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

COBEM-2017-2566
THERMAL PERFORMANCE OF A HEAT PUMP WITH SOLAR EVAPORATOR UNDER ZERO SOLAR RADIATION CONDITIONS

Hélio Augusto Goulart Diniz
Universidade Federal de Minas Gerais, Mechanical Engineering Department, 31270091, Belo Horizonte, MG, Brazil
helioufmg@gmail.com

Fernando Neves Quintino dos Santos
Centro Federal de Educação Tecnológica de Minas Gerais, Energy Engineering Department, 30510000, Belo Horizonte, MG, Brazil
fernandonquintino@gmail.com

Juan Jose Garcia Pabon
Ricardo Nicolau Nassar Koury
Luiz Machado
Universidade Federal de Minas Gerais, Mechanical Engineering Department, 31270091, Belo Horizonte, MG, Brazil
langa_27@hotmail.com
koury@demec.ufmg.br
luizm@ufmg.br

Abstract. *Heat pumps are a feasible way to heat water at low costs. In regions of high insolation, the use of solar evaporators raises the performance of the heat pump, however little data exists within cloudy and rainy environments. The objective of this paper is to analyze the thermal performance of a direct expansion heat pump assisted with a solar evaporator and a coaxial condenser, used to fill a 200 L tank with hot water. The prototype was tested under no direct solar exposure, in an in-door laboratory environment. The heating capacity, compressor power and a coefficient of performance showed little variations. The heat transfer rate from the water that condensate in the evaporator provided to be significant, approximately 46 % of energy the fluid absorbed in the evaporator. Due to the water condensate, the use of heat pumps with solar evaporators become feasible even cloudy environments or at night.*

Keywords: *heat pump, solar evaporator, water heating, R134a, coaxial condenser*

1. INTRODUCTION

Heat pumps are a feasible way to heat water at low cost. In most applications, the evaporator exchanges heat with the air via forced convection. In places with high insolation however, the usage of an integrated solar collector-evaporator is beneficial because the evaporator turns able to harness radiation heat to the refrigerant (Buker and Riffat, 2016). Direct expansion solar assisted heat pumps (DX-SAHP) have the evaporator integrated to a solar collector, where the refrigerant evaporates shortly after expansion. This system is interesting in relation to other SAHP configurations because it has better thermodynamic performance, lower cost, longer life, and allows for heat storage (Omojaro and Breitkopf, 2013).

Chatuverdi et al. (2014) performed a thermo-economic analysis for a DX-SAHP operating in Virgina (USA), using a life cycle cost method. Results pointed that the system is appropriated to heat water to 50-70 ºC. Li and Yang (2009) compared different solar water heating technologies to heat 280 L of water in Hong Kong weather conditions, and found that DX-SAHP provides second-to-best economic returns, prior only to the solar assisted air-source heat pump.

Literature contains several studies on DX-SAHP in sunny environment, with coefficient of performance (COP) ranging from 2-6 (Li et al., 2007; Anderson et al., 2007; Sun et al., 2015), however few considered zero solar radiation conditions and its effects (Scarpa and Tagliafico, 2016; Fernández-Seara et al., 2012). In this situation, it is common that the evaporator temperature falls below the dew point, increasing water condensation. The objective of this work is to evaluate experimentally the thermal performance and condensate heat transfer rate of a DX-SAHP operating under zero solar radiation conditions, for heating 200 L of water.

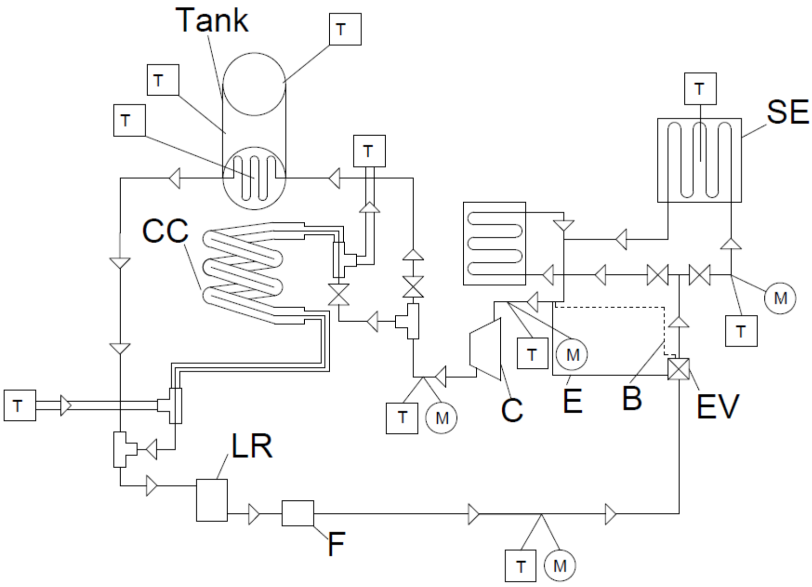

Figure 1. DX-SAHP diagram – SE: solar evaporator, EV: thermostatic expansion valve; C: compressor; CC: coaxial condenser; LR: liquid receiver, B: bulb, E: equalizer, F: filter dryer, M: bourdon gage, T: thermocouple

2. SYSTEM DESCRIPTION

Figure 1 depicts the schematic diagram of the heat pump in the present study. An 1/3 HP R134a hermetic compressor with volumetric displacement of 7.95 cm³, a thermostatic expansion valve, an unglazed flat-plate collector-evaporator (installed with a slope of 30º), a coaxial condenser (heating water from 30 °C to 45 °C), a 200 L isolated water reservoir and auxiliary components comprises the DX-SAHP system.

The evaporator coil is made of copper with internal and external diameters of 8.73 mm and 9.53 mm respectively. Straight tubes total length is 16.0 m, while the return bends outside the plate totals 1.28 m. The unglazed plate was made of aluminum, 1.03 m wide, 1.60 m long and 1 mm thick. The plate received black painting to maximize irradiation absorption. We measured an emissivity of 0.95 using a thermographic camera and assumed gray body behavior for absorptivity. Two pyranometers measured the flux of radiation in the solar evaporator, one horizontal and one inclined 30º (same plane of the collector).

The condenser is made of two concentric cooper tubes. It is 5.5 m long and 92.1 mm high, and spirals in 2.7 turns with a diameter of 0.65 m. The inner tube has internal and external diameters of 4.76 mm and 6.35 mm, while the outer tube has diameters of 11.11 mm and 12.70 mm respectively. R134a flows into the central pipe while water flows countercurrent into the annular space. A 9 mm thick polyethylene insulation tube covered the condenser with a thermal conductivity of 0.038 W/m.K, to reduce heat losses to the environment.

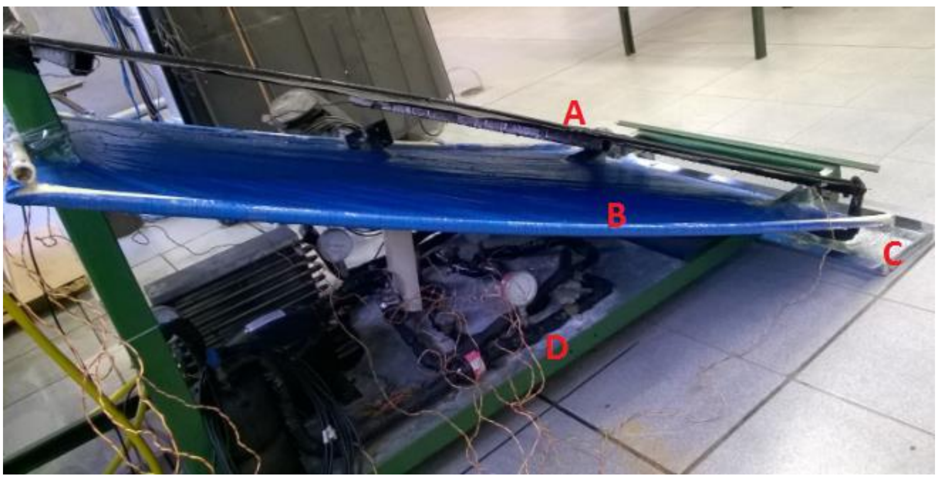

Figure 2. Side view of the heat pump (A) Solar evaporator (B) Condensate collector system (C) Condensate accumulator (D) Main components.

Figure 3 details the tarpaulin gutter devised to collect the condensate and a beaker used for measurement. Scarpa and Tagliafico (2016) used a similar device in their research. The condensate formed on both sides of the collector flowed down to the lower portion of the plate, effectively dripping onto the installed device.

Figure 3. Condensate storage in the tarpaulin gutter and the beaker used for periodic evaluation

Bourdon gauges, Type K thermocouples, psychrometer and an analog energy meter were used to measure pressure, temperature, humidity and energy consumption respectively. Data acquisition system consisted of a USB-9162 - 24 bits for the thermocouples, connected in Labview® environment, which read and treated all data signals. Table 1 shows the measurement uncertainties of the measuring instruments used.

Table 1. Uncertainty of the measurement instruments

Measuring instrument	Uncertainty
Type K thermocouple	$\pm$ 1 ºC
Bourdon tube (low pressure line)	$\pm$ 0,1 bar
Bourden tube (high pressure line)	$\pm$ 0,35 kgf/cm²
Digital psychrometer	$\pm$ 1 °C (room temperature)
	$\pm$ 2 °C (dew point temperature)
Energy meter	$\pm$ 1%
Beaker	$\pm$ 50 ml
Graduated test tube	$\pm$ 2 ml
Tank (water level)	$\pm$ 5%
Thermographic camera	$\pm$ 0,05 (emissivity)

3. METHODOLOGY

A test was conducted on 01/16/2016 (summer) starting at 13:30. Prior to it, the water flow was manually set so that the temperature at the inlet of the tank was 45 °C. This flow rate was measured by timing the time required to fill a 1 L beaker with circulating water in the system. Three measurements were taken and the mean value was adopted. Although not performed at the beginning of the test, the flow rate varied very little.

The assay contained 29 measurements performed at fixed intervals of 15 minutes. At each measurement interval, the solar irradiance at the evaporator, the temperature and pressure at the inlet and outlet of each component, the water temperature at the inlet and outlet of the coaxial condenser, the ambient temperature, the average plate temperature, and the electric power consumed by the compressor were measured. The test stopped when the tank was filled with 200 L of water.

The machine used had only one passage for water; thus, the principle of continuity ensures that the mass flow of water in all components is equal.

3.1 Condenser

Equation (1) provides an energy balance for the water in the condenser. The ‘w’ subscribe stands for water, while the inlet and outlet refers to the condenser.

$$\dot{Q}_w = \dot{m}_w c_p (T_{w,outlet} - T_{w,inlet}) \quad (1)$$

This equation gives the heat transfer rate received by the water in the passage through the condenser. The specific heat, c_p, was evaluated at the average inlet and outlet temperatures of the condenser.

3.2 Compressor

Equation (2) provides the compression work. The interval of measurement, Δt, equals 15 minutes.

$$\dot{W} = \Delta E/\Delta t \tag{2}$$

3.3 COP

The real COP is defined as the amount of energy absorbed by the water by the amount of energy expended. Both measured quantities. The energy absorbed by the water is the product of the heating capacity by the interval of measurement, in this case, 15 minutes.

$$COP = \frac{\dot{Q}_W \cdot \Delta t}{\Delta E} \tag{3}$$

3.4 Condensate evaluation

The solar evaporator has four main heat sources: heat transfer from direct solar radiation from the sun, heat transfer from convection with the environment air to the plate, heat transfer from radiation between the collector and the environment, and heat transfer from vapor condensation. In most heat pump applications under sun exposure, the direct solar radiation is the dominant mechanism; however, in cases when there is no such exposure (e.g. nightime), the collector temperature falls below dew point, allowing for water condensation. Scarpa and Tagliafico (2016) performed tests in a DX-SAHP under no solar radiation and found that water condensation represents almost 25 % of the total heat transfer. This result points that water condensation must be taken in account for systems operating below dew point.

Equations (4-6), originally from Scarpa and Tagliafico (2016), were used to model the condensate mass transfer coefficient and the condensation heat transfer rate. This approach consisted in calculating the mass transfer coefficient for the condensation of a very dilute system on a cold surface, based on the analogy of heat transfer and mass, from the knowledge of the convective heat transfer coefficient at the air interface and board.

$$h_m = \frac{h_{conv}}{c_p \cdot \rho \cdot R \cdot T}\left(\frac{P_{air}}{P_v - P_{sat}}\right) In \left[\frac{P_{air} - P_{sat}}{P_{air} - P_v}\right] \tag{4}$$

$$\dot{Q}_{cond,model} = h_m(2A_{pl} + A_{rb})(P_v - P_{sat})i_{lv} \tag{5}$$

$$P_v = \phi P_{sat} \tag{6}$$

Where c_p, ρ, T are the constant-pressure specific heat, density and temperature, all relatively to air, while R is the gas constant of the water vapor. A_{pl} and A_{rb} are the areas of the collector (1.65 m²) and the coil return bends outside the evaporator (0.0382 m²), respectively. P_v is the partial pressure of the vapor while P_{sat} is the vapor saturation pressure at the collector temperature. i_{lv} is the latent heat of the water while ϕ is the relative humidity.

Equation (7) provides the modeled condensate mass flow. i'_{lv} is the modified latent heat proposed by Rohsenow (1956). $c_{p,w}$ is the constant-pressure specific heat of the water. T_{dp} is the dew-point temperature while T_{pl} is the collector temperature.

$$\dot{m}_{cond,model} = \frac{\dot{Q}_{cond,model}}{i'_{lv_w}} \tag{7}$$

$$i'_{lv} = i_{lv} + 0{,}68 c_{p,w}(T_{dp} - T_{pl}) \tag{8}$$

The experimental condensation heat rate was evaluated using the condensate mass flow. Equation (9) gives mass flow of the condensate, as function of the condensate collected in the gutter. ρ_w is the density of the water, while Δt_w is the time interval between condensate measurements, which is 1 h. Equation (10) provides the condensate heat rate.

$$\dot{m}_{cond,exp} = \frac{\rho_w V_{cond}}{\Delta t_w} \tag{9}$$

$$\dot{Q}_{cond,exp} = \dot{m}_{cond,exp} \cdot i'_{lv} \tag{10}$$

4. RESULTS

Test started at 13:30, for twenty-nine measurements. The water flow rate adjusted before the test was 28.3 L/h, with few fluctuations in the on-going trial. The condenser heated water from an average temperature of 25.0 °C to an average temperature of 46.0 °C, however the mean temperature of the water in the tank was 41.7 °C. This temperature drop is partially explained because rained during this particular trial, increasing the relative humidity, even though the machine was in-door. Both the air and the collector temperatures varied less than 1 ºC, with mean values of 24.9 °C and 8.9 °C, respectively.

The compressor power was 304 W and almost did not vary because evaporation and condensation temperatures were almost constant. The heating capacity was 685.2 W with average COP of 2.225. Figure 4 indicates the COP of the heat pump. In the entire test spam, COP oscillation did not reach 5 %. Figure 5 shows the condensate heat transfer rate during the test. It had an average of 176.4 W, with few fluctuations. The variations occurred mainly due to difference in relative humidity, as it was raining.

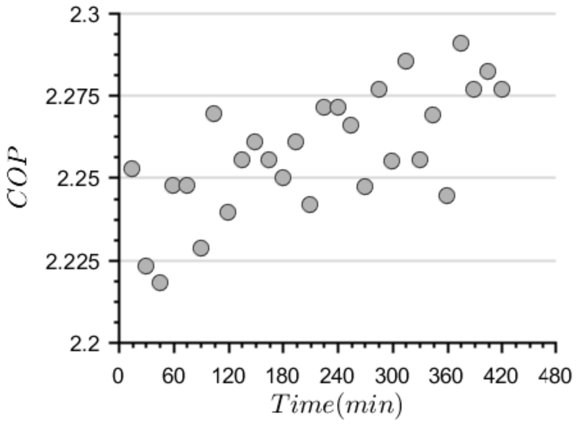

Figure 4. COP x time

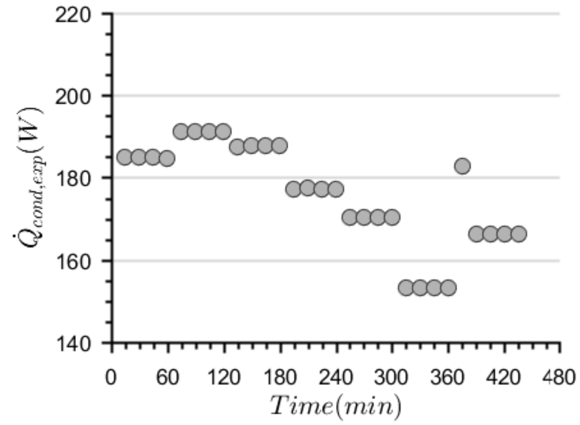

Figure 5. Condensate heat transfer rate x time

Figure 6 and 7 compares the modeled and experimental condensate mass flow and heat transfer rate. The model proposed by Scarpa and Tagliafica (2016) adjusted well to experimental data, however it presented a large deviation in some points. This could be due to experimental uncertainty, as small variations in the condensate mass flow led to large difference in the heat transfer rate. In spite of this dispersion, the matching tendency validates the condensate measuring procedure.

While the evaporator heat transfer rate was not investigated directly in this paper, it can be approximated by the heating capacity minus the compressor power, for an average of 381.3 W. That way, the condensate heat transfer rate becomes significant, representing 46 % of the heat absorbed by the evaporator fluid.

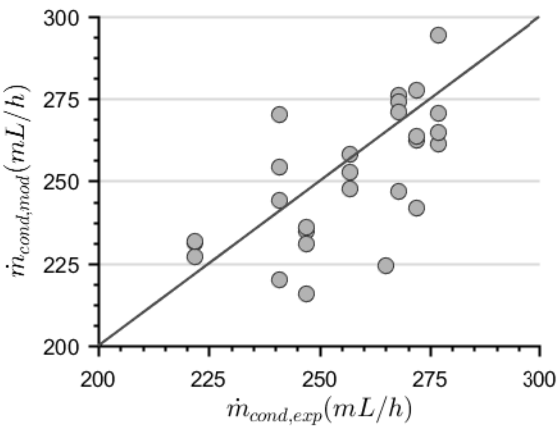

Figure 6. Condensate mass flow – Model x Experimental

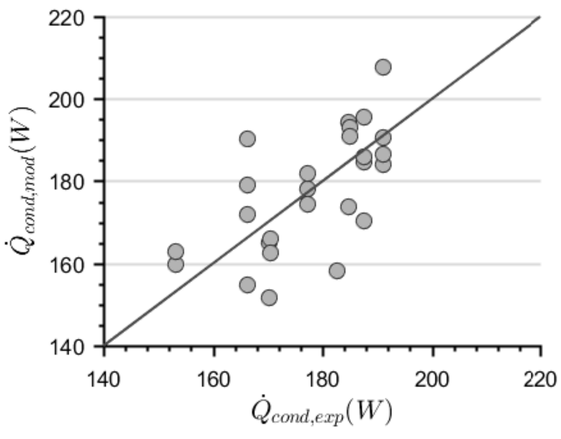

Figure 7. Condensate heat transfer rate – Model x Experimental

5. CONCLUSIONS

This study investigated the performance the thermal performance of a DX-SAHP with the solar evaporator operating below ambient temperature. The experiment took place on a partially rainy, in the summer of Belo Horizonte, which raised the relative humidity.

Main results are that water at 45.2 ºC filled a 200 L tank in 7.25 h, with a heating capacity of 685.2 W. The average heating COP was 2.225, low value mainly due to compressor poorly sized.

The mass flow and heat transfer rate of the water condensate were modeled using an empirical correlation from open literature. The good agreement validated the condensate measuring procedure used in this research. Based on these results, the condensate heat transfer rate reached almost 46 % of the total energy that the fluid absorbed in the evaporator.

The condensate heat transfer rate is significant when the evaporator works below ambient temperature, as is the case of cloudy and rainy environments, as well as nighttime. Due to this parcel, the use of a solar evaporator is feasible even when there is no sun exposure, making the DX-SAHP a viable alternative to direct solar water heating.

6. ACKNOWLEDGEMENTS

CAPES, FAPEMIG and UFMG's Mechanical Engineering Graduate Program for financial support.

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