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FEASIBILITY OF SOLAR GEOTHERMAL HYBRID SOURCE HEAT PUMP FOR PRODUCING DOMESTIC HOT WATER IN BRAZILIAN CLIMATES

Cleison Henrique de Paula
Sinthya Gonçalves Tavares
Luiz Machado
Antônio Augusto Torres Maia
Federal University of Minas Gerais (UFMG), Department of Mechanical Engineering, Av. Antonio Carlos, 6627, Belo Horizonte - MG, 31270-901, Brazil
cleissonhp@yahoo.com.br, tavaressinthya@gmail.com, aamaia@demec.ufmg.br,luizm@demec.ufmg.br

Tiago de Freitas Paulino
Federal Center of Technological Education of Minas Gerais (CEFET-MG), Department of Materials Engineering, Av. Amazonas, 5253, Nova Suiça, 30421-169, Belo Horizonte (MG), Brazil
tiagopaulino@cefetmg.br

Willian Moreira Duarte
University of Belo Horizonte (UniBH), Department of Mechanical Engineering, Av. Prof. Mário Werneck, 1685, Estoril, 30575-180, Belo Horizonte (MG), Brazil
willianmoreiraduarte@gmail.com

Abstract. *This work presents a study for examining the viability of Solar Geothermal Hybrid Source Heat Pump systems to produce domestic hot water in Brazilian climates. The system considered has an open loop ground underground (aquifer) water and an uncovered solar collector that can be use separately or connected in series topology. The refrigerant chosen for the heat pump was R290 (propane). This theoretical analysis was made using lumped models for each component of the system. The results show that the COP of a Hybrid Source Heat Pump is better than other type of heat pump for most environmental conditions but the best payback is obtained using a Solar Assisted Heat Pump. The results of COP and payback were compared with other papers and the maximum difference was 8.3% and 12%, respectively. Considering the scenario of the house that already has a water well, so there is no drilling cost, the best payback is obtained using a Ground Source Heat Pump.*

Keywords: *Hybrid Source Heat Pump (HSHP), Solar Assisted Heat Pump (SAHP), Ground Source Heat Pump (GSHP), R290, Economical analysis*

1. INTRODUCTION

According to forecasts by the International Energy Agency (IEA), world energy consumption will increase from 45% between 2006 and 2030. Besides, oil and gas will remain dominant in the matrix, accounting for 50% of total energy consumption in 2030. According to Rees (2016), although Brazil is the sixth country that least pollutes to produce electricity, about 100 grams of CO_2 are released into the atmosphere to produce 1 kWh of electricity in this country. Heating water represents a significant portion of household electricity consumption in Brazil. The heating of domestic hot water (DHW) through heat pumps reduces this consumption, since these devices use thermal energy available in the environment.

The performance of a heat pump can be improved through the integration with renewable energy sources such as solar thermal collector (Buker and Riffat, 2016), solar photovoltaic collector (Chow, 2010), geothermal heat exchangers (Sarbu and Sebarchievici, 2014) or wind turbines (Li *et al.*, 2018). One system that combines two renewable energy sources is the Solar Assisted Ground Source Heat Pump (SAGSHP) or Solar Geothermal Hybrid Source Heat Pump (HSHP). Reda (2017) presented a review of the studies assessing HSHP. In this review, it was presented twelve works considering cold and temperate climates and none study for hot climates. Although none studies were found for hot climate, Reda (2017) suggests that a HSHP parallel type can be applied in hot climate for space heating and cooling.

In the literature, HSHP has been used in greenhouses (Ozgener and Hepbasli, 2005b,a), for space heating (Hepbasli, 2007; Dai *et al.*, 2015), and for producing DHW and space heating/cooling simultaneously (Lazzarin, 2012; Rad *et al.*, 2013), but none study considered only the demand of DHW. Additionally, in the investigations of HSHP, the authors were not concerned about the refrigerant used. A commercial heat pump that uses refrigerants with non zero Ozone Depletion Potential (ODP), such as R22 (Dai *et al.*, 2015), or refrigerants with high Global Potential Warming (GWP), such as R134a (Bakirci *et al.*, 2011), were typically used in these studies. In this work was investigated the feasibility of HSHP for producing only DWH in tropical climates using a refrigerant with low GWP.

2. METHODOLOGY

The system considered in this study is shown in Fig. 1. The main components are the hermetic reciprocating compressor, two coaxial heat exchanger used in the evaporator and condenser, a thermostatic expansion valve (TEV), an uncovered flat type thermal solar collector, a control valve (V5), four solenoids and two water pumps. The control valve is required to maintain constant the temperature at the outlet of condenser during changes in the condenser inlet temperature and the evaporation temperature. This control can also be implemented using a small water pump, as described by Paulino *et al.* (2017).

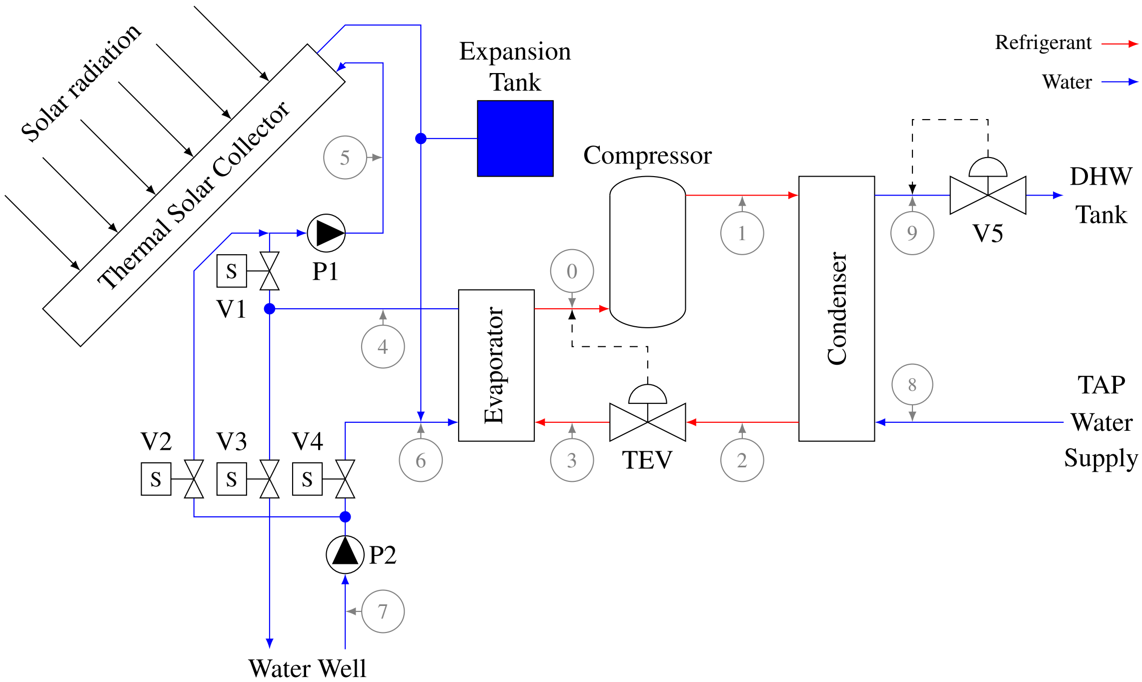

Figure 1. Schematic of the Hybrid Source Heat Pump

The system can operate in five different modes: **(i)** Solar Assisted Heat Pump (SAHP); **(ii)** Air Source Heat Pump (ASHP); **(iii)** Ground Source Heat pump (GSHP); **(iv)** Hybrid Source Heat Pump (HSHP); **(v)** Solar Ground Regeneration (SGR). In Table 1 is described which valves is open for each mode listed. The difference in the ASHP and SAHP modes are the presence of solar radiation in the collector. As described by Reda (2017), when extracted heat into the ground is greater than injected heat over the year, the SGR operating mode should be used to increase the system performance, enhancing the competitiveness of this particular setup. Thus, making proper use of solar heat becomes a fundamental issue in this system. In addition, the mode SGR is presented as a future possibility to this type of heat pump but it is not used in this work.

Table 1. Valve, pump and compressor status in each mode of operation

Mode	Compressor	P1	P2	V1	V2	V3	V4
SAHP	On	On	Off	Open	Closed	Closed	Closed
ASHP	On	On	Off	Open	Closed	Closed	Closed
GSHP	On	Off	On	Closed	Closed	Open	Open
HSHP	On	Off	On	Closed	Open	Open	Closed
SGR	Off	Off	On	Closed	Open	Open	Closed

2.1 Mathematical model

In order to evaluate the performance of a HSHP, a quasi-steady-state model was presented. For the heat exchangers, a lumped model was used and the pressure drop in the refrigerant side was considered negligible. The heat loss and pressure drop in the refrigerants ducts between components were assumed inappreciable and for the inventory charge of the refrigerant, the pipeline was considered two meters long. A thermostatic expansion valve (TEV) is used to maintain a fixed superheat at the outlet of the evaporator. The expansion process is assumed as isenthalpic. The compression process is modeled as isentropic and the mass flow rate ($\dot{m}_r$) in a constant rotation speed reciprocating compressor is given by:

$$\dot{m}_r = \rho_0 n V_c \eta_v \tag{1}$$

where ρ_0 is the refrigerant specific mass at the compressor inlet, n is the rotation speed, V_c is the compressor displacement volume and η_v is the volumetric efficiency. The numeric subscripts represents the points showed in figure 1. The electric power consumption in the compressor ($\dot{W}_c$), evaluated as follow:

$$\dot{W}_c = \frac{\dot{m}_r(i_1 - i_0)}{\eta_c} \tag{2}$$

where η_c is the compressor overall isentropic efficiency and i is the specific enthalpy. The overall and volumetric compressor efficiency was determinate fitting the equations 3 and 4 to the performance map provided by the compressor manufacturers as adopted by Duarte *et al.* (2019).

$$\eta_v = B_1 + B_2 \left(\frac{P_1}{P_0}\right) \tag{3}$$

$$\eta_c = B_3 + B_4 \left(\frac{P_1}{P_0}\right) + B_5 \left(\frac{P_1}{P_0}\right)^2 \tag{4}$$

In these equations P represents the pressure, and B_1 to B_5 are the regression coefficients. The heat transfer rate at the condenser ($\dot{Q}_{cd}$) at the condenser is given by:

$$\dot{Q}_{cd} = \dot{m}_r(i_1 - i_2) = \dot{m}_{cd} C_w (T_9 - T_8) \tag{5}$$

where T is temperature, C_w is the water heat capacity at constant pressure and $\dot{m}_{cd}$ is the water mass flow rate in the condenser. The heat transfer at coaxial condenser is modeled using the effectiveness-NTU method (Incropera *et al.*, 2007). The effectiveness of condenser is evaluated as follow:

$$\epsilon_{cd} = \frac{\dot{Q}_{cd}}{\dot{m}_{cd} C_w (T_1 - T_8)} \tag{6}$$

The effectiveness of coaxial counterflow heat exchanger is given by:

$$\epsilon = \frac{1 - exp[-NTU(1-r)]}{1 - exp[-NTU(1-r)]r} \tag{7}$$

The heat capacity ratio (r) given by:

$$r = \frac{\dot{m}_{cd} C_w}{\dot{m}_r \overline{C}_r} \tag{8}$$

The average refrigerant heat capacity ($\overline{C}_r$) is given by:

$$\overline{C}_r = \frac{\Delta i}{\Delta T} \tag{9}$$

The number of transfer units (NTU) is defined as:

$$NTU = \frac{\text{UA}}{\dot{m}_{cd} C_w} \tag{10}$$

where UA is evaluated by:

$$\text{UA} = \left(\frac{1}{h_i \pi D_i L_{cd}} + \frac{ln(D_o/D_i)}{2\pi k L_{cd}} + \frac{1}{h_o \pi D_o L_{cd}}\right)^{-1} \tag{11}$$

where L_{cd} is the condenser length, k is the thermal conductivity, D is the diameter, h is average convective coefficient and the subscripts i and o represents properties at inner and outer sides. For water flowing at annular region (h_o) the convective coefficient was calculated by correlations proposed by Rohsenow *et al.* (1998) considering a constant heat flux, for refrigerant side the Gnielinski (1976) and Shah (2016) correlations was used.

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

$$\dot{Q}_e = \dot{m}_r(i_0 - i_3) = \dot{m}_e C_b(T_4 - T_6) \tag{12}$$

where $\dot{m}_e$ is the water flow rate in the evaporator and C_b is the water heat capacity at constant pressure. The heat transfer at coaxial evaporator is also modeled using the effectiveness-NTU method. The effectiveness of evaporator is evaluated by Eq. 13, and Eq. 7 to 11 but using the properties of the evaporator. To calculate convective heat transfer in boiling in Eq. 11 the correlation of Shah (2017) was used.

$$\epsilon_e = \frac{\dot{Q}_e}{\dot{m}_b C_w(T_6 - T_3)} \tag{13}$$

The energy gain in a flat plate collector in steady-state condition is given by:

$$\dot{Q}_{col} = A_{col} R[S - U_{col}(T_5 - T_a)] \tag{14}$$

where A_{col} is the area of collector, R is the collector heat removal factor, S is the net radiation absolved per unit of area, U_{col} is overall heat loss coefficient and T_a is the ambient air temperature (Duffie and Beckman, 2013). The collector heat removal factor is given by:

$$R = \frac{\dot{m}_e C_w}{U_{col} A_{col}} \left[1 - exp\left(-\frac{U_{col} A_{col} F'}{\dot{m}_e C_w}\right)\right] \tag{15}$$

where F' is the collector efficiency, considering that the resistance to heat flow due the bond between the collector plate and tube can be neglected, is given by:

$$F' = \frac{1}{w U_{col}} \left\{\frac{1}{U_{col}[D_o + F(w - D_o)]} + \frac{1}{\pi D_i h_i}\right\}^{-1} \tag{16}$$

where the distance between the tubes is w and the fin efficiency is F that can be evaluated by:

$$F = \frac{tanh\left[(w - D_o)/2\sqrt{U_{col}/(k\delta)}\right]}{(w - D_o)/2\sqrt{U_{ev}/(k\delta)}} \tag{17}$$

where δ is the fin thickness (Duffie and Beckman, 2013). To evaluated the average convective heat transfer inside the tube (h_i) the correlation proposed by Gnielinski (1976) was used. The net radiation absolved is evaluated by Kong *et al.* (2011) using the fowling equation:

$$S = \theta I - \varepsilon q_0 \tag{18}$$

where θ is the solar absorptivity , I is the solar radiation intensity normal to evaporator, ε is the emissivity and q_0 is the difference between the emissive power from a black body and from the sky.

The overall heat loss coefficient compute the losses due the convection and radiation in uncovered flat plate collector is proposed by Kong *et al.* (2011) is determined by:

$$U_{col} = h_a + 4\varepsilon\sigma T_a^3 \tag{19}$$

where σ is the StefaneBoltzmann constant and h_a is the external convective coefficient calculated by correlation of McAdams, described by Kumar and Mullick (2010) and shown in EQ. 20, where the wind speed is u.

$$h_a = 5.7 + 3.8u \tag{20}$$

The electric power consumption in the pump ($\dot{W}_p$), evaluated as follow:

$$\dot{W}_p = \frac{\dot{m}_e}{\rho_w \eta_p} \sum \Delta P_f \tag{21}$$

where η_p is the pump overall efficiency and ΔP_f is the friction loss. At each stretch of piping the friction loss is evaluated by Eq. 22. To calculate the Darcy friction factor (f) was used the correlation of Churchill (1977) for flow in circular ducts, the correlation of Natarajan and Lakshmana (1973) for laminar flow in annular ducts and Jones and Leung (1981) for turbulent flow in annular ducts.

$$\Delta P_f = \frac{8 f L \dot{m}_e^2}{\pi^2 D_i^5 \rho_w} \tag{22}$$

The coefficient of performance (COP) is defined as follow:

$$COP = \frac{\dot{Q}_{cd}}{\dot{W}_c + \dot{W}_p} \tag{23}$$

The payback period ($\hat{P}$) of the HSHP over an electrical heater, if $\hat{P}$ is grater than one year, is given by:

$$\hat{P} = \frac{\hat{I}}{\hat{S}(1 + \text{IR})^{(\hat{P}-1)}} \tag{24}$$

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

$$\hat{S} = E\left(\frac{Q}{\eta_{eh}} - \frac{Q}{COP}\right) \tag{25}$$

where E is the electricity tariff, Q is annual heat demand and η_{eh} is the efficiency of electrical heater. The difference of initial investment is divided in two parts: (i) the fixed cost including the cost with compressor, DHW tank, expansion valve, electrical components, solar collector, heat exchanges, copper piping, assembly and installation; (ii) the variable cost which comprises the cost of drilling and PVC piping of the water well. Meanwhile, the initial investment for an instantaneous electrical heater is just the equipment itself.

The set of equations previously presented was solved in EES (Engineering Equation Solver). The EES can check units to avoid fail in the programming process. Automatically identifies the variables, regardless of the order in the program, and groups equations that must be solved simultaneously. In fact, this feature allows to change easily an input variable for an output variable and vise versa.

2.2 Simulation parameters

For the environmental parameters two strategies were adopted. As made by Hawlader *et al.* (2001), Deng and Yu (2016) and Kong *et al.* (2017) the first simulations considered fixed values of solar radiation, ambient temperature, atmospheric pressure, and wind speed. These values are described in Tab. 2. Additionally, as made by Chow *et al.* (2010), Reis *et al.* (2014) and Chaturvedi *et al.* (2014), a study of case was also made considered the average environmental conditions of a specific city. In this paper the city considered is Belo Horizonte (MG), Brazil. The climate data for Belo Horizonte, shown in Tab. 3, were obtained in the web site of Brazilian National Institute of Meteorology (INMET) for the years from 1961 to 2018, and in the software RETScreen Plus.

Table 2. Main simulation parameters

Parameter	Value	Parameter	Value	Parameter	Value
Collector size	1.65 m^2	Wind Speed	3 m/s	Inlet water temperature	25 ^{o}C
Collector tilt angle	30 ^{o}	Atmospheric Pressure	101.3 kPa	Outlet water temperature	65 ^{o}C
Collector emissivity	0.95	Solar radiation	700 W/m^2	Water tank size	0.2 m^3
Collector solar absorptivity	0.95	Ambient temperature	25 ^{o}C	Heating demand	350 days/year
Collector tube length	17.3 m	Compressor speed	3500 rpm	Compressor displacement	6.2 cm^3/rev
Collector plate thickness	1 mm	Superheating	10 ^{o}C	Electric heater efficiency	97%
Pump 1 overall efficiency	10%	Subcooling	5 ^{o}C	Electricity tariff	0.956 R$/kWh
Pump 2 overall efficiency	40%	Ground temperature	20 ^{o}C	Fixed cost of SAHP	5500 R$
Evaporator length	25.5 m	Water well depth	20 m	Fixed cost of GSHP	6000 R$
Condenser length	5.2 m	Water well cost	200 R$/m	Fixed cost of HSHP	6500 R$

Considering the studies presented by Makhnatch and Khodabandeh (2014), Ghoubali *et al.* (2014), Botticella and Viscito (2015), and Duarte *et al.* (2019) the refrigerant chosen was the R290 (propane). In addition, the R290 is used by many heat pump manufactures (Palm, 2008) and it can work at high evaporating temperature (Chaichana *et al.*, 2003). Despite of the flammability of the R290, the HSHP of this work is a small equipment and it will be installed in an open ventilated location. The compressor used is the same used by Duarte *et al.* (2019) for R290 manufactured by Embraco,

Table 3. Climate data from Belo Horizonte (Monthly averages)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Solar radiation (W/m^2)	700	812	614	589	500	481	500	534	662	700	837	812
Ambient temperature (^{o}C)	23.4	23.9	23.2	22.3	20.0	18.8	18.7	19.9	21.2	22.7	22.4	22.8
Wind speed (m/s)	3.0	3.0	2.9	2.9	2.7	2.7	2.9	3.2	3.4	3.5	3.2	3.1

model NEK2121U. For the compressor mentioned, the regression coefficients B_1 to B_5 of the Eq. 3 and 4 are 0.7664, -0.025, 0.2855, 0.0849 and -0.0089, respectively.

In order to compare the performance of the system proposed in this paper to others works, the collector and water tank size chosen are the same of those used in the SAHP described by Reis *et al.* (2014), Rabelo *et al.* (2019), Diniz *et al.* (2017a,b) and Duarte *et al.* (2019). A list of the main characteristics of uncovered flat collector used by Duarte *et al.* (2019) is presented in Tab. 2. The diameters was chosen considering the economics, pressure drop, noise, and oil entrainment establish feasible design velocities in refrigerant lines suggest by ASHRAE (2014) and economics design velocities in water lines suggest by Perroni *et al.* (2011). In the condenser and evaporator, the inner copper tube has external diameter of 6.35mm and thickness of 0.79mm. In the evaporator the external tube has external diameter of 15mm and thickness of 0.4mm and in the condenser the external tube has external diameter of 12.7mm and thickness of 0.4mm. The external diameter of the copper tube in the solar collector is 9.52mm and the thickness is 0.4mm and the external diameter of PVC tube used in the water well and between the collector and evaporator is 20mm and thickness of 1.5mm. The length of the heat exchangers was calculated considering the effectiveness of 95%, considering the environmental conditions of Tab. 2.

The final water temperature was chosen based in guideline of ASHRAE (2000) to minimize the risk of Legionellosis. The costs in Tab. 2 are based in the Belo Horizonte market in March of 2019. For the inflation rate was chosen the INPC (National Consumer Price Index), the mean value between 2008 to 2018 is 5.8%. The cost of electricity is considered constant during the day. This value represents the electricity tariff, including tax, households in Belo Horizonte during March of 2019. The heating demand is chosen considering 15 days of lay-off per year. The efficiency of the pumps was chosen based in the component available in the market. Pump 2 is an underwater pump, typically used in water wells.

3. RESULTS AND DISCUSSION

The COP of the ASHP, SAHP, GSHP and HSHP mode is 2.01, 2.73, 3.04, and 3.22, in the conditions presented in Tab. 2. The COP of HSHP and GSHP are 18% and 11% higher than the SAHP and the COP of ASHP is 26% lower than SAHP. The COP of SAHP mode is 13% higher than the payback of R290 SAHP reported by Duarte *et al.* (2019).

The variation of COP in function of ambient temperature for the HSHP and SAHP mode considering different values of solar radiation are shown in Fig. 2. The shape of the curves of SAHP mode are similar of those presented by Duarte *et al.* (2019) for R290 SAHP. The HSHP mode has better COP for solar radiation lower than 900 W/m^2 and ambient temperature of 35^{o}C. The increase of solar radiation or ambient temperature increase the outlet temperature of the solar collector and consequently the evaporating temperature. Since the condensation temperature is kept constant, the difference between the condensing and evaporating temperatures decreases, increasing the COP. Therefore, COP of SAHP mode is higher tan COP of GSHP mode for solar radiation of 900W/m^2 and ambient temperature over 27^{o}C and for solar radiation of 700W/m^2 and ambient temperature over 37^{o}C.

Considering the parameters listed in Tab. 2, the payback of the GSHP, SAHP and HSHP are 3.90, 2.45 and 4.00 years, respectively. The payback of HSHP and GSHP are 56% and 54% higher than the SAHP. The payback of SAHP is 16% higher than the payback of R290 SAHP, and 39% higher than the payback of R134a SAHP reported by Rabelo *et al.* (2019) for the same collector area. The variation of payback in function of ambient temperature for the HSHP and SAHP mode considering different values of solar radiation are shown in Fig. 3. It is important to notice that in these figures the scale of payback is different, so the payback of SAHP is, at least, one year lower than the payback of HSHP. The payback of the GSHP is no affected by solar radiation or ambient temperature, so the payback of GSHP is better or at least equal than HSHP and worse than SAHP.

The variation of COP and payback with water well depth for the HSHP and GSHP mode considering different values ground water temperature are shown in Fig. 4 and 5. The COP of HSHP is a little bit better but the payback of GSHP is better in any condition. The increase of the water well depth increases the friction losses and increases the work of pump, but in the Fig. 4 and 5 the maximum reduction of COP is 0.2%. The COP of HSHP is, in average, 6% higher than GSHP and the payback of HSHP is 2% higher.

Finally, a case study was performed considering the climate of Belo Horizonte, presented in Tab. 3. For this study the inlet water temperature was considered equal to the ambient temperature and the ground water temperature was assumed constant at 22.8^{o}C. This value represent the annual average earth temperature in the RETScreen Plus software. The variation of monthly COP of HSHP, SAHP and GSHP mode considering the climate of Belo Horionte are shown in Fig.

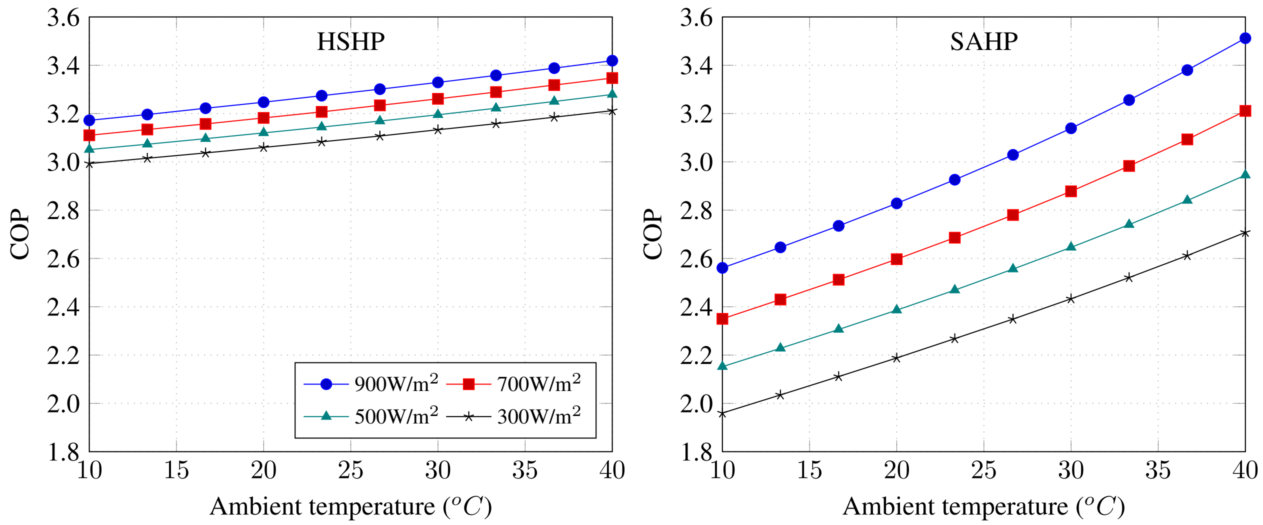

Figure 2. Variation of COP for different solar radiation and ambient temperature.

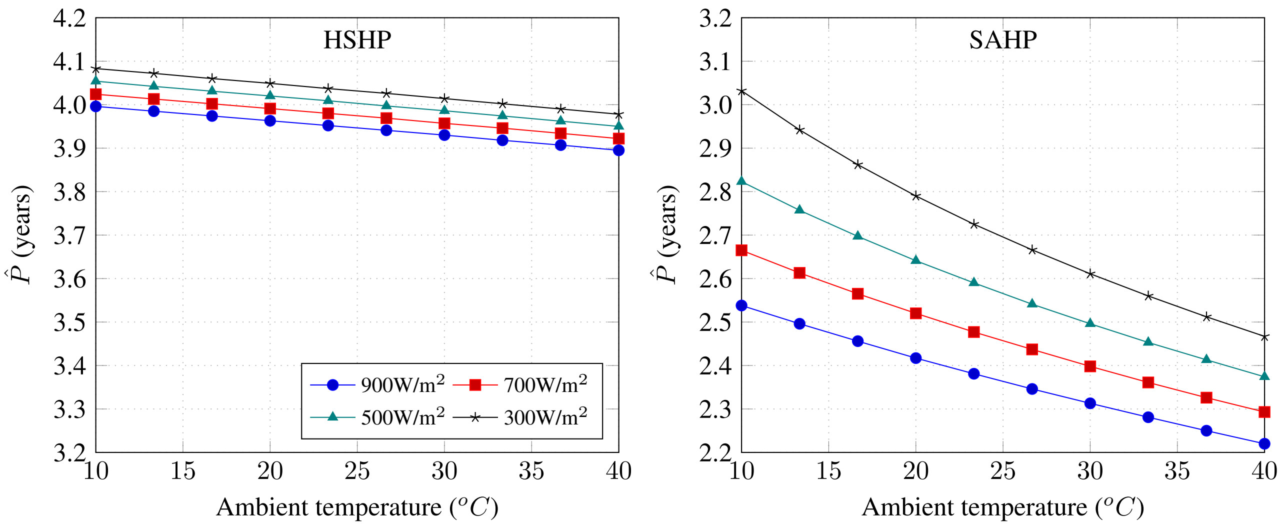

Figure 3. Variation of payback for different solar radiation and ambient temperature.

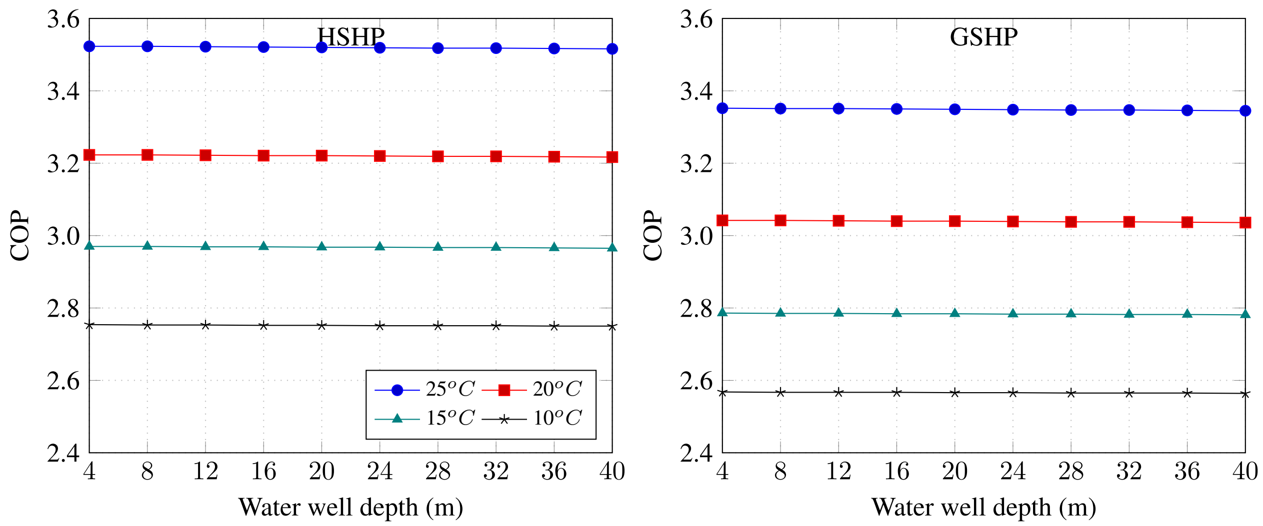

Figure 4. Variation of COP for different ground water temperature and water well depth.

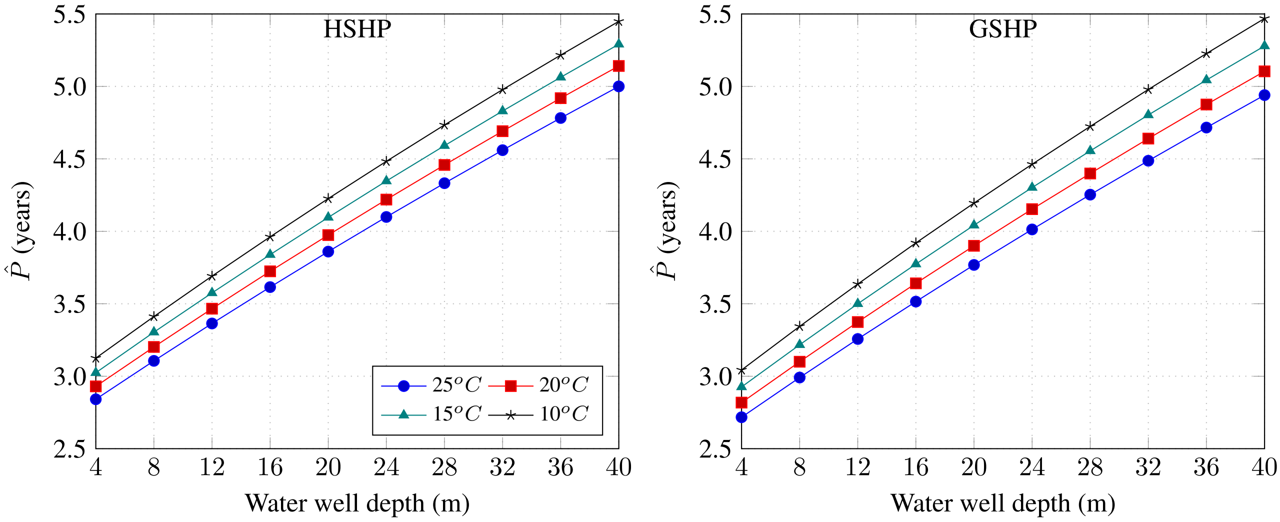

Figure 5. Variation of payback for different ground water temperature and water well depth.

6. The COP of HSHP mode is always better than GSHP mode and SAHP mode. The lower COP, in SAHP and HSHP in June is explained by the lower ambient temperature and lower solar radiation than other months. The higher COP of GSHP in June is explained by the lower inlet water temperature that increases the effectiveness of condenser and reduces the condensation temperature. The COP of SAHP mode in January is 1.5% higher than the COP reported by Diniz *et al.* (2017b) during an experimental test made in January of 2016 using a R134a SAHP in Belo Horizonte. The COP of SAHP mode in February is 3.0% higher than the mean COP reported by Diniz *et al.* (2017a) during an experimental test made in February of 2017 using a R134a SAHP in Belo Horizonte. The mean annual COP of SAHP, GSHP and HSHP mode are 2.60, 3.24 and 3.37, respectively. Lazzarin (2012) performed a study using a HSHP equipped with solar collector of $10.74m^2$ and GHE of 200m considering the Milan climate. The COP reported by Lazzarin (2012) is 8.3% higher than the one found in this work. A COP 3.8% lower than the COP of SAHP mode was found by Rabelo *et al.* (2019) for a SAHP using R290 and a solar collector of $2.3m^2$ and considering the Belo Horizonte climate.

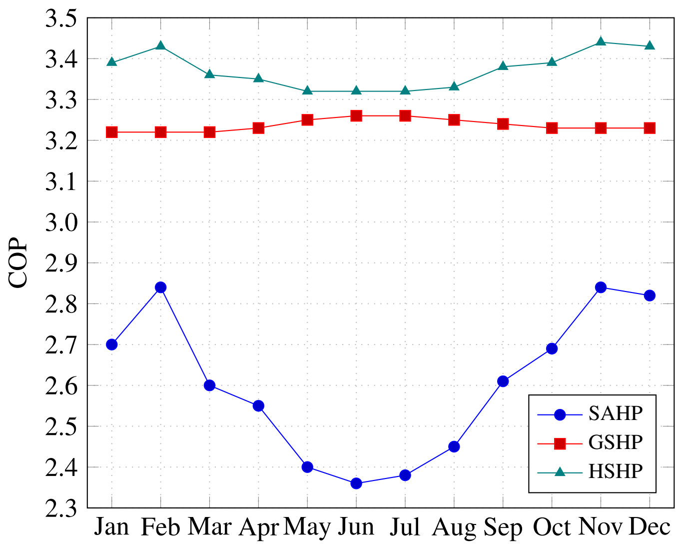

Figure 6. Variation of monthly COP of the system operating in Belo Horizonte.

The payback of the SAHP, GSHP and HSHP are 2.24, 3.49 and 3.40 years, considering the climate of Belo Horizonte. In the results presented by Reis *et al.* (2014), the payback of a R134a SAHP in the Belo Horizonte is 2.5 years, 11% higher than the SAHP used in this work. A payback 2.6% higher than the SAHP presented in this work was found by Rabelo *et al.* (2019) for a SAHP using R290 and a solar collector of $2.3m^2$ in the Belo Horizonte climate. A payback 12% lower than the SAHP presented in this work was found by Hawlader *et al.* (2001) for a SAHP using R134a and a solar collector of $4m^2$ in the Singapore climate. Considering the scenario of the house that already has a water well, so there is no drilling cost, the best solution is a GSHP with payback of 2.21 years, followed by SAHP with 2.24 years and HSHP with 2.34 years.

4. CONCLUSIONS

In this paper, an energetic and economic analysis of a R290 Hybrid Source Heat Pump for producing domestic hot water was carried out using a mathematical model. The system considered has an open loop ground underground (aquifer) water and an uncovered solar collector that can be use separately or connected in series topology. The refrigerant chosen for the heat pump was R290 (propane). This theoretical analysis was made using lumped models for each component of the system.

The results show that the COP of a HSHP mode is better than GSHP, ASHP, SAHP for most environmental conditions but the best payback is obtained using a SAHP. The case study performed considering the climate of Belo Horizonte showed that the mean annual COP are 2.60, 3.24 and 3.37 and the payback are 2.24, 3.49 and 3.40 years for SAHP, GSHP and HSHP respectively. The results of COP and payback were compared with other papers and the maximum difference was 8.3% and 12%, respectively. Considering the scenario of the house that already has a water well, so there is no drilling cost, the best payback is obtained using a GSHP.

5. ACKNOWLEDGEMENTS

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