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STUDY ON THE MAXIMUM PRESSURES OF A SOLAR-ASSISTED R744 DIRECT EXPANSION HEAT PUMP FOR WATER HEATING.

Jéssica Cristina Campos Miranda Silva
Luiz Machado
Willian Moreira Duarte
Post-graduate Program in Mechanical Engineering, Federal University of Minas Gerais (UFMG), Belo Horizonte, MG, Brazil
jc-cms@ufmg.br,willianmoreira@ufmg.br,luizm@ufmg.br

Abstract. *The use of heat pumps to heat water instead of electric heaters is a way to reduce energy consumption and consequently greenhouse gas emissions. In this context, carbon dioxide (CO2 or R744) as a refrigerant fluid has drawn the attention of several researchers in the refrigeration field. Several works in the literature evaluated the performance of the heat pump, economic performance, energy performance, exergy performance, influence of the geometry in the solar evaporator, but no work presents the behavior of the pressure in the evaporator. In this context, this paper presents a mathematical model for the heat pump evaporator (DX-SAHP) that evaluates the behavior of the pressure in the evaporator when it is not operating but exposed to the sun and the impacts of these pressures on the structural integrity of the components, considering an amount of CO2 mass trapped inside the evaporator varying between 8% and 12%. The meteorological data for solving the mathematical model were taken from INMET website for the day with highest solar radiation in 2022. The maximum R744 pressure for 12% mass inside the evaporator was around 10MPa whereas the maximum work pressure recommended by tube manufacture is 13.2MPa.*

Keywords: *Numeric model, DX-SAHP, pressure in the evaporator, Solar Assisted Heat Pump, R744 (CO2).*

1. INTRODUCTION

In Brazil, the energy sector is preeminent. Community and industrial growth, linked to the lack of natural resources has triggered the search for new ways to generate energy that are alternatives, less burdensome on the environment and new ways to transform energy that have a better use. According to research carried out by the Energy Research Company do Brasil (2021), in Brazil the use of renewable energies has been growing gradually, solar energy has increased by 61.5% between 2019 and 2020, which corresponds to a total of 4.2% of the renewable energies used. Despite the increase in solar energy as a source of residential energy, it still far behind other energy sources such as sugarcane derivatives, wind and biodiesel, which in 2020 reached a percentage of 48.4% of the domestic energy supply.

One of the alternatives found in the literature to generate energy with a lower environmental impact are heat pumps that generate thermal energy integrated with energy available in the environment, such as solar or geothermal. Several researches has been done to improve the performance of heat pumps using solar energy for different configurations (Buker and Riffat, 2016). It can be said that the publishing of the first studies on the heat pump are a direct consequence of the attempt to improve the population's quality of life, generating the least impact on the environment. According to Buker and Riffat (2016) the first studies that clarify the functioning of an SAHP came from Sporn and Ambrose (1955) and Jordan and Threlkeld (1954); since then, several other studies have been produced to contribute to the advancement of this technology.

A solar-assisted heat pump is basically composed of four elements: collector-evaporator (combined in direct series types), compressor, storage heat exchange tank, and thermal expansion valves. The system operation of a heat pump assisted by solar energy absorb the heat transferred by the solar energy, this absorbed energy is transmitted as heat to tubes in the absorber which transfers it to the storage tank, then returns to the compressor where the refrigerant gas returns to its initial state, completing the cycle. The refrigerant gas undergoes compression and expansion during the process, causing it to receive and release energy in an SAHP.

In this context, a direct expansion solar powered heat pump (DX-SAHP) is similar to the Rankine cycle coupled to a solar collector acting as an evaporator (Chyng *et al.*, 2003a), this means that the refrigerant gas undergoes a expansion inside the evaporator, thus absorbing heat through solar energy. A heat pump is considered direct expansion when the refrigerant gas absorbs heat directly in the evaporator, making the solar collector have the same function as the evaporator. Chaturvedi *et al.* (2014) propose a theoretical analysis where they present the ideal performance for a heat pump and collector using a bare collector and a variable frequency compressor, provided it is in a temperature range of 5 - 10ºC above ambient temperature, under favorable solar conditions.

The different types of heat pumps can be classified by the types of collectors, flat plates, encapsulated, types of refrigerant gas, etc. Currently, heat pumps that use thermal collectors have shown better results at a lower cost. Equally important in the performance of heat pumps are the climatic conditions to which they are subjected. Kuang *et al.* (2003) present results in their work where the performance of the heat pump is influenced by solar radiation, temperature of the external environment and the speed in the compressor. Direct expansion heat pumps present better results than conventional heat pumps (SAHP), such as thermodynamic performance, lower system cost and longer collector life (Kuang *et al.*, 2003).

In the literature there are several heat pump structures, Buker and Riffat (2016) present several types of heat pumps making a comparison between them. Among the pumps that were proposed, the authors selected three types that showed better performance for heating water at low temperatures: direct expansion heat pump DX-SAHP; indirect expansion heat pump (IX-SAHP), with association in parallel, as the collector is in parallel with the heat pump and indirect expansion heat pump (IX-SAHP), combined in series, the solar collector is in series with the heat pump. In IX-SAHP with parallel association, the water that is stored in the tank is pumped to the solar collector, which in turn receives heat through solar radiation. On days when the intensity of solar radiation is low, an air heat pump can be used to heat the water tank.

In IX-SAHP with series association, the collector heats the fluid, passes through the evaporator that removes heat from the water and the condenser produces hot water. it is necessary to remove heat from the environment, thus favoring the COP of the heat pump. When using a solar collector, the evaporation temperature increases, which ultimately reduces the COP of the heat pump.

In the direct expansion heat pump (SAHP) the refrigerant fluid passes directly to the solar collector. One of the advantages of using this configuration is the reduction of system components. In this configuration it is possible to couple the condenser to the heat pump, placing a tube at the bottom of the tank, so the fluid exchanges natural convection with the water. In this system, the pressure in the collector will be higher than in the IX-SAHP thermal collector. The evaporator is a component responsible for carrying out the heat exchange between the cold source and the refrigerant gas during expansion, causing it to pass from the liquid and vapor state to the superheated vapor state. This is composed of a serpentine-shaped tube integrated into a flat plate, this configuration favors the receipt of heat through solar radiation.

The characteristics of the refrigerant gas impact not only on the performance of the heat pump, but also on the consequences for the environment. Since it was established by the Montreal Protocol, 1996, which establishes a worldwide reduction in the emission of CFC gases, gases that harm the ozone layer. One of the viable options for replacement is the CO2 refrigerant fluid which, in addition to being abundant in nature, has a punctual property in relation to other refrigerants due to its low saturation temperature (Islam *et al.*, 2012).

In 1983, Professor Gustav Lorentzen proposed the use of CO2 as a natural refrigerant gas for refrigeration due to its low environmental impact, with GWP (Global Warming Potential) being equal to 1, and advantages offered by being replaced by CFC gases. However, CO2 is a greenhouse gas, responsible for global warming, and can change the environment if there is a change in its concentration in the atmosphere. Islam *et al.* (2012) presents satisfactory results when using CO2 refrigerant gas, showing an increase in COP when decreasing the compressor speed by 60%. CO2 has considerably higher vapor pressure than other gases, making its cooling capacity 3 to 10 times greater than CFC gases; this pressure near the critical point results in a smaller temperature change for a given pressure change. Thus, the temperature change associated with the pressure drop in the evaporator will become smaller (Kim *et al.*, 2004).

de Oliveira *et al.* (2016) present a dynamic model of a heat pump for heating residential water and conclude that CO2 has a relatively high volumetric cooling capacity, about five times greater than R-22. Rabelo et al., (2019) discuss the effects of opening the expansion valve in a DX-SAHP heat pump system where they use CO2 as a refrigerant gas, in this work they reach the conclusion that with the increase of the thermal radiation of 48 W/m2 to 715W/m2, the COP of the system has increased by 21%, also observed some effects such as: the high pressure and compressor outlet temperature decrease, the high pressure and the mass flow of CO2 increase and the difference of enthalpy in the evaporator and condenser decrease.

In this work, the refrigerant gas used will be CO2 (R744), this refrigerant fluid has drawn the attention of several researchers in the refrigeration field. As a natural refrigerant, R744 has zero net impact on climate change. In addition, it is a non-toxic, non-flammable and non-corrosive fluid, it is an inexpensive and readily available fluid (Nekså, 2002). However, one factor that deserves attention when using R744 as a refrigerant for heat pump systems is its low critical temperature value (31.1°C) and its high critical pressure (73.7 Bar), so, to provide heat at higher temperatures, the system must operate in a transcritical cycle.

Most of the studies found in the consulted literature have as main objective the evaluation of the energetic performance of the DX-SAHP. These studies are listed in Tab 2. Four different types of evaporators or collectors were presented in these studies: uncovered flat plate (UFP), covered flat plate (CFP) and photovoltaic thermal hybrid solar collectors (PV-T). The setups had two objectives: to produce domestic hot water (DWH) and/or space heating (SH). and energy performance for the solar assistant heat pump (SAHP). There other studies with other objectives, for example, Rabelo *et al.* (2019) and Chaturvedi *et al.* (2014) present in their works an analysis of the economic performance of a heat pump, Cervantes and Torres-Reyes (2002) and Paradeshi *et al.* (2018) study the exergetic performance, Buker and Riffat (2016) and Li *et al.* (2007) do an analysis of the influence of solar evaporator geometry on the performance of a heat pump. But there is no study on the pressures obtained in a sunny and hot day if the compressor is not running.

Table 1. Studies on energetic performance of DX-SAHP (Adaptade from Rabelo *et al.* (2019)).

Authors	Collector Type	Collector size (m^2)	Refrigerant	COP	Theoretical study	Experimental study	Heating Load	Location	Tank size (L)	Water Temp. (°C)
Chaturvedi and Shen (1984)	UFP	3.4	R12	2.0-3.0	✓	✓	DHW	Norfolk	-	-
Chaturvedi *et al.* (1998)	UFP	3.5	R12	2.5-4.5	✓	✓	SH	Norfolk	-	-
Ito *et al.* (1999)	UFP	3.2	R22	2.0-8.0	✓	✓	DHW	Japan	-	30-60
Torres-Reyes and Gortari (2001)	UFP	4.5	R22	5.6-4.4	✓	✓	SH	Guanajuato	-	-
Hawlader *et al.* (2001)	UFP	3.0	R134a	4.0-9.0	✓	✓	DHW	Singapore	250	55
Chyng *et al.* (2003b)	UFP	1.9	R134a	1.7-2.5	✓	✓	DHW	Taiwan	-	52-56
Kuang *et al.* (2003)	UFP	2.0	R22	4.0-6.0	✓	✓	DHW	Shanghai	150	50
Ito *et al.* (2005)	PVT	1.9	R22	4.5-6.5	✓	✓	SH, DHW	Japan	-	30-60
Kuang and Wang (2006)	UFP	10.5	R22	2.6-3.3	✗	✓	SH, DHW	Shanghai	150	50
Li *et al.* (2007)	UFP	4.2	R22	5.21	✓	✓	DHW	Shanghai	150	50
Xu *et al.* (2009)	PVT	2.3	R22	4.9-5.1	✓	✗	DHW	Nanjing	150	50
Chow *et al.* (2010)	UFP	12	R134a	6.5-10	✓	✗	DHW	Hong Kong	1500	50
Kong *et al.* (2011)	UFP	4.2	R22	5.2-6.6	✓	✗	DHW	Shanghai	150	50
Moreno-Rodríguez *et al.* (2012)	UFP	5.6	R134a	1.7-2.9	✓	✓	DHW	Madri	300	51
Fernández-Seara *et al.* (2012)	UFP	1.6	R134a	2.0-4.0	✗	✓	DHW	-	300	55
Zhang *et al.* (2014)	UFP	4.2	R22	3.5-6	✓	✗	DHW	-	150	50
Sun *et al.* (2015)	UFP	2.0	-	4.0-5.5	✓	✗	DHW	Shanghai	150	55
Deng and Yu (2016)	CFP	2.5	R134a	3.9-6.2	✓	✗	DHW	-	150	55
Kong *et al.* (2017)	UFP	4.2	R410A	5.2-6.6	✓	✗	DHW	-	150	50
Mohamed *et al.* (2017)	UFP	4.2	R407C	5.2-6.6	✓	✓	SH, DHW	Nottingham	200	50
Diniz (2017)	UFP	1.6	R134a	2.1-2.9	✗	✓	DHW	Belo Horizonte	200	45
Rabelo *et al.* (2018)	UFP	1.6	R744	3.5-5.5	✗	✓	DHW	Belo Horizonte	200	45-80
Kong *et al.* (2018)	UFP	2.1	R134a	3.6-5.6	✗	✓	DHW	Qingdao	200	60
Rabelo *et al.* (2019)	UFP	1.57	R744	2.58	✗	✓	DHW	Belo Horizonte	200	25-60
Kong *et al.* (2020)	UFP	2.1	R290	2.1 - 4.4	✗	✓	DHW	Qingdao	200	37.7-54.9
Duarte *et al.* (2021)	UFP	1.57	R744	3.2-5.4	✗	✓	DHW	Belo Horizonte	200	40-80

Based on the study shown above, there are still no works focusing on the behavior of the evaporator when it is turned off and exposed to the sun. The main aspect of this work is, therefore, to analyze the behavior of the evaporator and develop a mathematical model for the heat pump evaporator (DX-SAHP) that evaluates the behavior of the pressure in the evaporator with the compressor off.

2. SYSTEM DESCRIPTION

In Fig.1 an experimental setup of a DX-SAHP system that uses CO2 as a refrigerant is shown in this study is presented. The system consists of a compressor, a concentric tube gas cooler, a thermal expansion valve, a flat plate evaporator, an oil separator and a water reservoir. Together in the figure the operation of the heat pump is schematically represented. The system is designed to produce hot water in a 200 l storage tank in Belo Horizonte, Brazil. in tab.2 the components and parameters of the heat pump are described.

3. MATHEMATICAL MODEL

The mathematical model of the study is based on the energy balance (1st law of thermodynamics) in a solar evaporator of a heat pump. Duffie and Beckman (2013) present the following equation to evaluate the net heat transfer in the solar evaporator:

$$\dot{Q} = AF[S - U_L(T_f - T_a)] \tag{1}$$

where Q is the net heat transfer rate, A is the area of evaporator, F is the efficiency factor, S is the net radiation absolved,

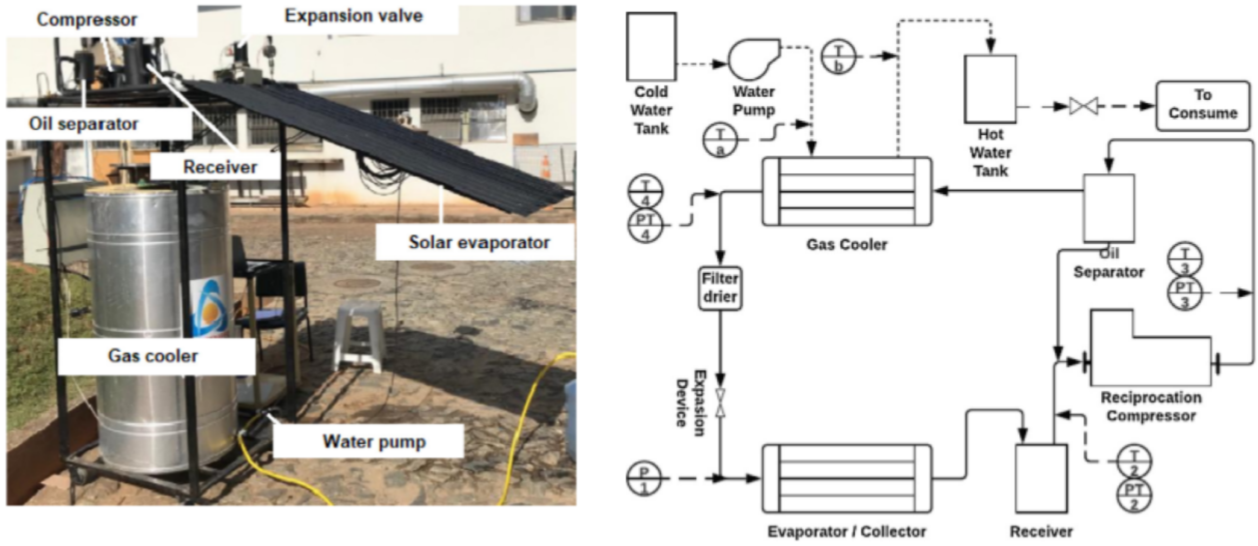

Figure 1. Experimental heat pump CO2 DX-SAHP Duarte *et al.* (2021)

Table 2. Characteristics and parameters of the evaporator.

Parameter	**Solar evaporator**
Tube and fins material	Copper
Tube diameters	$D_o = 6.34mm$ and $D_i = 4.66mm$
Length	L_e = 16.3 m
Distance between tubes	W = 0.1 m
Fin thickness	$\delta = 0.5mm$
Solar absorptance	$\alpha = 0.98$
Plate area	$A = 1.57m^2$
Emissivity	$\varepsilon = 0.95$
Copper conductivity (Incropera *et al.*, 2008)	$k_{cu} = 401W/m/K$
Copper density (Incropera *et al.*, 2008)	$\rho_{cu} = 8933kg/m^3$
Copper heat capacity (Incropera *et al.*, 2008)	$C_{cu} = 383J/kg/K$

T_f is the R744 temperature, T_a is the air temperature. The net radiation absolved is evaluated as made by Kong *et al.* (2017):

$$S = aI - \varepsilon\sigma(\bar{T}_f^4 - T_{sky}^4) \tag{2}$$

where a is the solar absorptance, I is the solar radiation, ε is the emissivity, σ is the Stefan-Boltzmann constant, T_{sky} is the sky temperature. The sky temperature was estimated by the method proposed by Gliah *et al.* (2011) (Eq. 3) using the correlation of Angstrom presented by Berdahl and Fromberg (1982) for sky emissivity (Eq. 4) that is function of the dew point temperature (T_{dp}).

$$T_{sky} = (\varepsilon_{sky} T_a^4)^{1/4} \tag{3}$$

$$\varepsilon_{sky} = 0.734 + 0.0061(T_{dp} - 273.15) \tag{4}$$

The efficiency factor is given by Duffie and Beckman (2013):

$$F = \frac{1/U_L}{W\left[\frac{1}{U_L[D_o+F_a(W-D_o)]} + \frac{1}{C_b} + \frac{1}{\pi D_i h_f}\right]} \tag{5}$$

where U_L is the overall heat transfer coefficient, W is the distance between the collector tubes, D_o is the outer tube diameter, , D_o is the inner tube diameter, F_a is the fin efficiency, C_b is the weld thermal conductance h_f is the transfer coefficient between the pipe wall and the fluid. In the present work the weld thermal resistance and internal convective resistance was neglected as made suggested by Deng and Yu (2016). The fin efficiency is given by:

The fin efficiency can be evaluated by Duffie and Beckman (2013):

$$F = \frac{tanh\left[(W - D_o)/2\sqrt{U_e/(k\delta)}\right]}{(W - D_o)/2\sqrt{U_e/(k\delta)}} \tag{6}$$

where δ is the fin thickness and k is the thermal conductivity. To evaluated U_L the wind-induced convection heat transfer coefficient is described by Kumar and Mullick (2010):

$$h_w = 6.9 + 3.87u_w \tag{7}$$

where u_w is the wind speed. From the balance of energy, the net heat transfer is equal to the energy stored in the system ASHRAE (2013):

$$\dot{Q} = \left(mC\frac{\partial T}{\partial t}\right)_{cu} + \left(m\frac{\partial u}{\partial t}\right)_f \tag{8}$$

where m is the mass, C is the heat capacity, t is time and u is the specific internal energy. The subscript cu and f refers to cooper and R744 properties, respectively. The specific internal energy of R744 cam be obtained using CoolProp Library (Bell *et al.*, 2014) and temperature and density as input. The R744 density is given by:

$$\rho_f = \frac{m_f}{L_e\pi D_i^2/4} \tag{9}$$

where L_e is the length of the evaporator tube. The mass of the cooper is given by:

$$m_{cu} = \left[\left(D_o^2 - D_i^2\right)\pi L_e/4 + A\delta\right]\rho_{cu} \tag{10}$$

The set of equations 1 to 10 is a system of differential equations that can be reduced to an algebraic equation system using the Euler method described Chapra *et al.* (2011). Algebraic equation system is nonlinear and transcendental, so the secant method, also described Chapra *et al.* (2011) as used to find the R744 temperature that give an error (E) lower 0.01% in the following equation:

$$E = \left|\frac{\dot{Q}_8 - \dot{Q}_1}{\dot{Q}_1}\right| \times 100 \tag{11}$$

where the subscripts 1 and 8 refer to the equations 1 and 8.

When the compressor is turned off, a part of the fluid remains inside the tube, this value is estimated by Humia *et al.* (2021), as a value between 10 - 12% of the mass that remains inside the tube. This value varies with the intensity of solar radiation, in this present work the percentage of fluid mass to be considered will be 10%. The mass of R744 in hole heat pump is 645g.

4. RESULTS

The meteorological data for solving the mathematical model were taken from INMET (National Institute of Meteorology), in Belo Horizonte (Pampulha whether station). Solar radiation, humidity and wind speed are factors that can have significant effects on the analysis results. The model was prepared using one of the days with the highest intensity of solar radiation in Belo Horizonte, which occurred on January 28, 2021. The whether data used in the model are shown in Fig. 2 and 3 using the Coordinated Universal Time (UTC) .

The analysis will assume that the pump was turned off, that is, the compressor was turned off and the minimum of refrigerant gas remained inside the evaporator. Murphy and Goldschmidt (1986) presents an expansion device model for optimal pump shutdown, so that the least amount of gas remains in the condenser until the pump restarts. The authors state that when the pump is turned off, due to thermal equilibrium with the ambient temperature, the refrigerant gas that remains in the condenser will slowly vaporize while exchanging heat with the environment. Therefore, the ideal model is one in which the refrigerant is optimally charged for steady-state operation without excess refrigerant, that is, with the least amount of mass possible inside the evaporator. Based on the work of Humia *et al.* (2021) on days with high radiation intensity, the estimated amount of refrigerant gas that remains in the evaporator is around 10%-12%.

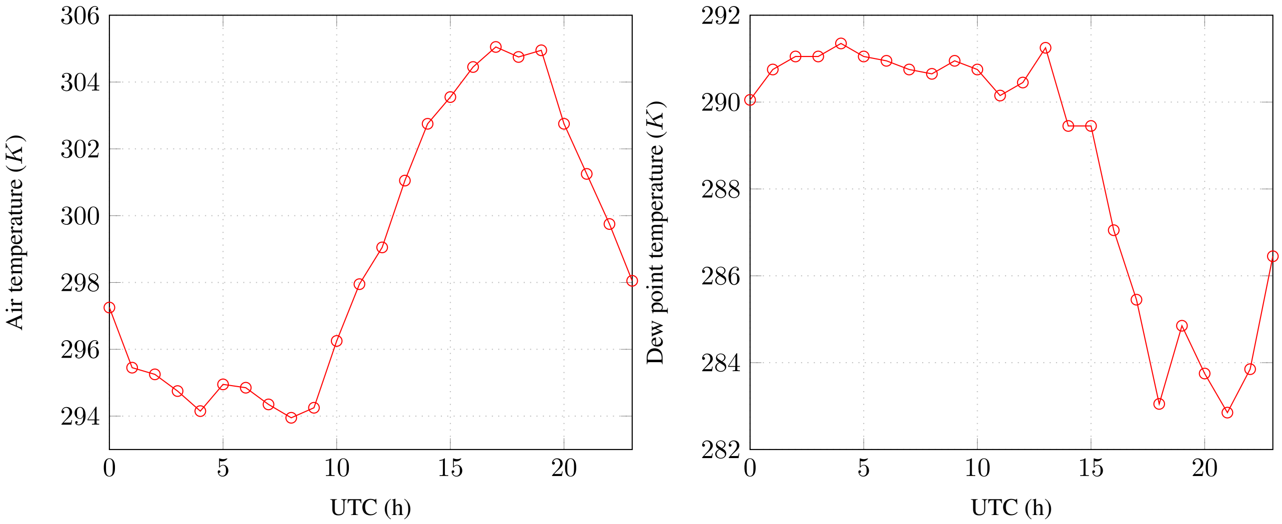

Figure 2. Air temperature and dew point temperature for January 28, 2021

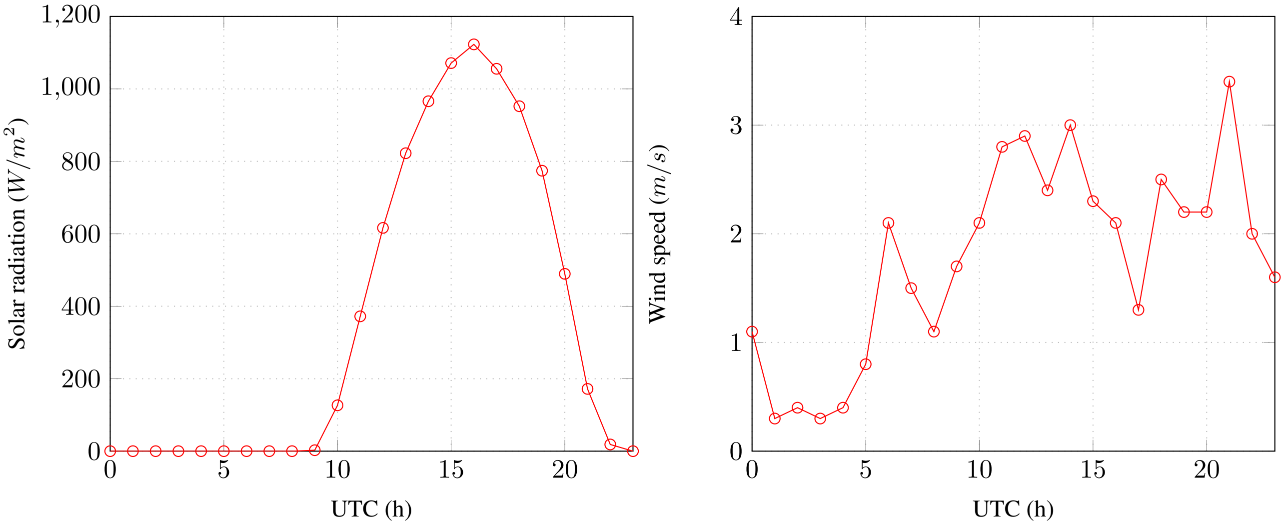

Figure 3. Solar radiation and wind speed for January 28, 2021

4.1 Grid Test

To solve the differential equation system, a mesh test was performed to find the smallest time variation to be considered for the calculation of interaction in the system. In Fig. 4 the mesh test for different dt is presented considering a total simulation time of one hour. Comparing the 22.5 s and 11.5 s simulations there are no significant variations in the result. In fact, the temperature variation in the last two simulations was approximately 0.5 °C, implying an error smaller than 0.06%.

4.2 Effect of solar radiation on the DX-SAHP performance

Initially in the simulation, using the interaction time of 22.5s it is possible to generate results of the amount of solar radiation that reaches the plate during a simulation day. The radiation intensity starts at 0 W/m^2, which is the period of night, at that time the plate temperature was around 293K. After 9 am (UTC), when the sun starts to fall on the plate, the radiation intensity gradually increases to a maximum value of approximately 1100 W/m^2. In Fig. 3 it is possible to notice that the variation of the solar intensity throughout the day is gradual, as well as the decrease of the radiation intensity.

The Fig. 5 presents the results of the pressure behavior throughout the simulation day. In the evaporator, heat exchange between the environment and the fluid takes place in two ways. The first is contact heat exchange, where the heat absorbed by the plate is transferred to the refrigerant gas, the second heat exchange is through convection, where the heat is evenly distributed between the convection waves of the fluid, thus increasing your temperature. In the present study, the temperature of the fluid throughout the day increases to a maximum value of 335 K, around 4 pm (UTC). This temperature value shows slight fluctuations throughout the day, so the fluid pressure also varies, which is a fact attributed

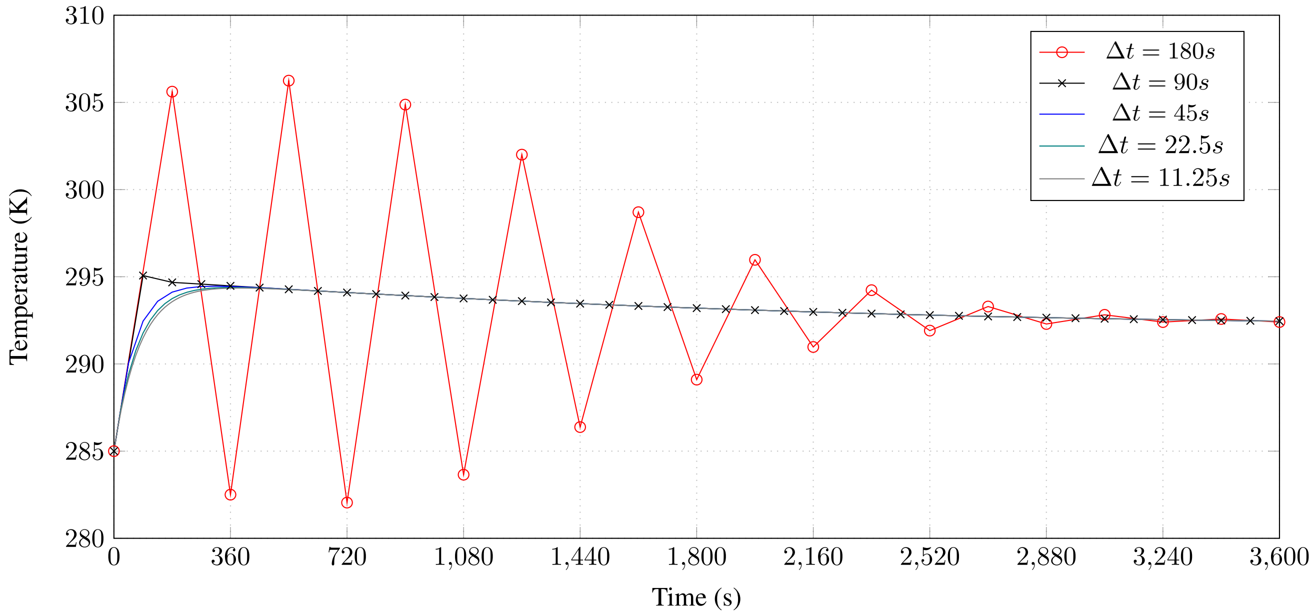

Figure 4. Grid test

to the wind speed, which, being common on summer days in Belo Horizonte, ends up unfavoring the exchange of heat in the plate. The maximum pressure established by the manufacturer that the ¼" copper tube can withstand is 13.2 MPa and the maximum pressure that the refrigerant fluid reaches on the day of the simulation is 9 MPa, which is within the expected.

The analysis of the study occurred as expected, as the pressure remained below what was established by the manufacturer in the analysis of just one day. During the day, the temperature increases to a maximum value of 335 K, and from that point on, the temperature drops. Note that the pressure fluctuates slightly close to its maximum value, this fact is attributed to the wind speed, as it ends up cooling the plate a little throughout the day. According to INMET data, on that day the wind speed was under 4 m/s, with its value varying throughout the day.

Therefore, the pressure of the fluid in the evaporator does not reach critical values when analyzed in just one day of high radiation intensity, but it cannot be said that over several days this fact does not impair the operation of the heat pump and possible plate damage. CO2 has a high critical pressure, so in heat pumps where the refrigerant gas is not CO2, the final result is different from the one presented in this study.

Considering an amount of mass that is trapped inside the evaporator of 8% (52g), 10% (65g) and 12% (72g). Through fig. 5, it is possible to see that by varying the amount of mass stored inside the evaporator, after turning it off, the pressure that the plate reaches also varies. For a mass quantity of 8% inside the evaporator the pressure reaches values considerably lower than for a mass quantity of 12% of the mass of CO2.

5. CONCLUSIONS

In this study, the effects of the solar radiation on a DX-SAHP CO2 heat pump were analyzed using a mathematical model, which was able to predict the value of the fluid pressure in the evaporator and the consequences that the pressure value can have in the DX-SAHP heat pump system.

The maximum pressure value that the fluid reaches when the pump is off is 90 bars. This value is about 66.6% of the maximum total pressure value that the fluid can reach within the manufacturer's standards. Which is an acceptable value within what is expected. This value can be lower if the amount of CO2 mass inside the evaporator is reduced as much as possible.

The maximum pressure is strongly influenced by the charge of refrigerant trapped in the evaporator. If the charge increases the maximum pressure also increases. It is important, therefore, that a proper procedure be done in the shutdown process of the machine so that a minimum amount of mass is trapped in the evaporator and this causes the least probability of damage to the system.

For future studies, other refrigerant gases can be used and evaluated to test whether their behavior in the heat pump, when it is off, will more or less affect the performance of the system. It is important to note that the study was carried out considering only one day that the pump was turned off. A more in-depth study could analyze what would happen if the pump was turned off for several days and what this would imply in the evaporator or even in the system.

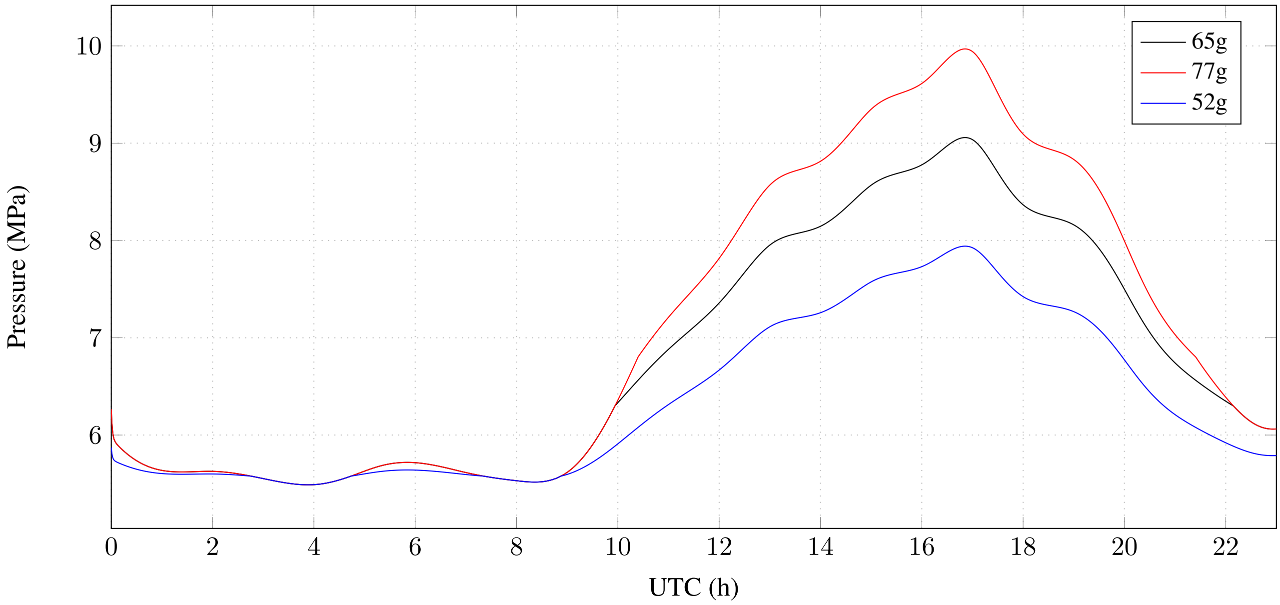

Figure 5. Fluid pressure fluctuation throughout the day for different R744 mass in the evaporator

6. ACKNOWLEDGEMENTS

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