Journal of Mechanical Research and Application
ISSN: 2251-7383, eISSN: 2251-7391
Vol. 3, No. 1, 2011, 9-20

Exergy Analysis of Evaporative Cooling for Optimum Energy Consumption in Diverse Climate Conditions

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Received: 9 Mar. 2011; Accepted: 12 Dec. 2011

Abstract: In this paper, exergy of conditioned air, exergy efficiency, irreversibility, and entropy generation of common models of evaporative cooling have been investigated in five cities of Iran. Direct evaporative cooling (DEC), indirect evaporative cooling (IEC), and two-stage IEC/DEC as the most popular methods of cooling have been modeled. Atmospheric conditions are considered as the dead state of each city. Exergy analyses of conditioned air are based on the output results of the theoretical modeling of evaporative cooling. Moreover, exergy balances of three cooling methods are derived. Thus, exergy destruction, reversible work, and entropy generatiom are calculated according to the exergy balances. The results obtained reveal that Bam, which is a hot city with medium relative humidity (24%RH), has the best exergy efficiency of direct evaporative cooling. The highest exergy efficiency of two-stage indirect/direct evaporative cooling belongs to Kerman. Kerman with the lowest dry-bulb temperature has medium relative humidity (24%RH). In addition, total output exergy of air in Yazd is more than other cities. Yazd is a hot-dried city with rather low relative humidity (19.5%RH).

Keywords: Exergy Analysis, Evaporative Cooling, Exergy Efficiency, Irreversibility, Entropy Generation

1. Introduction

Exergy analysis exploits the conservation of mass and energy principles as well as the second law of thermodynamics for the analysis and enhancement of energy. The exergy method is a useful means for promoting effectiveness of energy-resource use. The method enables immensities of losses to be identified and meaningful efficiencies to be determined [1]. Exergy is another name for the available energy in thermodynamic terms. Since the introduction of exergy concept, it has been burgeoned in a diversity of engineering fields, including air conditioning systems [2-4]. A considerable fraction of the energy is consumed in air conditioning. In order to appraise the available alternatives or to select the apt form of air conditioning systems, defining the effective cooling techniques are consequential. Although the efficiency of air conditioning systems has been evaluated by means of the first law of thermodynamics, the second law of thermodynamics is another apposite method [5].

In fact, due to the great consumption of energy in buildings, there are increasing demands to design building heating, ventilation, and air-conditioning (HVAC) equipment and systems energy efficiently. In addition, evaporative cooling system can be an economical alternative, or can be used as a pre-cooler in the conventional systems. Also, it is popular due to its zero pollution, easy maintenance, low energy consumption, simplicity, and good indoor air quality [6,7].

Direct evaporative cooling (DEC) is the oldest and the most widespread form of air conditioning. The underlying principle of DEC is the conversion of sensible heat to latent heat. Through a direct evaporative cooling system, hot outdoor air passes a porous wetted medium. Heat is absorbed by water as it evaporates from the porous wet medium, so the air leaves the system at a lower temperature. In fact, this is an adiabatic saturation

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process in which dry bulb temperature of the air reduces as its humidity increases (constant enthalpy). Some of the sensible heat of the air is transferred to water and become latent heat by evaporating some of the water. The latent heat follows the water vapor and diffuses into the air. The minimum temperature that can be obtained is the wet bulb temperature of the entering air [8].

Another alternative of evaporative cooling is indirect evaporative cooling (IEC) which has high potential for providing air conditioning demands at low energy costs. An indirect evaporative cooling system consists of two impervious separate air passages, primary and secondary air passages which are dry and wet, respectively. In the primary passages, outdoor air flow is sensibly cooled without adding water, while the secondary air and water flow in the secondary passages. The surface of the secondary passages is wetted by spray water, so that water film evaporates into the secondary air and this decreases the temperature of the wall. As a result, the cold wall removes the heat from the outdoor air. Consequently, the leaving air from the primary passages has a lower wet-bulb temperature than the entire air. In order to use this lower wet-bulb temperature, for instance, a proportion of the leaving air can be used as the secondary air. This method is called regenerative indirect evaporative cooler [9,10].

In addition, the lower wet-bulb temperature of the leaving air can be used in combining indirect and direct evaporative coolers to boost the cooling capacity. By adding an indirect stage to a direct stage, a two-stage indirect/direct cooler is capable of cooling the outdoor air more than a stand-alone DEC unit. The outdoor air is cooled through the IEC without an increase in its humidity ratio, and then it is further cooled through DEC [11].

Several research papers has been dedicated to explore the issues about evaporative cooling such as, Dai and Sumathy [12], Liao and Chiu [13], Al-Sulaiman [14], Camargo et al. [7], Chengqin and Hongxing [15] and Hettiarachchi et al. [16]. They have proposed mathematical modeling and done experimental tests in order to analyze efficiency or simulate direct and indirect evaporative cooling. El-Dessouky et al. [17] and Heidarinejad et al. [11] have studied a two-stage evaporative cooling to examine its efficiency on performance of air conditioning. However, none of them has investigated the second law of thermodynamics nor have they analyzed exergy in their research.

In the field of exergy analyses, Renchengqin *et al.* [4] have analyzed exergy changes in HVAC systems. They have presumed an unusual dead-state to eliminate the exergy calculation of water. Also, they have broken 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 [5] has calculated 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. Taufiq *et al.* [18] have studied the exergy analysis of the direct evaporative cooling in a Malaysian building. The average temperature and relative humidity were considered as dead state. The results obtained have shown that an increase in relative humidity increases exergy efficiency. Muangnoi *et al.* [19] have used an exergy analysis to demonstrate exergy and exergy destruction of water and air flow through the cooling tower. Aforementioned research papers on exergy analysis have not examined the exergetic efficiency on various climates. Indeed, multi-climate countries such as Iran necessitate an abroad consideration of exergy analysis on diverse environmental conditions.

To the best of our knowledge, little amount of investigation on exergy efficiency of evaporative cooling is available in the literature. In addition, different points of view and approaches are held about exergy analyses that have led to the consideration of different dead states. Also, information about the second law efficiency in different atmospheric conditions is not sufficient. Hence, the researchers provide some deep analyses about exergy efficiency of evaporative cooling in various climatic conditions.

In this paper, direct, indirect, and two-stage direct/ indirect evaporative cooling systems are firstly modeled. Then changes in exergy of cooled air in each system in various climatic conditions of some cities of Iran have been investigated. Moreover, reversible work, exergy destruction and entropy produced by irreversibilities are calculated by means of exergy balance. The atmospheric state of each city is selected as the dead state of that city.

2. Modeling of evaporative coolers

Modeling of direct evaporative cooling and indirect evaporative cooling have been described separately in the following sections.

2.1. Modeling of direct evaporative cooling

In a direct evaporative cooler, the transformation of the heat and mass between air and water causes decrease in the air dry bulb temperature (DBT) and an increase in its humidity, while the enthalpy is basically constant in a perfect process. The minimum temperature that can be attained is the wet bulb temperature (WBT) of the incoming air. Wet pads or porous materials equip a water surface in which the air is humidified and the pad is wetted by the dripping water.

Assuming the hot air flow near to a wet surface, according to Fig.1, heat transfer occurs due to the difference in surface temperature T_{SW} and the flow of air temperature T_{air}. Because the absolute humidity (concentration) of the air close to the surface ω_{air} is different from the humidity of the wet surface T_{SW}, mass transfer also occurs.

The total differential heat flow is achieved by Eq. (1).

$$dQ = m_a di_p = \left[h_c(T_{sw} - T) + h_m i_{vs}(\varpi_{sw} - \varpi)\right] dA \tag{1}$$

Using the specific enthalpy of the mixture as the sum of the individual enthalpies and assuming that air and vapor are perfect gases, Eq. (1) can be rewritten as:

$$dQ = \frac{h_c}{C_{pa}}[(i_{sw} - i) + h_m \frac{(\omega_{sw} - \omega)}{Le}(i_{vs} - i_v Le)]dA \tag{2}$$

Where, h_c is convective heat transfer coefficient, h_m is mass transfer coefficient, i_{vs}, and i_v are specific enthalpy of vaporization of the water and specific enthalpy of the vapor at surface temperature, respectively, C_{pa} is specific heat of the humid air, and $Le = \frac{h_c}{h_m C_{pa}}$ is Lewis number.

By considering $Le = 1$, the second term in bracket in Eq. (2) is negligible in presence of the first term. Therefore, by combining Eq. (1) with Eq. (2) and integrating then, the temperature of leaving air is derived as:

$$\frac{T_{a,out} - T_{sw}}{T_{a,in} - T_{sw}} = exp\left(\frac{-h_c A}{\dot{m}_a C_{pa}}\right) \tag{3}$$

It is assumed that the makeup water entering the sump to replace evaporated water is at the same adiabatic saturation temperature of the incoming air. Humidity ratio of leaving air is calculated from the following equation:

$$\frac{\omega_{a,out} - \omega_{sw}}{\omega_{a,in} - \omega_{sw}} = exp\left(\frac{-h_m A}{\dot{m}_a C_{pa}}\right) \tag{4}$$

Correlations to establish the convective heat transfer coefficient in a rigid cellulose pad can be found in literature.

2.2. Modeling of indirect evaporative Cooling

Non-adiabatic procedure as a result of three streams which are primary air (supply air), secondary air (working air), and water occurs in the indirect stage. In the indirect evaporative cooler, parallel plates form a series of primary and secondary passages. In cross flow configuration, primary air flows perpendicularly to secondary air and water. The water is sprayed on the top of the heat exchanger. The water and the secondary air flow downward along wall surfaces of the secondary passages (wet passages). The Primary air flows in the alternative passages. Fig. 2 shows the schematic diagram of IEC.

The balance of mass and energy of all streams should be taken into account. The following hypotheses are employed in the development of the mathematical model:

- The indirect evaporative cooler is insulated from the environment.
- Diffusion is considered negligible.
- Lewis factor is unity.

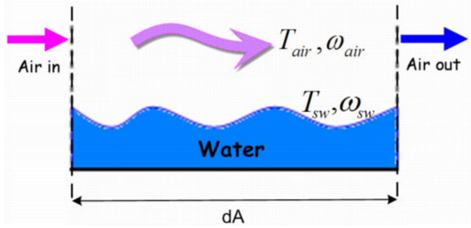

Fig. 1. A schematic of the direct evaporative cooling.

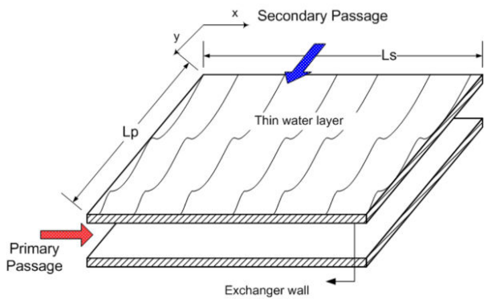

Fig. 2. Schematic diagram of the indirect evaporative cooling.

- The water wets uniformly all over the passages.
- Wall, bulk water, and air/water interface have an equal temperature.

Based on the above assumptions, a set of differential equations can be derived by applying principles of energy and mass conservation. For the primary air the governing energy balance equation is formulated asbelow:

$$\dot{m}_p C_{Pa} \frac{dT_p}{dx} = h_p L_p (T_w - T_p) \tag{5}$$

where, $\dot{m}$ is mass flow rate, C_{pa} air specific heat at constant pressure, h_p heat transfer coefficient, and L_p width of the primary passages. For conservation of energy and moisture in the secondary air, the following equations can be derived:

$$\dot{m}_s C_{Pa} \frac{dT_s}{dy} = h_s L_s (T_w - T_s) \tag{6}$$

$$\dot{m}_p \frac{d\omega_s}{dy} = h_m L_s (\omega_w - \omega_s) \tag{7}$$

Where, h_s and h_m are respectively heat and mass transfer coefficient and L_s is width of the secondary passage.

The energy balance equation for the water streams in the cooler is as follows:

$$\dot{m}_P C_{Pa} dT_p + \dot{m}_s C_{Pa} dT_s + \dot{m}_w C_{Pw} dT_w = 0 \tag{8}$$

By using multi step numerical integration coupled differential Eqs. (5), (6) and (8) can be solved simultaneously. Further information about solution method can be found in literature.

In a two-stage direct/indirect evaporative cooling, IEC and DEC stages are in a row, in a way that the leaving air from primary passages of IEC enters DEC. Lower temperatures can be achieved due to the lower wet-bulb temperature of entering air in the DEC. Furthermore, humidity ratio of leaving air in a two-stage IEC/DEC is not as much as leaving air in an individual DEC. The two-stage evaporative coolers have become popular in regions where dry-bulb temperature of outdoor air is high. Thus, exergy investigation of two-stage evaporative complements the analysis.

3. Exergy analysis

Exergy evinces the capability of energy to work. Exergy transfer happens in three ways, exergy transfer related to work, heat transfer, and matters entering and exiting a control volume. All processes in nature are irreversible. Irreversibilities within the system or the control volume destroy exergy [18]. Air-conditioning processes are basically steady-flow processes, so, exergy balances can be written as shown below [1]:

$$\sum_{in} E'x_Q + \sum_{in} \dot{m}(\psi) - \sum_{out} E'x_Q - \sum_{out} \dot{m}(\psi) - E'x_{dest} = 0 \tag{9}$$

In Eq. (9) kinetic and potential energies and work are assumed to be negligible. $\dot{E}x_Q$ is exergy transfer associated with heat transfer, $\dot{E}x_{dest}$ is the exergy destruction which is a positive quantity for any actual process and zero for a reversible process, and ψ represents specific exergy that is written as the following equation:

$$\psi = (i - i_0) - T_0(s - s_0) \tag{10}$$

Where i_0 and s_0 are specific enthalpy and entropy at the dead state, respectively.

Humid air can be considered as an ideal gas containing dry air and water vapor mixture. Wepfer et al. [20] has introduced the total flow exergy of humid air per kg dry air by the Eq. (11):

$$\psi_a = \underbrace{(C_{Pa} + \omega C_{Pv})\, T_0 \left(\frac{T}{T_0} - 1 - In\frac{T}{T_0} \right)}_{thermal} + \underbrace{(1 + 1.608\omega) R_a T_0\, In\frac{P}{P_0}}_{mechanical} \tag{11}$$
$$\underbrace{+ R_a T_0 \left[(1 + 1.608\omega)\, In\left(\frac{1 + 1.608\omega_0}{1 + 1.608\omega} \right) + 1.608\omega\, In\frac{\omega_0}{\omega} \right]}_{chemical}$$

Where, C_{pv} is water vapor specific heat at constant pressure, R_a is the air specific gas constant, and constant 1.608 is the ratio of molar mass of air to molar mass of water vapor. Chengqin et al. [4] have divided exergy of humid into three components: thermal, mechanical and chemical which are present as the first, second and third term in the Eq. (11). This division simplifies exergy analysis in air-conditioning systems.

The specific flow exergy of liquid water can be approximated by the following equation:

$$\psi_w = -R_v T_0 \, In\phi_0 \tag{12}$$

In the equation above, R_v is vapor specific gas constant, and ϕ_0 is the proportion of the water vapor pressure of the unsaturated atmospheric air to the saturated water vapor pressure at the dead-state temperature. Eq. (12) is a simplified form for exergy of liquid water. The complete form has been suggested by Wepfer et al. [20]. According to Eq. (12), relative humidity predominates exergy of water.

The exergy efficiency of an overall air-conditioning process may be formulated as [1]:

$$\eta_{ex} = 1 - \frac{exergy\ destroyed}{exergy\ sup\ plied} \tag{13}$$

In Eq. (13), exergy supplied is input exergy in the process, exergy destroyed represents the lost work potential and is also called the irreversibility or lost exergy. Also, it is proportional to the entropy generated. Irreversibility is expressed as:

$$T_0 S'_{gen} = \dot{I} = \dot{W}^{rev} - \dot{W}_{c.v.} \tag{14}$$

Where $\dot{S}_{gen}$ is the entropy generated in the process, $\dot{I}$ is irreversibility, $\dot{W}^{rev}$ is reversible work, and $\dot{W}_{c.v.}$ is axial work.

Reversible work presents the minimum amount of work necessary to perform a process. In the direct and indirect evaporative cooling no axial work is done on the control volume. So, irreversibility or exergy destroyed shows the amount of reversible work. Fig. 3 shows direct and indirect evaporative cooling processes on a psychometric chart.

As shown in Fig. 3, direct evaporative cooling is an adiabatic humidification in which dry-bulb temperature decreases while moisture increases. In other words, little amount of water on the pad evaporates into the air. Dry-bulb temperature decreases during indirect cooling process, and only a change in the sensible heat occurs. Based on the descriptions above and by employing Eq. (9), exergy balance for each process can be written as follows:

Direct evaporative cooling:

$$\dot{m}_a\psi_{a1} + \dot{m}_w\psi_w - \dot{m}_a\psi_{a2} - \dot{I} = 0 \tag{15}$$

$$\dot{m}_w = \dot{m}_a(\,\omega_2 - \omega_1\,) \tag{16}$$

In direct evaporative cooling, added moisture to air is considered as an input quantity.

Indirect evaporative cooling:

$$\dot{m}_a\psi_{a1} - \dot{m}_a\psi_{a2} - E'x_Q - \dot{I} = 0 \tag{17}$$

Exergy transfer associated with heat transfer can be derived by integration over input and output points.

$$\begin{aligned} E'x_Q &= -\int_1^2 \left(1 - \frac{T_0}{T}\right) dQ \\ &= \left(C_{Pa} + \omega C_{Pv}\right)\left[\left(T_1 - T_2\right) - T_0 \, In\frac{T_1}{T_2}\right] \end{aligned} \tag{18}$$

Two-stage indirect/direct evaporative cooling:

$$\dot{m}_a\psi_{a1} + \dot{m}_w\psi_w - \dot{m}_a\psi_{a2} - E'x_{Q_{1-2'}} - \dot{I} = 0 \tag{19}$$

Eqs. (16) and (18) can be used for water mass flow rate and exergy transfer associated with heat transfer, respectively.

By means of Eqs. (15), (17), and (19), irreversibilities as well as reversible work and entropy generation can be calculated.

4. Results and discussion

Exergy analyses of three models of evaporative cooling have been investigated in five cities of Iran. Iran has a wide variety of climatic conditions and evaporative cooling is the most popular method of cooling system in this country. Thus, examination of efficiency from the second law of thermodynamics point of view complements demand for energy saving and the requirements of comfort conditions. A list of cities and their climatic characteristics are tabulated in Table 1. Psychrometric specifications of outdoor conditions of each city are based on 1% design condition according to the ASHRAE standard [21,22].

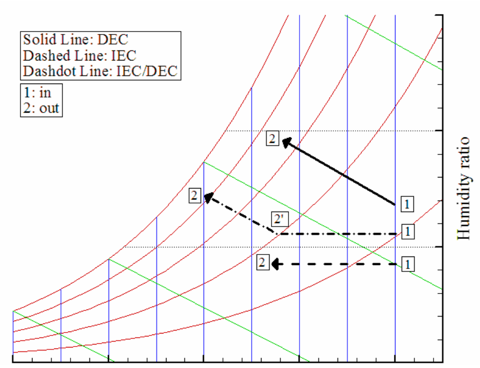

Fig. 3. Depiction of direct, indirect and indirect/direct evaporative cooling processes on a psychrometric chart.

Tehran has the highest wet-bulb temperature and relative humidity among the cities studied here. Yazd is a hot-dried city which its relative humidity is the lowest among the cities and its dry-bulb temperature is high. Bam is a hot city with medium relative humidity. Kerman with the lowest dry-bulb and wet-bulb temperature has medium relative humidity. Shiraz has medium dry-bulb temperature and relative humidity. Bam and Kerman are neighboring cities with different altitudes.

The following physical characteristic are presumed in order to conduct the investigation on exergy of conditioned air. An indirect evaporative heat exchanger with dimensions of 0.5×0.5×0.4 m^3 and plate spacing of 7 mm are considered. Mass flow rate of primary and secondary air are 0.56 kg/s and 0.28 kg/s, respectively. Dimensions of the direct evaporative pad are 0.5×0.5×0.15 m^3. 50 Pa is assumed as the pressure drop after each cooling stage.

The evaporative cooling modeling is validated by experimental investigation of Heidarinejad et al. [11]. Test setup contains three parts. Part one includes primary and secondary air simulators which provide air flow for both direct and indirect evaporative coolers. Part two consists of IEC/DEC unit including a plastic wet surface heat exchanger as IEC unit, a cellulose pad as DEC unit, a water circulating pump, flow meters, valves to control water flow rate of units. Part three is a control panel for adjusting steady-state temperature and relative humidity and flow rate of primary and secondary air.

Validation is based on a two-stage evaporative cooling which contains both the DEC and the IEC stages. Three cities of Tehran, Yazd and Bam are chosen as examples for this verification. As shown in Fig. 4 the theoretical modeling of evaporative cooling calculates temperature and the humidity ratio of the output air precisely.

For all the exergy analyses, atmospheric conditions of each city are assumed as the dead state. So, entrance air has zero exergy.

The most common form of evaporative cooling in Iran is the direct evaporative cooling system. Table 2 shows dry-bulb temperature depression and humidity ratio increase after passing through the direct evaporative pad.

Exergy of humid air can be divided into three components. Monitoring changes of the thermal and the chemical exergy of air simplifies the exergy analysis of evaporative cooling. An increase in the humidity ratio augments the chemical exergy of air. Also, a decrease in temperature increases the thermal exergy of air. Thus, variations in temperature and humidity ratio cause changes in exergy of air. Due to the same amount of pressure drop, the mechanical term of exergy almost equally lessens in all cities.

Fig. 5 depicts total exergy changes per kg air after the direct evaporative cooling in all the cities.

As shown in Fig. 5, the total exergy increases more than that of other cities in Yazd and Bam. Tehran has the lowest exergy change. Yazd and Bam have the most temperature depression and the humidity ratio increase (See Table 2). Consequently, exergy augmentation in these two cities is more than that of other cities. In contrary, Tehran has the lowest humidity ratio rise and its temperature does not drop significantly in comparison with other cities.

Therefore, the lowest exergy difference belongs to Tehran. However, as shown in Fig. 6 exergy efficiency of direct evaporative cooling in Tehran is best after Bam.

Exergy efficiency of direct evaporative cooling is roughly the ratio of exergy of leaving air to humidity ratio difference. Dry-bulb temperature depression in Bam is remarkably more than that of other cities, causing to have high total exergy.

Tehran has the lowest humidity ratio increase among other cities. Thus, total output exergy and increase in humidity ratio are two effective parameters in exergy efficiency of direct evaporative cooling. Based on this conception, Shiraz has the lowest exergy efficiency because its total output exergy is not so high, and in addition, its humidity ratio does not trivially increase. In other words, more output exergy and lower difference of humidity ratio raise exergy efficiency in direct evaporative cooling.

It is worth mentioning that the output humidity ratio increases the output exergy and also shows the amount of water absorbed by air as an input matter.

As far as indirect evaporative cooling is concerned, exergy changes are because of pressure and temperature drop which respectively cause decrease and an increase in exergy of leaving air. Fig. 7 illustrates the total output exergy of air after the indirect evaporative cooling unit.

During indirect evaporative cooling, the humidity ratio is constant. So, chemical exergy does not change. Mechanical exergy again reduces equally. So, the total output exergy depends on the dry-bulb temperature depression.

As shown in Fig. 7, dry-bulb temperature reduction in Yazd is more than that of other cities, and therefore, it has the highest total output exergy. The lowest total output exergy belongs to Tehran because of insignificant temperature depression.

The two-stage indirect/direct evaporative cooling is applicable in regions where a stand-alone direct evaporative cooling unit cannot provide comfort conditions. Table 3 lists dry-bulb temperature depressions, humidity ratio increase, thermal and chemical exergy of conditioned air in two-stage IEC/DEC.

In a two-stage heat exchanger increase in the humidity ratio is not as much as in a direct evaporative cooler. So, chemical exergy of air does not play a significant role in the total exergy. However, thermal exergy increases more than the direct evaporative cooler due to more temperature depression. Fig. 8 shows total output exergy of conditioned air after the two-stage IEC/DEC.

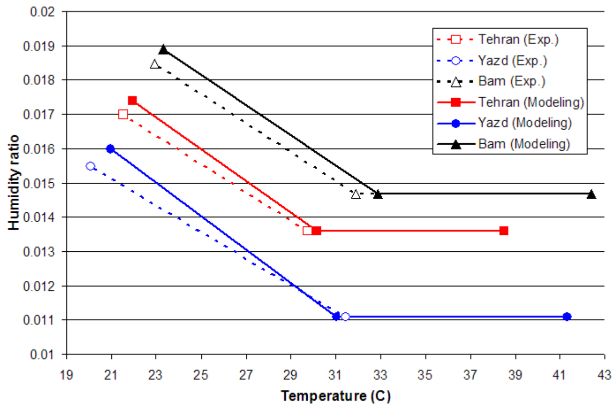

Fig. 4. Validation of evaporative cooling based on experimental investigation of Heidarinejad et al. [11].

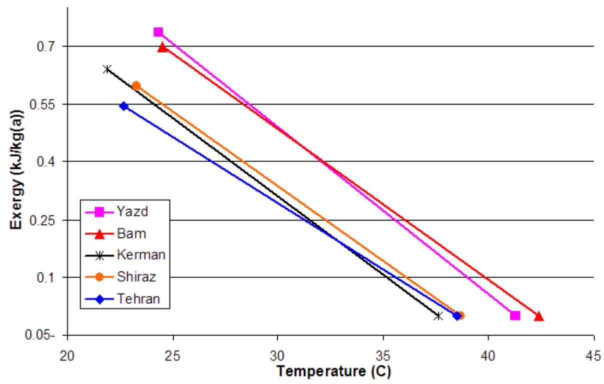

Fig. 5. Total specific exergy changes after direct evaporative cooling in five cities.

Table 1. Specifications of the cities evaluated for exergy analysis

City	Longitude	Latitude	Dry-bulb temperature (°C)	Relative humidity (%)	Altitude (m)
Tehran	51.4°	35.7°	38.5	27	1190
Yazd	55°	32°	41.3	19.5	1327
Bam	58.21°	29.6°	42.4	24	1067
Kerman	57.1°	30.3°	37.6	24	1754
Shiraz	37.29°	30.25°	38.7	22	1481

Table 2. Changes in temperature, humidity ratio, thermal exergy, and chemical exergy after the direct evaporative cooling process

City	Temperature decrease (°C)	Humidity ratio increase (g/kg)	Thermal exergy (J/kg(a))	Chemical exergy (J/kg(a))
Tehran	15.79	6.10	430.49	168.39
Yazd	16.95	7.50	491.79	298.61
Bam	17.86	7.00	548.18	205.81
Kerman	15.70	6.95	424.95	272.23
Shiraz	15.42	6.95	381.36	243.86

As shown in Fig. 8, during the DEC stage, exergy augments notably in comparison with the IEC stage. This is because of both higher dry-bulb temperature depression and increase in the humidity ratio.

Yazd has the most decrease in dry-bulb temperature and its humidity ratio accrues more than other cities. So, it has the highest total output exergy. Tehran, because of lower temperature depression and humidity ratio increase (See Table 3), has the lowest exergy augmentation.

Exergy efficiency of the two-stage IEC/DEC depends on total output exergy of air, exergy transfer associated with heat transfer in the IEC stage, and the amount of humidity ratio changes. Fig. 9 compares exergy efficiency of the two-stage IEC/DEC in aforementioned cities.

Exergy efficiency of the two-stage IEC/DEC is the proportion of sum of both exergy of the leaving air and the heat exergy transfer to the humidity ratio difference. As stated in Table 3, dry-bulb temperature in Kerman decreases considerably. It means that exergy transfer due to heat transfer is remarkable. Also, its total output exergy is high (See Fig. 8). As a result, exergy efficiency in Kerman is more than other cities. Although Yazd has the highest output exergy and temperature difference, its high humidity ratio growth derogates effectiveness.

Indeed, impact of humidity ratio changes enact essential role in exergy efficiency of evaporative cooling. Inferior efficiency belongs to Tehran due to the lowest output exergy and temperature depression in this city.

It is worth mentioning that exergy efficiency of a two-stage IEC/DEC is higher that a stand-alone DEC. Also, in cities like Yazd that has lower relative humidity, exergy of water is larger.

Irreversibility or exergy lost is the difference between input and output exergies in a controlled volume. It is also the difference between the reversible work and the axial work in a controlled volume. Because no axial work is done in the direct or indirect evaporative cooling, exergy lost is equal to the reversible work. Fig. 10 shows the amount of reversible work in the direct, indirect stages and the entire two-stage evaporative cooling in the cities.

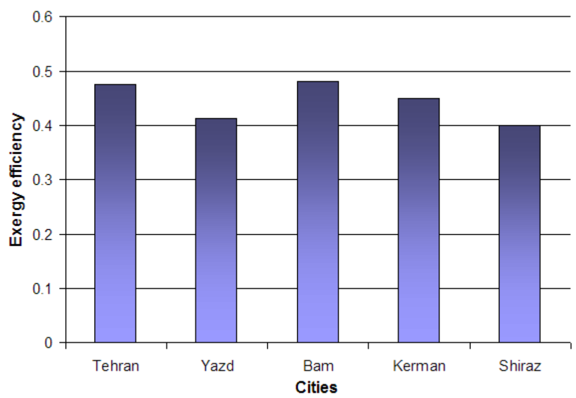

Fig. 6. Exergy efficiency of direct evaporative cooling in different cities of Iran.

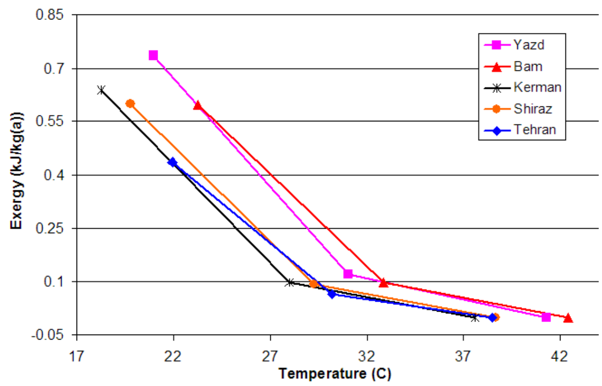

Fig. 8. Total specific exergy changes after a two-stage indirect/direct evaporative cooling in five cities.

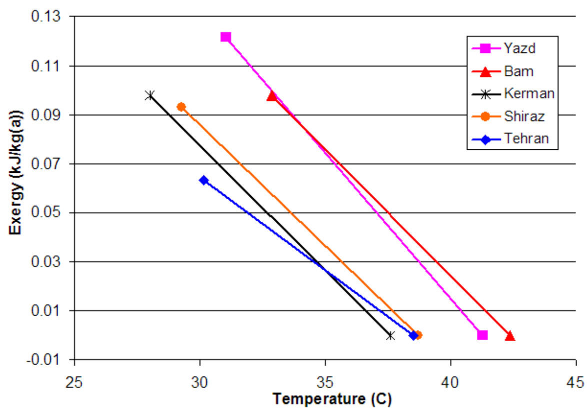

Fig. 7. Total specific exergy changes after indirect evaporative cooling in five cities.

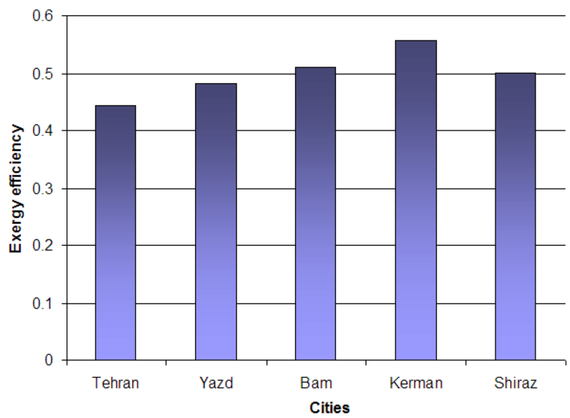

Fig. 9. Exergy efficiency of the two-stage indirect/direct evaporative cooling in the five cities.

Table 3. Changes in temperature, humidity ratio, thermal exergy, and chemical exergy after indirect/direct evaporative cooling.

City	Temperature decrease	Humidity ratio increase (g/kg)	Thermal exergy (J/kg(a))	Chemical exergy (J/kg(a))
Tehran	16.58	3.80	473.52	68.48
Yazd	20.32	4.90	708.77	135.18
Bam	19.10	4.20	625.51	78.07
Kerman	19.33	4.25	646.38	105.94
Shiraz	18.90	4.13	616.10	94.82

Although there is no attempt to produce work during evaporative cooling process, the potential to do work still exists, and the reversible work is a quantitative measure of this potential.

As a thumb rule, exergy of water is completely larger than humid air. So, in the exergy balances, exergy of water dominates equations.

As shown in Fig. 10, because of the high amount of water evaporation (increase in the humidity ratio) during the direct cooling in Yazd, the reversible work of the DEC is more than that of other cities.

Besides exergy of water, other parameters such as exergy transfer associated with heat transfer in the IEC and output exergy affects the reversible work. Kerman has the best exergy efficiency of the two-stage evaporative cooling system and its water evaporation is not significant. Thus, its reversible work is lowest among the cities.

Another parameter that can be calculated is the entropy produced in the evaporative cooling processes. Fig. 11 shows the entropy produced in separate processes for the direct evaporative cooling, indirect evaporative cooling, and a two-stage evaporative cooling.

The entropy produced also reveals the amount of reversible work in each individual cooling process. Figs. 11 and 10 demonstrate that irreversibility in the indirect evaporative cooling is completely lower than direct evaporative cooling. This is because the exergy of water is a large quantity in DEC. Again as shown in Fig. 11, exergy of water plays an important role in the entropy generated during the direct evaporative cooling. Yazd, due to the high amount of water evaporation, has the largest amount of the entropy generation in the direct evaporative cooling. Although a great amount of water evaporation in Bam increases entropy generation, significant temperature depression lessens produced entropy augmentation (Table 2).

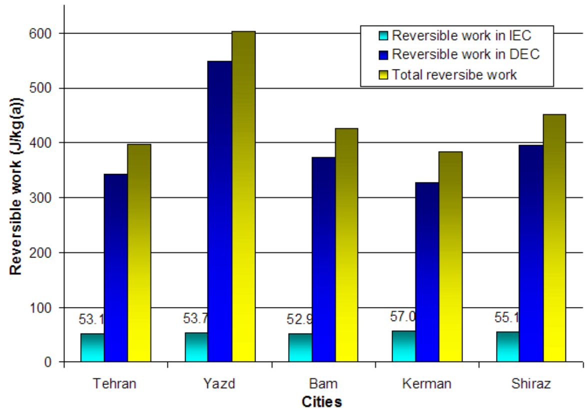

Fig. 10. Reversible work in the direct and indirect parts of a two stage evaporative cooling in five cities.

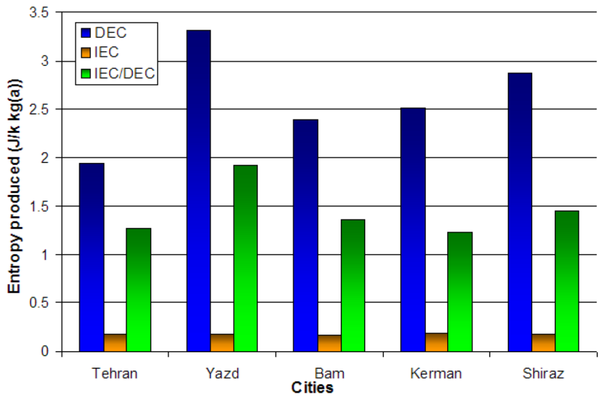

Fig. 11. Entropy produced in the separate processes of the direct evaporative cooling, indirect evaporative cooling and two-stage evaporative cooling in five cities.

Neither dry-bulb temperature depression nor water evaporation is noticeable in Tehran. In other words, output exergy does not considerably change (See Fig. 5). Also, the relative humidity in Tehran is higher than that of other cities. It means exergy of water is the lowest in Tehran. Thus, entropy produced during the direct evaporative cooling is low in Tehran.

Moreover, conditioned air after the stand-alone DEC is moistened more than the direct cooling in the two-stage evaporative cooling. So, the entropy produced or reversible work in the DEC is higher than that of the two-stage IEC/DEC.

5. Conclusions

Exergy analyses of common models of the evaporative cooling have been studied in five cities of Iran. Changes in exergy of conditioned air, exergy efficiency, irreversibility, and entropy produced in each city have been examined. The direct evaporative cooling, indirect evaporative cooling and a two-stage IEC/DEC are considered as the cooling systems. The results show that Yazd as a hot-dried city has the highest total output exergy of air in the three cooling systems. Exergy efficiency of the direct evaporative cooling unit depends on the amount of water evaporation in air as well as the total output exergy of air. Among those cities, Bam has the best exergy efficiency of direct evaporative cooling. Bam has hot dry-bulb temperature and relatively medium humidity.

The dry-bulb temperature depression determines the output exergy of air in indirect evaporative cooling. Moreover, exergy efficiency of the two-stage IEC/DEC unit depends on the total output exergy of air, exergy transfer associated with heat transfer in the IEC stage, and increase in humidity ratio. Kerman has the highest exergy efficiency of the two-stage evaporative cooling among all the cities. Kerman with the lowest dry-bulb and wet-bulb temperature has a relatively medium humidity. Also, due to high exergy efficiency in Kerman, the lowest amount of reversible work of the two-stage unit belongs to this city. Regarding exergy destruction, exergy of water is a large quantity in comparison with air; it is an input exergy in direct evaporative cooling. Thus, irreversibility during DEC is significantly higher than IEC. In addition, due to a higher increase in the humidity ratio of air after DEC, entropy generation or reversible work in DEC is more than other cooling systems.

Nomenclature

A	total wetted surface area (m^2)
C_p	specific heat at a constant pressure $(j/hg^{\circ}C)$
Ex	Exergy (j)
h	heat transfer coefficient $(w/m^{2\circ}C)$
h_m	mass transfer coefficient $(kg/m^{2\circ}C)$
i	specific enthalpy (j/kg)
I	Irreversibility (j)
L_e	Lewis number (dimensionless)
$\dot{m}$	mass flow rate (kg/s)
P	Pressure (ρa)
R	specific gas constant (j/kgK)
s	specific entropy (j/kgK)
T	Temperature $(^{\circ}C)$
W	axial work (j)
ϕ	relative humidity (dimensionless)
ω	humidity ratio (kg/kg)
ψ	specific exergy (j/kg)

Subscripts

a	air
c	convection
gen	generated
p	primary air
s	secondary air
sw	saturation
v	vapor
vs	vapor at a surface
w	water
0	dead state
number	state indices

Superscripts

rev	reversible

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