the moving boundary methodology. The model is able to provide important variables such as water outlet temperature of the gas cooler, COP, pressure drop along the gas cooler, the compressor power, the convection coefficients, among others. In order to validate the developed model, the model output data are compared with the data obtained by Paulino (2019), who performed experimental studies on the GREA CO_2 heat pump gas cooler.

Both in this work and in the literature, it was observed that the increase of the passage area of the expansion device causes drop in the pressures of discharge of the cycle. It was observed that changing the volumetric efficiency of the compressor, both the COP of the cycle and the temperature of the water outlet varies. It is also noted that the increase in the degree of overheating increases the COP of the cycle. The comparative study allowed to conclude that there is a good agreement between the model output data and the experimental data, with a maximum deviation of 4.8% for the outlet temperature and 0.5% for the high cycle pressures. Therefore, the developed model is a useful tool to predict the behavior of the gas cooler.

5. ACKNOWLEDGEMENTS

The authors appreciate the support of CAPES, CNPq, and FAPEMIG. We would also like to express our heartfelt gratitude to the late Dr. Ricardo Koury, Professor at Federal University of Minas Gerais and dear friend, who provided his exceptional insight and expertise while assisting us on our research.

6. REFERENCES

Braga, I. F. C. M., *Modelo dinâmico do resfriador de gás de uma bomba de calor solar a CO_2 pelo método de fronteira móvel.* Masters dissertation. Federal University of Minas Gerais, 2019.

Faria, R. N. *Projeto e construção de uma bomba de calor a CO_2 operando em ciclo transcrítico e modelagem dinâmica do conjunto evaporador solar-válvula de expansão*. Doctoral thesis. Federal University of Minas Gerais, 2013.

Hou, Y. et al., "Experimental investigation on the influence of EEV opening on the performance of transcritical CO_2 refrigeration system." *Applied Thermal Engineering*, v. 65, n. 1–2, p. 51–56, 2014.

Koury, R.N.N. et al. "Dynamic model and experimental study of an air-water heat pump for residential use." *International Journal of Refrigeration*, v. 1, p. JIJR 2403, 2012.

Martin, K., Rieberer, R., "Expansion devices for CO_2- results of measurements and simulation model." *International Congress of Refrigeration*, Beijing, 2007.

Oliveira, R. N. et al., "Dynamic model and experimental validation for a gas cooler of a CO 2 heat pump for heating residential water." *Science and Technology for the Built Environment*, v. 22, n. 1, p. 30–40, 2016.

Paulino, T. et al. "Modeling and experimental analysis of the solar radiation in a CO_2 direct-expansion solar-assisted heat pump". *Applied Thermal Engineering*, v. 148, n. October 2018, p. 160–172, 2019.

Ralney, N. F., *Projeto e construção de uma bomba de calor a CO_2 operando em ciclo transcrítico e modelagem dinâmica do conjunto evaporador solar-válvula de expansão*. Doctoral thesis. Federal University of Minas Gerais, 2013.

Span, R., Wagner, W. "A new equation of state for carbon dioxide covering the fluid region from the triple-point temperature to 1100 K at pressures up to 800 MPa". *Journal of physical and chemical reference*, v. 25, 1996.

Srinivasan, K. et al. "Optimum thermodynamic conditions for upper pressure limits of transcritical carbon dioxide refrigeration cycle". *International Journal of Refrigeration*, v. 33, n. 7, p. 1395–1401, 2010.

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