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ENERGY AND EXERGY ANALYSIS OF TWO-STAGE EVAPORATIVE COOLING SYSTEM

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ABSTRACT

In this paper exergy analysis of indirect-direct evaporative cooling system is presented. In this study ambient temperature, saturation specific humidity of the process and pressure at the end of the process is considered as the dead state. Variations in wet-bulb Saturation Efficiency, change in thermal exergy, chemical exergy, total inlet exergy, total outlet exergy and exergy efficiency is analysed for pads of different thickness with constant face velocity. It is observed that for a give thickness of pads Second Law Efficiency does not changes as wet bulb saturation efficiency changes. Irreversibility and entropy generation change almost same manner. Sustainability index curve is almost flat for pads of same thickness. However it increases slightly with the thickness of the pads.

Key words: Direct Evaporative Cooling, Exergy, Sustainability, EER, Exergy efficiency.

INTRODUCTION

Evaporative cooling had its birth around one thousand years ago in ancient Egypt. At that time, porous pots and ponds covered with a wet cloth were often used to preserve food against hot weather and some water chutes were also integrated into walls to keep the inside space cool, due to the evaporation of water when air flowed through. This technique was soon spread into other hot and arid places of the world.

The principle underlying Evaporative Air Conditioning (EAC) is the conversion of sensible heat of hot air to latent heat of water. Part of the sensible heat of the air is transferred to the water and becomes latent heat and evaporates the water. The wetted medium for the evaporation of water could be a porous wetted pad consisting of fibres, cellulose papers or a spray of water.

Two common types of evaporative cooling systems are the direct and indirect evaporative systems. In the direct evaporative cooler (DEC), the air comes into direct contact with water. The direct evaporative cooling system adds moisture to the cool air, while an indirect evaporative cooling system (IEC) provides only sensible cooling to the processed air without any addition of moisture. Depending on the climatic conditions and the application, combining indirect and direct evaporative coolers might be appropriate. If a first indirect stage is added to a second direct stage, a two-stage indirect/direct cooler is obtained which cools the air more than a stand-alone DEC unit.

Besides reducing the temperature EAC supply fresh air, use water as the working fluid instead of CFCs, simple manufacturing, easy maintenance, low power consumption, ability of achieve suitable level of humidity in rather dry regions. EAC technologies represent significant environmental benefits related to reducing CFC/HCFC use and for obviating CO_2 and other emissions, as well as for reducing peak electrical demand.

The minimum temperature that can be obtained is the wet bulb temperature of the entering air [1].

For the better understanding of EAC, exergy analysis method can be applied with energy analysis. Exergy analysis uses both conservation of mass and energy principles. This method is based on second law of thermodynamics for analysis, design and improvement of energy systems. Exergy is always evaluated with respect to a reference environment and it can be destroyed when the irreversibility process occurs. If an exergy analysis performed on a system, thermodynamic imperfections can be quantified as exergy destruction, which represent losses in energy quality or usefulness (Dincer and Rosen, 2007). The term exergy was introduced by Rant in 1953.

LITERATURE REVIEW

In the field of direct evaporative cooling several authors dedicated their researches. Watt [1] developed the first serious analyses of direct and indirect evaporative systems,

Halasz [2] presented a general dimensionless mathematical model to describe all evaporative cooling devices namely cooling water towers, evaporative condensers of fluid, air washes, dehumidification coils, etc. Camargo et al. [3] presented the principles of operation for direct and indirect evaporative cooling systems and the mathematical development of thermal exchanges equations, Koca et al. [4] have developed a procedure for testing evaporative cooling pads. Their results show that pad performance is affected by pad angle, pad thickness, face air velocity, and static pressure drop across the pad and can be expressed in terms of evaporative cooling efficiency and static pressure drop. Dai et al. [5] solved governing equations for cross-flow direct evaporative cooler using an integration method. They used honeycomb paper as packing material and assumed constant space between channels and they simply modelled the thin water film on surfaces. Their results showed that the performance improve by optimizing length of the air channel of honeycomb paper, mass flow rates of air and feed water.

Anyanwu [6] reported design, construction, and measured performance of a porous evaporative cooler for preservation of fruits and vegetables

Kulkarni et al [7] analyzed the performance of jute fibre ropes as alternative cooling media as ropes are capable of retaining high moisture and have a large wetted surface area. Hot and dry air is allowed to flow over the wet jute rope bank tightly held between two plates which are integral part of two tanks.

Kulkarni et al [8] theoretically analyzed the performance of evaporative cooler pads of rigid cellulose, corrugated paper, high density polythene packing and aspen fibre having rectangular, cylindrical and hexagonal shape.

Heidarinejad [9] presented a performance test of a direct evaporative cooler coupled with a ground circuit in Tehran. The investigation showed that the coupled system sufficiently provided the comfort condition with high cooling effectiveness and greatly reduce the electricity cost.

A.Joudi [10] to test its ability to eliminate the variable cold load of Iraqi. As a result, indirect evaporative cooling system provided residences with comfortable conditions for most of operation period with rather high efficiency owing to only fan and pump consuming power.

Delfani [11] utilized indirect evaporative cooler to pre-cool the supply air before it entered the mechanical cooling system. In 1952, Watt and Brown [12]. invented indirect/direct evaporative cooling system using aluminium plate heat exchanger After that, the studies of coupled indirect/direct systems have been performed widely. Heidarinejad [13] experimentally study the coupled evaporative cooling system in terms of thermal effectiveness and water consumption under various weather conditions in Iran, in which indirect evaporative cooling is followed by direct evaporative cooling. It is observed that the indirect/direct system is able to satisfy the demand of comfort conditions in more places where wet bulb temperature of ambient air is high which is not suitable for the operation of single evaporative cooler.

Kulkarni et al [14]theoretically analysed the performance of indirect-direct two-stage, indirect cooling stage consisting of plate type wet surface heat exchanger followed by direct cooling stage consisting of rigid cellulose and aspen fibre in rectangular, semi-cylindrical and semi-hexagonal shapes. Kulkarni et al [15] fabricated and tested the performance of rectangular, semi-cylindrical and semi-hexagonal shaped cooling pads made up of wood wool, rigid cellulose and aspen fibre are used as cooling media in direct stage.

In the field of exergy analysis, Dincer [16] reported the linkages between energy and exergy, exergy and the environment, energy and sustainable development, and energy policy making and exergy in detail. Chenqin et al. [17] analyze exergy changes in the HVAC systems. They presume an unusual dead state to eliminate the exergy calculation of water. Also, they break down the exergy of moist air to three components: thermal, mechanical, and chemical components. By assuming efficiency for each scheme of evaporative cooling, the results have shown that the regenerative scheme has the best performance. Alhamzy [18] calculates the minimum work of dehumidification in an air-conditioning process based on the second law of thermodynamics. In his research, the state of environment was chosen as the dead state.

Qureshi et al. [19] carried out study of various psychrometric processes on the basis of second law of thermodynamics. The relation between work and changes in entropy generation arises from the simultaneous treatment of the first and second laws referred to as exergy analysis. The study of each of the processes is carried out to determine the variation of second-law efficiency as a function of mass flow rate, relative humidity and temperature. Other trends such as variation of temperature with relative humidity are also shown where applicable. Irreversible losses are calculated by applying an exergy balance on each system. Kanoglua et al.[20] had studied the effect of ambient conditions on the first and second law performance of an open desiccant cooling process. The cooling system consists of a desiccant wheel, a rotary regenerator, two evaporative coolers, and a heating unit. Certain ideal operating characteristics based primarily on the first law of thermodynamics are assumed for each component. Qureshi et al. [21] carried out second-law-based parametric study of performance evaluation of cooling towers and evaporative heat exchangers, to determine the variation of second-law efficiency as well as exergy destruction as a function of various input parameters. Irreversible losses are determined on each of the systems investigated. They noticed that an increase in the inlet wet bulb temperature invariably increases the second-law efficiency of all the heat exchangers. Taufiq et al. [22] study the exergy analysis of the direct evaporative cooling in a Malaysian building. The average temperature and relative humidity were considered as the dead state. The results obtained have shown that increase in relative humidity increases exergy efficiency. Muangnoi et al. [23] use an exergy analysis to demonstrate exergy and exergy destruction of water and air flowing through the cooling tower. They show thermodynamics irreversibility is higher at bottom of a cooling tower. Kanoglu et al**.** [24] carried out studied on psychrometric processes on the basis of second law of thermodynamics. Mass, energy, entropy, and exergy balances and exergy efficiency relations are developed for common air-conditioning processes. The effects of air temperature and relative humidity at the inlet and exit, the temperature of steam used for humidification, and the dead state properties of exergy efficiency and exergy destruction are investigated. The results indicate that processes with low exergy efficiency and high exergy destruction have significant potential for improving performance. Chen et al. [25] analyzed and optimized several practical evaporative cooling systems based on the newly introduced moisture entransy theory. They realized that the most efficient evaporative cooling performance requires a minimum thermal resistance. Aforementioned research papers on exergy analysis have not examined exergetic efficiency on various climates. Indeed, multi-climate countries such as Iran necessitate an abroad consideration of exergy analysis on diverse environmental conditions.

Exergy analysis

The exergy balance of the evaporative cooling system for the control volume can be obtained by

$$\dot{E}x_{in} = \dot{E}x_{out} + \dot{E}x_{dest} + \dot{E}x_{lost} \quad (1)$$

where $\dot{E}x_{in}$, $\dot{E}x_{out}$, $\dot{E}x_{dest}$ and $\dot{E}x_{lost}$ are the exergy input rate, exergy output rate, exergy destruction rate and exergy loss rate, respectively.

The exergy input rate $\dot{E}x_{in}$,is defined by

$$\dot{E}x_{in} = \dot{E}x_{in,da} + \dot{E}x_{in,w} \quad (2)$$

$$\dot{E}x_{in} = \dot{E}x_{in,da} + \dot{E}x_{in,w} \quad (3)$$

$$\dot{E}x_{in,w} = \dot{m}_w e_w = \dot{m}_{da}\omega_{SI} e_w \quad (4)$$

Specific exergy rate of dry air is defined by

$$e_{da} = c_{pda}T_0\left[\frac{T_{SI}}{T_0} - 1 - \ln\frac{T_{SI}}{T_0}\right] + R_aT_0\ln\frac{P}{P_0} + R_aT_0\ln(1+\varpi_0) \quad (5)$$

and specific exergy rate of water at ambient temperature is defined by

$$e_w = (h_{f(T_{SI})} - h_{g(T_0)}) - T_0(s_{f(T_{SI})} - s_{g(T_0)}) + (P - P_{sat(T_{SI})})v_{f(T_{SI})} - R_{wv}T_0\ln\Phi \quad (6)$$

Where relative humidity is defined by

$$\Phi = \frac{P_0}{P_{\sup ply}} \quad (7)$$

The exergy output rate $\dot{E}x_{out}$,is defined by

$$\dot{E}x_{out} = \dot{m}_{da}e_t \quad (8)$$

$$e_t = (c_{pda} + \omega_{SI}c_{pw})T_0\left[\frac{T_{SI}}{T_0} - 1 - \ln\frac{T_{SI}}{T_0}\right] + (1+\varpi_{SI})R_aT_0\ln\frac{P}{P_0} + R_aT_0\left[(1+\varpi_{SI})\ln\frac{(1+\varpi_0)}{(1+\varpi_{SI})} + \varpi_{SI}\ln\frac{\varpi_{SI}}{\varpi_0}\right] \quad (9)$$

$$e_t = (c_{pda} + \omega_{SI}c_{pw})T_0\left[\frac{T_{SI}}{T_0} - 1 - \ln\frac{T_{SI}}{T_0}\right] + (1+\varpi_{SI})R_aT_0\ln\frac{P}{P_0} \quad (10)$$
$$+ R_aT_0\left[(1+\varpi_{SI})\ln\frac{(1+\varpi_0)}{(1+\varpi_{SI})} + \varpi_{SI}\ln\frac{\varpi_{SI}}{\varpi_0}\right]$$

Rate of exergy loss is defined by

$$\dot{E}x_{loss} = \dot{Q}_{cooling}\left(1 - \frac{T_0}{T_{SI}}\right) \quad (11)$$

Rate of exergy destruction is defined by

$$\dot{E}x_{dest} = \dot{E}x_{in} - \dot{E}x_{out} - \dot{E}x_{lost} \quad (12)$$

Rate of entropy generated

$$\dot{S} = \frac{\dot{E}x_{dest}}{T_0} \quad (13)$$

Exergy or second law of thermodynamics efficiency is defined by

$$\Psi = \frac{\dot{E}x_{out}}{\dot{E}x_{in}} = \frac{e_t}{e_{da} + \omega_{SI}e_w} \quad (14)$$

Thermal Specific exergy of moist air

$$ex_{th} = (c_{pa} + \omega c_{pv})T_0\left[\frac{T}{T_0} - 1 - \ln\frac{T}{T_0}\right] \quad (15)$$

Mechanical Specific exergy of moist air

$$ex_{mech} = (1 + 1.608\omega)R_aT_0\ln\frac{p}{p_0} \quad (16)$$

Chemical Specific exergy of moist air

$$ex_{chem} = R_aT_0\left[(1+1.608\omega)\ln\frac{(1+1.608\omega_0)}{(1+1.608\omega)} + 1.608\omega\ln\frac{\omega}{\omega_0}\right] \quad (17)$$

Total Specific exergy of moist air

$$ex = ex_{th} + ex_{mech} + ex_{chem} \quad (18)$$

Specific exergy of water

$$\psi_w = -R_vT_0\ln\phi_0 \quad (19)$$

Exergy Analysis for DEC

$$\dot{m}_a\psi_{a1} + \dot{m}_w\psi_w - \dot{m}_a\psi_{a2} - \dot{I} = 0 \quad (20)$$

$$\dot{m}_w = \dot{m}_a(\omega_2 - \omega_1) \quad (21)$$

Exergy Analysis for IEC

$$\dot{m}_a\psi_{a1} - \dot{m}_a\psi_{a2} - \dot{E}x_Q - \dot{I} = 0 \quad (22)$$

$$\dot{E}x_Q = -\int_1^2\left(1 - \frac{T_0}{T}\right)dQ$$

$$= (c_{pa} + \omega c_{pv})\left[(T_1 - T_2) - T_0\ln\frac{T_1}{T_2}\right] \quad (23)$$

Exergy Analysis for IDEC

$$\dot{m}_a\psi_{a1} + \dot{m}_w\psi_w - \dot{m}_a\psi_{a2} - \dot{E}x_{Q_{1-2'}} - \dot{I} = 0 \quad (24)$$

$$\sum_{in}\dot{E}x_Q + \sum_{in}\dot{m}(\psi) - \sum_{out}\dot{E}x_Q - \sum_{out}\dot{m}(\psi) - \dot{E}x_{dest} = 0$$

$$\psi = (h - h_0) - T_0(s - s_0) \quad (25)$$

$$\eta_{ex} = 1 - \frac{exergy\ destroed}{exergy\ \sup plied} \quad (26)$$

$$T_0\dot{S}_{gen} = \dot{I} = \dot{w}_{rev} - \dot{w}_{c.v.} \quad (27)$$

Variation in Saturation Efficiency

Wet-bulb saturation efficiency of IDEC decreases with increase in the air mass flow rate. This is expected because with increase in air mass flow rate, air has lesser contact time with water layer causing less evaporation of water. Saturation efficiency of IDEC increases with increase in the thickness of cooling pads. This is also expected because with increase in the thickness of the cooling air gets greater contact time with water layer causing higher evaporation of water. For 5 cm thick pads saturation efficiency varied from 74.9 to 88.3%, for 10cm thick pads saturation efficiency varied from 90.4 to 101.6% and for 15cm thick pads saturation efficiency varied from 98.6 to 106.9%. The overall variation in the saturation efficiency is from 74.9 to 106.9%.

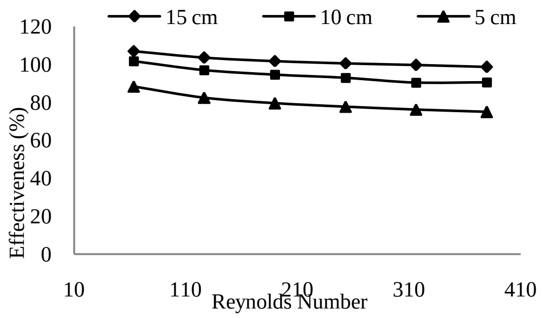

Fig. 1. Variations of saturation efficiency of pads of different thickness with Reynolds number.

Change in Thermal Exergy

For 5 cm thick pads Thermal Exergy varied from 136 to 344 for 10cm thick pads Thermal Exergy varied from 199 to 505 and for 15cm thick pads Thermal Exergy varied from 231 to 588. The overall variation in the Thermal Exergy is from 136 to 588.

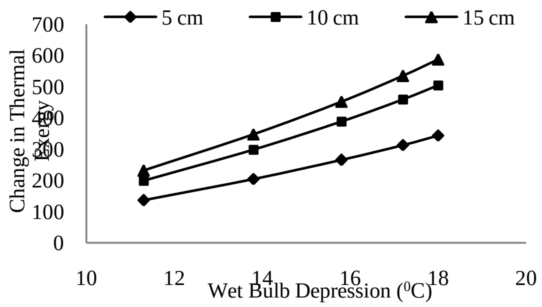

Fig. 2. Variations of change in Thermal Exergy of inlet air with Wet bulb Depression.

Change in Chemical Exergy

For 5 cm thick pads Chemical Exergy varied from 51 to 254 for 10cm thick pads Chemical Exergy varied from 27 to 83 and for 15cm thick pads Chemical Exergy varied from 94 to 260. The overall variation in the Chemical Exergy is from 51 to 260.

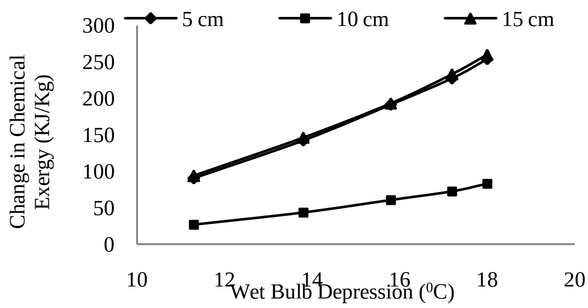

Fig. 3. Variations of change in Chemical Exergy of inlet air with Wet bulb Depression.

Total input Exergy

For 5 cm thick pads Total Output Exergy varied from 676 to 1437, 10cm thick pads Total Output Exergy varied from 584 to 2296 and for 15cm thick pads Total Output Exergy varied from 876 to 1939. The overall variation in the Total Output Exergy is from 676 to 1939.

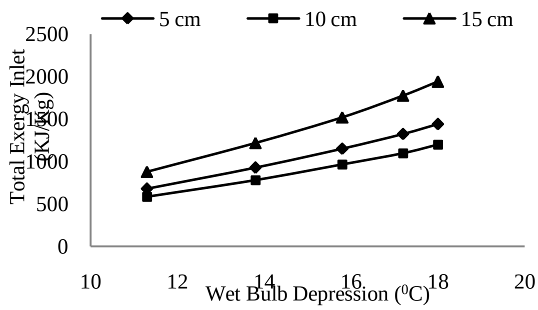

Fig. 4. Variations of Total Exergy of inlet air with Wet bulb Depression.

Total Output Exergy

For 5 cm thick pads Total Output Exergy varied from 262 to 670 for 10cm thick pads Total Output Exergy varied from 199 to 505 and for 15cm thick pads Total Output Exergy varied from 234 to 593. The overall variation in the Total Output Exergy is from 262 to 593.

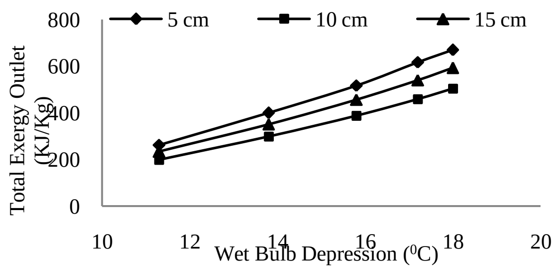

Fig. 5. Variations of Total Exergy of outlet air with Wet bulb Depression.

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Exergy Efficiency

For 5 cm thick pads Exergy Efficiency varied from 38.7 to 46.7 for 10cm thick pads Exergy Efficiency varied from 34.1 to 41.9 and for 15cm thick pads Exergy Efficiency varied from 26.7 to 30.4. The overall variation in the Exergy Efficiency is from 26.7 to 40.6.

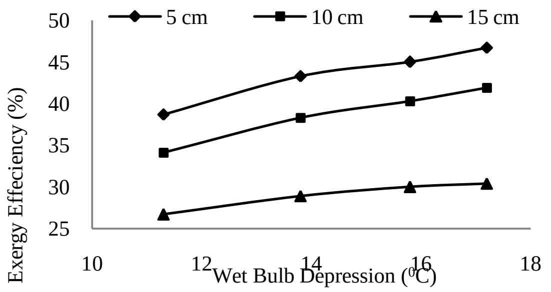

Fig. 6. Variations of Exergy Efficiency of DEC with Wet bulb Depression.

Conclusion

In this paper exergy analysis of direct evaporative cooling system is presented. Variations in wet-bulb Saturation Efficiency, change in thermal exergy, chemical exergy, total inlet exergy, total outlet exergy and exergy efficiency is analysed for pads of different thickness with constant face velocity. It is observed that for a give thickness of pads Second Law Efficiency does not changes as wet bulb saturation efficiency changes. Irreversibility and entropy generation vary in same manner for given thickness of pads. Sustainability index curve is almost flat for pads of given thickness, however it increases slightly with the thickness of the pads.

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