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ECONOMIC AND ENERGETIC ANALYSIS OF SOLAR COLLECTOR SIZE OF A DIRECT EXPANSION SOLAR ASSISTED HEAT PUMP

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Abstract. *One option to reduce the energy consumption to produce domestic hot water is replacing the electrical heaters by direct expansion solar assisted heat pumps. Although, there are many experimental and theoretical studies on the effect of environment conditions in energetic or exegetic performance, there are no studies on economic analysis of solar collector size. This theoretical analysis was made using lumped models for heat exchangers and a black box model for the hermetic R134a compressor. This modeling was validated using experimental results from DX-SAHP equipped with a 1.65 m^2 bare flat plate collector operating in different environment conditions. The average difference between experimental and theoretical COP is 1.7%, lower than the uncertainty of COP experimental that is around 5%. The results show that the increasing of collector size increases the COP but decrease the collector efficiency. An economic analysis comparing the payback of heat pump over an electrical heater was carried out. This economic analysis showed that there is a collector size that minimize a payback time. Additionally, the payback time and COP was calculated considered data from different Brazilian cities. The optimum payback was found for a collector size of 2.3 m^2 and a payback of 3.4 years.*

Keywords: *Economic analysis; Energetic analysis; Collector size; Solar assisted heat pump;*

1. INTRODUCTION

The use of heat pumps for heating water instead of electric heaters is one way to reduce energy consumption and consequently indirect emissions of greenhouse gases. Several studies show that Solar Assisted Heat Pump (SAHP) have better performance than Air Source Heat Pump (ASHP). Buker and Riffat (2016) presented different configuration for SAHP available in the literature for producing Domestic Hot Water (DHW) and for spacing heating. These authors define as Direct Expansion (DX) those equipment where the refrigerant flows in the solar collector (in that case also called as evaporator), and as Indirect Expansion (IX) those equipment where some brine flows in the solar collector.

Chata *et al.* (2005) analyze the COP in a DX-SAHP with different refrigerants. In this theoretical study, the condensing temperature was fixed at $60°C$ and the evaporating temperature range was between $0°C$ and $20°C$. The refrigerants analyzed were R12, R22, R134a, R404A, R407C and R410A. The results showed the best COP for R12, R22 and R134a respectively, but the difference between R-12 and R-134a were in the range of 2% to 4%. As the refrigerants R12 and R22 have Ozone Depletion Potential (ODP) not equal to zero. These authors also presented for R134a some charts to determine the collector area in for different evaporation temperatures and solar radiation.

There are studies presented by Kuang *et al.* (2003); Chow *et al.* (2010); Kong *et al.* (2011); Faria *et al.* (2016) with theoretical and experimental performance of DX-SAHP for DHW in different environmental conditions were investigated. In the works presented by Ito *et al.* (2005); Kuang and Wang (2006); Mohamed *et al.* (2017) performance of DX-SAHP for producing DHW for spacing heating are analyzed. In all of them the increase of evaporation temperature improve significantly the COP. One way to increase the evaporation temperature that are not well investigated are increasing the size of collector. In that case, the initial cost of equipment and the operational cost point out a optimum size of the collector. In the studies mentioned above, three types of collector has been tested: (i) uncovered flat plat collector, (ii) covered flat plat collector and (iii) evacuated tube collector, among them the most used is the first one.

In this study, a mathematical model is presented and validated experimentally. After that, simulations are performed by varying the size of the collector for different solar radiations. Then, the coefficient of performance, collector efficiency with and the payback period are analyzed.

2. TEST RIG

The DX-SAHP showed in Fig. 1 was used to validate the model. This heat pump is installed at UFMG, Belo Horizonte (MG), Brazil. The evaporator/solar collector was designed to operate as static air evaporator as described by Reis (2012). The collector is made by cooper and has as thickness of 1 mm, the evaporator tube has length of 17.3 m, and distance between the tubes is 103 mm. The inner and outer diameters are 8.73 mm and 9.53 mm, respectively for both the evaporator and immersed condenser. The heat pump has a R134a hermetic compressor model FFU100HAK manufactured by Embraco.

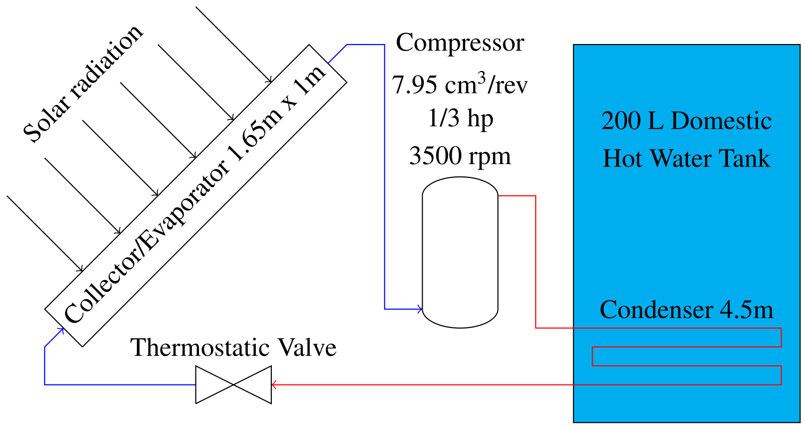

Figure 1. Schematic diagram of DX-SAHP used for a validation model.

3. MATHEMATICAL MODEL

In order to evaluate the performance of DX-SAHP for producing DHW a quasi-steady-state model was developed using the Equation Engineering Solver (EES). The losses in the tubes between components was considered negligible and for the inventory charge of the refrigerant, the pipeline was considered two meters long. The evaporator/solar collector and condenser was assumed as isobaric and a lumped model was used. The expansion device used is a thermostatic valve because of variation in solar radiation. The expansion valve is adjusted to maintain the superheat at evaporator outlet in 7^oC, and expansion process is modeled as isenthalpic. Following is described the modelling equation for each component.

3.1 Compressor model

The refrigerant mass flow rate ($\dot{m}$) in a constant rotation speed reciprocating compressor is given by:

$$\dot{m} = \rho_1 n V_s \eta_v \tag{1}$$

where ρ is the refrigerant density, n is the rotation speed, V_s is the compressor swept volume, η_v is the volumetric efficiency and the subscript 1 refers to compressor inlet or evaporator outlet. The compressor electric power consumption

($\dot{W}$), considering a isentropic compression process, is evaluated as follow:

$$\dot{W} = \frac{\dot{m}(i_2 - i_1)}{\eta_g} \tag{2}$$

where η_g is the global efficiency and i is the refrigerant specific enthalpy and the subscript 2 refers to compressor outlet under isentropic conditions or condenser inlet. The global and volumetric efficiency was determinate fitting equations proposed by Minetto (2011) to the compressor performance map available in Embraco website. The global and volumetric efficiency is given by:

$$\eta_v = -0.0143\left(\frac{P_2}{P_1}\right) + 0.915 \tag{3}$$

$$\eta_g = -0.0004\left(\frac{P_2}{P_1}\right)^2 + 0.0104\left(\frac{P_2}{P_1}\right) + 0.4839 \tag{4}$$

where P is the refrigerant pressure. The coefficient of determination (R^2) for volumetric efficiency is 97.6% and for global efficiency is 94.4%.

3.2 Direct expansion solar evaporator

The heat transfer rate received by the refrigerant in the evaporator ($\dot{Q}_e$) is given by:

$$\dot{Q}_e = \dot{m}(i_1 - i_4) \tag{5}$$

where the subscript 4 refers to thermostatic valve outlet or evaporator inlet. To evaluate the energy gain in a flat plate collector in steady-state condition Kong *et al.* (2011) suggest the following equation:

$$\dot{Q}_e = A_e F'[S - U_L(\overline{T}_r - T_a)] \tag{6}$$

where A_e is the area of evaporator of the solar collector, F' is the collector effectiveness factor, S is the net radiation absolved per unit of area, U_L is overall heat loss coefficient, $\overline{T}_r$ is the average temperature of the refrigerant fluid and T_a is the ambient air temperature.

The collector effectiveness factor proposed by Duffie and Beckman (2013), considering that the resistance to heat flow due the bond between the collector plate and tube can be neglected, is given by:

$$F' = \frac{1/U_L}{w\left[\frac{1}{U_L[D_o + F(w - D_o)]} + \frac{1}{\pi D_i h_i}\right]} \tag{7}$$

where the distance between the tubes in the evaporator is w, the fin efficiency is F, the outer diameter is D_o, the inner diameter is D_i, the internal convective coefficient is h_i that is calculated by the correlation proposed by Shah (2017) for two phase flow and by the correlation proposed by Gnielinski (1976) for single phase flow.

The fin efficiency can be evaluated by:

$$F = \frac{tanh\left[(w - D_o)/2\sqrt{U_L/(k\delta)}\right]}{(w - D_o)/2\sqrt{U_L/(k\delta)}} \tag{8}$$

where δ is the fin thickness and k is the thermal conductivity. The net radiation absolved is evaluated as made by Kong *et al.* (2017):

$$S = aI - \varepsilon\sigma(T_r^4 - T_s^4) \tag{9}$$

where the solar absorptivity is a, the solar radiation intensity normal to evaporator is I, the emissivity is ε, σ is the Stefane-Boltzmann constant and T_s is the sky temperature.

The overall heat loss coefficient proposed by Kong *et al.* (2011) is determined by:

$$U_L = h_o + 4\varepsilon\sigma T_a^3 \tag{10}$$

where the external convective coefficient (h_o) is calculated by the collection of correlations for free and forced convection, depending on wind speed, for tilted flat plate listed by Neils and Klein (2009).

3.3 Immersed condenser at hot water tank

Considering non-stratification in the water at the tank, the energy balance of water in the tank Q_t can by evaluated as follow:

$$\dot{Q}_t = \rho_w V_w C_w \frac{\partial T_w}{\partial t} \tag{11}$$

where V_w is the volume of water inside the tank, ρ_w is the water density, C_w is the water heat capacity at constant pressure, T_w is the water temperature at the tank and t is time.

The balance of energy is divided in three parts: gas flow region, condensation region and liquid flow region and the heat transfer in the three regions is given by:

$$\dot{Q}_c = \dot{m}\Delta i = UA(\overline{T}_r - T_w) \tag{12}$$

where Δi represents, in the gas flow region, the difference of the entalphy at condenser inlet and the saturated vapor enthalpy; in the condensation region, the enthalpy of vaporization; and in the liquid flow region, the difference of saturated liquid enthalpy and the entalphy at condenser outlet. The UA value is evaluated as follow:

$$UA = \left(\frac{1}{h_i \pi D_i L_c} + \frac{ln(D_o/D_i)}{2\pi k L_c} + \frac{1}{h_o \pi D_o L_c} \right)^{-1} \tag{13}$$

where L_c is the region length. The inner convective coefficient is obtained from the correlations proposed by Gnielinski (1976) and Shah (2016) for single and two phase flow, respectively. To outer convective coefficient is adopted the correlation for free convection in a horizontal cylinder presented by Rohsenow *et al.* (1998).

In order to consider the heat losses to the ambient through the wall and insulation in the hot water tank Kong *et al.* (2017) proposed a heat leakage coefficient (ζ) of 95% that is defined as follow:

$$\zeta = \frac{\dot{Q}_t}{\sum \dot{Q}_c} \tag{14}$$

3.4 Refrigerant charge

Another important parameter in the cost of the system is the refrigerant charge. The required mass of the refrigerant (m) is evaluated by the Eq. 15 for single phase flow and by the Eq. 16 for two phase flow.

$$m = \int \rho dV \tag{15}$$

$$m = \int \left[\alpha \rho_v + (1-\alpha)\rho_l\right] dV \tag{16}$$

where the subscripts l and v refers to the liquid and to the vapor, respectively, and the void fraction (α) is calculated by Hughmark (1965) correlation.

4. PERFORMANCE AND ECONOMIC INDICATORS

The coefficient of performance (COP) and the solar collector efficiency (η_{col}) proposed by Kong *et al.* (2011, 2017) and Kuang *et al.* (2003) is defined as follow:

$$COP = \int \frac{\dot{Q}_t}{\dot{W}} dt \tag{17}$$

$$\eta_{col} = \frac{\dot{Q}_e}{A_e I} \tag{18}$$

The payback period ($\hat{P}$) of the DX-SAHP over an electrical heater, in years, is given by:

$$\hat{P} = \frac{\hat{I}}{\hat{S}(1+\text{IR})^{(\hat{P}-1)}} \tag{19}$$

where $\hat{I}$ is the difference of initial investment between DX-SAHP and an electrical heater, IR is the annual inflation rate and the annual savings ($\hat{S}$) that is given by:

$$\hat{S} = Q_A \left(\frac{\hat{E}_e}{\eta_e} - \frac{\hat{E}_h}{COP} \right) \tag{20}$$

where $\hat{E}_h$ is the electricity tariff during heat pump running hours, $\hat{E}_e$ is the electricity tariff during electrical heater utilization period, Q_A is annual heat demand and η_e is the efficiency of electrical heater.

The difference of initial investment is divide in three parts: (i) the fixed cost, in a DX-SAHP comprises the cost with compressor, DHW tank, expansion valve and electrical components, in an instantaneous electrical heater is the all initial investment; (ii) the variable cost which comprises the collector cost, the piping of condenser cost, the refrigerant charge cost; and (iii) the service of installation and assembling cost, estimated as a percentage of materials costs.

5. SIMULATION PARAMETERS

In the following investigate the evaporator / collector length was fixed in 1.65 m and the width was change between 0.8 to 3.6 m to maintain the distance between the tubes in the collector constant. The evaporator tube length was calculated by linear proportion with collector area. For each collector area a new design for immersed condenser was made considering 20^{o}C as the maximum difference of condensation temperature and water temperature in the tank. The other geometrical parameters listed in section 2 was considered constant. A list of others parameters used in the simulations is presented in Tab. 1. The final water temperature was chosen based in guideline of ASHRAE (2000) to minimize the risk of Legionellosis. The costs in Tab. 1 are based in the Belo Horizonte market in March of 2018. For the inflation rate was chosen the INPC (National Consumer Price Index), and the value in Tab. 1 was measured between 2013 to 2017.

Table 1. Simulation parameters list

Parameter	Value	Parameter	Value	Parameter	Value
Heating demand	350 days/year	Inflation rate	6.34%	Initial water temperature	25 ^{o}C
Atmospheric Pressure	101.3 kPa	Collector tilt angle	30 ^{o}	Final water temperature	65 ^{o}C
Emissivity	0.95	Solar absorptivity	0.95	Ambient temperature	25 ^{o}C
Solar radiation	700 W/m^2	Superheating	10 ^{o}C	Subcooling	5^{o}C
Fixed cost	2500 BRL	Collector cost	300 BRL/m^2	Service cost	30%
Charge cost	40 BRL/kg	Condenser cost	10 BRL/m	Electric heater efficiency	97%

6. RESULTS AND DISCUSSION

6.1 Modelling validation

The model validation was performed comparing the COP of 10 experimental tests performed in the summer of 2017 and presented by Diniz (2017) with COP calculated using the model. In this test the solar radiation was between 0 and 1100 W/m^2, the ambient temperature between 24,8 and 34,5^{o}C, final water temperature around 45^{o}C and wind velocity between 0 and 2.4 m/s. The comparison between measured and calculated COP is showed in the Fig. 2.

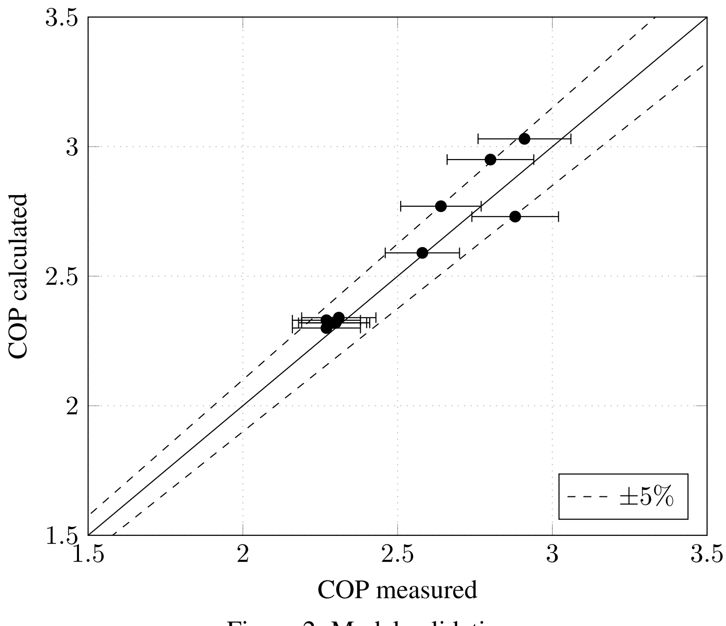

Figure 2. Model validation

The average uncertainty of COP measured is 5%. The mean and mean absolute difference between the measured COP and calculated COP are respectively 1.7% and 2.7%. The five points with COP around 2.3 was made inside the laboratory without solar radiation and in this case the heat pump operates as ASHP. The average COP of this heat pump in solar assisted mode is 21% bigger then air source mode.

6.2 Energetic analysis

The coefficient of performance and the collector efficiency for different collector size is showed Fig. 3. It can be observed that the increasing of collector size increases the COP and decreases the collector efficiency for any solar radiation. The rate of increase of the COP with collector area of 1.3 m^2 is 0.05 %/W/m^2 and for collector area of 5.9 m^2 is 0.12 %/W/m^2. Kong *et al.* (2011) have used a R22 DX-SAHP with aluminum unglazed solar collector with area of 4.22 m^2 and increase rate of COP found by these authors is 0.08 %/W/m^2, for the same collector area the increase rate of COP in Fig. 3 is 0.09 %/W/m^2. Kuang *et al.* (2003) investigated a R22 DX-SAHP with 2 m^2 aluminum collector, and increase rate of COP found by then was 0.05 %/W/m^2 and at the same area in Fig. 3 the increase rate is 0.06 %/W/m^2.

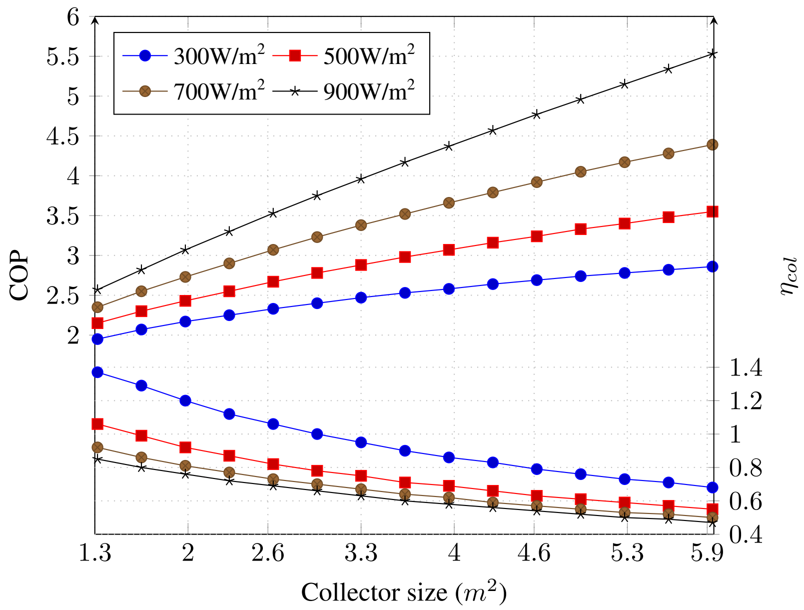

Figure 3. COP and collector efficiency vs collector size for different solar radiation

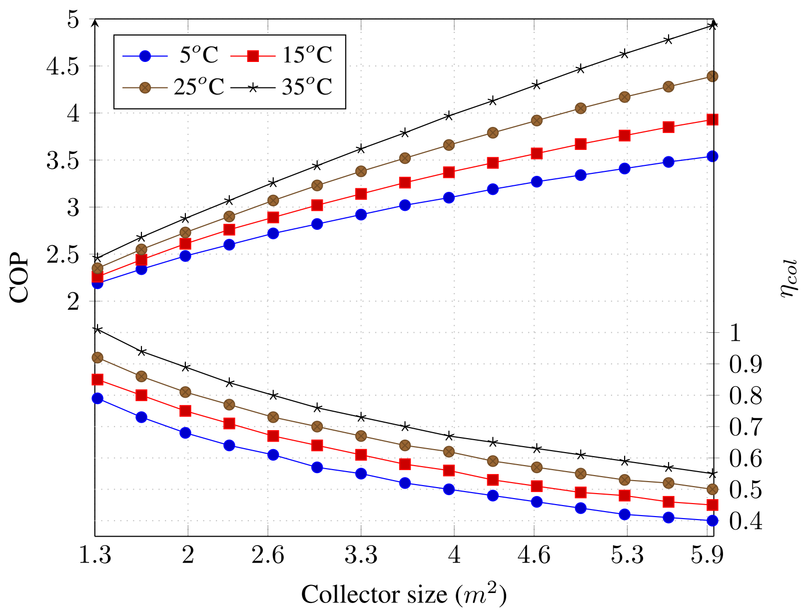

Figure 4. COP and collector efficiency vs collector size for different ambient temperatures

The decrease of the efficiency of the collector with the increase of solar energy in a DX-SAHP with uncovered collector have been discussed by some authors Kong *et al.* (2011, 2017); Kuang *et al.* (2003). For high solar radiation the difference between of the ambient temperature and evaporation temperature is positive and there is a heat loss for the ambient in the collector. On the other hand, if the solar radiation reduces the evaporation temperature decrease and there is less heat loss at the collector. If the evaporation temperature is lower than the ambient temperature the collector a certain heat is gain from ambient and the collector efficiency can be higher than one.

The effect of ambient temperature in the COP and collector efficiency is showed in Fig. 4. It can be noticed that the increasing of COP and the collector with the increase of the ambient temperature for any collector size. The increase rate of COP in Fig. 4 and at lower collector area is 0.4 %/^{o}C and at higher collector area is 1 %/^{o}C. Considering the first and last result of the simulation with ambient temperature equal to 35^{o}C the COP doubled and the collector area increases more than four times.

6.3 Economic analysis

Initially, the cost of electricity is considered 0.494 BRL/kWh and constant during the day. This value represents the electricity tariff for households in Belo Horizonte during March of 2018 obtained from Brazilian Electricity Regulatory Agency (ANEEL) website. The payback time in function of the collector size is showed in Fig. 5 for different solar radiation and ambient temperature. There is an optimum size of collector area, between 2 and 2.3 m^2, and this minimum point is approximately the same for any solar radiation and ambient temperature. The payback time is lower in condition with high solar radiation or ambient temperature. The difference between optimum payback in the simulations with 500 and 900W/m^2 is 10% and in the simulations with 35^{o}C and 15^{o}C are 4%.

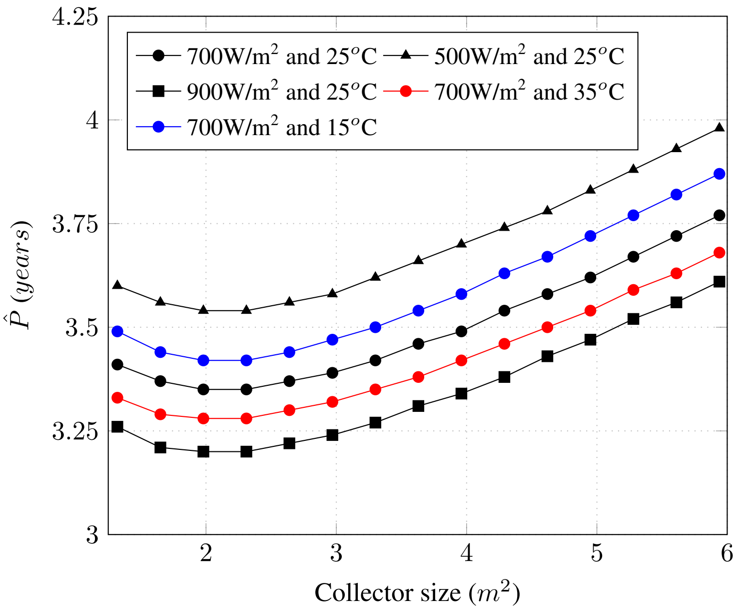

Figure 5. Payback period versus collector size for different solar radiation and ambient temperature

A sensitive analysis was made changing all the costs, the main results was found changing the parameters presented in Tab. 2. The cases 0, 1 and 2 of Fig. 6 shows the effect of collector cost in the payback. The decrease of this cost to 400 for 200 BRL/m^2 reduces in 10% the payback time but increase in 60% the collector area need. The analysis of the cases 0, 3 and 4 shows the effect of fixed cost in the payback time. The difference of optimum payback time in the cases 3 and 4 is 42% and the collector area of this optimum point in case 4 is twice of case 3. The costs with the condenser, refrigerant charge and service change the payback but do not affect significantly the area of the minimum payback time.

Table 2. Cases of a parametric study

Case	0	1	2	3	4
Collector cost (BRL/m^2)	300	200	400	300	300
Fixed cost (BRL)	2500	2500	2500	1500	3500

Results considering different electricity tariff is showed in Fig. 7 for collector area of 2 m^2. The electricity tariff range was select based on data from ANEEL in March of 2018. The lowest tariff was found in Santa Catarina state (0.31 BRL/kWh) and the highest tariff in São Paulo state (0.71 BRL/kWh). ANEEL started a program called "Tarifa Branca" in

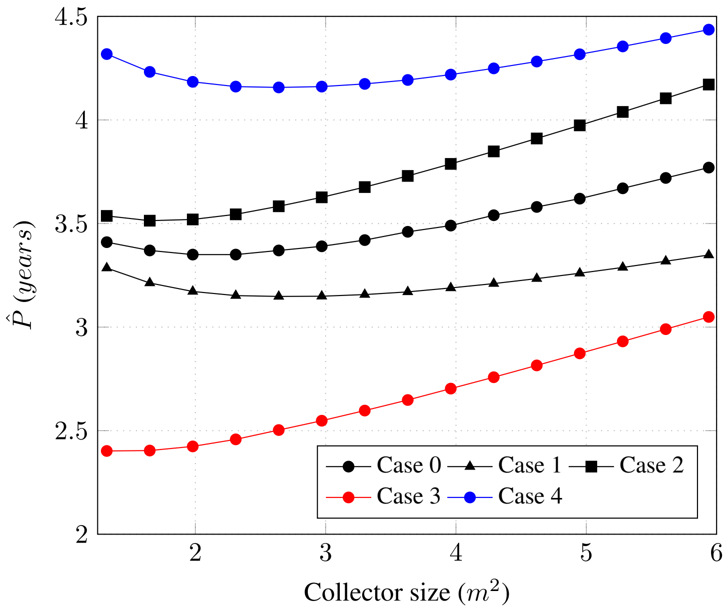

Figure 6. Payback period versus collector size for different investment cost

January 2018 that provide discounts in the electricity price during the day. However, the value of the discount is defined by each power distribution company. Then, in the Fig. 7 further the scenario with constant electricity tariff, three other results are plot considering different ratio between the electricity tariff during electrical heater utilization and the heat pump utilization. The reduction of 30% in the electricity cost during the day reduces the payback in 12%. The collector size that minimizes the payback was not affect by the electricity cost in the range displayed in Fig. 7.

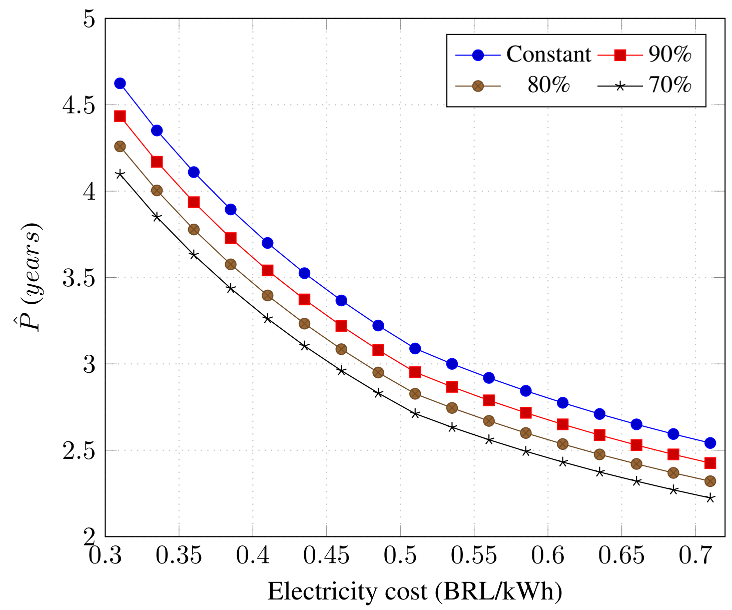

Figure 7. Payback period versus electricity tariff for collector area of 2 m^2

In fact, a larger collector reduces the time of operation need to heat the water then the heat pump can be programmed to operate during the hours with higher solar radiation and a better COP and payback can be found. To carry out this analyses, the annual average solar radiation for three different cities were determined: (i) Belo Horizonte (MG) at latitude 19^{o}49'S; (ii) Florianópolis (SC) at latitude 27^{o}35'S; and (iii) Natal (RN) at latitude 5^{o}47'S. These cities were chosen based in the results of Tiba (2001) to represents cities with average, maximum and minimum annual average solar radiation per day in Brazil, respectively. The available mean solar radiation curves in Fig. 8 was obtained using the weather data from meteorological institute of Brazil (INMET) considering that at noon the heat pump takes the half time to heat the water. Figure 8 also shows the solar radiation and payback period (for constant electricity tariff) for different collector area and the COP curve for any collector area.

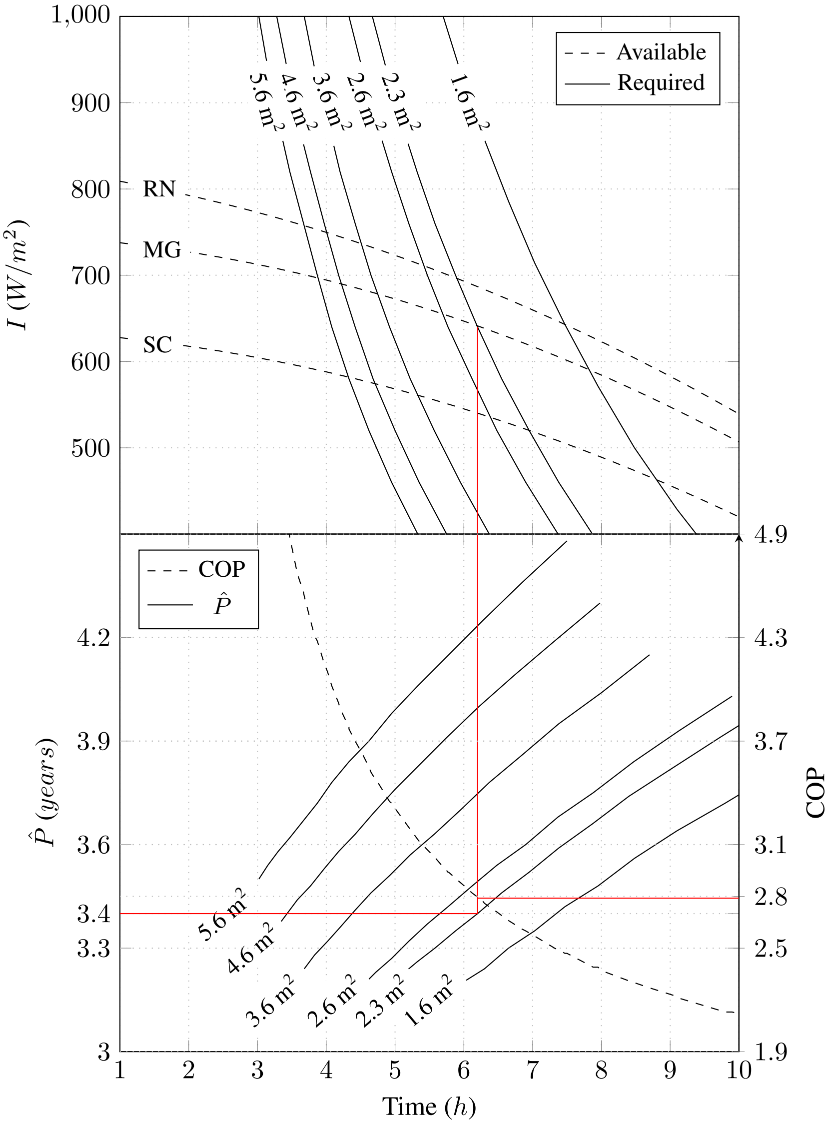

Figure 8. Required and available solar radiation, COP and payback in function of time of operation

The intercession of the available and required curves in Fig. 8 represents the required time to operate a heat pump with a specific collector size in each city. For example, to Belo Horizonte (MG) with a colector at 2.3 m^2 the intercession between the red lines show the average of solar radiation (640 W/m^2), the time of operation (6.2 hours), the COP (2,8) and the payback time (3.4 years). The maximum and minimum COP in Brazil, considering the collector areas showed in Fig. 8 are 4.4 and 2.2. For Natal (RN) and Florianópolis (SC), considering the cost presented in Tab. 1 and electricity cost in march of 2018, the minimum payback are 4.0 and 3.8 and the best collector are 2.3 m^2. The payback in these cities are bigger then Belo Horizonte because of the lower electricity tariff (0.418 and 0.46 BRL/kWh).

7. CONCLUSIONS

In this paper, an energetic and economic analysis of a R134a DX-SAHP for producing domestic hot water was carried out using a mathematical model. The model was successfully validate using a heat pump equipped with a 1,65 m^2 bare flat plate collector operating in different conditions of solar radiation, ambient temperature and wind velocity. The difference between experimental and theoretical results are lower than the uncertainty of experimental results.

The results show that the COP is almost proportional to the collector area and solar radiation. Additionally, there is an economic optimum size of collector area for the solar assisted heat pump. The best collector size is significantly affect only by the initial investment cost. The analysis for different electricity cost showed that the reduction of 30% in the electricity cost reduces the payback time in 12%. Furthermore, this minimum payback period point achieved with a collector of 2.3m^2, and for Belo Horizonte, Natal and Florianópolis the payback are 3.4, 3.8 and 4 years, respectively.

8. ACKNOWLEDGEMENTS

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