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PERFORMANCR EVALUATION AN AIR-COOLED OF AN AIR-COOLED HEAT EXCHANGER BASED ON ENTROPY GENERATION

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บทคัดย่อ

ในการศึกษานี้ ประสิทธิภาพของเครื่องแลกเปลี่ยนความร้อนด้วยอากาศถูกประเมินจากการสร้างเอนโทรปี (entropy generation) ภายใต้สภาพการใช้งานที่แตกต่างกัน การสร้างเอนโทรปีที่ถูกนำมาอธิบายเกี่ยวกับการผันกลับไม่ได้ (irreversibility) ด้วยการถ่ายเทความร้อนเท่านั้น การคาดการถูกยืนยันและแสดงผลด้วยผลการวิเคราะห์และผลการวัดจากเครื่องทดสอบ ผลการวิจัยพบว่า อุณหภูมิที่แตกต่างระหว่างน้ำร้อนที่ไหลเข้าและอากาศโดยรอบ หรือความแตกต่างของอุณหภูมิเริ่มต้น (ITD) สามารถใช้เป็นพารามิเตอร์ เพื่อกำหนดผลของสภาวะการทำงานได้ การถ่ายเทความร้อนและจำนวนการสร้างเอนโทรปีเพิ่มขึ้นตาม ITD ที่เพิ่มขึ้น จำนวนการสร้างเอนโทรปีเพิ่มขึ้นเล็กน้อยตามอัตราส่วนการไหลของน้ำต่ออากาศ ($\frac{L}{G}$) และจะน้อยลงเมื่อ ITD ลดลง ITD ระดับกลางถูกใช้สำหรับการทำงานของเครื่องแลกเปลี่ยนความร้อนด้วยอากาศ คือ limiting temperature ($ITD_{\min}$) จะได้รับเมื่ออุณหภูมิอากาศออกลดน้อยลงถึงขีดที่สุด การทำงานภายใต้ค่านี้จะส่งผลให้เปลี่ยนแปลงเล็กน้อย

ABSTRACT

In this study, the performance of an air-cooled heat exchanger was evaluated based on entropy generation under variations in operating conditions. The entropy generation was taken into account the irreversibility associated with heat transfer only. The predictions were validated and showed good agreement with analytical results and experimental measurements. The results reveal that the difference temperature between the inlet hot water and the ambient air or the initial temperature difference (ITD) can be used as the parameter to determine the effect of operating conditions. The heat transfer and entropy generation number increase with increasing ITD. The entropy generation number slightly increases with mass flow ratio of water to air ($\frac{L}{G}$) and the effecte is less as the ITD is reduced. The moderate ITD is required for the operation of the air-cooled heat exchanger. Its limiting temperature ($ITD_{\min}$) is obtained when the outlet air temperature is minimize. Working below this value results in over-sensitivity to small change.

KEYWORDS: air-cooled heat exchanger; entropy generation; exergy analysis; thermal capacity; operating parameters; *ITD*

1. Introduction

Cooling system performance is a function of a wide variety of factors including ambient weather conditions, cooling system type and capacity, etc. Therefore, when working on an air-cooled heat exchanger, we can ask what operating conditions are conductive to the highest heat transfer process. What we are looking for is more than just a high value of heat exchanger effectiveness. In fact, Bejan [1] demonstrated that the effectiveness (ε) is not a relevant measure of the thermodynamic performance of a heat exchanger. According to thermodynamics laws, an equal amount of heat is transfers from a high-temperature fluid (e.g. water) to a low-temperature fluid (e.g. air). However the output energy is always less available to do useful work than the input due to the effect of irreversibility that can be referred as entropy generation. This means that irreversibility is the reason why the available energy (or exergy) received by cold fluid is less than that supplied by hot fluid. The three main causes of irreversibility are heat transfer between the flows, pressure losses due to friction, and dissipation of energy to the environment [2]. Bejan [3] – [5] applied the entropy generation principle to the analysis and design of heat exchangers. He showed that the design of heat exchangers should be based on the concept of entropy generation minimization. The dissipation of energy to the environment or heat loss frequently represents a small fraction of the total heat transfer ($\approx$ 10%) and is usually neglected [2]. On the other hand, the contribution of fluid heat transfer and friction can be quantified by a single number which is the irreversibility distribution ratio, ϕ [6]. It is defined as the ratio of fluid flow irreversibility (friction) to heat transfer irreversibility. Bejan concluded that an optimum in entropy generation takes place for ϕ_{opt} = 0.168. Narayan et al. [7] developed a control volume procedure by which the entropy generation of combined heat and mass transfer devices (e.g. cooling towers) could be applied. In their study, the fluid streams were incompressible and the pressure change was negligible between the inlet and outlet.

The thermal capacity of an air-cooled heat exchanger can be evaluated by Logarithmic Mean Temperature Difference (LMTD) method or effectiveness-Number of Transfer Units ($\varepsilon - NTU$) method [8]. Reviews of the methods are presented by many authors (e.g. Incropera [9], Thomas [10], and Bejan [11]). However, the methods are mathematically equivalent to each other [12]. Many authors (e.g., Oilet et al. [13], Wang et al. [14], and Choi and Glicksman [15]) have successfully employed the effectiveness-NTU method to predict the performance of air-cooled heat exchangers. This method was extended by Asvapoositkul and Kuansathan [8] to predict the behavior of an air-cooled heat exchanger in which the effectiveness (ε) varies in proportion to $(\frac{L}{G})^n$ where $\frac{L}{G}$ is mass flow ratio of water to air. This approach can be used as a design tool for predicting an air-cooled heat exchanger's characteristics.

In this analysis, performance of air-cooled heat exchanger was evaluated to determine its operating conditions in steady flows of a fix geometry with entropy generation solely due to heat transfer. The entropy generation is used as a quantitative measure of irreversibility. The various operating parameters that account for irreversibility compete with one another. This means that the system generates the least entropy while still performing its fundamental engineering function.

2. Model descriptions

A heat exchanger is used to transfer heat from one fluid stream to another due to the temperature difference. Here, we assume that there is no phase change in fluid streams and that the heat exchanger has no heat loss to the environment. The conservation equations for the system can be written as follows.

The mass balance for the system

Air: $G_i = G_e = G$ (1)

Water: $L_i = L_e = L$ (2)

The energy balance for the system

$$Q = L(h_{L,i} - h_{L,e}) = G(h_{G,e} - h_{G,i}) \tag{3}$$

The exergy balance for the system

$$Q = L(\psi_{L,i} - \psi_{L,e}) = G(\psi_{G,e} - \psi_{G,i}) + \chi_{destroy} \tag{4}$$

where ψ_L = specific exergy of water

ψ_G = specific exergy of air

Neglect changes in kinetic energy and potential energy, exergy values of the fluids may be defined in the following equations.

The specific total flow exergy of humid air per kg dry air is [16]

$$\psi_e = (C_{p,G} + \omega C_{p,v})T_0(\frac{T_G}{T_0} - 1 - \ln\frac{T_G}{T_0}) + (1+\varpi)R_G T_0 \ln\frac{P_G}{P_0} + R_G T_0[(1+\varpi)\ln\frac{1+\omega_0}{1+\bar{\omega}} + \varpi \ln\frac{\omega}{\omega_0}] \tag{5}$$

$$\psi_{G,e} - \psi_{G,i} = C_{p,G}(T_{G,e} - T_{G,i}) - C_{p,G}T_0(\ln\frac{T_{G,e}}{T_{G,i}}) + C_{p,v}(\omega_e T_{G,e} - \omega_i T_{G,i}) - C_{p,v}T_0(\omega_e \ln\frac{T_{G,e}}{T_0} - \omega_i \ln\frac{T_{G,i}}{T_0}) - \tag{6}$$

$$C_{p,v}(\omega_e - \omega_i)T_0 + R_G T_0 \ln\frac{P_{G,e}}{P_{G,i}} + R_G T_0(\varpi_e \ln\frac{P_{G,e}}{P_0} - \varpi_i \ln\frac{P_{G,i}}{P_0} + R_G T_0 \ln\frac{1+\varpi_i}{1+\varpi_e} + R_G T_0(\varpi_e \ln\frac{1+\varpi_0}{1+\varpi_e} - \varpi_i \ln\frac{1+\varpi_0}{1+\varpi_i}) +$$

$$R_G T_0(\varpi_e \ln\frac{\varpi_e}{\varpi_0} - \bar{\omega}_i \ln\frac{\varpi_i}{\varpi_0})$$

The specific enthalpy of moist air, h_G, equals the sum of the specific enthalpy of dry air and the specific enthalpy of saturated water vapor at the temperature of the mixture [17].

$$h_G = h_{da} + \omega h_g \approx C_{p,G}T_G + \omega C_{p,v}T_G \tag{7}$$

For air-cooled heat exchanger or dry-cooling tower, the moisture content of the air remains unchanged ($\omega_e = \omega_i = \omega_0$). Therefore, we can write the equation (6) in the form

$$\psi_{G,e} - \psi_{G,i} = (h_{G,e} - h_{G,i}) - C_{p,G} T_0 (\ln \frac{T_{G,e}}{T_{G,i}}) - C_{p,v} T_0 \omega_0 \ln \frac{T_{G,e}}{T_{G,i}} - R_G T_0 \ln \frac{P_{G,e}}{P_{G,i}}) \tag{8}$$

To simplify the equation, we shall consider that no pressure change throughout the tower ($P_{G,i} \approx P_{G,e} \approx P_0$) and $C_{p,v}\omega_0 << C_{p,G}$. The effects from the last two terms can be ignored. With these simplifications, the specific exergy of air can be written as

$$\psi_{G,e} - \psi_{G,i} = (h_{G,e} - h_{G,i}) - C_{p,G} T_0 \ln \frac{T_{G,e}}{T_{G,i}} \tag{9}$$

The specific total flow exergy of liquid water is [16]

$$\psi_L = h_L(T_L, P_L) - h_0(T_0, P_0) - T_0[s_L(T_L, P_L) - s_0(T_0, P_0)] + [P_L - P_{sat}(T_L)]v_f(T_L) - R_v T_0 \ln \phi_0 \tag{10}$$

In the case where the fluid is incompressible and the pressure change is negligible between the inlet and outlet ($P_{L,i} \approx P_{L,e} \approx P_0$, $v_{f,i} \approx v_{f,e} \approx v_{f,0}$), the rate of exergy change is

$$\psi_{L,i} - \psi_{L,e} = (h_{L,i} - h_{L,e}) - T_0(s_{L,i} - s_{L,e}) \tag{11}$$

Substitute Eqs. (3), (9), (11) in Eq. (4) and rearrange, we get

$$\chi_{destroy} = L[-T_0(s_{L,i} - s_{L,e})] + GC_{p,G} T_0 \ln \frac{T_{G,e}}{T_{G,i}} \tag{12}$$

The rate of entropy generation during the process can be determined from its definition

$$\chi_{destroy} = T_0 \dot{s}_{gen} \tag{13}$$

Therefore,

$$\dot{s}_{gen} = L[-(s_{L,i} - s_{L,e})] + GC_{p,G} \ln \frac{T_{G,e}}{T_{G,i}} \geq 0 \tag{14}$$

The temperature of water in this application ranges from about 20 °C to about 60 °C. In this range, water can be treated as an incompressible substance with a constant $C_{p,L}$ values of 4.186 kJ/kg-K with negligible error. The change in entropy for water is determined to be

$$s_{L,i} - s_{L,e} = C_{p,L} \ln \frac{T_{L,i}}{T_{l,e}} \tag{15}$$

Entropy generation rate is generally used in a dimensionless form. There are a number ways of non-dimensionalizing entropy generation rate. The most frequently used entropy generation number is divided the entropy generation rate by the capacity flow rate [18]. According to Narayan et. al. [7], the entropy generation number is

$$s_{gen}^{*}=\frac{\dot{s}_{gen}}{GC_{p,G}}=-\frac{L}{G}\frac{C_{p,L}}{C_{p,G}}\ln\frac{T_{L,i}}{T_{l,e}}+\ln\frac{T_{G,e}}{T_{G,i}} \tag{16}$$

The heat exchanger effectiveness has received considerable attention by several researchers [2, 9-11]. By definition

$$\varepsilon=\frac{T_{G,e}-T_{G,i}}{T_{L,i}-T_{G,i}}=\frac{L}{G}\frac{C_{p,L}}{C_{p,G}}\frac{T_{L,i}-T_{L,e}}{T_{L,i}-T_{G,i}}=\frac{L}{G}\frac{C_{p,L}}{C_{p,G}}\frac{R}{ITD} \tag{17}$$

The initial temperature difference (ITD) is the difference temperature between the inlet hot water and the ambient air. The range (R) is the difference temperature between the inlet and the outlet of cooling water. The above equations are presented the relationship of various parameters such as temperatures and the mass flow rate ratio within the system.

3. Air-cooled heat exchanger simulation calculations

Prediction of air-cooled heat exchanger performance at various operating conditions is presented differently by various authors. One such approach is the effectiveness-NTU method, which is accurate and simple to implement, and can be applied to evaluate the thermal performance of an air-cooled heat exchanger. The heat exchanger effectiveness (ε) is a function of mass flow ratio of water to air ($\frac{L}{G}$) by the following relationship [8]:

$$\varepsilon=C(\frac{L}{G})^{n} \tag{18}$$

The values of C and n may be determined from an experiment. This is an air-cooled heat exchanger supply curve. The tower effectiveness (ε) is also determined from the effectiveness-NTU method [8, 15]:

Case I: $\frac{L}{G}<\frac{C_{p,G}}{C_{p,L}}$

$$NTU=-\ln[1+\frac{G}{L}\frac{C_{p,G}}{C_{p,L}}\ln(1-\varepsilon\frac{L}{G}\frac{C_{p,L}}{C_{p,G}})] \tag{19}$$

$$\varepsilon=\frac{G}{L}\frac{C_{p,G}}{C_{p,L}}\frac{T_{G,e}-T_{G,i}}{T_{L,i}-T_{G,i}}=\frac{T_{L,i}-T_{L,e}}{T_{L,i}-T_{G,i}}=\frac{R}{ITD} \tag{20}$$

Case II: $\frac{L}{G}>\frac{C_{p,G}}{C_{p,L}}$

$$NTU = -\frac{L}{G}\frac{C_{p,L}}{C_{p,G}}\ln[1+\frac{G}{L}\frac{C_{p,G}}{C_{p,L}}\ln(1-\varepsilon)] \tag{21}$$

$$\varepsilon = \frac{L}{G}\frac{C_{p,L}}{C_{p,G}}\frac{T_{L,i}-T_{L,e}}{T_{L,i}-T_{G,i}} = \frac{L}{G}\frac{C_{p,L}}{C_{p,G}}\frac{R}{ITD} \tag{22}$$

This is an air-cooled heat exchanger demand curve. The intersection of the supply curve and the demand curve is its operating point. The solution of simulation calculations using this method for evaluation of an air-cooled heat exchanger at various conditions is given by Asvapoositkul and Kuansathan [8]. A previous study [8] has shown that the thermal performance capacity of air-cooled heat exchanger is a strong function of mass flow rate ratio, and system inlet temperatures. Therefore, this study looks at the same parameters but also closely considers the entropy generation under variations in operating parameters.

4. Performance Predictions

In this analysis, an air-cooled heat exchanger is designed to cool 136 lpm from 50 °C to 48 °C (R= 2 °C) at a dry-bulb temperature of 30 °C (A = 18 °C) with an effectiveness of 70.71% ($\frac{L}{G}$ = 1.7 and $T_{G,e}$ = 44.13 °C). It transfers all heat by forced convection. The following case study illustrates the use of energy and entropy generation analysis in the determination of the heat exchanger's operating characteristics.

The heat exchanger design conditions: $\frac{L}{G}$ = 1.7, $T_{L,i}$ = 50 °C, $T_{G,i}$ = 30 °C, R = 2 °C

Case 1: Changes in the ambient dry-bulb temperature $T_{G,i}$ with constant $T_{L,i}$ and $\frac{L}{G}$

$\frac{L}{G}$ = 1.7 or ε = constant

$T_{L,i} = 50 \pm 3$ °C

$T_{G,i}$ = vary from 20 – 50 °C

In this case the heat exchanger effectiveness (ε) or $\frac{L}{G}$ is constant at all operating conditions for the system being evaluated. The value of R is determined from Eq. (17) with the given $T_{L,i}$, $T_{G,i}$, and ε or $\frac{L}{G}$. The outlet temperatures ($T_{L,e}$, $T_{G,e}$) under the condition of constant effectiveness ε and $\frac{L}{G}$ are then determined. From Eq. (17) as $T_{G,i}$ decreases, both ITD and R will increase. Therefore Q will be increased in this case (e.g. $Q \propto R \propto ITD$).

These and the following performance curves were taken from data calculated by the governing equations using the individual design parameters. The heat load Q_{design} at the design conditions is used as a basis. Figure 1 compares heat rejection capabilities ($\frac{Q}{Q_{design}}$) of the various ambient temperatures for fixed values of $\frac{L}{G}$, ε , and $T_{L,i}$. The heat load is transferred by convection which decreases linearly to zero as $T_{G,i} = T_{L,i}$. The heat exchanger could transfer heat at a rate Q under one set of ambient conditions. Its performance curve sensitive to ambient temperature. Thus the heat rejection rate of $T_{L,i}$ = 50 °C at $T_{G,i}$ = 33 °C, ITD = 17 °C, would be 0.85 of that at $T_{G,i}$ = 30 °C, ITD = 20 °C, etc. At higher ambient temperatures the heat exchanger is unable to carry the entire heat load alone (e.g. $\frac{Q}{Q_{design}} < 1$). It should be noted that the value of ITD is constant for $\frac{Q}{Q_{design}}$ = constant, regardless of $T_{L,i}$. For instance ITD would be 20 °C in all inlet water temperatures at $\frac{Q}{Q_{design}} = 1$. The results are shown in Figure 2, where $\frac{Q}{Q_{design}}$ data for an air-cooled heat exchanger has been plotted against ITD at various operating conditions. This is exactly the same data as plotted in Figure 1. The relationship of the two parameters ($\frac{Q}{Q_{design}}$ and ITD) reduces the amount of data scatter. Both are functions of $\frac{L}{G}$, $T_{L,i}$, R, and $T_{G,i}$. The heat transfer rate under these conditions is linearly dependent on ITD. Its capacity increases with ITD (see Figure 2). A low ITD means either $T_{L,i}$ is decreased or $T_{G,i}$ is increased. A high ITD means either $T_{L,i}$ is increased or $T_{G,i}$ is decreased. The effect of $\frac{L}{G}$ on the relationship of the heat rejection capabilities ($\frac{Q}{Q_{design}}$) and the ITD is also shown in Figure 2. In this case $G = G_{design}$ + 0.58 kg/s, - 0.56 kg/s and L is constant (L = 2.28 kg/s). The high air mass flow rate decreases $\frac{L}{G}$ to 1.2 and the low air mass flow rate increases $\frac{L}{G}$ to 3. It can be seen that $\frac{Q}{Q_{design}}$ increases with G. However it has less affected as the ITD is reduced. These relationships will later be discussed in detail.

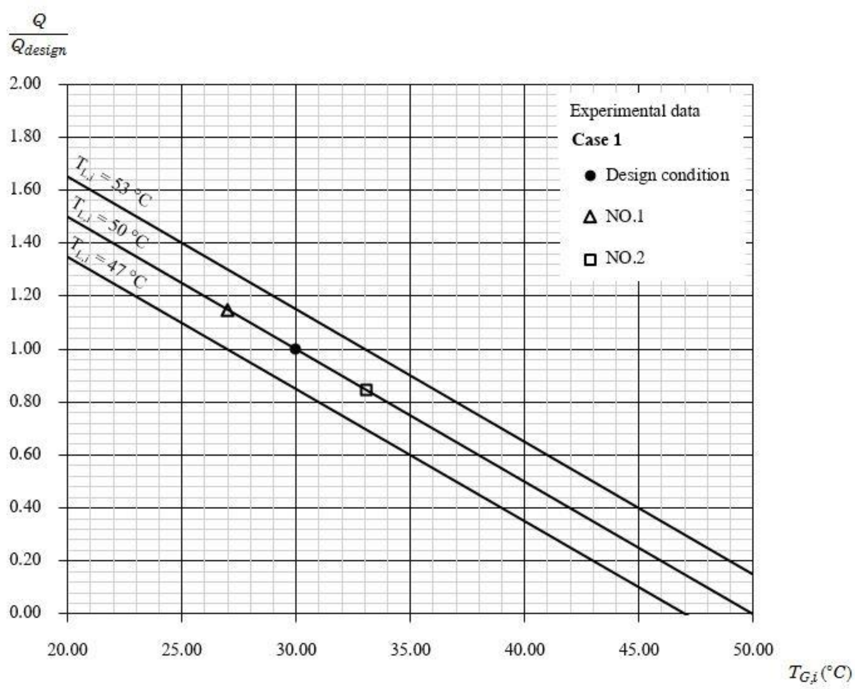

Figure 1 The heat rejection capabilities ($\frac{Q}{Q_{design}}$) of the various ambient temperatures for fixed values of $\frac{L}{G}$ = 1.7 or ε = 71% and $T_{L,i}$ = 50 °C

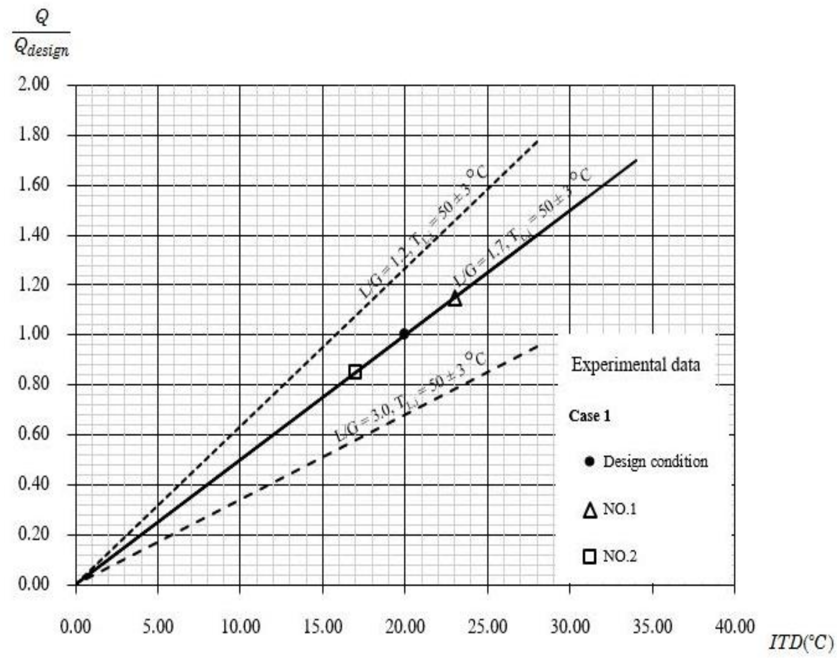

Figure 2 The relationship of the heat rejection capabilities ($\frac{Q}{Q_{design}}$) and the ITD for fixed values of $\frac{L}{G}$ = 1.7 or ε = 71% and $T_{L,i}$ = 50 °C

The air outlet temperature can be obtained from the definition of heat exchanger effectiveness. Then the values of non-dimensional entropy generation or entropy generation number (s^*_{gen}) can be calculated. Figure 3 illustrates the effect that the ITD has on entropy generation number. It can be observed that the entropy generation number increases with increasing ITD. The minimum entropy generation is achieved at $ITD = 0$ or $T_{G,i} = T_{L,i}$. It can also be seen that s^*_{gen} slightly increases with $\frac{L}{G}$ and the effecte is less as the ITD is reduced.

Case 2: Changes in the ambient dry-bulb temperature $T_{G,i}$ with constant $T_{L,i}$ and R

$\frac{L}{G}$ = vary or ε = vary

$T_{L,i}$ = 50 ± 3 °C

$T_{G,i}$ = vary from 20 – 50 °C

R = 2.0 or Q (heat load) = constant

In this case the heat exchanger is to handle the fixed heat load at all operating conditions. The given heat load is a function of $C_{p,L}$, R and L . The heat exchanger effectiveness (ε) is a function of $\frac{L}{G}$ according to Eq. (18). The value of ε or $\frac{L}{G}$ is determined from Eq. (17) with the given $T_{L,i}$, $T_{G,i}$, and R. Once the value of $\frac{L}{G}$ is known, the outlet temperatures ($T_{L,e}$, $T_{G,e}$) and required G can be determined. The heat load is held constant at Q_{design} and the required air mass flow rate (G_{design}) at this design condition is used as a basis. In figure 4, the value of $\frac{G}{G_{design}}$ for the heat exchanger has been plotted against ITD at various ambient temperatures $T_{G,i}$ and inlet water temperatures $T_{L,i}$. The relationship of the two parameters reduces data into a single curve. This means that when the ITD is reduced (i.e. due to the increase of ambient temperature), the required air mass flow rate must be increased to maintain the same level of heat rejection. The required air flow rate is increased until it reaches the temperature for which the tower was sized. Point T_D on the curve is the design point with $\frac{G}{G_{design}} = 1$ and ITD_{design} = 20 °C (R = 2 °C, A = 18 °C). It is seen that the quantity of air necessary increases rapidly as ITD decreases. On the other hand, it decreases slowly when ITD is high. It is of advantage, to keep to a moderate figure, and especially not to decrease below a certain value for ITD. It should be noted that air mass flow is proportional to fan power consumption and inversely proportional to $\frac{L}{G}$. Point T_L on the curve represents ITD = 18 °C (R = 2 °C, A = 16 °C) and $\frac{G}{G_{design}}$ = 1.16. This point will be discussed in the following.

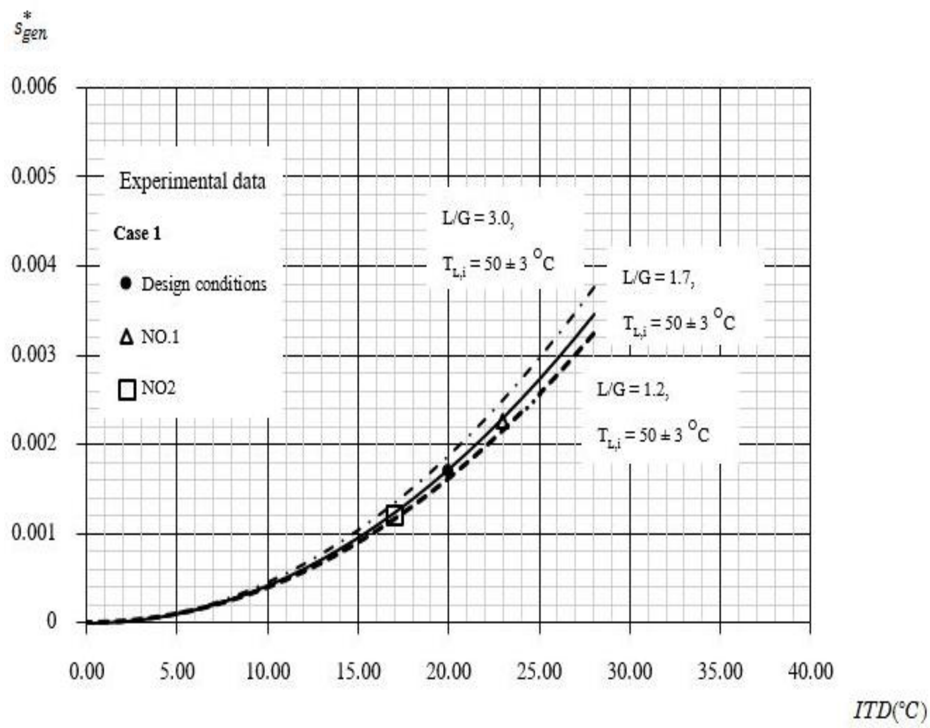

Figure 3 The effect of ITD on entropy generation number (s^*_{gen}) for fixed values of $\frac{L}{G}$ or ε

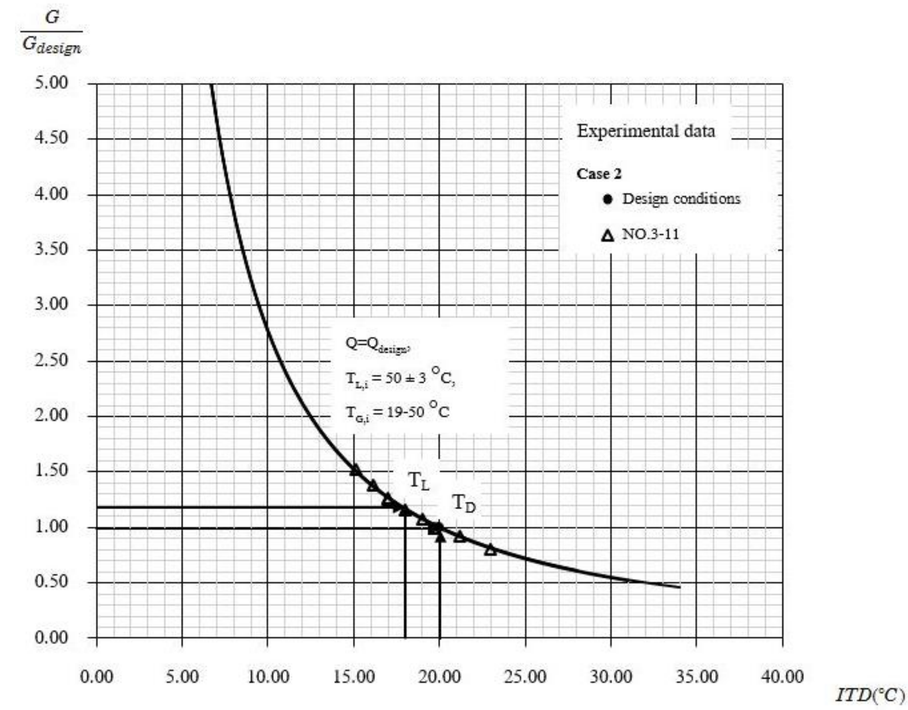

Figure 4 The relationship of ($\frac{G}{G_{design}}$) and the ITD for fixed values of heat load

The calculated outlet air temperature at these various operating conditions was also shown in Figure 5. The outlet air temperatures decreased and increased as the ITD is reduced as shown in Figure 5. These curves also showed minimum outlet air temperatures where ITD = 18 °C (R = 2 °C, A = 16 °C) in all inlet water temperatures at Q_{design} . The relationship among the temperatures is

$$T_{G,e} = T_{G,i} + R\frac{C_{p,L}}{C_{p,G}}\frac{L}{G} = T_{L,i} - ITD + R\frac{C_{p,L}}{C_{p,G}}e^{\left\{\frac{\ln\left(\frac{C_{p,L}}{C_{p,G}}\frac{R}{ITD}\frac{1}{c}\right)}{n-1}\right\}} \tag{23}$$

By setting $\frac{dT_{G,e}}{dITD} = 0$, the value of $T_{G,e}$ is minimum, and we get

$$ITD_{\min} = \frac{R}{1-n}\frac{C_{p,L}}{C_{p,G}}e^{\left\{\frac{\ln\left(\frac{C_{p,L}}{C_{p,G}}\frac{R}{ITD_{\min}}\frac{1}{c}\right)}{n-1}\right\}} \tag{24}$$

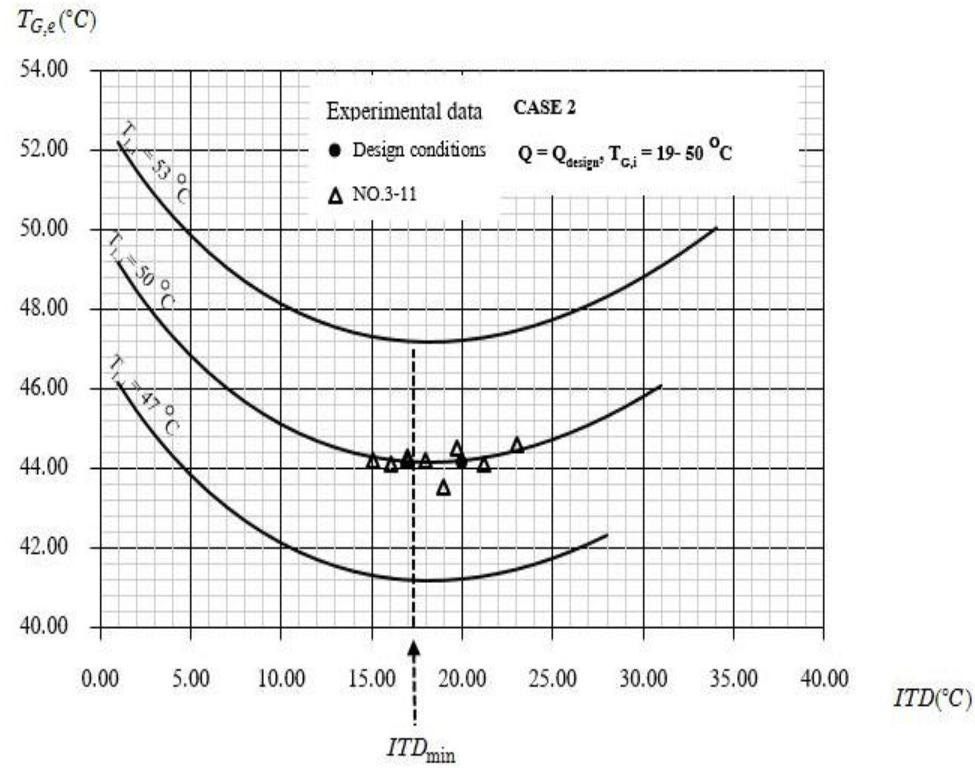

Figure 5 The relationship of the outlet air temperatures $T_{G,e}$ and the ITD for fixed values of heat load

The minimum outlet air temperature is obtained by setting $ITD = ITD_{\min}$ in the equation (23). Using Eq (24) with R = 2 °C, we get $ITD_{\min}$ = 18 °C and $\frac{G}{G_{design}}$ = 1.16. This is point T_L on the curve of Figure 4. Below this point, the necessary air mass flow increases rapidly as the ITD is reduced.

Figure 6 shows the effect that the ITD has on s^*_{gen} for the same set of operation conditions. The values of s^*_{gen} obtained from cases 1 (Figure 3) and that from case 2 (Figure 6) are similar with different corresponding values of ITD.

5. Model validation

The purpose of this work has been to test and analyze the dry-cooling tower previously developed to evaluate its performance based on entropy generation. The model tower tested for this report is essentially the same as described in Reference [8]. The tower performance has been measured under a variety of operating conditions. The results were compared to further validate the model presented here. The triangle symbols and others, as shown in Figs. 1 – 6 in each condition, were obtained from the experiments. Instrumentation and testing of the tower produced data with an average experimental error of 10%. In 12 test runs the predicted total heat transfer differed from the measured total heat transfer by no more than 6.5% with most runs coming well within 1%. With the model validity established, it may now be used to help predict the performance of the dry-cooling towers. The results summarized for both cases in Table 1.

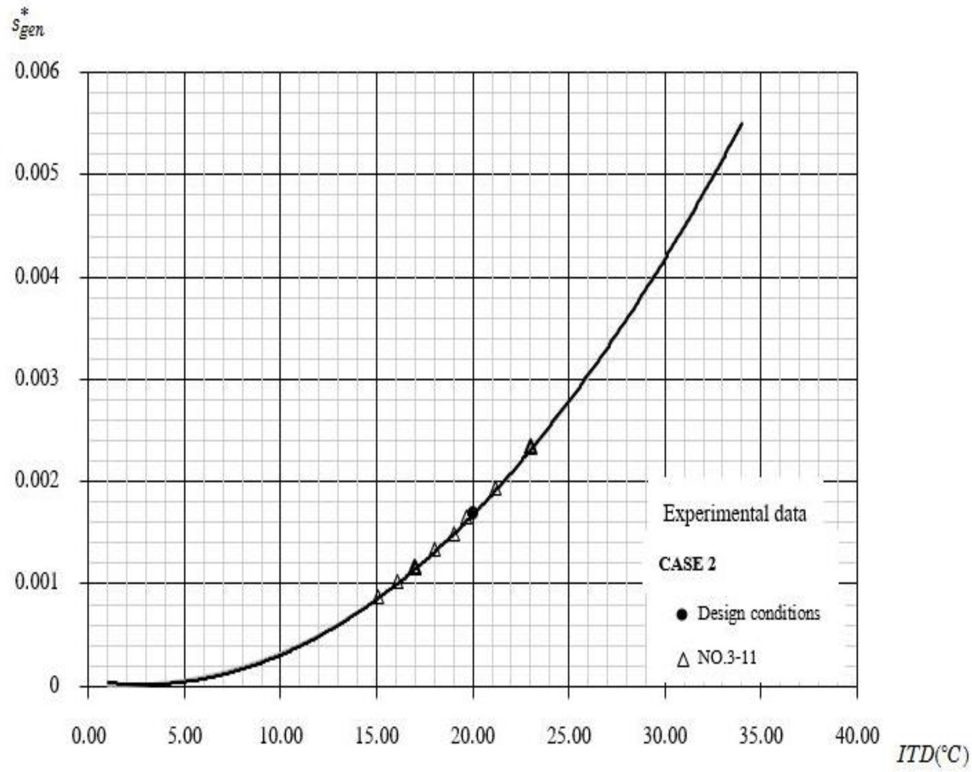

Figure 6 The effect of ITD on entropy generation number (s^*_{gen}) for fixed values of heat load

6. Result and discussion

The influence of different working conditions ($\frac{L}{G}$, $T_{L,i}$, $T_{G,i}$ and R) has been determined. The variations of $\frac{Q}{Q_{design}}$ and s^*_{gen} for various conditions among these parameters are calculated. The results reveal that the amount of data scatter is reduced when plotted against ITD at various operating conditions. In case 1 with constant ε or $\frac{L}{G}$, it can be observed from Figure 2 and Figure 3 that the calculated $\frac{Q}{Q_{design}}$ and s^*_{gen} are increased with increasing ITD. When $\frac{L}{G}$ is reduced by either L is decreased or G is increased with constant ITD, the calculated $\frac{Q}{Q_{design}}$ is increased and that of s^*_{gen} is decreased (see Figure 2 and Figure 3). However it has less affected as the ITD is reduced. We also see that s^*_{gen} slightly changes with $\frac{L}{G}$.

In case 2 with constant R or Q, it can be observed from Figure 4 that the required $\frac{G}{G_{design}}$ is increased with decreasing ITD. At low ITD (i.e. due to the increase of ambient temperature), the approach temperature would be reduced to maintain the range temperature. In practice, air-cooled heat exchanger are designed with an approach of 10 to 15 °C [9]. Below this value, small changes in ITD yield large changes in $\frac{G}{G_{design}}$. The outlet air temperatures would be allowed to "float" with ambient conditions, rising or falling until ITD closes to $ITD_{\min}$. The quantity of s^*_{gen} is increased with increasing ITD.

Table 1 Comparison between the corresponding predicted and measured values of the dry-cooling tower.

Conditions	**Design conditions**	**Case 1**		**Case 2**								
Description	**Working at design conditions**	**Varying ambient condition at temperature of ± 3 $°C$ of design $T_{G,i}$ with constant $\frac{L}{G}$**		**Varying ambient condition at temperature of ± 3 $°C$ of design $T_{G,i}$ with constant R**								
Experimental conditions		NO.1	NO.2	NO.3	NO.4	NO.5	NO.6	NO.7	NO.8	NO.9	NO.10	NO.11
$T_{L,i}\,(°C)$	50	50	50	50	50	50	50	50	50	50	50	50
$T_{G,i}\,(°C)$	30	27	33	27	33	28.8	30.3	31.0	32.0	33.0	33.9	34.9
$R(°C)$	2	2.3	1.7	2	2	2	2	1.9	2	2	2	2
$ITD(°C)$	20	23	17	23	17	21.2	19.7	19.0	18.0	17.0	16.1	15.1
$\frac{L}{G}$	1.7	1.7	1.7	2.12	1.36	1.84	1.71	1.59	1.4700	1.35	1.23	1.12
$\frac{G}{G_{design}}$	1.000	1.000	1.000	0.802	1.250	0.924	0.994	1.069	1.156	1.259	1.382	1.518
Effectiveness	70.71	70.71	70.71	76.67	66.55	72.93	70.71	69.60	67.87	66.05	64.11	62.22
Experimental results												
$T_{G,e}\,(°C)$	44.13	43.25	45.01	44.62	44.30	44.09	44.51	43.55	44.21	44.22	44.12	44.21
$\frac{Q}{Q_{design}}$	1.00	1.15	0.85	1.00	1.00	1.00	1.00	0.95	1.00	1.00	1.00	1.00
s^*_{gen}	0.001695	0.002259	0.001216	0.002350	0.001168	0.001935	0.001655	0.001483	0.001332	0.001164	0.001018	0.000875
$\frac{s^*_{gen}}{s^*_{gen,design}}$	1.000000	1.332443	0.717149	1.386463	0.689214	1.141323	0.976201	0.874879	0.785717	0.686942	0.600648	0.516116
Equation predicted results												
$T_{G,e}\,(°C)$	44.13	43.35	45.09	44.47	44.20	44.29	44.21	44.19	44.18	44.20	44.24	44.31
$\frac{Q}{Q_{design}}$	1.00	1.16	0.86	1.01	1.01	1.01	1.01	1.01	1.01	1.01	1.01	1.01

Table 1 Comparison between the corresponding predicted and measured values of the dry-cooling tower. (Continuous)

Conditions	Design conditions	Case 1		Case 2								
Description	Working at design conditions	Varying ambient condition at temperature of ± 3 $°C$ of design $T_{G,i}$ with constant $\frac{L}{G}$		Varying ambient condition at temperature of ± 3 $°C$ of design $T_{G,i}$ with constant R								
		NO.1	NO.2	NO.3	NO.4	NO.5	NO.6	NO.7	NO.8	NO.9	NO.10	NO.11
s^*_{gen}	0.001695	0.002263	0.001218	0.002342	0.001163	0.001943	0.001640	0.001508	0.001329	0.001163	0.001023	0.000879
$\frac{s^*_{gen}}{s^*_{gen,design}}$	1.000000	1.335190	0.718639	1.381905	0.685939	1.146355	0.967592	0.889624	0.784253	0.685939	0.603468	0.518478
Errors in predicted values												
% in error of $T_{G,e}$ (%)	0.00	-0.23	-0.18	0.34	0.23	-0.45	0.67	-1.47	0.07	0.05	-0.27	-0.23
% in error of $\frac{Q}{Q_{design}}$ (%)	0.00	-0.87	-1.18	-1.00	-1.00	-1.00	-1.00	-6.32	-1.00	-1.00	-1.00	-1.00
% in error of s^*_{gen} (%)	0.00	-0.18	-0.16	0.34	0.43	-0.41	0.91	-1.69	0.23	0.09	-0.49	-0.46
% in error of $\frac{s^*_{gen}}{s^*_{gen,design}}$ (%)	0.00	-0.21	-0.21	0.33	0.48	-0.44	0.88	-1.69	0.19	0.15	-0.47	-0.46

7. Conclusion

The performance of the heat exchanger was calculated under a wide range of operating conditions. The performance has been evaluated based on thermal capacity as well as on entropy generation. Comparison of the model tower test results with those of a computer simulation has demonstrated the validity of that simulation and its use as a design tool. The correlation of data for various conditions among the operating conditions shows that ITD could be used as the parameter to determine the effect on thermal capacity and entropy generation. The DRY tower is sensitive to ambient temperature and requires moderate ITD. The variation in an ambient temperature with constant $\frac{L}{G}$ results in heat rejection changes. A larger heat rejection with increasing ITD yields a higher entropy generation than the case of lesser heat rejection with decreasing ITD. Consequently, what is good for increasing the heat transfer is bad for entropy generation, and vice versa. The variation in an ambient temperature with constant heat rejection results in $\frac{L}{G}$ changes. The required air mass flow rate G relates to ITD. A lesser G with increasing ITD yields a higher entropy generation than the case of larger G with decreasing ITD. Since the quantity of s^*_{gen} is increased with increasing ITD and decreasing ITD yield large change in G. The operating conditions for the heat exchanger will be limited when ITD approach $ITD_{\min}$ where low irreversibility is expected. Working below this value results in over-sensitivity to small change.

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Nomenclature

C	constant
$C_{p,L}$	specific heat of water (constant pressure) (kJ/kg-K)
$C_{p,G}$	specific heat of air (constant pressure) (kJ/kg-K)
G	air mass flow rate (kg/s)
h	enthalpy (kJ/kg)
ITD	initial temperature difference (°C) (= $T_{L,i} - T_{G,i}$)
L	water mass flow rate (kg/s)
n	constant
NTU	number of transfer units
P	pressure (kPa)
Q	heat transfer rate (kW)
R	range (°C) (= $T_{L,i} - T_{L,e}$)
R_G	gas constant of air (kJ/kg-K)
s	entropy (kJ/kg-K)
$\dot{s}_{gen}$	rate of entropy generation (kJ/s-K)
s^*_{gen}	entropy generation number (= $\frac{S_{gen}}{G\ C_{p,G}}$)
T	temperature (°C)

Greek symbol

ε	effectiveness
ϕ	relative humidity
$\chi_{destroy}$	exergy destruction due to entropy generation (kW)
ψ	specific exergy (kJ/kg)
ω	humidity ratio (kg of water vapor / kg of dry air)
ϖ	specific humidity ratio on a mole basis (= 1.608 ω)

Subscripts

0	environment, restricted dead state
approx.	approximated value
e	exit or outlet
G	air
i	inlet
L	water
min	minimum