24th ABCM International Congress of Mechanical Engineering
December 3-8, 2017, Curitiba, PR, Brazil

COBEM-2017-2575
THERMAL PERFORMANCE ANALYSIS OF A HEAT PUMP WITH SOLAR EVAPORATOR

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Abstract. *Water heating for residential use is traditionally done using electric resistance in Brazil. However due to the rising electrical energy production costs and consumption, the search for more efficient equipment is important. Among other solutions, is the use of HP (Heat Pump) as a substitute to the electric shower. The purpose of this paper is to analyze the thermal performance (based on experimental trails) of an air-water HP using refrigerant fluid R-134a. This equipment operates with a submerged condenser in a water tank with 200 liters of volume, and a solar evaporator exposed to radiation from the sun. The heat rates of the solar evaporator and its efficiency, as well as the system coefficient of performance (COP) were analyzed. The HP COP was 2.76, considering the average of the five experiments done. The water heating time for 200 liters with an average temperature of 29.2ºC to 45.3ºC was 3 hours and 21 minutes. In addition to that, the evaporator dimensioning was properly done, presenting efficiency of 89%. This work allowed the analysis of the thermal performance of a HP operating under a high availability of thermal energy, which is, exposed to the sun. This situation represents an optimal set up that increases the COP, making it possible to predict the residential operation for most of the days in a region with high sun incidence e climatic regularity.*

Keywords: *Heat Pump, solar evaporator, water heating, R134a, submerged condenser*

1. INTRODUCTION

According to the study of energy efficiency and distributed generation for 2014 to 2024, electric showers were responsible for 16.1% of the electrical energy consumed in Brazilian households in 2014. In 2015, this figure decreased slightly, to 16.0%. This figure is predicted to keep shrinking and it can reach 14.5% in 2019 and 12.2% in 2024 (EPE, 2016a). Table 1 presents data for 2014 and 2015 and projections for 2019 and 2024 of the electrical energy consumption by appliance in the residences of Brazil. There is a perspective of evolution in the percentage of residences with solar water heating until 2024, exemplifying the evolution of this energy source in the Brazilian residential scenario (EPE, 2016b). The lack of solar energy in some days (rainy or cloudy weather) is a restriction to the use of water heating systems that rely on solar collectors only, making the heating process unfeasible. Due to this phenomenon, the shared use of solar collectors and an auxiliary heating system, for days with low solar energy available, becomes necessary. Recent studies have found significant advantages of combining a HP equipment with a solar collector system when environmental conditions are not favorable (Rodríguez et al., 2015).

Table 1: Electrical energy consumption on the Brazilian residential scenario. Source: Adapted from EPE (2016a)

Appliance (GWh)	**2014**	**2015**	**2019**	**2024**
Air conditioner	17126	18658	26230	36216
Refrigerator	22396	22609	23329	24685
Freezer	5964	5865	5508	5428
Lights	18552	18019	12670	9856
Electric shower	21324	21671	22708	23979
Washing machine	2914	2999	3385	3729
Television	19232	20883	23152	25472
Others	24542	24642	39285	67829
Total	**132050**	**135346**	**143597**	**197194**
	16.1%	**16.0%**	**14.5%**	**12.2%**

The predominant way of heating water for showering in Brazil is using the electric shower. Due to the rising cost of producing electrical energy as well as the growing consumption caused by demographic growth, it is important to develop studies about energy efficiency in order to find more efficient equipment (EPE, 2016b). Among the most promising solutions to this issue is using the HP to warm-up residential water as an alternative to the electric shower. This way, the HP can work alone reducing the electrical energy consumption when compared to the electric shower or as auxiliary equipment in a solar collector heating system (Rodríguez et al., 2015; Silva et al., 2007). Results for the monthly Coefficient de Performance (COP) ranging from 1.40 to 2.54, that is, average annual COP of 1.88 were obtained by Rodríguez et al. (2015). Results for the average COP of 2.01 were found by Silva et al. (2007).

Recent studies have found the HP with a solar evaporator to be a beneficial strategy (Buker and Riffat, 2016; Omojaro and Breitkopf, 2013). The evaporator is exposed to solar radiation, therefore, it has higher thermal energy availability when compared to indoor equipment. The COP of the solar exposed HP also becomes higher than the one of the HP with non-exposed, conventional evaporators. As shown in table 1, in 2015, the electric shower was responsible for 16.0% of the total energy consumed. According to recent studies, this figure can decrease significantly if the HP with a sun exposed solar evaporator is used individually. This way, the HP with solar evaporator is able to perform the function of a solar collector regardless of the weather conditions, since it has good efficiency if not exposed to solar radiation and excellent efficiency if exposed to it (Willem et al., 2017; Li et al., 2007a; Li et al., 2007b). Results of the COP from 4 to 4.9 on winter, and from 7 to 9 on summer for the water heating HP operating with CO_2 were found by Willem et al. (2017). Results of the COP of 6.61 with an average seasonal value of 5.25 were discovered by Li et al. (2007a) and Li et al. (2007b). Regarding the efficiency of the solar evaporator used in this type of system, Kong et al. (2011) found the efficiency of the solar collector, also treated as a solar evaporator, to be from 0.88 to 0.91.

The purpose of this paper is to analyze the system thermal performance by experimental trials of an air-water HP with solar evaporator. This equipment operates exposed to the characteristic sun´s radiation of Belo Horizonte, MG, Brazil and works with the refrigerant fluid R-134a as its primary fluid, air as its secondary fluid in the evaporator and water as the secondary fluid in the underwater condenser (serpentine shaped tube submerged in a tank with the water to be heated).

2. METHODOLOGY

In this section all the HP system components are presented, including the instruments used in experimental trials. Additionally, the mathematical models used to determine several parameters of thermal performance of the system are presented.

2.1 Experimental procedures

The HP operates according to the vapor compression cycle, as the refrigeration machines. In this system, the fluid that is the hot source is the water stored in the reservoir and the cold source is the ambient air exposed to solar radiation. The cycle begins when part of the ambient energy flows to the solar evaporator, transferring heat to the fluid that flows in this component. This fluid goes to the compressor, where it receives compression work in the form of heat, has its pressure raised, and then flows to the condenser. In this device, the refrigerant fluid, having received heat from the ambient and the compressor, transfers energy to the water, heating it up and reducing the refrigerant temperature. Afterwards, the fluid is directed to the expansion valve, where its pressure drops, returning to the same state in which it entered the evaporator, completing the cycle.

The analyzed HP has two types of evaporators, a solar one (positioned in a fixed 30º angle) and one with finned tubes and forced ventilation. It also has two types of condensers, a coaxial counterflow one and a submerged one. In addition to that, the HP is equipped with an alternative compressor hermetically sealed (nominal power of 1/3 HP), a thermostatic expansion valve and a backup thermal bottle with refrigerant fluid. The HP circuit has several valves that control whether or not the fluid goes through the heat exchangers in such a way that the system operator manages to select only one evaporator and one condenser to be activated. Finally, the HP has a thermal reservoir (tank), with capacity of 200 liters for storing water. The tank is filled to its maximum volume and the submerged condenser heats the water from around 30ºC up to approximately 45ºC. Figure 1 presents the side view of the HP, equipped with Bourdon type manometers and K type thermocouples, with the main components except the thermal reservoir.

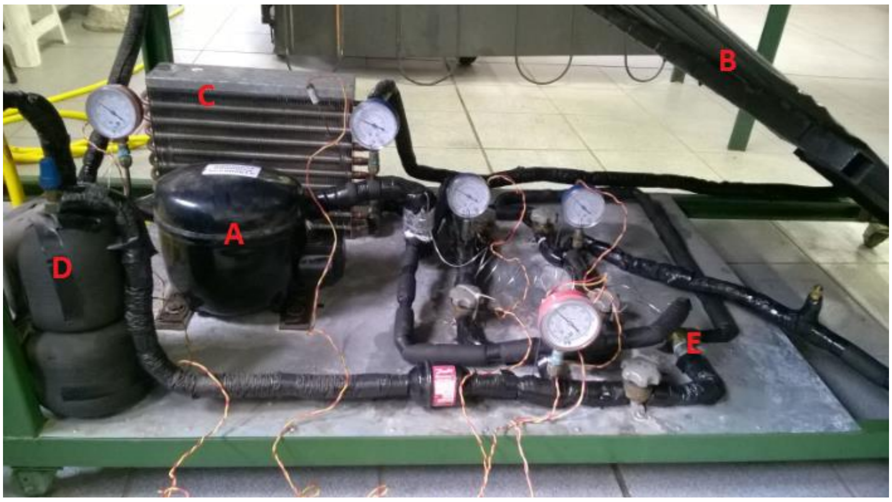
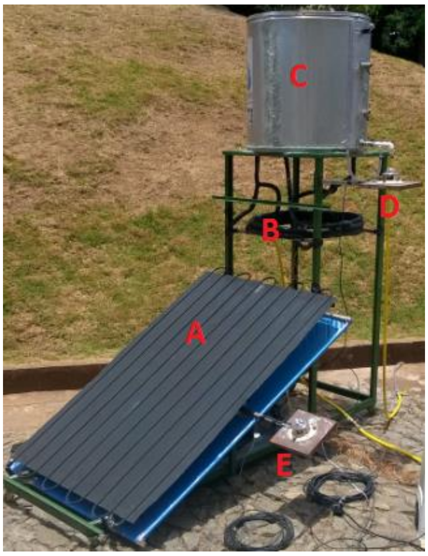


Figure 1: Side view of the HP. (A) Compressor, (B) solar evaporator, (C) evaporator with finned tubes, (D) backup thermal bottle (E) and thermostatic expansion valve.

Figure 2 presents a broader view of the HP, front and side, in addition to the constructive data of the solar evaporator. It shows the thermal reservoir, both the horizontal and the 30° inclined pyranometers (instantaneous solar radiation flow measuring instruments), the coaxial conterflow condenser and the solar evaporator (103x160 mm).



Tube material	Copper
Plate material	Aluminum
Tube internal diameter	8.73 mm
Tube external diameter	9.53 mm
Plate thickness	1 mm
Tube length (Except curves)	16.0 m
Curves length	1.28 m
Plate width	1.03 m
Plate length	1.60 m
Plate area($A_{pl)}$	1.65 m²
Plate emissivity(ε)	0.95

Figure 2: View of the HP and constructive data of the solar evaporator. Solar evaporator (A), coaxial condenser (B), thermal water reservoir (C), horizontal pyranometer (D), and inclined pyranometer (E).

The solar evaporator is a serpertine fin assembly (black plate collector with copper tubes) built to explore natural convection imposed by ambient air, all types of solar radiation (beam, diffuse and ground radiation) and also the condensation latent heat of the water vapor present in the atmospheric air. The plate temperature was registered by three type K thermocouples, two of them closed to its extremities and one positioned in its central region.

Figure 3 shows in detail the bottom of the tank inside (where the submerged condenser is visible) and the component constructive data. A psychrometer was used to determine the ambient temperature and dew point, an energy measuring device to follow the compressor real energy consumption, and an anemometer with blades to measure the wind speed.

Tube material	Copper
Tube internal diameter	8.73 mm
Tube external diameter	9.53 mm
Tube length	4.5 m

Figure 3: Inside view of the bottom of the tank where the serpentine shaped underwater condenser is located and constructive data of the component.

The temperature of the reservoir water was determined by the average values of three different measuring points (botttom, middle and top) with one type K thermocouple in each of them. The time between readings of the variables involved in the system operation was 15 minutes. The parameters measured were: temperatures and pressures of the thermodynamics vapor compression cycle, temperatures of the water, the air, the plate, the dew point, wind speed, incoming radiation flow at the evaporator (also treated as a collector) and the electrical energy consumed by the compressor.

For analysis of results, the uncertainties of measurements were compiled using the manufacturers information for each of the instruments used in the tests. This information is shown on Table 2.

Table 1: Uncertainty of measurement of the instruments used

Measuring instrument	Uncertainty
K type thermocouple	± 1 ºC
Bourdon type manometer (low pressure)	± 0.1 bar (1% of full scale)
Bourdon type manometer (high pressure)	± 0.35 kgf/cm² (1% of full scale)
Pyranometers	± 5%
Anemometer	± 0.9 m/s (3% of full scale)
Digital psychrometer	± 1 °C (ambient temperature)
	± 2 °C (dew point)
Energy meter	± 1%
Tank	± 5%

2.2 Modelling procedures

The global thermal performance of the heat pump, called in this work $\mathrm{COP_{global}}$, was determined using Eq. (1), where m_w is the mass of water, ΔT_w is the water temperature variation (final and initial states), c_{p_w} is the water specific heat under constant pressure, $\dot{W}_{comp_{real}}$ is the real compressor work and t is the measuring time.

$$\mathrm{COP_{global}} = \frac{m_w c_{p_w} \Delta T_w}{\dot{W}_{comp_{real}} t} \tag{1}$$

All the heat given by the refrigerant fluid ($\dot{\mathrm{Q}}_{\mathrm{cond}}$) when passing through the condenser was considered to be absorbed by the water in the tank. This way, all heat losses throughout the tank were not considered (adiabatic tank). The heat transfer ($\dot{\mathrm{Q}}_{\mathrm{cond}}$) is given by Eq. (2), where $\mathrm{h_{f_{cond_{ext}}}}$ and $\mathrm{h_{f_{cond_{ent}}}}$ are, respectively, the fluid specific enthalpy at the exit and entrance of the condenser. In addition to that, $\dot{Q}_{cond}$ can be understood as the system heating capacity. The mass flow of the refrigerant fluid is $\dot{m}_f$.

$$\dot{Q}_{cond} = \dot{m}_f (h_{f_{cond_{ent}}} - h_{f_{cond_{ext}}}) \tag{2}$$

The heat pump´s thermal performance COP_{cycle} is given by Eq. (3),in which $\dot{W}_{comp_{ideal}}$ is the compressor ideal work given by Eq. (4), where $h_{f_{evap_{ext}}}$ is the specific fluid enthalpy at the evaporator exit.

$$\mathrm{COP}_{cycle} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp_{ideal}}} \tag{3}$$

$$\dot{W}_{comp_{ideal}} = \dot{m}_f (h_{f_{cond_{ent}}} - h_{f_{evap_{ext}}}) \tag{4}$$

The compressor ideal work considers the fluid´s thermodynamic cycle only. It does not take into account the energy losses inside the compressor, the heat losses to the surrounding ambient and the losses due to irreversibilities. This way, $\dot{W}_{comp_{ideal}}$ is lower than $\dot{W}_{comp_{real}}$, therefore, COP_{cycle} is greater than the global thermal performance $\mathrm{COP}_{\mathrm{global}}$ of the heat pump. The thermal performance that really matters in these systems is given by $\mathrm{COP}_{\mathrm{global}}$, like in all articles from the literature. Based upon it, it is possible to analyze how efficient is the system being used. Moreover, the global compressor efficiency (η_{comp}) is given by Eq. (5).

$$\eta_{comp} = \frac{\dot{W}_{comp_{ideal}}}{\dot{W}_{comp_{real}}} = \frac{\mathrm{COP}_{global}}{\mathrm{COP}_{cycle}} \tag{5}$$

Preliminary studies with the HP operating under solar radiation have shown no formation (or tiny momentaneus formation) of condensed water vapor on the plate, therefore the air´s latent heat rate ($q_{cond_{\mathrm{vapor}}}$) was ignored. The contribution of the air´s sensible heat (q_{conv}) and the ambient radiation (q_{rad}) were not significant in comparison with the solar radiation heat rate. Thus, the mathematical modelling of these portions, except for the solar radiation, is dispensable.

The solar heat rate (q_{solar}) received by the evaporator is given by Eq. (6). This rate was considered to be 100% responsible for supplying the collector during the operation of the HP.

$$q_{solar} = \varepsilon I_{solar} A_{pl} \tag{6}$$

The solar radiation heat transfer was measured by a pyranometer positioned at the same plane of the plate, reading the heat rate from instantaneous solar radiation (I_{solar}) (W/m²) incident around the HP during all the experiment.

The collector efficiency (η_{col}), given by Eq. (7), is evaluated by the relation between the heat rate absorbed by the fluid passing through the evaporator ($\dot{Q}_{evap_{real}}$), given by Eq. (8), and the ambient available heat rate that could theoretically be absorbed by the collector ($\dot{Q}_{evap_{ideal}}$), given by Eq. (9). The value of $h_{f evap_{ent}}$ is the fluid specific enthalpy at the evaporator entrance. Furthermore, $\dot{Q}_{evap_{real}}$ can be understood as the fluid´s capacity to absorb ambient energy, when it flows through the evaporator.

$$\eta_{\mathrm{col}} = \frac{\dot{Q}_{\mathrm{evap_{real}}}}{\dot{Q}_{\mathrm{evap_{ideal}}}} \tag{7}$$

$$\dot{Q}_{evap_{real}} = \dot{m}_f (h_{f evap_{ext}} - h_{f evap_{ent}}) \tag{8}$$

$$\dot{Q}_{evap_{ideal}} = q_{cond_{\mathrm{vapor}}} + q_{rad} + q_{conv} + q_{solar} \tag{9}$$

The parameter η_{col} is an important reference to evaluate the energy performance of the collector, representing the percentage of liquid heat coming from the ambient that is received by the collector and used by the fluid in its expansion when it passes in the evaporator. Also, it indicates how well dimensioned is the collector size for the system operating under given weather conditions.

Lastly, the software EES (Engineering Equation Solver) was used to obtain all the proprieties of the fluids and solids described along the paper.

3. RESULTS AND DISCUSSION

Five experimental water heating trials were made. The values of the parameters shown represent the average of the figures obtained in these trials. Moreover, the uncertainty propagation analysis of the COP_{real}, the system´s most important variable, was done.

The water was heated from an average temperature of 29.2 °C to an average temperature of 45.3 °C, with an ambient average of 30.7 °C. The average temperature of the plate was 29.6 °C. In addition to that, the average trial duration was 3 hours and 21 minutes and an average of 14 measurements was done in each trial.

Figure 4 shows the water heating curve throughout the trial. The heating rate had, on average, an asymptotic tendency at the end of the heating time. Additionally, the trial of 02/18/2017 had an abnormally higher duration for the water heating, since the average solar radiation was very low. This way, this trial´s graph differs from the general tendency of the others.

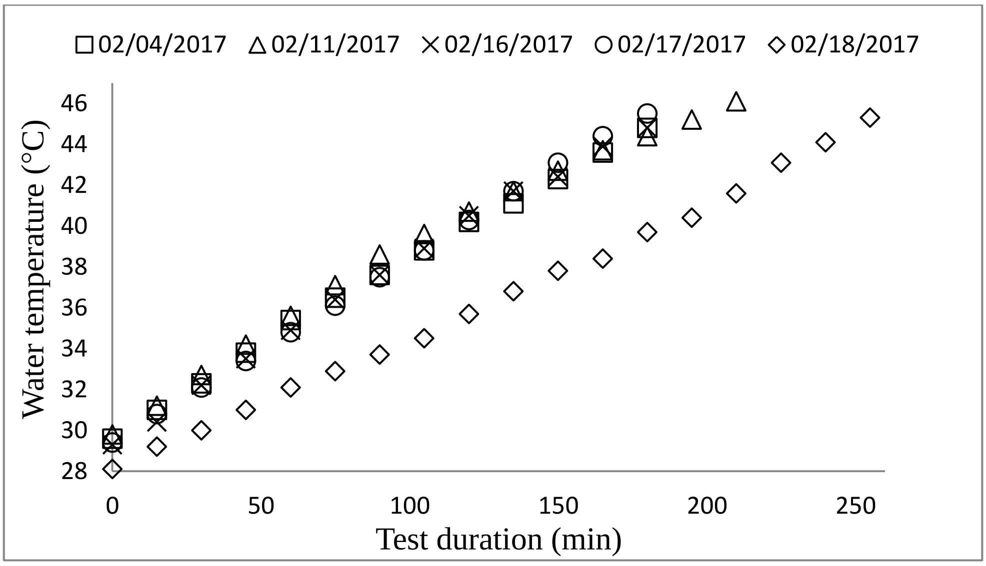


Figure 4: Water heating curves

The cycle Coefficient of Performance, on average, was 5.36 and the global one, on average, was 2.76±0.06, with the uncertainty propagation of the latter determined. Figure 5 presents the behavior of the variables with respect to the water temperature. The system thermal performance, both the cycle and global ones, decreased with the water heating, ranging from 6.31 to 4.83, on average for the COP of the cycle, and ranging from 3.18 to 2.22, on average, for the global COP.

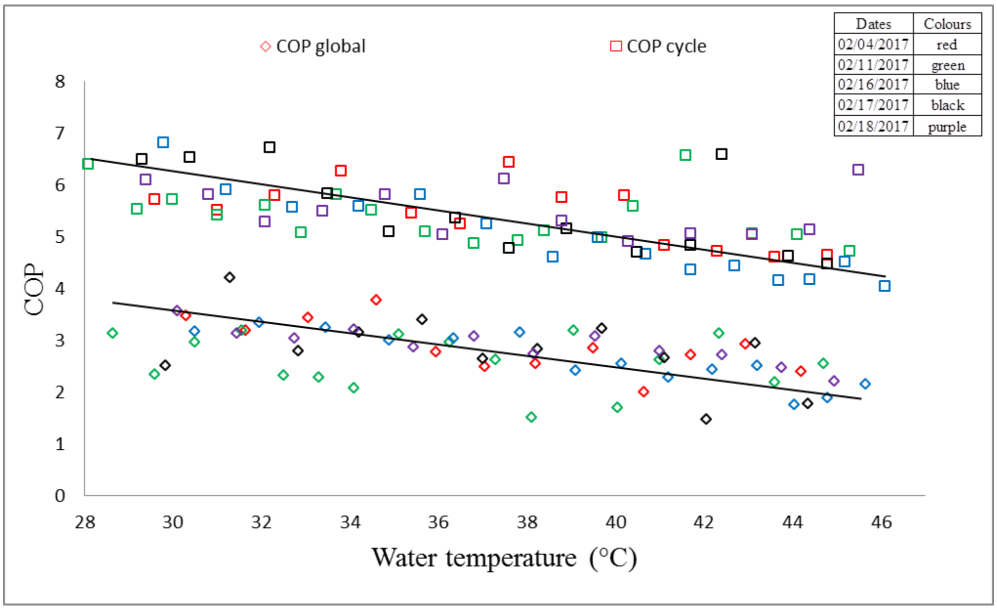


Figure 5: Thermal performance results

Both the cycle and global COPs presented a decreasing tendency with throughout the water heating process, as shown in the tendency curve. Therefore, it is possible to conclude that the higher the water temperature, the less efficient will be the HP. This can be a limiting factor in applications in which the water must be heated to temperatures higher than the ones presented in these trials.

The average global COP was around 52% of the value of the average cycle COP. This is due to the heat losses in the compressor as well as irreversible processes inside of it. That is, the compressor global efficiency was, on average, 52% and had a general stability tendency. The compressor low efficiency was the main cause of the small thermal efficiency of the system when compared to the literature. Figure 6 shows the compressor ideal and real power versus water temperature.

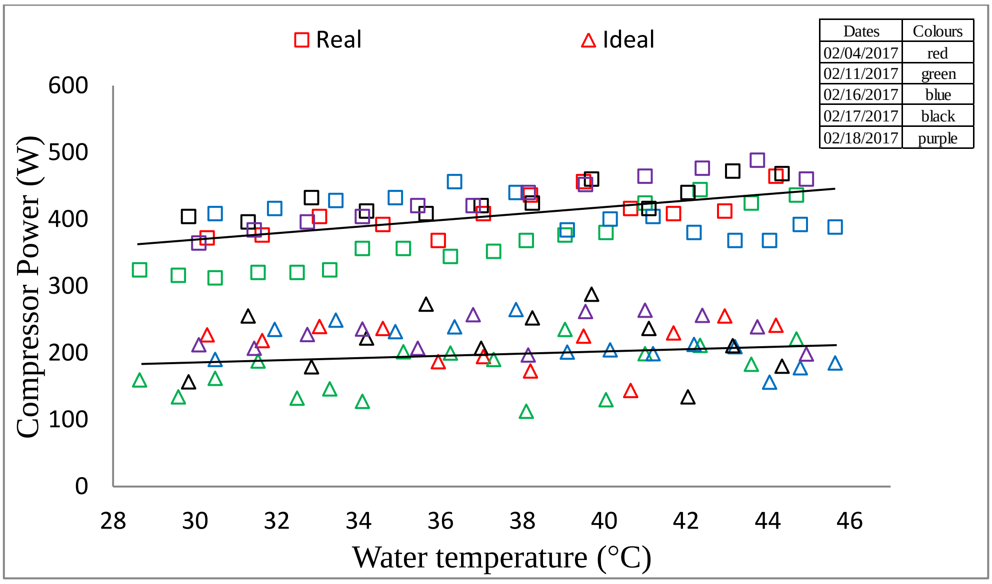


Figure 6: Results of compressor ideal and real power

The average ideal power of the compressor was 209 W, the average real power of the compressor was 407 W. As shown on Figure 6, the ideal power of the compressor had a slightly increasing tendency. The real power of the compressor had a growing tendency, ranging, on average, from 374 W to 443 W. This fact was directly responsible for the decreasing the system COP.

The collector efficiency was, on average, 89%, coherent with the values found on the introduction. It is a number that indicates a proper dimensioning of the collector for the system, taking into account the average incident solar radiation flow during the trial days. This average was 595 W/m², relatively low figure when compared to sunny days, when the average beats 1000 W/m². This way, for partially cloudy days, such as the ones when this research experiments were made, the collector is well dimensioned and suitable with the compressor work and the condenser heating capacity. However, the collector efficiency tends to shrink on sunny days.

The system heating capacity and its energy absorbing capacity (real and theoretical) by the evaporator presented significant fluctuations. The system heating capacity was, on average, 1125 W and its energy absorbing capacity by the fluid was, on average, 916 W.

The sensible heat rate of the air (convecction) and the ambient radiation heat rate were calculated using equations from the literature (Incropera et al., 2007). However, the results indicated their sum represents only around 1% of the solar value. Therefore, the mathematical model of these heat rates was neglected. Figure 7 shows the behavior of the solar radiation rate, the convection rate and ambient radiation rate. The convection and radiation heat rates are sometimes positive and sometimes negative. This is due to the fact that the collector is warmer or colder than the ambient at each measuring time. The ambient temperature was considered for the temperatures of the air and the surroundings, that is, they were all equal.

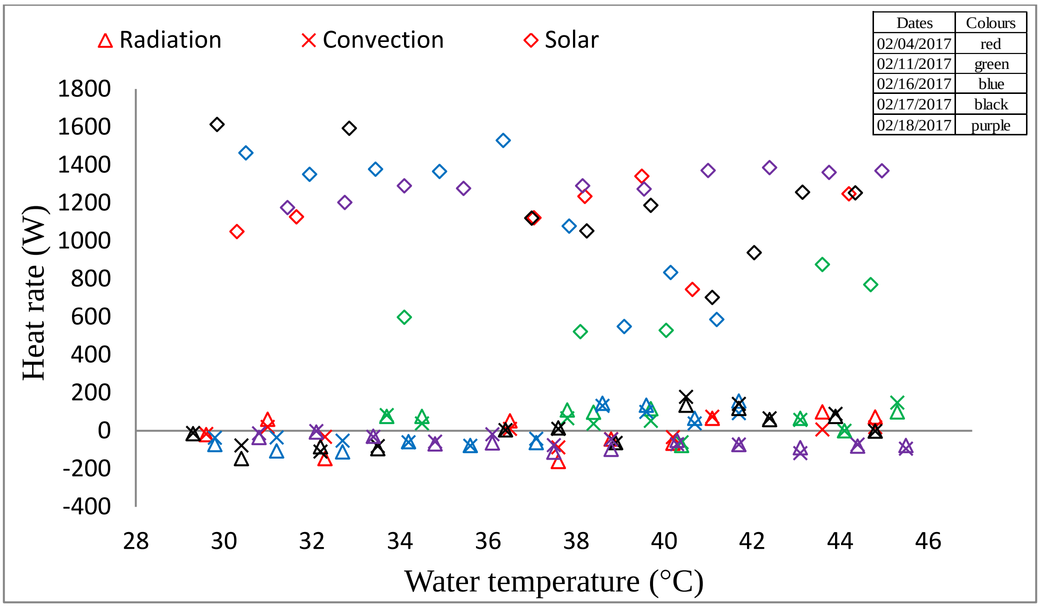


Figure 7: Heat rates changed with the collector

4. CONCLUSIONS

This paper made it possible to analyze the thermal performance of a HP operating under high thermal energy availability through the exposition of the solar evaporator to the sun. This replicates the conditions of a sunny or slightly cloudy day, in which the solar thermal availability is very significant and tends to increase the performance of the HP when compared to a cloudy or rainy day.

In the five trials done and analyzed in this work, a satisfactory operation of radiation exposed the HP was found, heating 200 L of water from around 29.2 °C to 45.3 °C in an average time of 3 hours and 21 minutes. The average heating capacity was 1125 W and thermal performance of 2,76±0,06. The collector was found to be well dimensioned for the weather conditions of the trials (partially cloudy), presenting an efficiency of 89%. However the compressor presented low efficiency, 52% on average, compromising the performance of the system. Finally, the air convection and ambient radiation heat rates were negligible in comparison to the direct solar radiation rate.

The results of this paper reinforce the advantages of the energy efficiency increase of the heat pump water heating system when it operates with the evaporator exposed to solar radiation. The system was advantageous when compared to the traditional residential water heating system in Brazil (electrical shower), as well as when compared to the same heat pump operating with the solar evaporator without direct exposition to solar radiation, in accordance with literature.

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