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DIRECT EXPANSION HEAT PUMP CONDENSERS MATHEMATICAL MODELING FOR THE SELECTION OF A REFRIGERANT WITH OPTIMIZED ENVIRONMENTAL PERFORMANCE

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Abstract. *The mathematical modeling of heat exchangers is explored by the scientific community in studies of thermal equipment to evaluate the feasibility of replacing traditional refrigeration fluids with ecological fluids and their impact on the system energy efficiency. The present work aims to present Condensers mathematical modeling (coaxial counter-current and immersed) of a Direct-Expansion Solar Assisted Heat Pump (DX-SAHP) used to residential water heating combined with the selection procedure for a friendly refrigerant. The methodology is based on dimensioning the heat exchanger and evaluating, through computer simulations, the global energy efficiency of the system and the Total Equivalent Warming Impact (TEWI) using the following refrigerants that have low GWP (Global Warming Potential) and ODP (Ozone Depleting Potential) zero: R152a, R744, R1234yf, R1234ze (E), R1233zd (E), R170, R290, R600 and R600a. The main result is the selection of R290 to operate the proposed system. This study has allowed the development of a useful tool for the selection of the appropriate ecological refrigerant of a low cost residential solar heat pump.*

Keywords: *DX-SAHP, TEWI, ecological fluids, condensers, mathematical modeling.*

1. INTRODUCTION

The use of solar heat pumps for water heating has stood out due to the significant energy savings compared to electrical resistance and gas heaters. Within this context, a solar heat pump, with the purpose of heating water for residential use, was designed and built by the Refrigeration and Heating Group (GREa) of the Federal University of Minas Gerais (UFMG), resulting in master's and PhD thesis (Reis, 2012; Rodríguez, 2015; Diniz, 2017).

The replacement of traditional synthetic fluids, HFCs (hydrofluorocarbons), with fluids with low environmental impact, zero ODP and low GWP, has started to be explored more intensely in recent decades by the industry and academic community looking for better energy efficiency and environmental efficiency in cooling and heating systems and equipment. This issue was further intensified in 2016 by the Kigali Amendment (UNEP, 2016) of the Montreal protocol that defined a strict timetable for reducing the production and consumption of HFCs (Ruas, 2018). Among the current refrigeration fluids with low environmental impact, the following stand out: natural fluids, especially carbon dioxide

(CO₂) due to the absence of toxicity in relation to other natural fluids. The CO₂ has the lowest GWP of all refrigerants, being the reference for this index. Moreover, it was one of the first refrigerant fluids applied in refrigeration. It was forgotten for decades by the advent of synthetic fluids, however, has returned to be applied on a large scale due to the restrictions of the Montreal protocols and Kyoto (Faria, 2013; Oliveira, 2013). HCs (hydrocarbons), which have a similar historical context to natural fluids and which are currently being widely applied by the industry and explored by researchers, especially R600a (isobutane) and R290 (propane). And finally, the Hydrofluoroolefins (HFOs), belonging to the fourth generation of fluorinated fluids, especially the R1234yf, developed for the retrofit of the R134a in the air conditioning system of new cars produced in Europe (Bobbo et al., 2014).

Some works in this research segment have already been carried out, evaluating the energy and environmental performance of heat pumps for heating with replacement of traditional fluids by fluids with low environmental impact, according to Table 1.

Table 1. Works on fluid replacement in heat pumps.

Authors	Thermal Systems	Fluids evaluated
Park e Jung (2009)	Heat pump	R170/R290/ R22
Barve e Cremaschi (2012)	Residential heat pump	R32/R1234yf/R410A
Zhang e Muehlbauer (2012)	Residential heat pump	R410A/R134a/R1234yf
Makhnatch e Khodabandeh (2014a); Makhnatch e Khodabandeh (2014b)	Air-water heat pump	R410A/R290/R1270/ R152a/R1234yf
Ju et al. (2017)	Heat pump for water heating	R1233zd(E)/ R22/R134a/ R1270/R290
Nawaz et al. (2017a)	Heat pump for water heating	R1234yf/R1234ze(E)/ R134a
Nawaz et al. (2017b)	Heat pump for water heating	R290/R600a/ R134a
Koyama et al (2018)	Water-to-water heat pump	R32/R1123
Ju et al. (2018a); Ju et al. (2018b)	Water-to-water heat pump	R744/R290/R22
Duarte et al. (2019)	DX-SAHP for water heating	R290/R600a/R744/R1234yf/ R134a
Xiao et al. (2020)	DX-SAHP for water heating	R290/R600a/R131I/R134a
Bai et al. (2020)	DX-SAHP for water heating	R32/R290/R600a/ R134a

The purpose of this article is to present Condensers mathematical modeling (coaxial counter-current and immersed) of a DX-SAHP for residential use and describe the procedure for selecting a friendly refrigerant. This work simulates and critically evaluates energy efficiency, through the performance coefficient (COP) and TEWI, of the following selected fluids: R152a, R744, R1234yf, R1234ze(E), R1233zd(E), R170, R290, R600 e R600a.

2. METHODOLOGY

The mathematical modeling of the heat exchangers will enable the construction of the system. The proposed heat pump prototype is a steam compression heating system, with the primary working fluid being a low GWP and TEWI refrigerant, and as secondary fluids, air in the evaporator and water in the condenser. The machine is basically composed of a hermetic compressor with fixed rotation, a coil-shaped flooded condenser, a countercurrent coaxial condenser, a flat plate solar evaporator and three capillary tubes. In addition, it has a thermal reservoir with a storage capacity of 200 L where the condenser is installed at the bottom by immersion.

Heat exchangers are sized (in this case only the condensers is sized) and the necessary mass of refrigerant for the correct functioning of the system is calculated for the different refrigerants. The EES software (Klein and Alranrado, 2015) was used for the heating pump project and for all fluids and solids properties calculation.

2.1 Environmental performance indicators

Different environmental indicators are used to facilitate the decision-making process for selecting a refrigerant with low global warming potential. The three conventional and most applied environmental indicators in the literature are: GWP, TEWI and life cycle climate performance (LCCP) (Makhnatch and Khodabandeh, 2014b).

The TEWI indicates the global warming impact of direct and indirect emissions and it is calculated as a sum of both: the direct effect of the refrigerant released during the life of the equipment and the indirect impact of CO₂ emissions from fossil fuels used to generate energy to operate the equipment throughout its life depending on the type of energy matrix in the country. The TEWI is simpler to be used as environmental indicator than LCCP and more correct than GWP when selecting a low GWP refrigerant that is environmentally friendly. The TEWI is influenced by the system energy performance. Thus, the efficiency of the cooling or heating system may be the most important parameter in estimating

the system environmental impact (Zhang and Muehlbauer, 2012; Makhnatch and Khodabandeh, 2014b). The TEWI (kg-CO₂) can be calculated using Eq. 1 to Eq. 5 (Antunes and Bandarra Filho, 2016; Paula and Duarte, 2019).

$$TEWI = TEWI_{Direct} + TEWI_{Indirect} \quad (1)$$

$$TEWI_{Direct} = m_f \cdot L_{rate} \cdot L_{time} \cdot GWP + m_f \cdot (1 - \alpha_{TEWI}) \cdot GWP \quad (2)$$

$$TEWI_{Indirect} = E_{Annual} \cdot \beta_{TEWI} \cdot L_{time} \quad (3)$$

$$E_{Annual} = 365 \cdot t_{oper} \cdot \frac{\dot{Q}_{evap}}{COP} \quad (4)$$

$$E_{Annual} = 365 \cdot t_{oper} \cdot \frac{\dot{Q}_{cond}}{COP} \quad (5)$$

On what m_f is the system refrigerant mass (kg), L_{rate} is the annual rate of refrigerant emitted (by replacement and leakage of the system) (%/year), L_{time} is the life of the system (year), α_{TEWI} is the recovery rate of refrigerant life (%), E_{Annual} is the system electricity annual consumption (kWh/year), β_{TEWI} is the carbon dioxide emission factor per kWh of energy produced by the country's energy matrix (kg-CO₂/kWh) and t_{oper} is the equipment daily operation time (h/day). Eq. (4) is applicable to a cooling machine with cooling capacity \dot{Q}_{evap} (kW) and Eq. (5) is applicable to a heat pump with heating capacity \dot{Q}_{cond} (kW). To determine the TEWI of a heat pump operating in Brazil, the values shown in Table 2 were assumed.

Table 2. Values and references for TEWI parameters.

Parameter	Considerations	Reference
$L_{rate} = 12,5\%$	Centralized system, normal operation, catastrophic losses during service and maintenance.	AIRAH (2012); Antunes e Bandarra Filho (2016); Paula e Duarte (2019).
$L_{time} = 15 \text{ years}$	Equipment that operates with economical useful life.	Makhnatch e Khodabandeh (2014a); Makhnatch e Khodabandeh (2014b); Paula e Duarte (2019).
$\alpha_{TEWI} = 70\%$	Refrigerant mass less than 100 kg.	AIRAH (2012); Antunes e Bandarra Filho (2016); Paula e Duarte (2019).
$\beta_{TEWI} = 0,082 \text{ kgCO}_2/\text{kWh}$	Reference value for Brazil.	Antunes and Bandarra Filho (2016).
$t_{oper} = 12 \text{ hours}$	Estimated value in daily use.	Paula e Duarte (2019).
$\eta_{comp} = 0,50$ (overall compressor efficiency)	Value assumed for all refrigerants.	Diniz (2017); Paula e Duarte (2019).

Computer simulations are performed using the EES software (2015) for the ecological refrigerants indicated in Table 1 and refrigerants with GWP below 150, a procedure analogous to the methodology assumed by Makhnatch and Khodabandeh (2014a) and Makhnatch and Khodabandeh (2014b). It should be noted that only refrigerants with ODP equal to zero and available in the thermophysical properties software library were analyzed. The GWP values were consulted in ASHRAE (2017). The following refrigerants were evaluated: R152a, R744, R1234yf, R1234ze (E), R1233zd (E), R170, R290, R600 and R600a. R134a is also analyzed for comparative reasons with the works of Reis (2012), Rodríguez (2015) and Diniz (2017).

2.2 Modeling coaxial condenser and immersion condenser

There are two types of situations in which a DX-SAHP can be used. The first refers for the system operating indoor, inside the laboratory, out of the influence of wind and solar radiation. The second situation refers to when the system operates outdoors, in an open environment exposed to solar radiation and the action of the wind. For sizing the condensers, a scenario with medium environmental conditions to indoor and outdoor conditions was considered. The average values of the parameters involved in the problem are based on the extensive test bench available in the work of Diniz (2017), which performed tests in both conditions with a DX-SAHP (operating with R134a) similar to the one dimensioned in this work. However, the system has two types of condensers, requiring a certain criterion in the definition of the input data for the project. Therefore, the aforementioned averaging procedure is performed for each condenser and then the value considered for the design of the system includes the average value of the two condensers. For sizing the condensers for

the specific case of CO₂ and R170, in this case the correct term is gas coolers, the methodology of Oliveira (2013) was assumed.

The prototype has two types of Condensers installed in parallel, with simple opening / closing valves enabling only one Condenser to operate at time. It is worth mentioning that the DX-SAHP dimensioned in this article has a heating capacity of 900 W. This condition is the starting point for the design of the entire system.

2.2.1 Coaxial condenser

The heat transfer in a condenser takes place in three different stages, corresponding to the refrigerant desuperheating, condensing and subcooling stages. For each of these modes it is necessary to calculate the corresponding length at which the respective process takes place. It should be noted that this analysis is applicable for refrigerants for which the heat exchange process with the hot source takes place in a subcritical regime. To do this, the critical temperature of the fluid is basically observed, it must be higher than the condensing temperature. For the specific case of CO₂, whose critical temperature is 31 °C, the heat exchange of the refrigerant with the hot source takes place at the expense of cooling the CO₂ in a transcritical cycle. Thus, in the gas cooler (suitable nomenclature for this case) the CO₂ remains in the vapor state throughout the cooling process. The same occurs for the R170, because its critical temperature is 32.7 °C. For the sizing of the condensers, main parameters are defined and presented in Table 3.

Table 3. Parameters for sizing the condensers.

Parameters	Characteristics and values
Type	Concentric tubes
Number of passages through the condenser	Only 1
Tube material	Copper
Primary Fluid (refrigerant)	Central pipe
Secondary Fluid	Annular region
Operation mode	Counter-current
Thermal load rate	900 W
Refrigerant initial temperature (Except CO ₂ and R170)	$T_{f_2} = 74 \text{ }^\circ\text{C}$
Refrigerant initial temperature (CO ₂ and R170 only)	$T_{f_2} = 65 \text{ }^\circ\text{C}$
Refrigerant final temperature (Except CO ₂ and R170)	$T_{f_3} = 45 \text{ }^\circ\text{C}$
Refrigerant final temperature (CO ₂ and R170 only)	$T_{f_3} = 30 \text{ }^\circ\text{C}$
Refrigerant condensation temperature (Except CO ₂ and R170)	$T_{f_{cd}} = 52 \text{ }^\circ\text{C}$
Water initial/inlet temperature	$T_{w_{initial}} = T_{w_{in}} = 25 \text{ }^\circ\text{C}$
Water final/outlet temperature	$T_{w_{final}} = T_{w_{out}} = 45 \text{ }^\circ\text{C}$
Subcooling (Except CO ₂ and R170)	$T_{sc} = 7 \text{ }^\circ\text{C}$
High Pressure (CO ₂ only)	$P_{high} = 7600 \text{ kPa}$
High pressure (R170 only)	$P_{high} = 5000 \text{ kPa}$
Internal pipe diameter	$d_{int_f} = 7.94 \text{ mm}$
External diameter of inner tube	$d_{ext_f} = 9.53 \text{ mm (3/8 inch)}$
Internal diameter of external pipe	$D_{int_w} = 14.3 \text{ mm}$
External diameter of external pipe	$D_{ext_w} = 15.9 \text{ mm (5/8 inch)}$
Pipe wall thickness	0.795 mm
Heated water volume	200 L

For the case of the CO₂ refrigerant, Oliveira (2013) highlights the work of Cavallini and Zilio (2007), who recommend that the temperature difference between the fluids at the inlet of the cooler should correspond to a value around 20 °C and, at the outlet, 5°C. Therefore, $T_{f_2} = 65 \text{ }^\circ\text{C}$ and $T_{f_3} = 30 \text{ }^\circ\text{C}$ are assumed. In addition, it is necessary to define the upper pressure, in which the cooling process occurs, = 7600 kPa, according to Oliveira (2013). The same procedure is applied to R170, except for its upper pressure, whose value considered is $P_{high} = 5000 \text{ kPa}$.

The Eq. (1) allows the calculation of the water mass flow \dot{m}_w where c_{p_a} is its specific heat. The refrigerant mass flow rate \dot{m}_f is calculated by Eq.(2), depending on its enthalpies at the inlet h_{f_2} and outlet h_{f_3} of the heat exchanger. It is considered the index "2" related to the condenser inlet (compressor outlet) and the index "3" related to the condenser outlet (expansion device inlet).

$$\dot{Q}_{cond} = \dot{m}_w c_{p_w} (T_{w_{in}} - T_{w_{out}}) \quad (1)$$

$$\dot{Q}_{cond} = \dot{m}_f (h_{f_2} - h_{f_3}) \quad (2)$$

The desuperheating region consists in the region of change of the operating fluid from the overheated steam state to the saturated steam state, i.e. point 2 to point 2' in Chart 1. This chart presents the cooling cycle for the refrigerant R134a, but its configuration is similar for the other fluids in which the heat exchange with the hot source takes place through a subcritical cycle. For the specific case of CO₂, Chart 2 illustrates the transcritical cycle in the upper pressure. The path 2 to 3, passing through 2' and 3' is the path taken by the refrigerant, while the path taken by the water is exactly the opposite. The configuration of this graph is valid for the case of R170 as well. Charts 1 and 2 are based on the consideration that the evaporation temperature is 5 °C with 7 °C of overheating, being these some of the necessary parameters for the evaporator design.

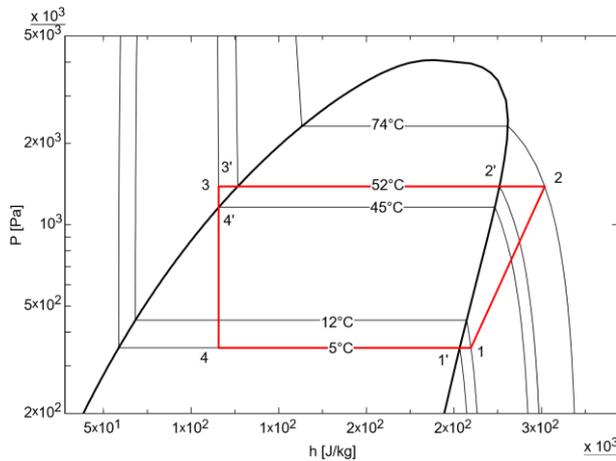


Chart 1: R134a cooling cycle.

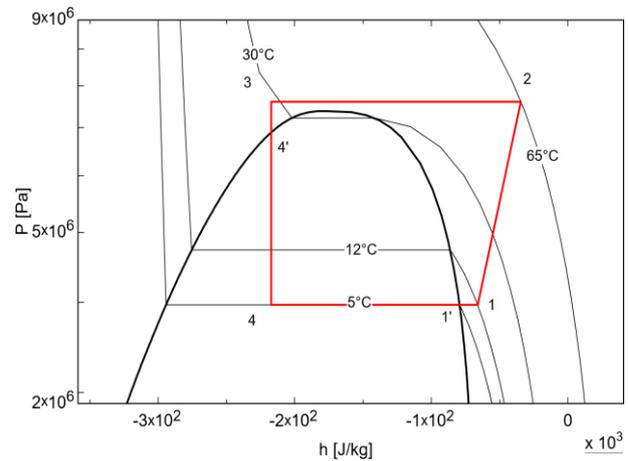


Chart 2: CO₂ cooling cycle.

The refrigerant enters the condenser at point 2 while the water is leaving the condenser. Then, the refrigerant moves to point 2' with temperature $T_{f_2'}$, releasing heat that is absorbed by the water, at this point the water temperature is $T_{w_{out}}$. The thermal charge in the region of desuperheating \dot{Q}_{des} is calculated by an equation analogous to Eq. (2), using the saturated steam enthalpy at temperature $T_{f_2'}$ and then $T_{w_{out}}$ is calculated by an equation analogous to Eq. (1). To determine the length of the desuperheating region L_{des} the Eq.(3) is used. According to Incropera et al. (2007), the concept of logarithmic average temperature $\Delta T_{ml_{des}}$ of counter-current heat exchanger given by Eq.(4) can be applied.

$$\dot{Q}_{des} = U_{des} \pi d_{ext} L_{des} \Delta T_{ml_{des}} \quad (3)$$

$$\Delta T_{ml_{des}} = \frac{(T_{f_2} - T_{w_{out}}) - (T_{f_2'} - T_{w_{out}})}{\ln\left(\frac{T_{f_2} - T_{w_{out}}}{T_{f_2'} - T_{w_{out}}}\right)} \quad (4)$$

Where U_{des} is the overall convective coefficient in the region of desuperheating given by Eq.(5), where k_t is the thermal conductivity of the tube at wall temperature in the region, which in this case is $T_{t_{des}}$ given by Eq.(6). It is important to note that the effect of heat conduction through the wall of the central tube is taken into consideration.

$$\frac{1}{U_{des}} = \frac{1}{H_{w_{des}}} + \frac{d_{ext}}{H_{f_{des}} d_{int}} + \frac{d_{ext} \ln\left(\frac{d_{ext}}{d_{int}}\right)}{2k_t} \quad (5)$$

$$T_{t_{des}} = \frac{H_{f_{des}} \left(\frac{T_{f_2} + T_{f_2'}}{2}\right) + H_{w_{des}} \left(\frac{T_{w_{out}} + T_{w_{out}'}}{2}\right)}{H_{f_{des}} + H_{w_{des}}} \quad (6)$$

Where $H_{f_{des}}$ and $H_{w_{des}}$ are the internal convective coefficient of refrigerant and external convective coefficient of water, respectively, in the region of desuperheating. According to Ghiaasiaan (2008), to determine $H_{f_{des}}$, the correlation Eq. (7) of Dittus and Boelter (1930) is used in case of cooling, considering a turbulent flow with low temperature difference between the wall and the fluid.

$$Nu_{des_f} = 0,023 Re_f^{0,8} Pr_f^{0,3} \quad (7)$$

Where Nu_{des_f} is the Nusselt number given by Eq.(8), valid for Reynolds number of the fluid Re_f higher than 2300 (turbulent). In addition, Pr_f is the Prandtl number and k_f is the thermal conductivity of the refrigerant, in addition Re_f is given by Eq.(9). It should be emphasized that all thermo-physical properties of the fluid, when applying the Dittus and Boelter (1930) correlation, are based on the average temperature of the fluid in the analyzed region where μ_f is the dynamic viscosity of the fluid and A_f is the flow section area of the internal pipe.

$$Nu_{des_f} = \frac{H_{fdes} d_{int_f}}{k_f} \quad (8)$$

$$Re_f = \frac{m_f d_{int_f}}{A_f \mu_f} \quad (9)$$

The external heat transfer coefficient of the water to this region is determined using Eq.(10) if the flow is turbulent or Eq.(11) if the flow is laminar (number of Reynolds less than 2300), besides Eq.(12), where Nu_{des_w} is the Nusselt number for the water flowing through the annular area. It should be noted that Eq.(11) is presented by Incropera et al. (2007) for the case of an isolated tube (external tube) and the other with constant temperature (internal tube). D_{hyd_w} is assumed as hydraulic diameter for water flow given by Eq.(13), taking into consideration the annular area. Where Pr_w is the number of Prandtl and k_w is the thermal conductivity of water in the region. The Reynolds number for water in the region Re_w is given by Eq.(14).

$$Nu_{des_w} = 0,023 Re_w^{0,8} Pr_w^{0,3} \quad (10)$$

$$Nu_{des_w} = 4,43 \quad (11)$$

$$Nu_{des_w} = \frac{H_{wdes} D_{hyd_w}}{k_w} \quad (12)$$

$$D_{hyd_w} = D_{int_w} - d_{ext_f} \quad (13)$$

$$Re_w = \frac{m_w D_{hyd_w}}{A_w \mu_w} \quad (14)$$

Where μ_w is the dynamic viscosity of the water in the region and A_w is the area of the water flow section between the pipes (ring area). The second region is the condensation zone of the refrigerant, which corresponds to the interval between points 2' and 3' of Chart 1. This stage is characterized by the phase change from saturated steam fluid to saturated liquid, transferring a large amount of energy to water \dot{Q}_{cd} , due to the high heat transfer coefficient by condensation. In this phase, the temperature of the refrigerant does not vary, being in points 2' and 3' equal to $T_{f_{cd}}$. The water temperature is $T_{w_{in}}$ when reaching point 3'. To determine the length of the condensation region L_{cd} the Eq.(15) is used. Again the logarithmic average temperature in the region $\Delta T_{ml_{cd}}$ given by Eq.(16) is applied.

$$\dot{Q}_{cd} = U_{cd} \pi d_{ext_f} L_{cd} \Delta T_{ml_{cd}} \quad (15)$$

$$\Delta T_{ml_{cd}} = \frac{(T_{f_{3'}} - T_{w_{in}}) - (T_{f_{2'}} - T_{w_{out}})}{\ln\left(\frac{T_{f_{3'}} - T_{w_{in}}}{T_{f_{2'}} - T_{w_{out}}}\right)} \quad (16)$$

Where U_{cd} is the overall convective coefficient of the refrigerant in the region of condensation given by Eq.(17) and the wall temperature $T_{t_{cd}}$ in the region is given by Eq.(18). The calculation of $T_{w_{in}}$ is shown later, for the subcooling region.

$$\frac{1}{U_{cd}} = \frac{1}{H_{w_{cd}}} + \frac{d_{ext_f}}{H_{f_{cd}} d_{int_f}} + \frac{d_{ext_f} \ln\left(\frac{d_{ext_f}}{d_{int_f}}\right)}{2k_t} \quad (17)$$

$$T_{t_{cd}} = \frac{H_{f_{cd}} T_{f_{cd}} + H_{w_{cd}} \left(\frac{T_{w_{in}} + T_{w_{out}}}{2}\right)}{H_{f_{cd}} + H_{w_{cd}}} \quad (18)$$

Where $H_{f_{cd}}$ is the internal heat transfer coefficient of the refrigerant in the condensing region and $H_{w_{cd}}$ is the external heat transfer coefficient of the water to this region. For the calculation of $H_{f_{cd}}$, Ghiaasiaan (2008) presents the Shah (1979) correlation according to Eq.(19). Ghiaasiaan (2008) states that the Shah (1979) correlation is valid for a wide range of mass flow per area G_f in the range of $10.8 \leq G_f \leq 1599$ kg/s.m², this being given in Eq.(20).

$$H_{f_{cd}} = 0,023 Re_{f_l}^{0,8} Pr_{f_l}^{0,4} \frac{k_{f_l}}{d_{int_f}} \left[(1-x)^{0,8} + \frac{3,8x^{0,76}(1-x)^{0,04}}{P_{red}^{0,38}} \right] \quad (19)$$

$$G_f = \frac{\dot{m}_f}{A_f} \quad (20)$$

Where P_{red} is the reduced pressure defined by the relation between the saturation pressure (corresponding to $T_{f_{cd}}$ and the critical pressure of the refrigerant, x is the vapor equity, and finally Pr_{f_l} and k_{f_l} are, respectively, the Prandtl number and the thermal conductivity of the refrigerant in the saturated liquid state. The Reynolds number of the saturated liquid Re_{f_l} is given by Eq.(21), where μ_{f_l} is the dynamic viscosity of the saturated liquid. The thermophysical properties of the refrigerant relative to the saturated liquid state are related to the condensation temperature.

$$Re_{f_l} = \frac{\dot{m}_f d_{int_f}}{A_f \mu_{f_l}} \quad (21)$$

However, as the equity of the refrigerant varies along the length of the analyzed region, the Shah (1979) correlation is applied considering the equity varying linearly. In this region, the equity was divided in N parts, enough to obtain a stable value (200 divisions are considered). Therefore, the value of $H_{f_{cd}}$ assumed in the project is given by Eq.(22).

$$H_{f_{cd}} = \frac{\sum_{i=1}^N H_{f_{cd}i}}{N} \quad (22)$$

Where the index i represents the i -th variable or parameter analyzed. The $H_{w_{cd}}$ is determined in the same way as in the region of desuperheating, using equations analogous to Eq.(10) to Eq.(14), taking into account the new conditions of that region.

The region of subcooling corresponds to the reduction of the temperature of the refrigerant, corresponding to the level of subcooling, after having reached the liquid phase, from point 3' to 3, according to Chart 1. This step ensures that the expansion device does not receive the fluid in the steam state. The methodology for calculating the length of this region is similar to that used for the desuperheating region, but considering that the working fluid passes from the temperature $T_{f_{3'}}$ to T_{f_3} , which depends on the level of subcooling selected. The water temperature goes from $T_{w_{in'}}$ to $T_{w_{in}}$. The thermal rate in the sub-cooling region \dot{Q}_{sc} is calculated by an equation analogous to Eq.(2) using the enthalpy of the saturated liquid at temperature $T_{f_{3'}}$ and then $T_{w_{in'}}$ is calculated by an equation analogous to Eq.(1). To determine the length of the subcooling region L_{sc} , Eq.(23) is used. The logarithmic average temperature $\Delta T_{ml_{sc}}$ is applied in the region given by Eq.(24).

$$\dot{Q}_{sc} = U_{sc} \pi d_{ext_f} L_{sc} \Delta T_{ml_{sc}} \quad (23)$$

$$\Delta T_{ml_{sc}} = \frac{(T_{f_{3'}} - T_{w_{in'}}) - (T_{f_3} - T_{w_{in}})}{\ln\left(\frac{T_{f_{3'}} - T_{w_{in'}}}{T_{f_3} - T_{w_{in}}}\right)} \quad (24)$$

Where U_{sc} is the overall coefficient in the region of subcooling given by Eq.(25) and the wall temperature $T_{t_{sc}}$ in the region is given by Eq.(26).

$$\frac{1}{U_{sc}} = \frac{1}{H_{w_{sc}}} + \frac{d_{ext_f}}{H_{f_{sc}} d_{int_f}} + \frac{d_{ext_f} \ln\left(\frac{d_{ext_f}}{d_{int_f}}\right)}{2k_t} \quad (25)$$

$$T_{t_{sc}} = \frac{H_{f_{sc}} \left(\frac{T_{f_3} + T_{f_{3'}}}{2}\right) + H_{w_{sc}} \left(\frac{T_{w_{in}} + T_{w_{in'}}}{2}\right)}{H_{f_{sc}} + H_{w_{sc}}} \quad (26)$$

Where $H_{f_{sc}}$ and $H_{w_{sc}}$ are the convective coefficients internal to the refrigerant and external to the water, respectively, in the region of sub-cooling. The internal coefficient of heat transfer of the refrigerant $H_{f_{sc}}$ for that region is determined

in the same way as in the region of desuperheating, using equations analogous to Eq.(7) to Eq.(9). The external heat transfer coefficient of the water H_{wsc} for that region is determined in the same way as in the region of desuperheating, using equations analogous to Eq.(10) to Eq.(14), taking into account the new conditions of that region.

The total length L_{cond} of the condenser is the sum of the lengths obtained for the three regions, given by Eq.(27). The exchange power of the condenser, and consequently the heating power of the Heating Pump using this condenser is given by Eq.(28).

$$L_{cond} = L_{des} + L_{cd} + L_{sc} \quad (27)$$

$$\dot{Q}_{cond} = \dot{Q}_{des} + \dot{Q}_{cd} + \dot{Q}_{sc} \quad (28)$$

For sizing the CO₂ or R170 coaxial gas cooler, Graph 2 helps to understand the process. The heat exchange process of the refrigerant is entirely single-phase and takes place with the steam cooling from point 2 to point 3. It is ideally located at a constant pressure, known as high pressure.

The sizing of the cooler is simpler than the design of the condenser. By Eq.(1) and Eq.(2), the mass flow of water and refrigerant is obtained. To determine the length L_{gc} of the cooler Eq.(29) is used which provides the heat rate exchanged in the \dot{Q}_{gc} . According to Incropera et al. (2007), the logarithmic average temperature concept $\Delta T_{ml_{des}}$ of counter-current heat exchanger given by Eq.(30) can be applied.

$$\dot{Q}_{gc} = U_{gc} \pi d_{ext_f} L_{gc} \Delta T_{ml} \quad (29)$$

$$\Delta T_{ml} = \frac{(T_{f2} - T_{wout}) - (T_{f3} - T_{win})}{\ln\left(\frac{T_{f2} - T_{wout}}{T_{f3} - T_{win}}\right)} \quad (30)$$

Where U_{gc} is the overall convective coefficient given by Eq.(31) and the wall temperature T_t is given by Eq.(32).

$$\frac{1}{U_{gc}} = \frac{1}{H_w} + \frac{d_{ext_f}}{H_f d_{int_f}} + \frac{d_{ext_f} \ln\left(\frac{d_{ext_f}}{d_{int_f}}\right)}{2k_t} \quad (31)$$

$$T_t = \frac{H_f \left(\frac{T_{f2} + T_{f3}}{2}\right) + H_w \left(\frac{T_{win} + T_{wout}}{2}\right)}{H_f + H_w} \quad (32)$$

Where H_f and H_w are the internal convective coefficients of refrigerant and external water, respectively. To determine H_f , Incropera et al. (2007) present the correlation of Gnielinski (1976), according to Eq.(33), as an alternative to the correlation of Dittus and Boelter (1930), which provides a smaller deviation. Gnielinski (1976) is assumed because it is a long length with a greater variation in the thermophysical properties of the refrigerant.

$$Nu_f = \frac{\frac{f}{8}(Re_f - 1000)Pr_f}{1 + 12,7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(Pr_f^{\frac{2}{3}} - 1\right)} \quad (33)$$

Where Nu_f is the number of Nusselt given by an equation analogous to Eq. (8). Eq.(33) is valid for $3000 \leq Re_f \leq 5 \times 10^6$ and $0,5 \leq Pr_f \leq 20000$. The properties should be estimated at the average temperature of the refrigerant. Finally, f is the friction factor and for $Re_f \geq 10000$ is given by Eq.(34). The external heat transfer coefficient of the water is determined using equations analogous to Eq.(10) to Eq.(14).

$$f = (0,79 \ln Re_f - 1,64)^{-2} \quad (34)$$

2.2.2 Immersion condenser

This section presents the equations that allow the design of the condenser by immersion. Some considerations of Oliveira (2013) for the design of the CO₂ gas cooler were assumed. The development is based fundamentally on the procedures assumed by Maia (2007) and Reis (2012) for the sizing of the condenser for the other fluids. The sizing procedure is quite similar to the one performed for the coaxial condenser, and only the differences are highlighted and detailed in this section. The Eq.(35) allows the calculation of the time t needed for heating a certain mass of water m_w .

$$\dot{Q}_{cond} = \frac{m_w c_{pw} (T_{wfinal} - T_{winitial})}{t} \quad (35)$$

The water heating process is permanently transient, starting from the initial temperature until reaching the final temperature. The average value of the water is considered for sizing and simplification of the problem. To determine the length of the desuperheating region the Eq.(36) is used. Where $T_{waverage}$ is the average water temperature given by Eq.(37) and the wall temperature in the region is given by Eq.(38).

$$\dot{Q}_{des} = U_{des} \pi d_{extf} L_{des} \left(\frac{T_{f2} + T_{f2'}}{2} - T_{waverage} \right) \quad (36)$$

$$T_{waverage} = \frac{T_{wfinal} + T_{winitial}}{2} \quad (37)$$

$$T_{t_{des}} = \frac{H_{f_{des}} \left(\frac{T_{f2} + T_{f2'}}{2} \right) + H_{w_{des}} T_{waverage}}{H_{f_{des}} + H_{w_{des}}} \quad (38)$$

To determine $H_{w_{des}}$, Incropera et al. (2007) recommend the correlation of Churchill and Chu (1975) given by Eq.(39). This correlation is appropriate for free convection of a fluid around an isothermal cylinder and is valid for $Ra_w \leq 10^{12}$, where Ra_w is the Rayleigh number for water (fluid in question). The tube is considered divided into three parts, each corresponding to one of the lengths of the characteristic regions of the condenser. Each part of the tube is in an isothermal situation, allowing the application of the above mentioned correlation.

$$Nu_{w_{des}} = \left\{ 0,6 + 0,387 Ra_{w_{des}}^{\frac{1}{6}} \left[1 + \left(\frac{0,559}{Pr_w} \right)^{\frac{9}{16}} \right]^{\frac{8}{27}} \right\}^2 \quad (39)$$

Where $Ra_{w_{des}}$ is Rayleigh's number for water in the region. It should be noted that for the application of the Churchill and Chu (1975) correlation, all thermophysical properties of the fluid must be obtained at the temperature of the film in the region (average between the temperature of the wall and that of the water). Still regarding Eq.(39), $Ra_{w_{des}}$ is given by Eq.(40) and $Nu_{w_{des}}$ is given by Eq.(41).

$$Ra_{w_{des}} = g \beta_w (T_{t_{des}} - T_{waverage}) \frac{d_{extf}^3}{\nu_w \alpha_w} \quad (40)$$

$$Nu_{w_{des}} = \frac{H_{w_{des}} d_{extf}}{k_w} \quad (41)$$

Where β_w , ν_w , α_w and k_w are, respectively, the coefficient of thermal volumetric expansion, kinematic viscosity, thermal diffusivity and thermal conductivity of water. To determine the length of the condensation region, Eq.(42) is assumed. Where the wall temperature in the region is given by Eq.(43).

$$\dot{Q}_{cd} = U_{cd} \pi d_{extf} L_{cd} (T_{f_{cd}} - T_{waverage}) \quad (42)$$

$$T_{t_{cd}} = \frac{H_{f_{cd}} T_{f_{cd}} + H_{w_{cd}} T_{waverage}}{H_{f_{cd}} + H_{w_{cd}}} \quad (43)$$

It was assumed 200 quality values to determine $H_{f_{cd}}$. $H_{w_{cd}}$ is given by equations analogous to Eq.(39) to Eq.(41). To determine the length of the subcooling region, Eq.(44) was assumed. Where the wall temperature in the region is given by Eq.(45) and $H_{w_{sc}}$ is given by equations analogous to Eq.(39) to Eq.(41).

$$\dot{Q}_{sc} = U_{sc} \pi d_{extf} L_{sc} \left(\frac{T_{f3} + T_{f3'}}{2} - T_{waverage} \right) \quad (44)$$

$$T_{t_{sc}} = \frac{H_{f_{sc}} \left(\frac{T_{f3} + T_{f3'}}{2} \right) + H_{w_{sc}} T_{waverage}}{H_{f_{sc}} + H_{w_{sc}}} \quad (45)$$

For the design of the CO₂ or R170 type gas cooler the procedures used for the design of the coaxial cooler are valid. Only the differences are highlighted in this section. With Eq.(35) the time spent for heating a certain mass of water is obtained. To determine the length of the chiller Eq.(46) is used. The wall temperature is given by Eq.(47). Where H_w is given by equations analogous to Eq.(39) to Eq.(41).

$$\dot{Q}_{gc} = U_{gc} \pi d_{ext} L_{gc} \left(\frac{T_{f2} + T_{f3}}{2} - T_{w_{average}} \right) \quad (46)$$

$$T_t = \frac{H_f \left(\frac{T_{f2} + T_{f3}}{2} \right) + H_w T_{w_{average}}}{H_f + H_w} \quad (47)$$

In the use of any of the condensers, all the heat given away by the refrigerant is considered to be fully absorbed by the water. Therefore, all the heat losses to the external environment that occurred in the condensers were neglected. Therefore, the thermal performance of the heat pump when operated with any of the condensers is given by Eq.(48). Where \dot{W}_{comp} is the work performed by the compressor.

$$COP = \frac{\dot{m}_f (h_{f2} - h_{f3})}{\dot{W}_{comp}} \quad (48)$$

2.3 Determination of refrigerant mass

The mass of refrigerant in the heat exchangers and in the equipment as a whole is an impacting factor in determining the TEWI of the system. The calculation of the total mass of fluid in the heat exchangers is divided into two regions: single-phase and two-phase. The single-phase region appears in the condenser in the sections of desuperheating and sub-cooling. In the case of the gas cooler, the whole process takes place by cooling the CO₂ in single-phase. The Eq.(49) allows the determination of the m_{mon} mass in the single-phase regions of the system.

$$m_{mon} = \sum_{i=1}^N \rho_{mon_i} \forall_i \quad (49)$$

Where ρ_{mon_i} and \forall_i are, respectively, the specific mass in the single-phase region and the volume occupied by the refrigerant in each division of the tube. The index i represents the i -th variable. The two-phase region appears in the condenser in the condensation section. The Eq.(50) allows the determination of the mass m_{bip} in the biphasic regions of the system. It is considered $N = 200$ divisions for all biphasic regions of the system.

$$m_{bip} = \sum_{i=1}^N [\alpha_{void_i} \rho_v + (1 - \alpha_{void_i}) \rho_l] \forall_i \quad (50)$$

Where ρ_l and ρ_v are, respectively, the specific masses of the refrigerant in the liquid saturated state and saturated steam. The term α_{void_i} represents the void fraction of the refrigerant in each division of the pipe and can be determined by the Rouhani and Axelsson (1970) modified Steiner (1993) correlation given by Eq.(51) and Eq.(52), where C_o is a parameter of the correlation.

$$\alpha_{void} = \frac{x}{\rho_v} \left[C_o \left(\frac{x}{\rho_v} + \frac{1-x}{\rho_l} \right) + \left(\frac{1,18(1-x)[g\sigma(\rho_l - \rho_v)]^{0,25}}{G_f \rho_l^{0,5}} \right) \right]^{-1} \quad (51)$$

$$C_o = 1 + 0,12(1 - x) \quad (52)$$

Where σ is the refrigerant surface tension and g is the gravity acceleration. The sizing of the heat exchangers makes it possible to determine the required mass when it operates with a given refrigerant. The average of the masses required by the two condensers was considered in order to simplify the simulations. The solar evaporator was also dimensioned and the mass of refrigerant required was also calculated.

3. RESULTS

Table 4 shows the total mass in the heat exchangers and the result of the TEWI and COP. The refrigerants are arranged in ascending order of TEWI.

Table 4. TEWI for Heat Exchangers.

Fluid	T crit. (°C)	P crit. (kPa)	GWP	Mass Heat exchan. (kg)	COP	TEWI direct	TEWI indirect	TEWI
R744	31	7377	1	0.2636	2.90	0.57	1532	1532
R152a	113.3	4517	140	0.1186	2.80	36.12	1582	1618
R1233zd(E)	165.6	3580	1	0.1230	2.18	0.27	2038	2038
R290	96.7	4251	5	0.0758	2.12	0.82	2089	2090
R600	152	3796	4	0.0583	1.98	0.51	2242	2243
R1234ze(E)	109.4	3632	1	0.1467	1.88	0.32	2361	2362
R600a	134.7	3629	20	0.0632	1.88	2.75	2365	2368
R170	32.7	4872	5.5	0.1084	1.85	1.30	2397	2398
R134a	101	4059	1300	0.1625	2.21	459.53	2008	2467
R1234yf	94.7	3382	1	0.1447	1.70	0.31	2607	2607

Regarding the environmental performance of the refrigerants applied in the heat exchangers, the R744 presents itself as the best option, besides providing the best energy efficiency. Then there is R152a, also responsible for the second best COP. In third place comes R1233zd(E), in fourth place comes R290 and in fifth place comes R600. Adopting R134a as the refrigerant to be replaced, R744 presents TEWI 38% better, while R152a, R1233zd(E), R290 and R600 present environmental performance, respectively, 34, 17, 15 and 9% better when applied in heat exchangers.

For refrigerants with very low GWP, the mass has a negligible influence, since it only influences the direct TEWI, and this value is close to 1. The indirect TEWI is preponderantly high in relation to the direct TEWI and depends directly on the COP, being very sensitive to the influence of this parameter. This explains why R744 is the best option, since it provides the best COP even with the largest mass request.

Considering the results of the simulations and from an environmental performance point of view, the R744 is the best choice to operate in a small Heat Pump (heating power around 1 kW). However, from a logistical point of view, no compressor is for sale in the country that operates with this fluid. The same happens with R152a and R1233zd(E). As the design premise is a low cost heat pump and the import of components from other countries makes the manufacture of the system significantly more expensive, these fluids are not considered. Thus, R290 is chosen, having low cost and wide availability in the national market, both in terms of the refrigerant itself, and in terms of compressor and auxiliary components.

3.1 Coaxial condenser design results

The length of the coaxial condenser required to operate with R290 is 5.70 m and the refrigerant mass required is 42 g. The concentric tubes are arranged in a helix shape with an average diameter of 605 mm, totaling 3 turns (number of spirals). The outer tube is covered with thermal insulation composed of a 32.5 mm thick shielded polyethylene sponge tube.

3.2 Immersion condenser results

The length of the immersion condenser required to operate the R290 is 4.55 m. By simulation, the refrigerant mass required by the condenser to operate with R290 is 40 g. Figure 1 presents the coaxial condenser, Figure 2 presents the coaxial condenser section and Figure 3 presents the immersion condenser.



Figure 1. Coaxial condenser.

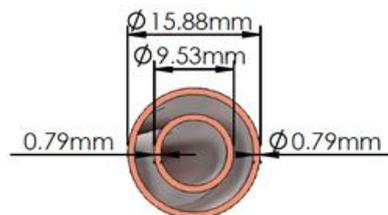


Figure 2. Detail of the coaxial condenser section.



Figure 3. Immersion condenser.

4. CONCLUSIONS

The application of a DX-SAHP in water heating proves to be an optimal sustainable solution and the best alternative among heat pumps for this purpose. The sizing of the heat exchangers and the determination of the mass required by them and the rest of the system enables the analysis of the environmental and logistic performance (relative to the construction of the system) of 10 selected refrigerants (R134a, R152a, R744, R1234yf, R1234ze(E), R1233zd(E), R170, R290, R600 and R600a). As a result, R290 is selected to operate the proposed system.

The coaxial condenser and the immersion condenser are dimensioned in length of 5.70 and 4.50 m, respectively. The refrigerant masses required by these exchangers are 42 and 40 g, respectively.

This work allowed the development of a useful tool to size the condensers, making it possible to select the appropriate ecological refrigerant from a solar heat pump for the required operating conditions. Furthermore, the application of this tool provides the analysis of the thermal and environmental performance of a DX-SAHP and its retrofit, simulating the application of different refrigerants.

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