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Energy and Exergy Analyses of HVAC systems

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Abstract In this study, evaporative air cooling systems are taken into account as HVAC systems. The direct, indirect and novel Maisotsenko cycle evaporative air cooling systems are explained and the first and second laws of thermodynamics analyses of the evaporative air cooling systems are illustrated. A case study of Maisotsenko cycle based air cooling system is studied under energetic and exergetic perspective. The wet bulb effectiveness, cooling capacity and energetic coefficient of performance (COP) of the system are found in the energy analysis. On the other hand, the exergy input rates of dry air and water, exergy output rate, exergy loss rate, exergy destruction rate, useful output exergy rate, exergetic COP (COP_{ex}), exergy efficiency of the system are determined in the exergy analysis. It is found that the energetic COP of the system is very high as 8.43, while the exergetic COP (COP_{ex}) rate is low as 0.08. Furthermore, the second law of thermodynamic (exergy) efficiency of the system is determined to be 25%.

Key words: COP, Energy & exergy, Evaporative air cooling, HVAC, Maisotsenko cycle

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1. Introduction

Heating, Ventilation and Air Conditioning (HVAC) is the technology that provides thermal comfort and especially important for residential sector to supply acceptable indoor air quality [(1)]. HVAC systems consume 13% of all primary energy generated around the world with commercial buildings [(2)]. So, utilization of energy/free energy is important. Effective evaluation of HVAC systems plays major role in improving the utilization of energy and indoor air quality [(3)]. HVAC systems are generally evaluated by the first law of thermodynamics. However, the energy to maintain the operation of HVAC system is available energy, also known as "exergy" [(4)]. The energy analysis is not sufficient to understand the HVAC systems holistically. In order to overcome the limitation of this situation, exergy (availability) analysis is applied for a better understanding of HVAC systems [(5)].

Since 1970s, exergy analysis is widely used to evaluate energy systems to find the most rational use of energy. The main reasons to perform exergy analysis on HVAC system can be seen in its definition: (i) the useful work, (ii) the quality of energy, and (iii) a measure of the potential of a system to produce work in given environment [(6)]. Another importance of the exergy is that it can support the selection, improvement and diffusion of the most promising existing/new technologies in order to minimize exergy consumption, giving a quantitative evaluation of the minimization potential [(7)].

Exergy analysis bases on not only first, but also second laws of thermodynamics. So, it is a useful tool for HVAC and other energy systems. In recent years, exergy analysis approaches are widely applied in many areas such as assessment of physical criteria, local optimization and operational diagnosis [(8)].

2. Evaporative air coolers as HVAC systems

Refrigeration is one of the main process in HVAC systems and it is added to the HVAC field's abbreviation as HVAC&R or HVACR [(1)]. The outside air temperature effects the energy utilization of air-cooled HVAC systems. If the weather is hot, more energy is necessary for the hard work of the equipment. In this regard, evaporative air coolers that cools air through the evaporation of water are better option for efficient HVAC systems. There are generally two types of evaporative air coolers: direct and indirect. However, there are also some novel technologies for advanced indirect evaporative air cooling such as Maisotsenko cycle that creates comfort air conditions for the buildings with lowest energy consumption and highest efficiency.

In direct evaporative air cooling process; when air is transferred through a water filled medium, the water is evaporated and this process make the air moisture and cooler. In the indirect evaporative air cooling process; a secondary air stream is cooled by water in the wet channel. The cooled secondary air goes through a heat exchanger, where it cools the primary air stream in the dry channel. The cooled primary air stream is circulated by a blower. Indirect evaporative cooling does not add moisture to the primary air stream [(9)]. Maisotsenko cycle air coolers have got the wet and dry sides of a plate like indirect evaporative coolers, but with a much different airflow creating a new thermodynamic cycle (resulting in product temperatures which approach the dew point temperature of the air).

This cycle uses the enthalpy difference of the air at dew point temperature and the air saturated at a higher temperature to reject the heat from the product, and also it allows the product fluid to be cooled in to the dew point temperature of the incoming air ideally. The air is then pre-cooled before passing into the heat rejection stream where the water is evaporated [10]. The schematic layout of the direct, indirect and Maisotsenko cycle based evaporative air cooling systems can be seen in Figure 1.

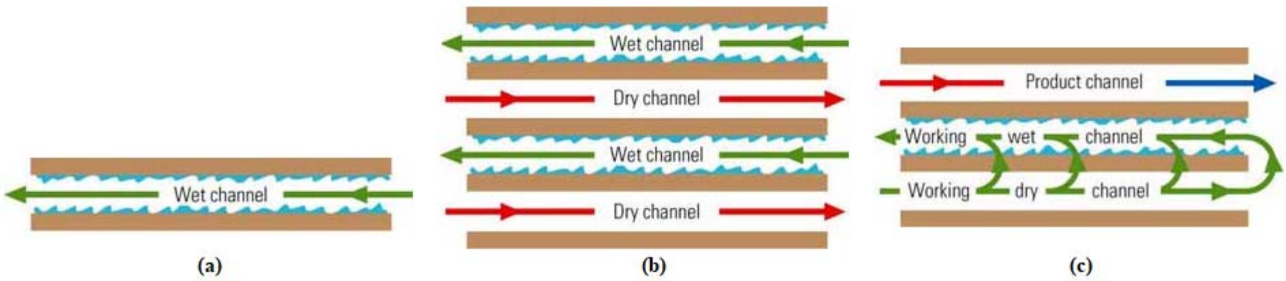

Fig. 1 Schematic layout of the (a) direct, (b) indirect and (c) Maisotsenko cycle based evaporative air cooling systems [11].

As a case study Maisotsenko cycle evaporative air cooling system is taken into account. The detail of the Maisotsenko cycle evaporative air cooling system is given in Figure 2. Also, the necessary system data of the case study is taken from Ref. [10].

3. Analysis

In the thermodynamic analysis energy analysis is applied first. Then, exergy analysis can be conducted. In the energy analysis, wet bulb effectiveness, cooling capacity and coefficient of performance (COP) are calculated. The performance of the air coolers can be expressed by the wet bulb effectiveness " ε_{wb} ", which is defined as the ratio of temperature depression of the device to the potential wet bulb depression as given below:

$$\varepsilon_{wb} = \left(T_{SI,db} - T_{SO,db}\right) / \left(T_{SI,db} - T_{SI,wb}\right) \tag{1}$$

where " $T_{SI,db}$ " is the supply inlet dry bulb temperature of air (37.77°C), " $T_{SO,db}$ " is the supply outlet dry bulb temperature of air (17.5°C) and " $T_{SI,wb}$ " is the supply inlet wet bulb temperature of air (18.33°C).

The cooling capacity rate of the system " $\dot{Q}_{cooling}$ " is found by

$$\dot{Q}_{cooling} = \dot{m}_{da}\left(h_{in} - h_{out}\right) \tag{2}$$

where " h_{in} " is the enthalpy of supply inlet air (311.4 kJ/kg), and " h_{out} " is the enthalpy of supply outlet air (cooled room air) (291 kJ/kg).

The Coefficient of Performance (COP) for cooling is the ratio of the heat removed from the cold reservoir to input work. The C*OP* (also the so-called energetic *COP*) of the system can be determined from

$$COP = \dot{Q}_{cooling} / \dot{W}_{consume} \tag{3}$$

where " $\dot{W}_{consume}$ " is the consumed power (electricity) (0.276 kW).

In the exergy analysis, exergy inputs, outputs, losses and destruction are calculated. The exergy balance of a control volume can be written as follows:

$$\dot{E}x_{in} = \dot{E}x_{out} + \dot{E}x_{loss} + \dot{E}x_{dest} \tag{4}$$

or

$$\dot{E}x_{in,da} + \dot{E}x_{in,w} = \dot{E}x_{ha} + \dot{E}x_{loss} + \dot{E}x_{dest} \tag{5}$$

where " $\dot{E}x_{in}$ ", " $\dot{E}x_{out}$ ", " $\dot{E}x_{loss}$ " and " $\dot{E}x_{dest}$ " are the exergy input rate, useful exergy output rate, exergy loss rate and exergy destruction rate, respectively. Also, " $\dot{E}x_{in,da}$ "," $\dot{E}x_{in,w}$ " and " $\dot{E}x_{ha}$ " are the exergy input rate of dry air, exergy input rate of water and exergy rate of humid air, respectively.

The exergy input rate of dry air " $\dot{E}x_{in,da}$ " is expressed by

$$\dot{E}x_{in,da} = \dot{m}_{da}\, ex_{da} \tag{6}$$

where " $\dot{m}_{da}$ " is the mass flow rate of dry air (kg/s) and " ex_{da} " is the specific exergy flow of dry air (kJ/kg) as follows:

$$ex_{da} = c_{p,da}\, T_0 \left[\left(T_{SI,db} / T_0 \right) - 1 - \ln\left(T_{SI,db} / T_0 \right) \right] + R_{da}\, T_0 \ln\left(P_1 / P_0 \right) + R_{da}\, T_0 \ln\left(1 + \bar{\omega}_0 \right) \tag{7}$$

where “ $c_{p,da}$ ” is the specific heat capacity of dry air (1.003 kJ/kgK), “ T_0 ” is the dead state (reference environment) temperature (35°C), “ $T_{SI,db}$ ” is the supply inlet dry bulb temperature (37.77°C), “ R_{da} ” is the specific ideal gas constant of dry air (0.287 kJ/kgK), “ P_1 ” is the supply inlet pressure (1 atm), “ P_0 ” is the dead state (reference environment) pressure (1 atm), “ $\bar{\omega}_0$ ” is the mole fraction ratio of the dead state (reference environment) condition (0.0072).

The exergy input rate of water “ $\dot{E}x_{in,w}$ ” is determined by

$$\dot{E}x_{in,w} = \dot{m}_w\, ex_w = \dot{m}_{da}\, \omega\, ex_w \tag{8}$$

where “ $\dot{m}_w$ ” is the mass flow rate of water (0.000627 kg/s) and “ ex_w ” is the specific exergy flow of water (kJ/kg) which is found from

$$ex_w = \left(h_{f(T_{SI;db})} - h_{g(T_0)} \right) - T_0 \left(s_{f(T_{SI;db})} - s_{g(T_0)} \right) + \left(P_1 - P_{sat(T_{SI;db})} \right) v_{f(T_{SI;db})} - R_v\, T_0 \ln(\phi_0) \tag{9}$$

where “ $h_{f(T_{SI;db})}$ ” is the enthalpy (fluid) of saturated water at inlet air temperature ($T_{SI,db}$) (158.2 kJ/kg), “ $h_{g(T_0)}$ ” is the enthalpy (gas) of saturated water vapor at dead state (reference environment) temperature (T_0) (2564.53 kJ/kg)), “ $s_{f(T_{SI;db})}$ ”is the entropy (fluid) of saturated water at inlet air temperature ($T_{SI,db}$) (0.5425 kJ/kgK), “ $s_{g(T_0)}$ ” is the entropy (gas) of saturated water vapor at dead state (reference environment) temperature (T_0) (8.3516 kJ/kgK), “ P_1 ” is the supply inlet pressure (1 atm), “ $P_{sat(T_{SI;db})}$ ” is the saturated water pressure at inlet air temperature ($T_{SI,db}$) (0.064651 atm), “ $v_{f(T_{SI;db})}$ ” is the specific volume rate of saturated water (fluid) at inlet air temperature ($T_{SI,db}$) (0.001007 m^3/kg), “ R_v ” is the specific ideal gas constant of water vapor (0.4165 kJ/kgK), “ ϕ_0 ” is the relative humidity of dead state (reference environment) condition (12.8%).

The useful exergy output rate “ $\dot{E}x_{out}$ ” is equal to the exergy rate of humid air “ $\dot{E}x_{ha}$ ” as given below:

$$\dot{E}x_{out} = \dot{E}x_{ha} = \dot{m}_{ha}\, ex_{ha} \tag{10}$$

where “ $\dot{m}_{ha}$ ” is the mass flow rate of humid air (0.114627 kg/s) and “ ex_{ha} ” is the specific exergy flow of humid air (kJ/kg) which is found by

$$ex_{ha} = \left(c_{p,da} + \omega\, c_{p,v} \right) T_0 \left[\frac{T_{SO,db}}{T_0} - 1 - \ln \frac{T_{SO,db}}{T_0} \right] + \left(1 + \bar{\omega} \right) R_{da}\, T_0 \ln \frac{P_2}{P_0} + R_{da}\, T_0 \left[\left(1 + \bar{\omega} \right) \ln \left(\frac{1 + \bar{\omega}_0}{1 + \bar{\omega}} \right) + \bar{\omega} \ln \left(\frac{\bar{\omega}}{\bar{\omega}_0} \right) \right] \tag{11}$$

where “ $c_{p,v}$ ” is the specific heat capacity of water vapor (1.872 kJ/kgK), “ $T_{SO,db}$ ” is the supply outlet dry bulb temperature (cooled air temperature) (17.5℃), “ P_2 ” is the supply outlet pressure (1 atm), “ P_0 ” is the dead state (reference environment) pressure (1 atm) and “ $\bar{\omega}$ ” is the mole fraction ratio of the inlet condition (0.008844).

The heat loss occurs due to temperature differences between inlet and outlet conditions. Also, exergy loss bases on this heat loss considering dead state (reference environment) temperature. The exergy loss rate “ $\dot{E}x_{loss}$ ” is calculated by

$$\dot{E}x_{loss} = \dot{Q}_{cooling} \left[1 - \left(T_0 / T_{SI,db} \right) \right] \tag{12}$$

The exergy destruction can be thought to be irreversibility of the systems, which occurs inside the control volume and cannot be gained again. The exergy destruction rate “ $\dot{E}x_{dest}$ ” is determined by

$$\dot{E}x_{dest} = \dot{E}x_{in} - \dot{E}x_{out} - \dot{E}x_{loss} \tag{13}$$

The exergetic coefficient of performance (C*OP*$_{ex}$) of the system is found by

$$COP_{ex} = COP \left[1 - \left(T_0 / T_{SI,db} \right) \right] \tag{14}$$

The exergy efficiency is defined as

$$\Psi = \dot{E}x_{out} / \dot{E}x_{in} = \dot{E}x_{ha} / \left(\dot{E}x_{in,da} + \dot{E}x_{in,w} \right) \tag{15}$$

4. Results and Conclusion

The energy and exergy analyses are applied to the Maisotsenko cycle based air cooling system. In the energy analysis, the wet bulb effectiveness of the system is found as 1.04, while the cooling capacity and energetic COP are 2.33 kW and 8.43, respectively. In the exergy analysis, the exergy input rates of the dry air and water are determined to be 0.07 kW and 0.17 kW, respectively. Also, the exergy output, loss and destruction rates of the system are 0.06 kW, 0.02 kW and 0.16 kW, respectively. Finally, the exergetic coefficient of performance (COP_{ex}) is determined as 0.08 and the exergy efficiency is found to be 25%. The exergy loss and flow (Grassmann) diagram of the system is illustrated in Figure 2.

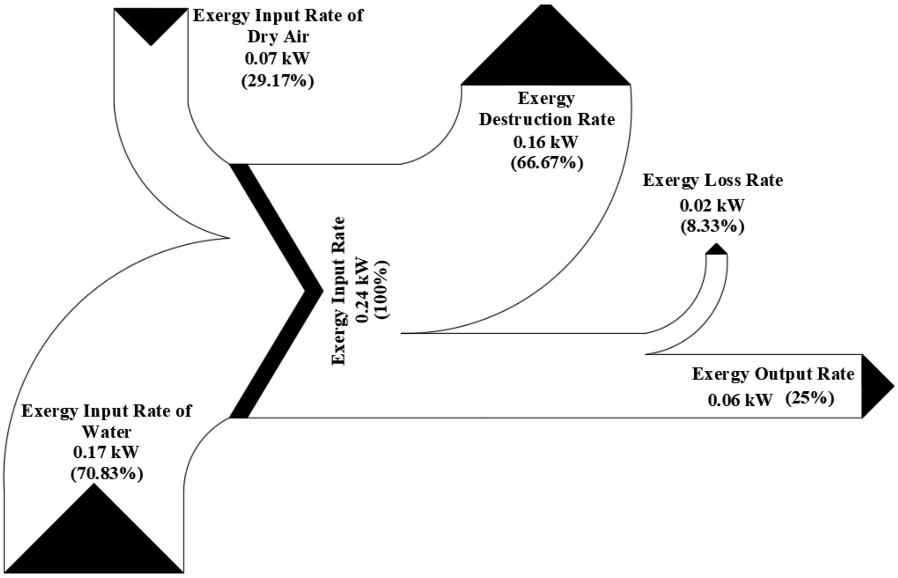

Fig. 2 Exergy loss and flow (Grassmann) diagram of the system.

As is seen in Figure 2, 29.17% of the total exergy input is obtained from dry air, while exergy input rate of the water is responsible for 70.83% of total exergy input. The most of the exergy (66.67%) of the system is destructed due to irreversibility in the system and only one fourth of the exergy is obtained as a useful output (25%), while exergy loss is 8.33% of total energy input.

The energetic COP of the system is 8.43 which is very high and it shows that the novel Maisotsenko cycle based air cooling system has very efficient technology. On the other hand, the exergetic COP (COP_{ex}) of the system is 0.08 which is low and mostly depends on the dead (reference environment) and supply inlet dry bulb air (indoor air) temperatures alongside cooling capacity and consumed electricity power. So, it is low in hot climates. Furthermore, the second law of thermodynamic efficiency of the system is 25% which can be considered as medium level for this kind of cooling system in hot climates.

As a result, the COP, which is used to assess the cooling systems' performance, is vary depending on the first and second laws of thermodynamics, and exergy play important role to show this variety.

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