

Energy evaluation of an evaporative cooling system using water driven ejector

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Abstract

The search for efficient and accessible cooling systems has increased worldwide. This study aims to build and evaluate an evaporative cooling system using a water driven ejector, allowing it to be installed in places with plenty of water. The system was investigated varying the flow rate and temperature of the circulating water, temperature of the replacement water, and coefficient of performance. The best vacuum obtained was 8.5 kPa at nominal operating conditions of 4.1 ± 0.1 m³/h and 5 ± 0.5 °C for the circulating water reaching the temperature of 9.7 ± 0.5 °C. The pulse-like disturbance generated by replacing the cooling water at different periods of times did not result in significant affect vacuum destabilization and the temperature rise in the cooling tank. The coefficient of performance of the system at the highest thermal power of 92.27 W was 0.077, which was underestimated due to possible problems related to pump efficiency. The system evaluated under the conditions proposed can be very efficient for cooling fluids at higher temperatures, and it can be complementary to main refrigeration systems.

Keywords: thermal storage; evaporative cooling; ejector.

1 Introduction

Industries and researchers have searched for more efficient and economical cooling systems. Thus, devices that modify the conventional cooling system or that are combined with it to improve efficiency and viability have emerged on the market.

The use of storage heaters to store energy at low temperatures during off-peak hours for later use during peak-demand periods has been subject of various studies, especially focusing on large-scale industrial processes. This system allows for financial savings of up to 70%, depending on the segment and on the working hours of the company (Thermoambiental, 2013).

An evaporative cooling system using ejectors is an alternative to a cooling system with viable cost. Ejectors are simple devices governed by Bernoulli's principle, in which a flow of air, water, or steam as fluid movement produces a pressure drop that can be used to suck up the vapor from the primary refrigerant; evaporation then occurs cooling it. Cooling occurs by removing the sensible heat of the fluid by evaporation (vaporization enthalpy).

Evaporative cooling system with steam ejectors are the most widely used, and therefore data about their dimension and geometry are already know making it easy the use of software for operation simulations. In contrast, water driven ejectors or ejectors that use liquid water as circulating fluid, are little used, which led to the investigation of its potential, allowing the possibility of developing a simple and viable cooling system in locals with plenty of water from natural waterfalls or obtained at a low cost.

These systems can be used in rural or industrial properties with abundant water resources. Depending on how the resource is available on the property, the system under study can produce

cooling without extra use of electric power by using, for example, the existing runoff water for producing low pressures.

The evaporative cooling system with water driven ejector can be used under limited conditions to replace conventional cooling systems, vacuum pumps, and steam ejectors among others, where there is fluid flow to be used implemented in the system. It is a system that requires low financial investment that offers environmental sustainability since it does not generate hazardous waste. It uses non-toxic, non-flammable, non-carcinogenic refrigerant; it uses water, an abundant resource in the country.

The studied system can be used for cooling any case in which there is indirect contact with the water cooled by the system. Unlike Rodrigues et al. (2012) and Cavalheiro et al. (2013), who investigated the direct contact of the vacuum and the food being cooled, in the present study, the water is cooled in the evaporative cooling system when circulating water passes through the ejector obstruction; this cooled water may be used for indirect cooling of a solid or liquid food, such as milk and coffee among others, in a jacketed tank.

This study aimed to develop, build, and evaluate an evaporative cooling system using a water driven ejector by assessing the effect of the operating conditions such as flow rate and temperature of the circulating water and the replacement water on the water temperature in the cooling tank. The coefficient of performance of the system was also evaluated.

2 Material and methods

The evaporative cooling system consisted of the following devices: tank for cooling water at a low pressure, which can also be used as storage tank; tank for circulating water (volume 80 L);

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tank for replacement of water (volume 60 mL); centrifugal pump (Thebe Th16) to pump the circulating water to the water driven ejector; water driven ejector with two compartments (convergent and divergent) (Figure 1); thermostatic bath for indirect cooling of the circulating water; vacuum hose for connecting the nozzle of the water driven ejector to the cooling tank; and an electrical resistance for thermal load simulation.

Figure 2 shows the apparatus used to assess the operating conditions.

Description of instrumentation and assembly: TQ-1, Circulating Tank; TQ-2, Cooling tank; TQ-3, Replacement Tank; V-1, Outlet ball valve in the circulating tank; V-2, Needle valve located after the centrifugal pump; V-3, Outlet ball valve in the replacement tank; V-4, Ball valve in the cooling tank;

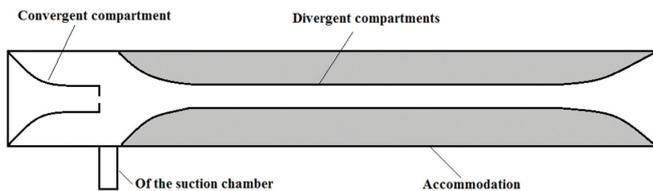


Figure 1. Water driven ejector diagram.

TT 101, Temperature transducer in the circulating tank 01; TT 102, Temperature transducer in the circulating tank 02; TT 103, Temperature transducer in the circulating tank 03; TT 104, Temperature transducer located after the centrifugal pump; TT 105, Temperature transducer located before the water driven ejector; TT 106, Temperature transducer located after the water driven ejector; TT 107, Temperature transducer in the cooling tank 01; TT 108, Temperature transducer in the cooling tank 02; TT 109, Temperature transducer in the cooling tank 03; TT 110, Temperature transducer in the cooling tank 04; TT 111, Temperature transducer in the environment; PT 101 and PI 101, Pressure transmitter and pressure indicator (pressure gauge); IT 101 and II 101; Current transformer and Power Indicator (Multimeter); and FT 101, Flow Transmitter.

The system was equipped with pressure sensors (model TPI-Press, 78703 series); temperature sensors (resistance thermometers Pt100 sensors) with accuracy of 0.2 °C and linearity ($R^2 = 0.99$); magnetic flow meter and transmitter; variable analog voltage attenuator (model W5MT3); and AC current transducer with input 0-5A and output 4-20 mA. The data were recorded using the FieldChart 1.76 software in 30-second intervals.

According to the design of the water driven ejector, it produces a pressure drop as the fluid travels through the

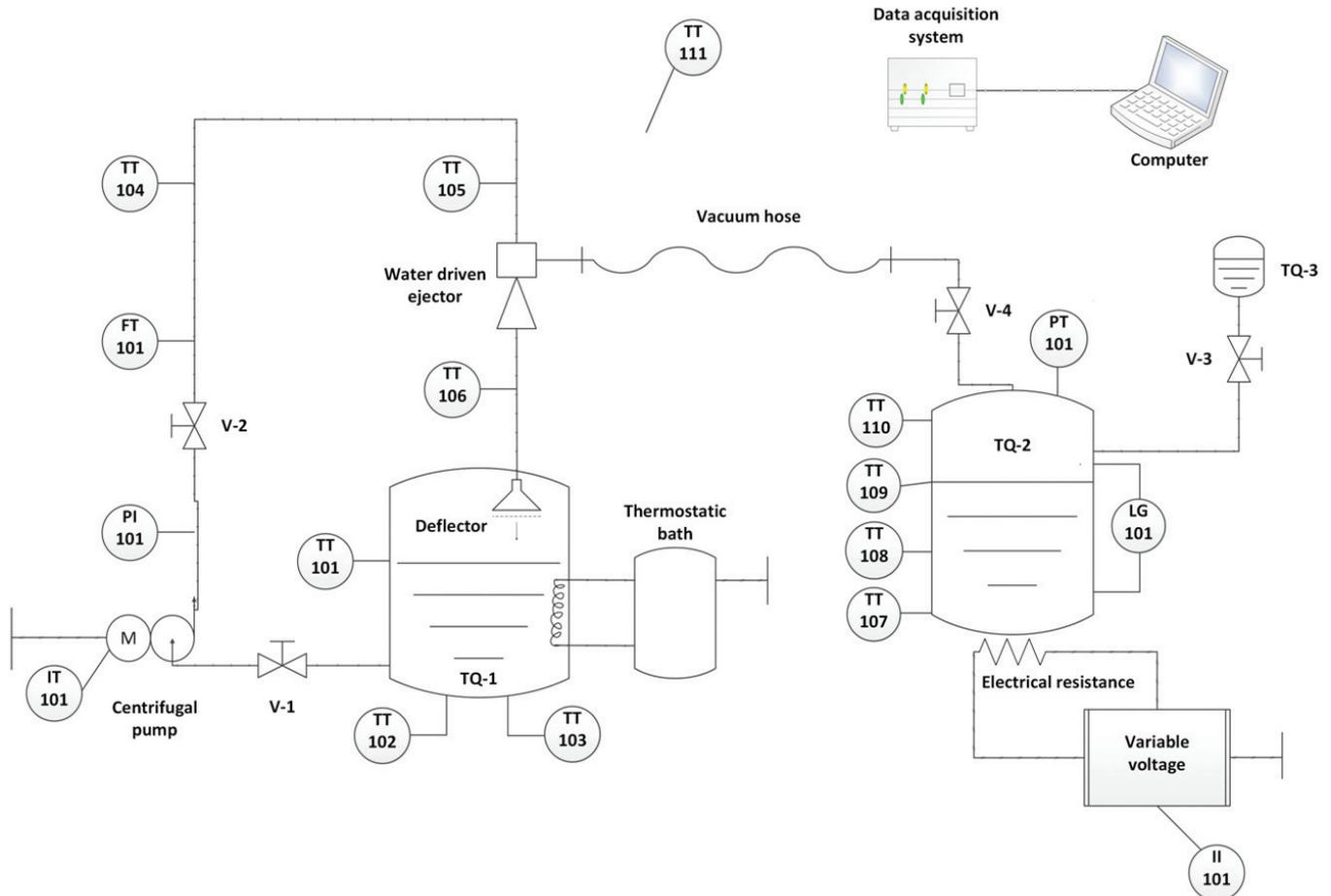


Figure 2. Schematic diagram of the evaporative cooling system with water driven ejector.

convergent compartment, in which the flow rate increases while pressure decreases. Since the nozzle is connected to the cooling tank through the suction chamber, a pressure reduction is observed in the cooling tank. At first, the moist air from the cooling tank is sucked, thus decreasing the partial pressure of the water vapor in the cooling tank to values below the saturation pressure of the liquid water. This phenomenon leads to evaporation of the water in the cooling tank, which causes cooling by heat and mass transfer. Moist air or water vapor at low pressure are sucked from the cooling tank by the circulation water and mix in the mixing chamber of the water driven ejector. Then, they regain enough pressure to balance the atmospheric pressure in the divergent diffuser increasing its diameter and reaching the circulating tank.

The tests assessed the effects on the temperature of the cooling water under various operating conditions, thermal loads, and temperature of the replacement water.

2.1 Operating conditions

The effects of the most important variables, temperature of the circulating water and water flow (speed flow in the convergent section) in the nozzle of the water driven ejector, evaluated in terms of the vacuum created in the cooling tank and the consequent stabilization of the cooling water temperature. The circulating water temperatures were: 5, 10, 15, 20, and 25 ± 0.5 °C, which were obtained by the cooling system working in the circulation tank (thermostatic bath and spiral heat exchanger) or by placing ice to help lower the temperature. The volumetric flow rates were set according to the pressure at which the water in the cooling tank was cooled down. Thus, the nominal volumetric flow rates were 3.7, 3.9, and 4.1 ± 0.1 m³/h, which were modulated by the needle valve and recorded by the magnetic flow meter.

Based on the initial temperature of the circulating water at 30 °C, the tests were performed with the nominal temperature of the circulating water varying from 5 to 25 ± 0.5 °C, and each nominal volumetric flow rate was provided by opening the needle valve.

2.2 Thermal load

After reaching steady state at the lowest water temperature in the cooling tank, when the system was exposed to the best operating conditions for obtaining low-vacuum, the behavior of the temperature of the cooling water subjected to varying degrees of permanent thermal loads was evaluated by thermally activating it with the electrical resistance wire. The different thermal loads imposed on the system were obtained by varying the electrical resistance voltage to 20, 25, 30, 40, 45, and 60 V, resulting in thermal power of 12.61; 27.85; 31.14; 49.92; 56.68; and 92.27 Watts, respectively.

2.3 Replacement of cooling water

The effect on the temperature of the cooling water due to the use of known volumes of replacement water at different temperatures and its effects on the system were evaluated. The

temperatures of the replacement water were 40, 50, 60, 70, and 80 °C stored in a 60 mL replacement tank.

These temperatures were higher than the room temperature in order to provide a progressive power level in the same replacement volume, 60 mL.

2.4 Energy assessment

The coefficient of performance (COP) was assessed, and it determines the ratio of the effect desired by the energy used (Equation 1) under stationary conditions:

$$COP = \frac{Q_a}{W} \quad (1)$$

Where:

Q_a = Thermal power imposed (or cooling) to the water stored in the cooling tank (kJ/s)

W = Work supplied to the system (pump power) (kJ/s)

The effective thermal power (Q_a) was calculated by Equation 2:

$$Q_a = \dot{m}_{water_evaporated} \times \Delta H_{vaporization} \quad (2)$$

Where:

$\dot{m}_{water_evaporated}$ – mass flow rate of water evaporated by the heat provided by the electrical resistance (kg/s)

$\Delta H_{vaporization}$ – variation of specific enthalpy of the evaporated water at the pressure and temperature reached (kJ/kg) (Potter & Scott, 2006)

The work (W) provided to the evaporative cooling system was calculated as the product of the nominal voltage (220 V), intensity of electric current ($3.15 \text{ A} \pm 0.01$), and square root of three (three-phase motor) because both remain approximately constant during operation of the system (Equation 3):

$$W = V \times I \times \sqrt{3} \quad (3)$$

Where:

V – Voltage (Volt)

I – Intensity of electric current (Ampere)

3 Results and discussion

3.1 Operational conditions of the circulating water

To ensure the cooling of the liquid at low-pressure values, both the temperature and the flow rate of the circulating water may be combined with the water driven ejector, together with a proper design of the compartments. According to Jeong et al. (2005), the gradual decrease in the circulating water temperature is intended to reduce the pressure in the ejector since, in general, water density is temperature dependent. The flow rate of the circulating water influences the ejector nozzle pressure since it is governed by the Bernoulli's principle. To maintain the mechanical energy constant in the convergent compartment,

the increase in kinetic energy of the fluid causes a decrease in the pressure energy in the nozzle, considering the variation in potential energy as insignificant once the distance between two points in the control volume was very small. The system was subjected to the nominal volumetric flow rates of 3.7; 3.9 and $4.1 \pm 0.1 \text{ m}^3/\text{h}$ at the nominal temperatures of 5, 10, 15, 20, and $25 \pm 0.5 \text{ }^\circ\text{C}$ for each flow rate under study.

In all tests carried out it was observed that the lower the temperature of the circulating water, at the same nominal volumetric flow studied, the lower the temperature of the water stored in the cooling tank due to the lower pressure at the ejector nozzle reached by the circulating water. The same behavior was observed when the circulating water reached the same temperature. The increase in the flow rate of circulating water at the ejector nozzle created greater pressure decrease. Figure 3 shows the pressure behavior at three volumetric flow rates and in five different circulating water temperatures.

Table 1 shows the cooling water temperatures under different combinations of nominal flow rate and temperature of the circulating water.

The cooling water temperatures at nominal flow rate of $3.7 \pm 0.1 \text{ m}^3/\text{h}$ were much higher than those at flow rates of $3.9 \pm 0.1 \text{ m}^3/\text{h}$ and $4.1 \pm 0.1 \text{ m}^3/\text{h}$ at the same circulating water

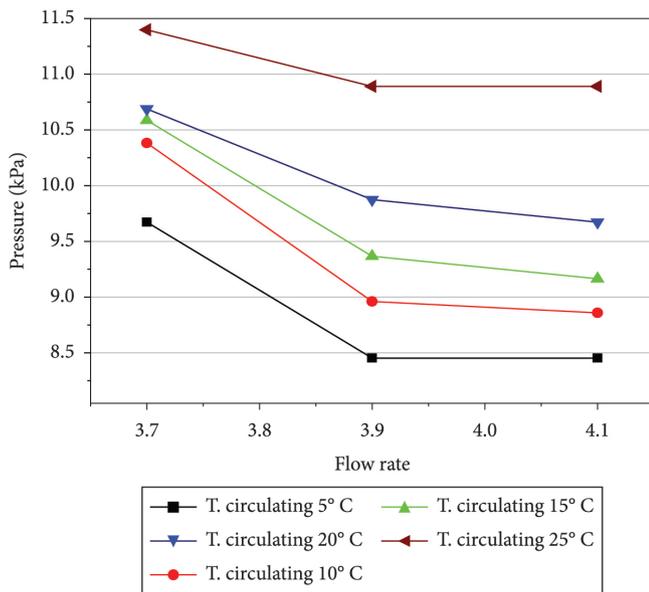


Figure 3. Pressure behavior at different nominal temperatures of the circulating water in the nominal flow rates of 3.7, 3.9 and $4.1 \pm 0.1 \text{ m}^3/\text{h}$.

Table 1. Cooling water ($^\circ\text{C}$) temperatures in the cooling tank under different nominal conditions without thermal load at initial temperature of $30 \text{ }^\circ\text{C}$.

Nominal Flow rate of the Circulating Water (± 0.1)	Nominal Circulating Water Temperature ($\pm 0.5 \text{ }^\circ\text{C}$)				
	5 $^\circ\text{C}$	10 $^\circ\text{C}$	15 $^\circ\text{C}$	20 $^\circ\text{C}$	25 $^\circ\text{C}$
3.7 m^3/h	20.4	22.3	23.5	24.5	27.9
3.9 m^3/h	10.5	13.6	16.6	21.4	25.9
4.1 m^3/h	9.7	12.7	16.3	21.4	26.2

temperature. At flow rates of $3.9 \pm 0.1 \text{ m}^3/\text{h}$ and $4.1 \pm 0.1 \text{ m}^3/\text{h}$, the cooling water temperatures were similar when the system was subjected to the same circulating water temperature, differing from each other by less than $1 \text{ }^\circ\text{C}$. This small variation occurred because the flow rate of $3.9 \pm 0.1 \text{ m}^3/\text{h}$ approached $4.1 \pm 0.1 \text{ m}^3/\text{h}$ for the respective circulating water temperatures.

Considering that the initial cooling water temperature was $30 \text{ }^\circ\text{C}$, the system was capable of cooling water with $\Delta T = 10 \text{ }^\circ\text{C}$ at a nominal temperature of $5 \pm 0.5 \text{ }^\circ\text{C}$ and at nominal flow rate of $3.7 \pm 0.1 \text{ m}^3/\text{h}$, while for nominal flow rates of $3.9 \pm 0.1 \text{ m}^3/\text{h}$ and $4.1 \pm 0.1 \text{ m}^3/\text{h}$, ΔT was around $20 \text{ }^\circ\text{C}$. As the circulating water temperature increased, the difference between the initial and final cooling water temperatures decreased. Similar results were obtained by Jeong et al. (2005), who showed that the pressure decreases sharply with an increase in the flow rate of the ejector nozzle, and the greater the pressure drop, the lower the fluid temperature since the specific weight of water is temperature dependent.

3.2 Analysis of the cooling water subjected to different thermal loads

The experimental tests were carried out applying thermal loads in the cooling water by means of an electrical resistance placed in the bottom of the tank and aimed to study the behavior of both the water and the vapor phase with an accuracy of 0.01 A and 0.5 V, under the best nominal circulating water operating conditions, $4.1 \pm 0.1 \text{ m}^3/\text{h}$ and $0.5 \pm 5 \text{ }^\circ\text{C}$, to obtain lower pressure and lower cooling temperature. Table 2 shows the thermal power imposed to the cooling water and the temperatures before and after the application of the thermal load to steady state process.

It can be seen that with increased thermal load, the system stabilized at increasing and different cooling water temperatures. Therefore, due to the step-like thermal load disturbance, it was observed that the higher the thermal load, the higher the temperature stabilization.

The increase in the temperature with the increase in the thermal load tends to reach a point at which the amount of evaporated water will be so great that it will hinder the suction of the vapor phase by the water driven ejector destabilizing the system. As a consequence of the non-functionality of the ejector, the cooling water may be heated until complete evaporation if the electrical resistance remains on, or the cooling tank may be inundated by the circulating water due to the pressure difference initially generated.

3.3 Analysis of the cooling water subjected to different replacement water temperatures

The variation and stabilization of the cooling water temperature after addition of replacement water at different temperatures at nominal operating conditions of $4.1 \pm 0.1 \text{ m}^3/\text{h}$ and $5 \pm 0.5 \text{ }^\circ\text{C}$ of circulating water was also investigated. Temperatures higher than those commonly used for replacement water were studied, namely 40, 50, 60, 70, and $80 \text{ }^\circ\text{C}$ to investigate more abrupt temperature shifts.

Figure 4 shows the pressure and temperature variations in the cooling tank during the replacement of the cooling water at 40, 50, 60, 70, and $80 \text{ }^\circ\text{C}$ after the system reaches a steady state.

Although during water replacement, pulse like disturbance, the pressure in the cooling tank increased with the increase in the temperature of the gas phase and due to the presence of air in the replacement tank, it decreased to its initial pressure. A small increase in the circulating water temperature was observed, which returned to the initial values after some time. With regard to the replacement of cooling water, at an average evaporation rate of $1.4 \text{ mL}/\text{min}$ and operating conditions of $4.1 \pm 0.1 \text{ m}^3/\text{h}$ and $5 \pm 0.5 \text{ }^\circ\text{C}$, without application of thermal load, the volume of 60 mL was replaced every 43 minutes.

In contrast, applying the thermal power of 92.27 W, the replacement of 60 mL would take place at a shorter time, 26 minutes, with an evaporation rate of $2.27 \text{ mL}/\text{min}$.

3.4 Evaluation of the coefficient of performance (COP)

The coefficient of performance (COP) was calculated at the work supply of $1200.3 \text{ J}/\text{s}$ for the evaporative cooling system using water driven ejector. Table 3 presents the COP values reached by the system with their respective thermal power absorbed by the cooling water.

It was observed that the greater the thermal power absorbed by the cooling water, the higher the COP. The low COP values obtained in the present study were expected when compared to the COP values found for cooling systems using a steam ejector since the velocity of the steam is higher than that of the liquid state of aggregation. Another factor that may have influenced negatively the COP values was the use of a centrifugal pump that heated the circulating water when passing through its rotor. The centrifugal pump exhibited an unexpected behavior, thus it may have worked harder, losing its efficiency and using more energy to transport water. Therefore, the energy required by the centrifugal pump may be overestimated due to pump malfunction.

Although the system under study did not show high COP values, they were similar to the COP values of other cooling systems. Although Vargas et al. (2009) expected a COP value of 0.292, based on the empirical correlations of Huang & Chang (1999) for an evaporator using 91W and R141b fluid at $10 \text{ }^\circ\text{C}$, the authors found a COP value of 0.221, which is 25% lower than expected, using a secondary fluid rate of $0.39 \text{ g}/\text{s}$ and primary fluid rate of $1.68 \text{ g}/\text{s}$. Jain et al. (2012) reported a COP value of 0.13 for a solar power cooling cycle using an ejector. The

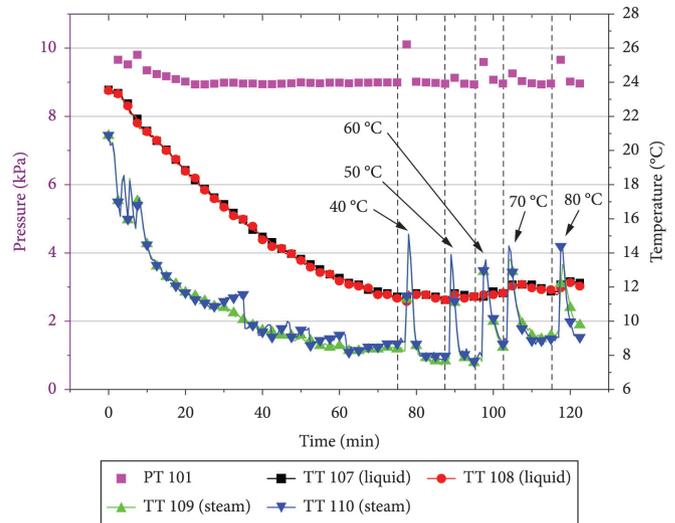


Figure 4. Variation of pressure and temperature in the cooling tank due to the replacement of 60 mL cooling water at temperatures of 40, 50, 60, 70, and $80 \text{ }^\circ\text{C}$ at nominal operating conditions of $4.1 \pm 0.1 \text{ m}^3/\text{h}$ and $5 \pm 0.5 \text{ }^\circ\text{C}$ of circulating water.

Table 2. Average temperature reached by the cooling water ($^\circ\text{C}$) when subjected to different thermal loads at nominal operating conditions of $4.1 \pm 0.1 \text{ m}^3/\text{h}$ and $5 \pm 0.5 \text{ }^\circ\text{C}$.

Thermal Power Absorbed (Watts)	Initial temperature of the cooling water ($^\circ\text{C}$)	Final temperature of the cooling water ($^\circ\text{C}$) after stabilization
12.61	10.2	11.6
27.85	10.2	13.3
31.14	9.3	15.3
49.92	11.5	20.8
56.58	9.7	20.7
92.27	8.9	23.7

Table 3. Coefficient of performance of the evaporative cooling system using a water driven ejector under different thermal power absorbed by the cooling water.

Thermal power absorbed by the cooling water (J/s)	COP _{reached}
12.61	0.011
27.85	0.023
31.14	0.026
49.92	0.042
56.68	0.047
92.27	0.077

system was exposed to solar radiation of $700 \text{ W}/\text{m}^2$ with cooling capacity of 5 kW generating temperature of $90 \text{ }^\circ\text{C}$, condensing temperature of $37 \text{ }^\circ\text{C}$, and evaporator temperature of $10 \text{ }^\circ\text{C}$.

4 Conclusion

Based on the results, for a cooling system using water driven ejector, it was concluded that: a) the higher the flow rate and the lower the circulation water temperature, the lower the pressure

at the nozzle of the suction device, which consequently will lead to lower cooling water temperatures in the steady state, b) the replacement of cooling water can occur at any time because it produces minimal effects under the conditions studied; c) the cooling water temperature is directly proportional to the thermal load applied and d) a proper design of the water driven ejector is fundamental to achieve low pressures and mitigate the operational conditions.

The evaporative cooling system using water driven ejector showed a coefficient of performance of 0.077 at nominal operational conditions of 4.1 ± 0.1 m³/h and 5 ± 0.5 °C circulating water. One of the reasons for obtaining low COP values is the malfunctioning of the centrifugal pump used for pumping and circulating water through the ejector, resulting in low equipment efficiency and therefore greater energy demand for operation.

Systems with abundance of circulating water and in which water cooling can occur at higher temperatures tend to be appropriate since there is no need to have very low pressure, and thus cooling water at room temperature can be used. The system proposed in the present study can be used combined with a cooling tower or as a thermal storage system.

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