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FEASIBILITY OF THERMAL STORAGE FOR AIR CONDITIONER: A CASE OF STUDY FOR OFFICE APPLICATIONS

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Abstract. *The high cost of electricity has done that the air condition sector search for alternatives to produce cold efficiently. It is a necessity, especially in Brazil, where 17% of the total electricity consume is due air condition systems. This work presents a comparative analysis of conventional and an air conditioning with thermal storage system. The systems are design for cooling an office considering the climate of Belo Horizonte, Brazil. The refrigerant chosen was the R410A due availability of commercial components. In order to evaluate the performance of the air conditioner with and without a thermal storage a quasi-steady-state model was developed using the Equation Engineering Solver (EES). The simulations was made considering the climate of Belo Horizonte, Brazil using weather data obtained from June of 2019 to May of 2020. The results show that the use of thermal storage increases the average COP in 6% and cooling capacity in 1.7%, reduces the number of running hours in 2% but increase the energy consumption in 42%.*

Keywords: *Thermal storage, HVAC, Mathematical model, Payback*

1. INTRODUCTION

Two currently goal in air condition sector are to produce cold efficiently and if possible off-peak times. It occurs because of the high electricity consumption in the air condition sector. In 2017, the air condition system electricity consume represents 17% of the total electricity consume in in Brazilian residences (EPE, 2018). In addition, there is a study that informed that the air conditioning systems accounts for 85% of consumption of residential electricity at the peak of summer (Li *et al.*, 2019). Furthermore, air conditioning systems consumes 40.3% of electricity in commercial sector (PROCEL, 2008).

Focus on this large electricity demand, different propositions are studied, among them combined air condition and thermal storage systems (Zhou *et al.*, 2012; Zhai *et al.*, 2013; Li *et al.*, 2019). Through thermal storage systems it is possible that the cold production occurs outside of peak time (Dincer, 2002) while simultaneously it could happens at times when the thermal energy available in the environment increases the efficiency.

Upshaw *et al.* (2015) developed a modeling to study peak load reduction and energy consumption for residential air conditioning systems that integrated thermal energy storage in water. The results showed a reduction between 29 and 53% on-peak compressor power demand and an increase between 5 and 15% in the total compressor energy consumption, compared the purpose system with a traditional air conditioning. The authors highlighted that the additional energy consumption occurs out off-peak hours. Said and Hassan (2018) presented a study that coupled air conditioning and thermal energy storage. The phase change material (PCM) is used in thermal energy storage. They integrated PCM with the condenser of the air-conditioning unit. At night the PCM storage cold that will used during the day in the air conditioning unit. The results indicate that it saves 6.7% of power per ton refrigeration for each PCM kilogram,

considering inlet air velocity and temperature of 0.96m/s and 40^{o}C, respectively, compared its with a conventional system.
This paper studies the feasibility of integrated a thermal energy storage with an air conditioning system that is used in an office.

2. METHODOLOGY

The two system considered in this study are shown in Fig. 1. In the conventional system the main components are the hermetic reciprocating compressor, two cross flow unmixed air-cooled heat exchanger and a capillary tube. The data used in the model are based in the air conditioner model HAFE09B2 manufactured by Elgin. The main difference of the conventional system and proposed system is the coaxial water-cooled condenser that is cooled by water pumped from a cold water storage. An important step of this work is the economic analysis, therefore the refrigerant chosen is the R410A due availability of commercial components although there is studies (Tian *et al.*, 2015) to replace the use R410A in this type of device due the its high impact in global warming.

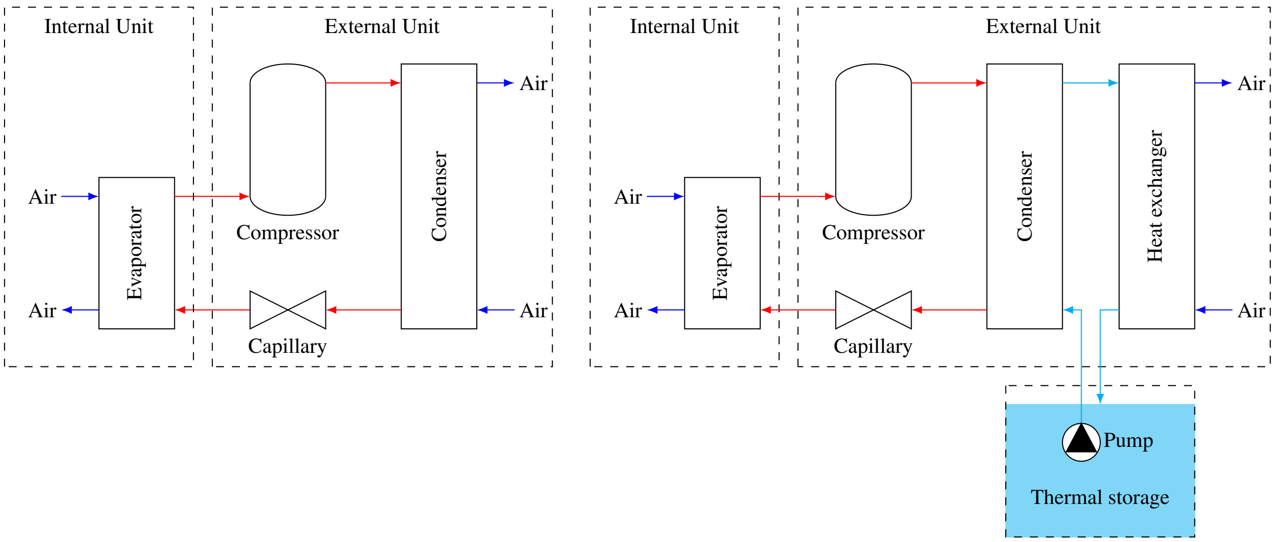

Figure 1. Main components of conventional and proposed air conditioner system.

2.1 Air conditioner mathematical model

In order to evaluate the performance of the geothermal and conventional air conditioner a quasi-steady-state model was developed using the Equation Engineering Solver (EES). The losses in the tubes between components was considered negligible. The evaporator and condenser was assumed as isobaric and a lumped model was used. Following is described the modelling equation for each component.

2.1.1 Compressor model

The hermetic compressor selected for this work is the model HGA5494BXD manufactured by Tecumseh. In the literature, there are some very detailed model of compressors (Duprez *et al.*, 2007; Ndiaye and Bernier, 2010; Duarte *et al.*, 2019) but these models required many parameters and geometrical details that are not provide by the manufactures of hermetic compressors. Additionally, the compressor model used in the complete model of the refrigeration system is a simple one, assuming a isentropic compression process, as those employed in other works (Minetto, 2011; Rabelo *et al.*, 2019; de Paula *et al.*, 2020). The manufacturer of the compressor presents the following equation to evaluate the R410A mass flow rate ($\dot{m}$) and the electric input power ($\dot{W}_{cp}$) in function of the evaporation temperature (T_e) and condensation temperature (T_c).

$$\dot{m}_m = B_1 + B_2T_e + B_4T_e^2 + B_7T_e^3 + \left(B_3 + B_5T_e + B_8T_e^2\right)T_c + \left(B_6 + B_9T_e\right)T_c^2 + B_{10}T_c^3 \quad (1)$$

$$\dot{W}_m = B_1 + B_2T_e + B_4T_e^2 + B_7T_e^3 + \left(B_3 + B_5T_e + B_8T_e^2\right)T_c + \left(B_6 + B_9T_e\right)T_c^2 + B_{10}T_c^3 \quad (2)$$

The coefficients of the Eq. 1 and 2 are listed in Tab. 1. To obtain the Eq. 1 the manufacturer performed test with a fixed temperature at the inlet of the compressor (35^{o}C). For lower evaporation temperature, this condition is not achieve in air conditioner. In order to correct the mass flow rate in the Eq. 1, AHRI (2005) suggest the Eq. 3 to evaluate the refrigerant

Table 1. Parameters of the equations of the compressor

	B_1	B_2	B_3	B_5	B_5
$\dot{m}_m$	1.975964E-2	-9.623655E-5	-1.159600E-4	-8.356414E-6	2.575877E-5
$\dot{W}_{cp}$	1.485628E+2	-9.748167E+0	2.360980E+1	-1.372568E+0	4.168670E-01
	B_6	B_7	B_8	B_9	B_{10}
$\dot{m}_m$	-1.557530E-6	6.290759E-8	5.195857E-7	-3.186213E-7	2.287479E-8
$\dot{W}_{cp}$	-3.283619E-1	4.620434E-2	4.604102E-3	-9.753143E-4	2.993434E-3

corrected mass flow rate ($\dot{m}_r$). AHRI (2005) comment that the variation on super-heating has negligible effects in power consumption.

$$\dot{m}_r = \frac{\rho_1}{\rho_m}\dot{m}_m \tag{3}$$

In this equation, ρ_1 is the density at the inlet of the compressor, and ρ_m is the density calculated using the fixed temperature (35^{o}C) give by the manufacturer.

2.1.2 Heat exchangers

The evaporator and condenser are modeled using the using the effectiveness-NTU method as made by Rabelo *et al.* (2019). The balance of energy in the refrigerant and air in the heat exchangers are given by:

$$\dot{Q} = \dot{m}_r(i_o - i_i) = \dot{m}_a C_{pa}(T_i - T_o) \tag{4}$$

where $\dot{Q}$ is the heat transfer rate, i is the refrigerant specific entalphy, $\dot{m}_a$ is the air mass flow rate, C_{pa} is the air heat capacity at constant pressure and T is the air temperature. The subscripts i and o refers to inlet and outlet of the heat exchanger. In the water-cooled condenser, the properties of air are replaced by properties of water. The maximum heat transfer rate calculate by:

$$\dot{Q}_{max} = \dot{C}_{min}(T_{hi} - T_{ci}) \tag{5}$$

where $\dot{C}_{min}$ is the smaller heat capacity rate, T_{hi} is inlet temperature of hot fluid and T_{ci} is inlet temperature of cold fluid. The heat capacity rate and refrigerant heat capacity at constant pressure (C_{pr}) can be expressed as:

$$\dot{C} = C_p\dot{m} \tag{6}$$

$$C_{pr} = \frac{i_i - i_o}{T_i - T_o} \tag{7}$$

The effectiveness (ε) of heat exchanger is evaluated as follows Incropera *et al.* (2007):

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{8}$$

The effectiveness can also be expressed by Eq. 9 for cross-flow, single-pass, both fluids unmixed and by Eq. 10 for double pipe counter-flow heat exchanger (ASHRAE (2013)).

$$\varepsilon = 1 - \exp\left[\left(\frac{\dot{C}_{max}}{\dot{C}_{min}}\right)\mathrm{NTU}^{0.22}\left\{\exp\left[-\left(\frac{\dot{C}_{min}}{\dot{C}_{max}}\right)\mathrm{NTU}^{0.78}\right] - 1\right\}\right] \tag{9}$$

$$\varepsilon = \frac{1 - \exp[-\mathrm{NTU}(1 - \dot{C}_{min}/\dot{C}_{max})]}{1 - \exp[-\mathrm{NTU}(1 - \dot{C}_{min}/\dot{C}_{max})]\dot{C}_{min}/\dot{C}_{max}} \tag{10}$$

In these equation NTU is number of transfer units and is expressed as:

$$\mathrm{NTU} = \frac{UA}{\dot{C}_{min}} \tag{11}$$

where A is the heat transfer surface area of the heat exchanger and U is the overall heat transfer coefficient given by:

$$U = \left(\frac{1}{\bar{h}_a} + \frac{1}{\bar{h}_r}\right)^{-1} \tag{12}$$

the mean refrigerant heat transfer coefficient ($\bar{h}_r$) is calculated integrating numerically in function of specific entalphy as made by Zhang *et al.* (2014). In the condenser the correlation of Gnielinski (1976) is used if $i \geq i_V$ or $i \leq i_L$ and the correlation of Shah (2016) is used if $i_L < i < i_V$. In the evaporator the correlation of Gnielinski (1976) is used if $i \geq i_V$ and the correlation of Shah (2017) is used if $i_L < i < i_V$. To evaluate the mean air heat transfer coefficient ($\bar{h}_a$), Incropera *et al.* (2007) suggest the correlation of Grimson (1937), that it is given by:

$$\bar{h}_a = B_0 Re_{max}^m Pr^{1/3}(k/D) \tag{13}$$

where the k is the thermal conductivity, D is external diameters of the tube and Pr is the Prandtl number. The coefficients B_0 and m are listed for aligned or staggered tubes and for different transverse pitch and longitudinal pitch. In this work, due the geometry of the heat exchangers the B_0 coefficient is 0.4545764 for the condenser and for thermal storage heat exchange, 0.38307 for the evaporator and the m coefficient is 0.568 for both heat exchanger. The Re_{max}, for the specific geometry of heat exchanger used in this work, is given by:

$$Re_{max} = \frac{2\rho V D}{\mu} \tag{14}$$

where V is the air velocity at the inlet of evaporator and μ is the viscosity. In the water-cooled condenser, air heat transfer coefficient are replaced water heat transfer coefficient in the Eq. 12. The mean water HTC ($\bar{h}_w$) is calculated using the correlations described by Rohsenow *et al.* (1998) for flow in annular regions.

In fact the pressure of the refrigerant is not known and not appear in none of the equations present. Therefore a value for the pressure of the refrigerant is estimated and an error is calculated comparing the effectiveness of Eq. 8 and 9 and secant method describe by Chapra and Canale (2008) is used to find a pressure that gives an error lower than 0.1%. For the water-cooled condenser, the error is calculated comparing the effectiveness of Eq. 8 and 10.

The balance of energy in the air in the evaporator is evaluated from EQ. 15 to compute the condensation of water present in the air.

$$\dot{Q}_e = \dot{m}_a C_p (T_o - T_i) + \dot{m}_{ce} i_{fg} \tag{15}$$

where $\dot{m}_{ce}$ is the mass flow rate of condensation water in the evaporator. The data of heat exchangers are listed in Tab. 2.

2.1.3 Pump

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

$$\dot{W}_p = \frac{\dot{m}_w}{\rho_w \eta_p} \sum \Delta P_f \tag{16}$$

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. 17. 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.

$$\Delta P_f = \frac{8 f L \dot{m}_w^2}{\pi^2 D_i^5 \rho_w} \tag{17}$$

The pump selected for this work is manufactured by Dancor model DS4. The pump head (H) and overall efficiency (η_p) is given by EQ. 18 and 19, that were adjusted using the data from manufactured catalog. The mass flow of the pump is calculate equalling the losses in Eq. 17 to the head in Eq. 18.

$$H = 2.1739\dot{m}_w^2 - 1.0519\dot{m}_w + 6 \tag{18}$$

$$\eta_p = 1.0319\dot{m}_w - 0.7023\dot{m}_w^2 \tag{19}$$

2.2 Thermal zone and thermal storage

To compare the performance of the system with thermal storage with a convectional system, an office cooling was chosen as application. The office chosen has length of 3m, width of 3m and wall height of 2.8m. The power generate ($\dot{Q}_g$) inside the office is given by:

$$\dot{Q}_g = N_p F_p + \dot{Q}_{eq} \tag{20}$$

where N_p is the number of people, F_P the power generation per person factor, A_f in the ceiling or floor area and $\dot{Q}_{eq}$ the power generation by electrical equipment. F_P and $\dot{Q}_{eq}$ is listed in ABNT (2008) for different applications and equipment,

Table 2. Main simulation parameters

Parameter	Value	Parameter	Value	Parameter	Value
Atmospheric Pressure	92 kPa	Evaporator fan power	33W	Evaporator fan power	45W
Superheating	10^oC	Evaporator fan mass flow	500kg/h	Condenser fan mass flow	1100kg/h
Sub-cooling	5^oC	Evaporator heat transfer area	$3.5m^2$	Condenser heat transfer area	$5\ m^2$
Electricity tariff	0.95 R$/kWh	Evaporator frontal area	$775cm^2$	Condenser frontal area	$876\ cm^2$
Wall overall heat transfer	$2.58\ W/(m^2K)$	Moisture removal	1kg/h	Water storage size	$2m^3$
Floor overall heat transfer	$1.59\ W/(m^2K)$	Number of people	4	Evaporator/Condenser diameter	5/16 in
Ceiling overall heat transfer	$1.92\ W/(m^2K)$	Ceiling thermal delay	3.6 h	Ceiling solar absorptivity	0.3
Storage overall heat transfer	$0.77\ W/(m^2K)$	AC + thermal storage cost	3000R$	AC Convectional cost	1800R$
Storage size	$2\ m^3$	Storage heat transfer area	$8.4\ m^2$	Simulation time step	5 min

the values selected are 130W/person and 720W. The power generation by electrical equipment take account computers, printers and lighting. The heat transfer rate through the wrapper ($\dot{Q}_{wr}$) (walls, floor and ceiling) is evaluated by:

$$\dot{Q}_{wr} = UA(T_{ex} - T_{in}) \tag{21}$$

where A is area of heat transfer, T_{ex} is the external temperature and T_{in} is the internal temperature. The values of overall heat transfer U listed by ABNT (2005b,c) different materials and constructive system typically used in Brazil. The properties of isolating materiel in the themal storage is listed by Incropera *et al.* (2007). The ABNT (2005b) suggest the EQ. 22 to evaluate heat gain due solar radiation.

$$\dot{Q}_{sol} = 0.04UA\alpha I \tag{22}$$

In this equation α is the solar absorptivity and I is the solar radiation. ABNT (2005b,c) presents values for solar absorptivity and thermal delay (ϕ). The thermal delay is defined by ABNT (2005a) as time elapsed between a thermal variation in a medium and its manifestation on the opposite surface of a constructive component subjected to a periodic regime of heat transmission. Therefor the solar radiation used in Eq. is evaluate at time $t = t - \phi$. Finally, the variation of internal temperature in given by:

$$T_{in} - T_{in}^0 = \frac{\Delta t}{C}(\dot{Q}_g + \dot{Q}_{sol} + \dot{Q}_{wr} - \dot{Q}_e) \tag{23}$$

where T_{in}^0 is the internal temperature evaluated at time $t = t - \Delta t$ and C thermal capacitance. The value of thermal capacitance used is for the office is 750kJ/K that was estimate take account desks, chairs, computers, printers, air, evaporator unit, cabinet, paper, etc. The terms in the right of Eq. 23 were calculate at using T_{in}^0. The thermal storage model uses only the EQ. 21 and 23 but with the water temperature (T_w) instead of T_{in}^0. The value of thermal capacitance in the thermal storage is the product of mass and heat capacity of water in thermal storage.

2.3 Performance indicators

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

$$COP = \frac{\dot{Q}_e}{\dot{W}_{cp} + \dot{W}_e + \dot{W}_{cd}} \tag{24}$$

where $\dot{W}_e$ is the power consumed by the evaporator fan and $\dot{W}_{cd}$ is the power consumed by the condenser fan or pump that were assumed constant and they are listed in Tab. 2. The air mass flow rate and fan power were assumed constant. The costs in Tab. 2 are based in the Belo Horizonte market in March of 2020.

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3. RESULTS

To compare the performance of a convectional air conditioner and an air conditioner with thermal storage, the hourly climate data recorded in the Pampulha (Belo Horizonte, MG) whether station were considered. The period between June of 2019 and May of 2020 was analyzed. The sub-model of thermal zone was used to obtained the month cooling capacity. The month with highest cooling capacity (2.9kW) was September and the month with lowers cooling capacity was May. The solar radiation and external air temperature obtained from INMET website for the mentioned whether station is shown in Fig. 3 and 2. The storage temperature (T_w) and air conditioner COP is also shown in Fig. 3 and 2 for both setup.

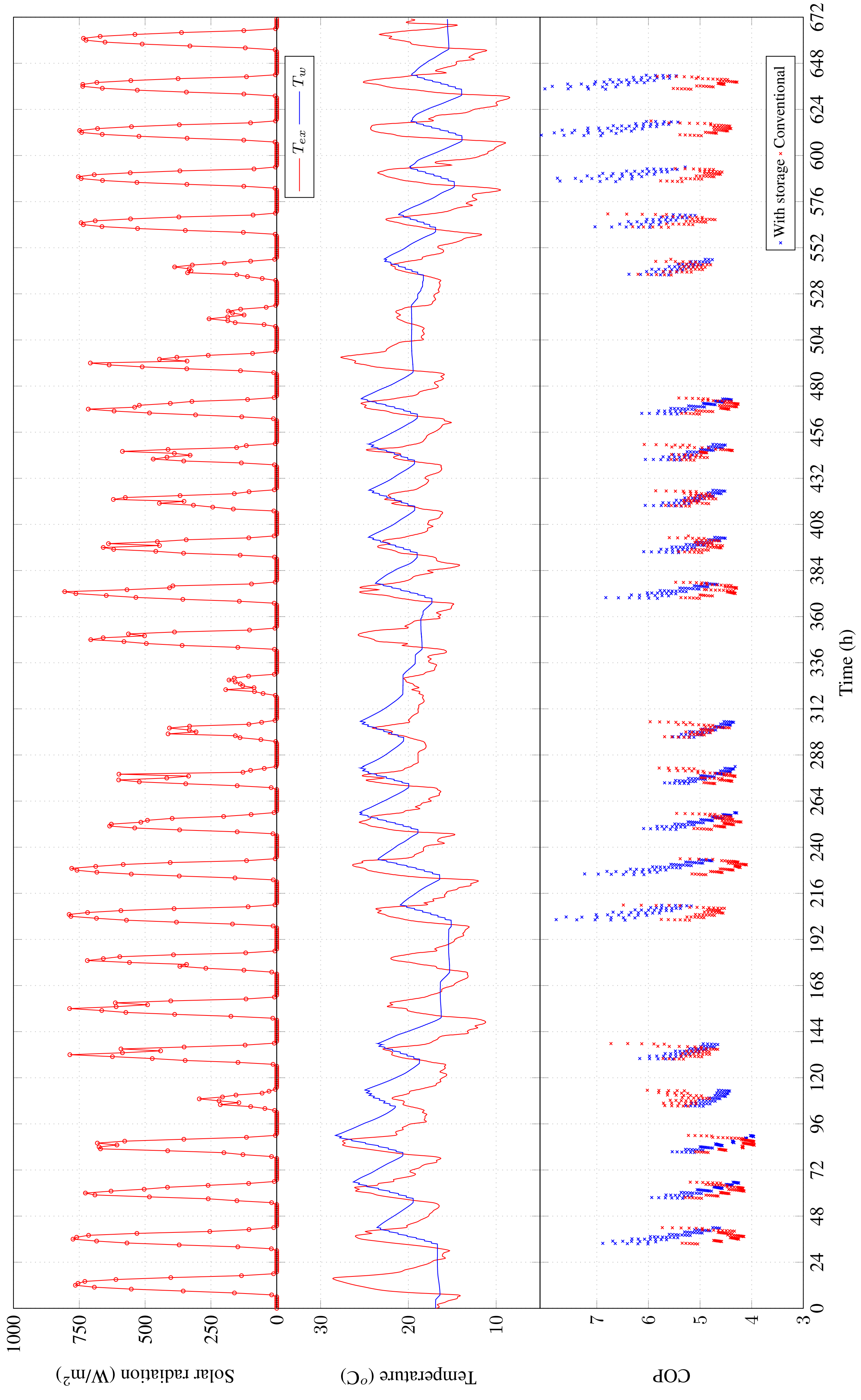

Figure 2. Solar radiation, temperatures and COP for May 3 to May 30, 2020

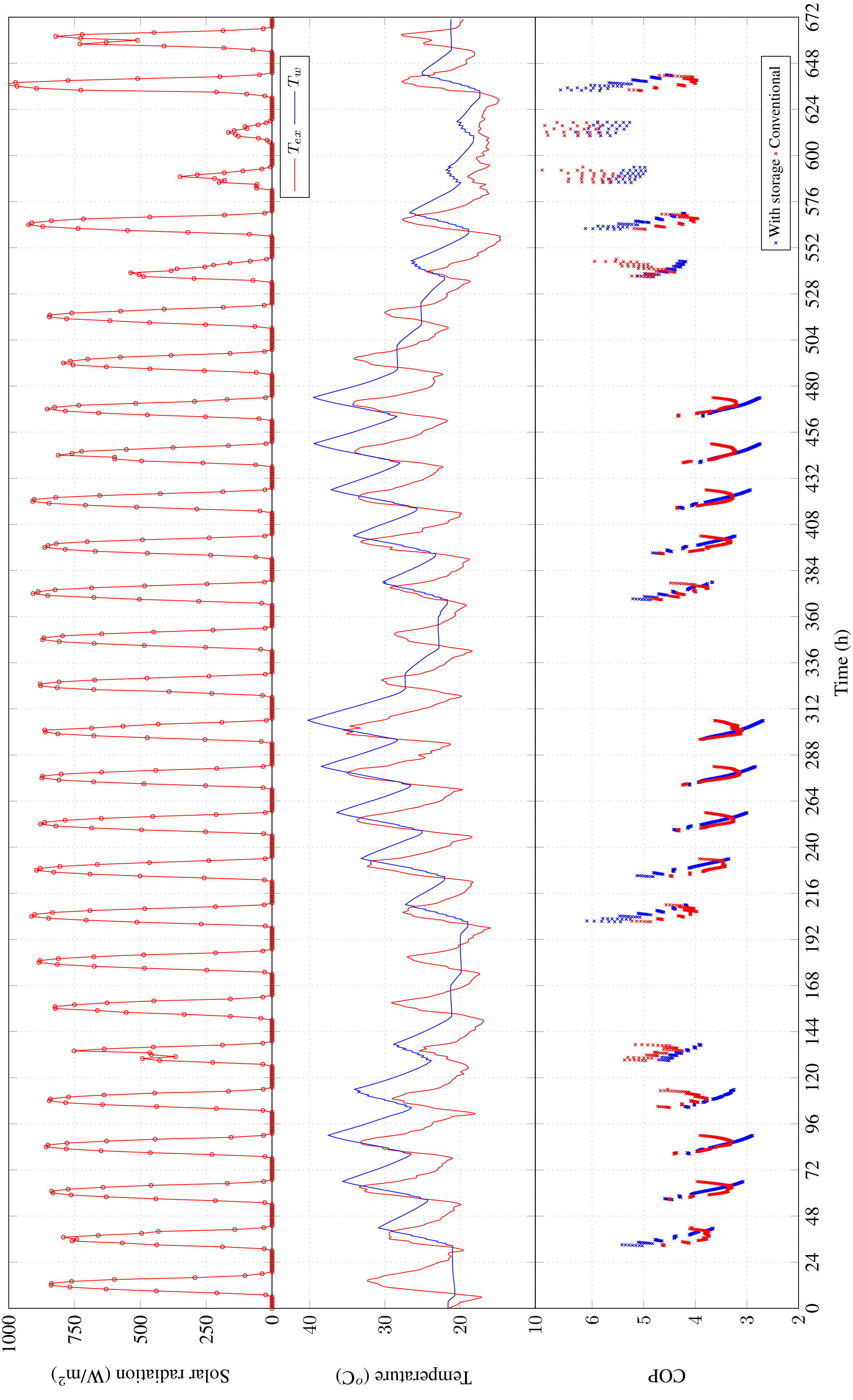

Figure 3. Solar radiation, temperatures and COP for September 1 to 28, 2019

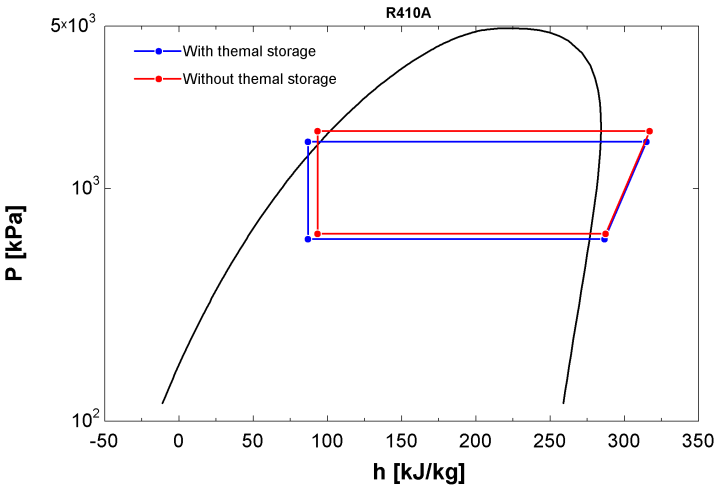

Figure 4. Diagram pressure versus specific enthalpy for the first time that the air conditioner starts at May of 2020.

In figure 2, most of the time the COP of system with thermal storage is higher than the COP of system without thermal storage. The mean COP of proposed system is 5.36 and the COP of conventional system is 4.85, a difference of 11%. The higher COP for system with thermal can be explain using FIG. 2 where most of the time the water temperature is lower than external air temperature. Unlike figure 2 in figure 3 the is an equilibrium, in some moments the COP of system with thermal storage is higher than the COP of system without thermal storage and in some moments a better COP is obtained conventional system. In September, the mean COP of proposed system is 3.89 and the COP of conventional system is 3.87, a difference of 1%.

The number of running hours in May for proposed system is 78.75 and the COP of conventional system is 81.8, a reduction of 4%. The difference of number hours is due the difference of cooling capacity. For example, the first time that the air conditioner starts, the water temperature is 16.7^oC and the air temperature is 20.8^oC, the diagram pressure versus for that moment is shown in FiG. 4. In this figure the difference of entalphy in the evaporator is lower without the thermal than with the thermal storage. In fact, this figure show a increase of pressure ratio (P_2/P_1), comparing the system without the thermal and with the thermal storage, that reduces the mass flow rate of compressor and also reduces the cooling capacity. In figure 4 the cooling capacity is 6% lower for conventional AC than AC with the thermal storage. The mean cooling capacity of proposed system is 3.4% higher than convectional system in May of 2020. The number of running hours in September for proposed system is 140.4 and the COP of conventional system is 139.2. The mean cooling capacity of proposed system is the same that convectional system in September, 2.7kW.

The consumption of energy in May is 49% higher in the system with thermal storage than without thermal storage. This result is justified due energy required to the pump and fan that operate in the hours that cooling capacity is not produced but they still running to reduce the water temperature. The consumption of energy in May is 34% higher in the system with thermal storage. An increase of energy consumption with the use of thermal storage was also found by Upshaw *et al.* (2015). A sensitivity analysis was made to verify the influence of the thermal storage size, a new simulation was made considering the climate data of September and storage size of 3m^3 the COP increased only 1% and the energy consumption increased 2.5%.

4. CONCLUSIONS

In this paper, an energetic and economic analysis of a R410A air conditioner with and without a thermal storage was carried out using a mathematical model. To compare the performance of both system, an office cooling was used as application. The simulations was made considering the climate of Belo Horizonte, Brazil using weather data obtained from June of 2019 to May of 2020. The results show that the use of thermal storage increases the average COP in 6% and cooling capacity in 1.7%, reduces the number of running hours in 2% but increase the energy consumption in 42%.

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