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MODELING A DIRECT-EXPANSION SOLAR ASSISTED HEAT PUMP CONDENSER WITH EMPHASIS ON CONVECTIVE HEAT TRANSFER COEFFICIENTS

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Abstract. Used in several areas of engineering either for heating or cooling, heat exchangers are equipment responsible for the process of the heat exchange between fluids. In this process, knowledge of the convective heat transfer coefficients is very important. The objective of this work is to dimension the coaxial condenser of a Direct-Expansion Solar Assisted Heat Pump (DX-SAHP) used for domestic water heating, emphasizing the methodology used to determine the convective heat transfer coefficient in the single-phase regions – desuperheating and subcooling – and in the two-phase region – condensation – of the heat exchanger. In the single-phase regions, the quality of the refrigerant fluid does not change, therefore the correlation of Gnielinski was used. In these regions, it was considered that the temperature varies linearly over the heat exchanger. However, in the two-phase region, the quality variation must be considered, so the Shah correlation was adopted. The quality was assumed to vary linearly during the fluid condensation. The three control volumes were divided into 100 parts and the convective heat transfer coefficient for each one was determined using an arithmetic average. For the thermal load and pressure conditions considered in this work, the condenser were dimensioned in length of 9.6 m and the propane convective heat transfer coefficients found for the desuperheating, condensation and subcooling regions were 333.9 W/m²-K, 1209.8 W/m²-K and 334.2 W /m²-K, respectively. For the water, the heat transfer coefficient is almost constant in the three regions at about 715.3 W/m²-K.

Keywords: refrigeration, condenser, convective coefficient, heat transfer.

1. INTRODUCTION

In refrigeration and heating, mathematical models help in the sizing of equipment, since they can describe the behavior of a system and the properties of fluids during the thermodynamic processes of a refrigeration cycle, in a faster and more financially viable way in relation to experimental procedures. In addition, modeling is a very useful tool in studies of replacement of refrigerant fluids that harm the environment and in the optimization of the refrigeration capacity of a system.

In recent years, several research works in the area of heat pumps for water heating using fluids with low environmental impact have been developed. Xiao et al. (2020) study a refrigerant mixture with low Global Warming Potential (GWP) and no ozone depletion potential (ODP) as drop-in replacement for R134a in an air source heat pump water heater

(ASHPWH). Duarte et al. (2019) proposed a numerical model for the heat exchangers of a direct-expansion solar assisted heat pump (DX-SAHP) using refrigerants with low GWP. Braga developed a mathematical model for the condenser of a DX-SAHP operating with CO_2.

The objective of this work is to design the condenser of a DX-SAHP operating with propane for domestic water heating and to analyze the convective heat transfer coefficient of the propane along the heat exchanger.

2. METHODOLOGY

The condenser was designed to meet the heating demand corresponding to 200 L of water at 40ºC for domestic use. The heat exchanger chosen is a concentric tube type with fluids flowing in countercurrent, with propane flowing through the inner tube and water in the annular space. The condenser was dimensioned following the methodology applied by Diniz (2017), Faria (2013) and Oliveira (2013). To perform the condenser sizing, some design parameters were considered, such as: the inlet and outlet temperatures of the water, the inlet and outlet temperatures of the propane, the high pressure, the water flow and the internal and external diameters of the tubes. These parameters are shown in Tab. 1.

Table 1. Parameters for sizing the condenser.

Parameters	Description
Type	Concentric tubes with countercurrent fluids
Number of passages	1
Tube material	Copper
Primary fluid – Propane	
High pressure (P_f)	1434 kPa
Critical pressure (P_c)	4247 kPa
Initial temperature (T_{f1})	70 ºC
Final temperature (T_{f4})	37 ºC
Secondary fluid – Water	
Initial temperature (T_{w1})	25 ºC
Final temperature (T_{w4})	40 ºC
Mass flow	60 kg/h
Inner tube – Propane	
Internal diameter (d_i)	7.94 mm
External diameter (d_e)	9.53 mm
External tube – Water	
Internal diameter (D_i)	14.3 mm
External diameter (D_e)	15.9 mm

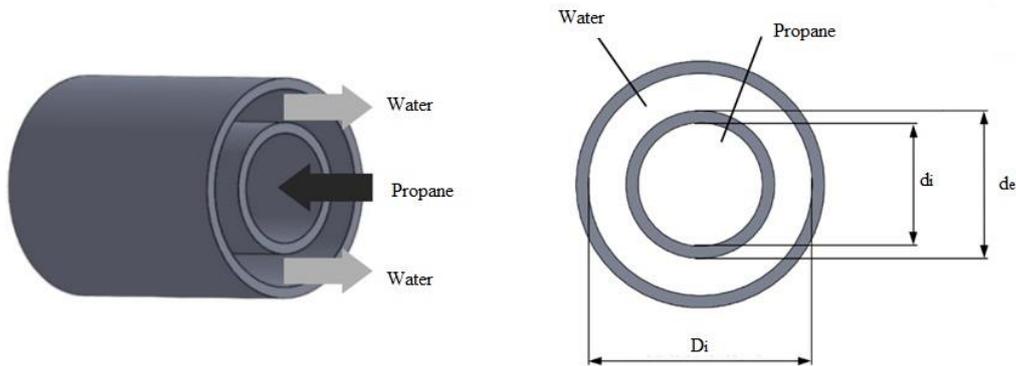The representation of the condenser under study can be seen in Fig. 1.

Figure 1. Condenser representation.

An energy balance was applied in the heat exchanger, considering that the water absorbs all the heat lost by the propane. That is, heat loss to the environment is neglected. Thus, the Eqs. (1) and (2) allows the calculation of the propane mass flow.

$$\dot{Q} = \dot{m}_w c p_w (T_{w4} - T_{w1}) \tag{1}$$

$$\dot{Q} = \dot{m}_f \, (h_{f1} \text{-} h_{f4})$$ (2)

where $\dot{Q}$ is the thermal load rate, $\dot{m}_w$ is the water mass flow, cp_w is the water specific heat, $\dot{m}_f$ is the propane mass flow and h_{f1} and h_{f4} are the propane enthalpies at the inlet and outlet of the condenser, respectively.

Heat transfer in a condenser can be divided into three regimes: desuperheating, condensing and subcooling. Therefore, it is necessary to know the length corresponding to each of these regions to obtain the total length of the heat exchanger, given by Eq. (3).

$$L = L_{ds} + L_{cd} + L_{sc}$$ (3)

where L_{ds}, L_{cd} and L_{sc} are the length of region of desuperheating, condensation and subcooling, respectively.

The desuperheating region consists in the region where the fluid change from overheated steam state to saturated steam. The second region is where the fluid change from saturated steam to saturated liquid. Finally, in the subcooling region the temperature of the fluid reduces until it reaches the liquid phase. The water that is flowing in countercurrent absorbs the heat released by the fluid during each region. Figure 2 shows the temperature variation of propane and water during each step of the heat transfer at the condenser.
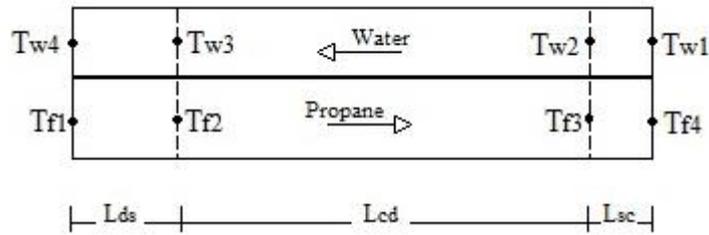

Figure 2. Temperature distribution of propane and water in the three regions of the heat exchanger.

The length of the desuperheating region is determined by Eq. (4).

$$L_{ds} = \frac{\dot{Q}_{ds}}{U_{ds} \, \pi \, d_e \, \Delta T_{ml_{ds}}}$$ (4)

$\dot{Q}_{ds}$ is the thermal rate in the desuperheating region, given by Eq. (5), considering the fluid enthalpies at the inlet and the outlet of the region.

$$\dot{Q}_{ds} = \dot{m}_f \, (h_{f1} \text{-} h_{f2})$$ (5)

The logarithmic average temperature ($\Delta T_{ml_{ds}}$) is calculated according to Incropera et al. (2007) by Eq. (6).

$$\Delta T_{ml_{ds}} = \frac{(T_{f1} - T_{w4}) - (T_{f2} - T_{w3})}{\ln\left(\frac{T_{f1} - T_{w4}}{T_{f2} - T_{w3}}\right)}$$ (6)

T_{w3} is calculated by Eq. (7).

$$\dot{Q}_{ds} = \dot{m}_w cp_w \, (T_{w4} \text{-} T_{w3})$$ (7)

U_{ds} is the overall convective coefficient in the region of desuperheating given by Eq. (8), where k_{tw} is the tube wall thermal conductivity in the region and $T_{wt_{ds}}$ is the wall temperature, calculated by Eq. (9).

$$\frac{1}{U_{ds}} = \frac{d_e}{H_{f_{ds}} d_i} + \frac{d_e ln\left(\frac{d_e}{d_i}\right)}{2k_{tw}} + \frac{1}{H_{w_{ds}}}$$ (8)

$$T_{wt_{ds}} = \frac{H_{fds} \frac{T_{f1}+T_{f2}}{2} + H_{wds} \frac{T_{w4}+T_{w3}}{2}}{H_{fds}+H_{wds}} \tag{9}$$

where H_{fds} and H_{wds} are the are the convective coefficient of the refrigerant fluid and the convective coefficient of water, respectively, in the desuperheating region.

According to Incropera (2007), the convective coefficient of water flowing in annular space is given by Eq. (10) if the flow is laminar and by the Dittus-Boelter equation (Eq.(11)) if it is turbulent flow.

$$H_{w_{ds}} = 5.634 \frac{k_w}{D_h} \qquad \rightarrow \qquad Re_w \leq 2300 \tag{10}$$

$$H_{w_{ds}} = 0.023 Re_w^{0.8} Pr_w^{0.3} \frac{k_w}{D_h} \qquad \rightarrow \qquad Re_w \geq 10000 \tag{11}$$

where Re_w, Pr_w, k_w is the Reynolds number (Eq. (12)), number of Prandtl and the thermal conductivity for water in the region, respectively. D_h is the hydraulic diameter given by Eq. (13). A_w is the cross-sectional area of the water flow between the tubes and μ_w is the dynamic viscosity of water in the region.

$$Re_w = \frac{\dot{m}_w D_h}{A_w \mu_w} \tag{12}$$

$$D_h = D_i - d_e \tag{13}$$

The convective heat transfer coefficient of the fluid is determined by Eq. 14 for laminar flow and for turbulent flow, it is used the Gnielinsky (1976) correlation, given by Eq. (15). It is assumed that the temperature decays linearly along the region. The region is divided into 100 parts and for each temperature a value of Re, Pr and f is calculated and, consequently, a value for H_{fds}. The H_{fds} of the superheating region is determined by an arithmetic average of the values found.

$$H_{f_{ds}} = 4.364 \frac{k_f}{d_i} \qquad \rightarrow \qquad Re_f \leq 2300 \tag{14}$$

$$H_{f_{ds}} = \frac{\left(\frac{f}{8}\right)(Re_f - 1000)Pr_f}{1+12.7\left(\frac{f}{8}\right)^{0.5}(Pr_f^{2/3}-1)} \frac{k_f}{d_i} \qquad \rightarrow \qquad 3000 \leq Re_f \leq 5.10^6 \tag{15}$$

$$f = (0.79 \ln Re_f - 1.64)^{-2} \tag{16}$$

The second region is the condensation region. In this region, there is no change in the temperature of the refrigerant ($T_{f2} = T_{f3}$), it only changes the state. To determine the length of the condensation region, Eq. (17) is used.

$$L_{cd} = \frac{\dot{Q}_{cd}}{U_{cd} \pi d_e \Delta T_{ml_{cd}}} \tag{17}$$

$\dot{Q}_{cd}$ is the thermal rate in the condensation region, given by Eq. (18), considering the fluid enthalpies at the inlet and the outlet of the region. The logarithmic average temperature ($\Delta T_{ml_{cd}}$) is calculated again according to Incropera et al. (2007) by Eq. (19).

$$\dot{Q}_{cd} = \dot{m}_f (h_{f2} - h_{f3}) \tag{18}$$

$$\Delta T_{ml_{cd}} = \frac{(T_{f2}-T_{w3})-(T_{f3}-T_{w2})}{\ln\left(\frac{T_{f2}-T_{w3}}{T_{f3}-T_{w2}}\right)} \tag{19}$$

T_{w2} is calculated by Eq. (20).

$$\dot{Q}_{cd} = \dot{m}_w cp_w \ (T_{w3} - T_{w2}) \tag{21}$$

U_{ds} is the overall convective coefficient in the region of condensation given by Eq. (22), where k_{tw} is the tube wall thermal conductivity in the region and $T_{wt_{ds}}$ is the wall temperature, calculated by Eq. (23).

$$\frac{1}{U_{cd}} = \frac{d_e}{H_{f_{cd}} d_i} + \frac{d_e ln\left(\frac{d_e}{d_i}\right)}{2k_{tw}} + \frac{1}{H_{w_{cd}}} \tag{22}$$

$$T_{wt_{cd}} = \frac{H_{fcd}\,T_{f2} + H_{wcd}\,\frac{T_{w3}+T_{w2}}{2}}{H_{fcd} + H_{wcd}} \tag{23}$$

H_{fcd} and H_{wcd} are the are the convective heat transfer coefficient of the refrigerant fluid and the convective coefficient of water, respectively, in the condensation region. H_{wcd} is determined in the same way as in the desuperheating region, using equations analogous to Eqs. (10), (11), (12) and (13), taking into account the new conditions in this region.

For the calculation of the convective coefficient of the refrigerant (H_{fcd}), the Shah (1979) correlation, given by Eq. (24) is used.

$$H_{fcd} = 0.023 Re_{fl}^{0.8} Pr_{fl}^{0.4} \frac{k_{fl}}{d_i} \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{\left(\frac{P_f}{P_c}\right)^{0.38}} \right] \qquad \rightarrow \qquad 10.8 \leq G_f \leq 1599 \text{ kg/s-m}^2 \tag{24}$$

$$G_f = \frac{\dot{m}_f}{A_f} \tag{25}$$

where G_f is the mass velocity, A_f is the cross section area of the inner tube, x is the quality, Pr_{f_l} and k_{f_l} are the Prandtl number and the thermal conductivity of the refrigerant at the saturated liquid state, respectively. The Reynolds number of the saturated liquid (Re_{fl}) is given by Eq. (26), where μ_{fl} is the dynamic viscosity of the saturated liquid.

$$Re_{fl} = \frac{\dot{m}_f d_i}{A_f \mu_{fl}} \tag{26}$$

However, as the refrigerant fluid quality varies along the length of the analyzed region, the Shah (1979) correlation is applied considering the quality varying linearly. The region is divided into 100 parts and H_{fcd} is calculated using an arithmetic average of the values found.

The methodology for calculating the length of the subcooling region is similar to that used for the desuperheating region, but considering the temperatures corresponding to the last step of the process (shown in Fig (2)). To determine the length of the subcooling region, Eq. (27) is used. The thermal rate in the subcooling region ($\dot{Q}_{sc}$) is determined by Eq. (28) and the logarithmic average temperature ($\Delta T_{ml_{sc}}$) by Eq. (29). The overall convective coefficient (U_{sc}) is given by Eq. (30), where k_{tw} is the tube wall thermal conductivity calculated at the wall temperature ($T_{wt_{sc}}$) determined by Eq. (31)

$$L_{sc} = \frac{\dot{Q}_{sc}}{U_{sc}\,\pi\,d_e\,\Delta T_{ml_{sc}}} \tag{27}$$

$$\dot{Q}_{sc} = \dot{m}_f \ (h_{f3} - h_{f4}) \tag{28}$$

$$\Delta T_{ml_{sc}} = \frac{(T_{f3} - T_{w2}) - (T_{f4} - T_{w1})}{ln\left(\frac{T_{f3} - T_{w2}}{T_{f4} - T_{w1}}\right)} \tag{29}$$

$$\frac{1}{U_{sc}} = \frac{d_e}{H_{f_{sc}} d_i} + \frac{d_e ln\left(\frac{d_e}{d_i}\right)}{2k_{tw}} + \frac{1}{H_{w_{sc}}} \tag{30}$$

$$T_{wt_{sc}} = \frac{H_{fsc} \dfrac{T_{f3} + T_{f4}}{2} + H_{wsc} \dfrac{T_{w2} + T_{w1}}{2}}{H_{fsc} + H_{wsc}} \tag{31}$$

H_{fsc} and H_{wsc} are the are the convective coefficient of the refrigerant fluid and the convective coefficient of water, respectively, in the subcooling region. These coefficients are also calculated in a similar way to the desuperheating region coefficients. The Gnielinsky correlation is used to obtain H_{fsc} and Dittus-Boelter to find H_{wsc}. The region is divided into 100 parts and the temperature is assumed to decay linearly along the region. The convective coefficient is obtained through an arithmetic average of the values found for each of the 100 parts.

3. RESULTS

3.1 Length of the condenser

The length of the coaxial condenser required to operate with propane is 9.6 m, which 2.1 m corresponds to the desuperheating region, 7.0 m to the condensation region and 0.5 m to the subcooling region.

3.2 Analysis of the refrigerant fluid heat transfer coefficients

Figure 3 shows the heat transfer coefficient of propane as a function of fluid temperature across the heat exchanger. It is observed that the coefficient remains practically constant in the subcooling (37 – 41.9ºC) and desuperheating (42.1 – 70ºC) regions, while in the condensation region (42ºC) it reaches its maximum value.

The average convective heat transfer coefficient for the desuperheating, condensation and subcooling regions are 333.9 W/m²-K, 1209.8 W/m²-K and 334.2 W /m²-K, respectively.
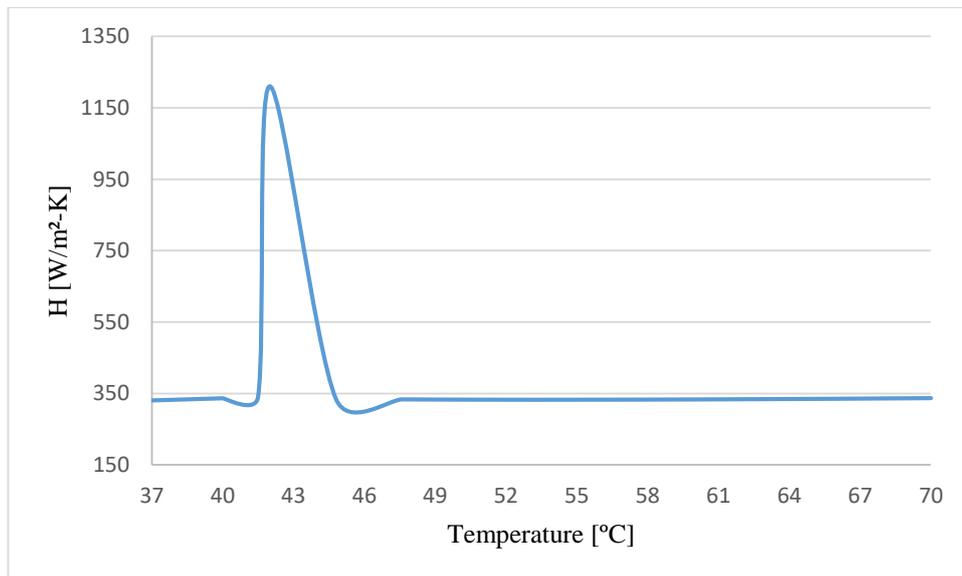

Figure 3. Convective coefficient of propane as a function of temperature.

Figure 4 shows the convective coefficient of propane in the condensation region. In this region, the refrigerant fluid transfers the greatest amount of energy to the water due to the high value of the heat transfer coefficient.
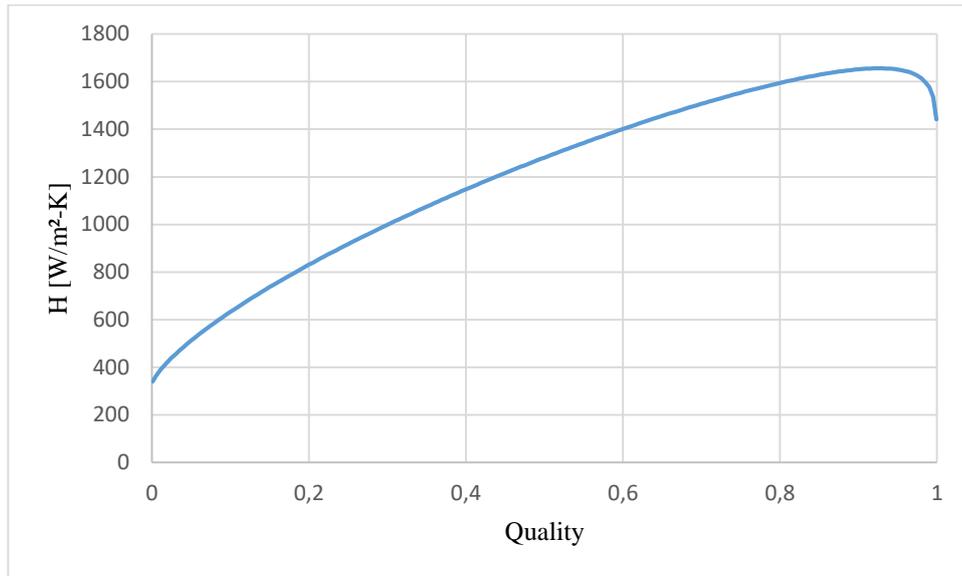
Figure 4. Convective coefficient of propane as a function of condensation region quality.

3.3 Fluid temperature distribution in the condenser

Figure 5 shows the value of water and propane temperatures along the heat exchanger. The length values are out of scale. The figure is just a representation of how the temperature varies along the heat exchanger. As expected, as the condensation region is where the propane transfers more heat to the water due to the high heat transfer coefficient, this is the region where the temperature of the propane decreases the most and, consequently, where the temperature of the water increases the most.
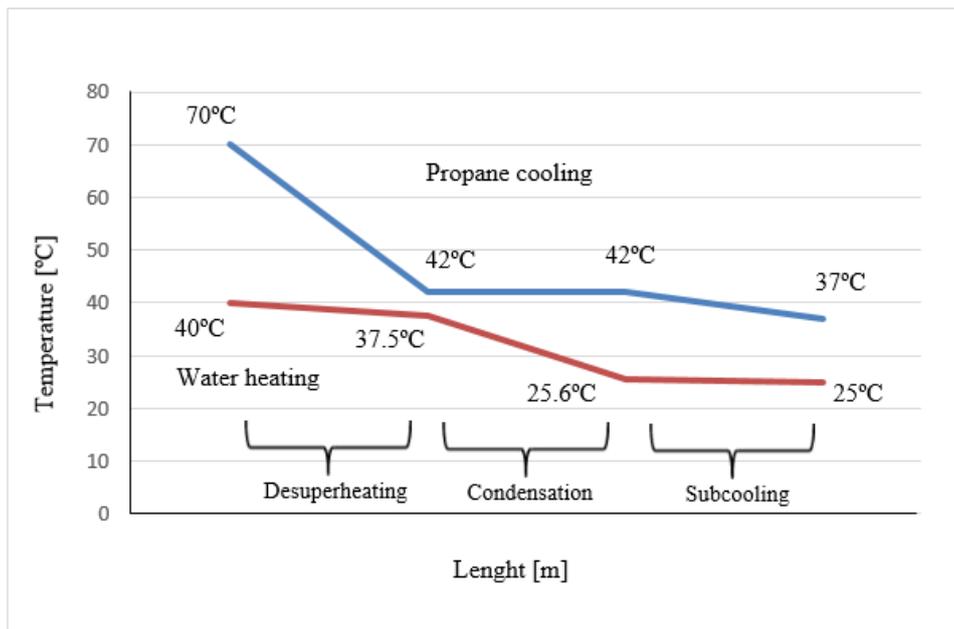

Figure 5. Fluid temperature distribution.

4. CONCLUSIONS

The use of solar heat pumps to heat water for domestic use proves to be an efficient and ecological option. So, the sizing of the heat exchangers is an important step for the implementation of this type of system. In addition, the analysis of the heat transfer coefficients of the fluids along the condenser is also necessary.

For the working conditions adopted in this study, the condenser were dimensioned in length of 9.6 m and the propane convective heat transfer coefficients found for the desuperheating, condensation and subcooling regions were 333.9 W/m²-K, 1209.8 W/m²-K and 334.2 W /m²-K, respectively. For the water the heat transfer coefficient is almost constant in the three regions, at about 715.3 W/m²-k.

Finally, the mathematical model of a coaxial condenser proposed in this work is an important tool that can be used to design heat pumps operating under different conditions and with different fluids.

5. ACKNOWLEDGEMENTS

6. REFERENCES

Braga, I. F. C. M., 2019. *Modelo dinâmico do resfriador de gás de uma bomba de calor solar a CO₂ pelo método de fronteira móvel.* Master's thesis, Post-Graduate Program in Mechanical Engineering, Federal University of Minas Gerais, Belo Horizonte.

Diniz, H.A.G., 2017. *Estudo comparativo da eficiência energética de uma bomba de calor assistida por energia solar operando com condensadores por imersão e coaxial.* Master's thesis, Post-Graduate Program in Mechanical Engineering, Federal University of Minas Gerais, Belo Horizonte.

Dittus, F.W., and Boelter, L.M.K., 1930. *"Heat transfer in automobile radiators of the tubular type"*. University of California Publication on Engineering, vol. 2, No. 13, Berkeley.

Duarte, W.M.; Paulino, T.F.; Pabon, J.J.G.; Sawalha, S. and Machado, L., 2019. "Refrigerants selection for a direct expansion solar assisted heat pump for domestic hot water". Solar Energy, vol. 184, p. 527-538.

Faria, R.N., 2013. *Projeto e construção de uma bomba de calor a CO2 operando em ciclo transcrítico e modelagem dinâmica do conjunto evaporador solar-válvula de expansão.* Ph.D. thesis, Post-Graduate Program in Mechanical Engineering, Federal University of Minas Gerais, Belo Horizonte.

Gnielinski, V., 1976. "New equations for heat and mass transfer in turbulent pipe and channel flow". *International Chemical Engineering*, Vol.16, pp. 359-368.

Incropera, F.P.; Dewitt, D.P.; Bergman, T.L. and Lavine, A.S., 2007. *Fundamentals of Heat and Mass Transfer.* John Wiley and Sons, Hoboken, 6th edition.

Oliveira, R. N., 2013. *Modelo dinâmico e estudo experimental para um resfriador de uma bomba de calor operando com CO₂ para aquecimento de água residencial.* Ph.D. thesis, Post-Graduate Program in Mechanical Engineering, Federal University of Minas Gerais (UFMG), Belo Horizonte, Brazil.

Shah, M. M., 1979. "A general correlation for heat transfer during film condensation insidepipes". *International Journal of Heat and Mass Transfer*, Vol. 22, pp.547-556.

Xiao, B .; Chang, H .; He, L .; Zhao, S. and Shu, S., 2020. "Annual performance analysis of an air source heat pump water heater using a new eco-friendly refrigerant mixture as an alternative to R134a". Renewable Energy, vol. 147, p. 2013-2023.

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