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Technical Note

A comparison between the entropy generation in terms of thermal conductance and generalized thermal resistance in heat exchanger analyses

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ABSTRACT

This paper reports the comparison on the applicability between the concepts of entropy generation in terms of thermal conductance and generalized thermal resistance to analyzing heat exchangers. Six typical two-stream heat exchangers and one three-stream heat exchanger are analyzed. When the heat capacity flow rates of the discussed heat exchangers are fixed, the minimum generalized thermal resistance always corresponds to the maximum heat exchanger effectiveness, while the minimum entropy generation in terms of thermal conductance does not always. When the heat capacity flow rates of the discussed heat exchangers are not fixed, smaller generalized thermal resistance always leads to larger heat transfer rate, while smaller entropy generation in terms of thermal conductance does not always. Therefore, the concept of generalized thermal resistance is more appropriate than the concept of entropy generation in terms of thermal conductance for describing the performance of the discussed heat exchangers.

1. Introduction

Heat exchangers are widely used in industry, and their analyses are very important for the improvement of heat exchanger performance and energy utilization efficiency [1–5]. In the past decades, some theories have been developed for the heat exchanger analyses, such as the entropy generation minimization [6–12] and the minimum generalized thermal resistance principle [2,13].

From the thermodynamic viewpoint, the best heat exchanger performance was related to the entropy generation minimization [4–12,14,15]. However, it was found that the entropy generation rate and entropy generation number do not always decrease with the performance improvement of the heat exchangers [2,4,16]. To solve this problem, some modified normalized parameters were proposed, such as the revised entropy generation number [14,15,17] and entropy generation in terms of heat conducting capacity [18].

The minimum generalized thermal resistance principle was proposed from the viewpoint of the entransy theory, which has been used to analyze many heat transfer problems [2,13,17,19–25]. When generalized thermal resistance was applied to analyzing heat exchangers, it was always found that smaller thermal resistance results in better heat exchanger performance [2,13,17].

In the past several years, the applicability of the entropy generation minimization and the minimum generalized thermal resistance principle to the analyses of heat exchangers was compared [2,17], and the results showed that the revised entropy generation number does not always decrease with increasing heat transfer rate of heat exchangers, either. However, there is still no report on the comparison between the generalized thermal resistance and the entropy generation in terms of heat conducting capacity. In this paper, we focus on this topic.

2. Definitions of the discussed parameters

The entropy generation in terms of heat conducting capacity was defined as [18]

$$\Gamma = S_g/U, \tag{1}$$

where S_g is the entropy generation rate, and U is the thermal conductance of the heat exchanger. As U is the thermal conductance of the heat exchanger, we call Γ the entropy generation in terms of thermal conductance in this paper. For a two-stream balanced counter flow heat exchanger, when the heat exchanger

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effectiveness $\varepsilon \ll 1$ and inlet temperatures of the streams are fixed, both the theoretical analyses and numerical results showed that Γ decreases monotonically with increasing ε [18].

On the other hand, the concept of entransy was also introduced [19]. For an object with internal energy E and temperature T, its entransy is

$$G = ET/2 = cmT^2/2, \tag{2}$$

where c is the specific heat capacity, and m is the mass. With this concept, it is found that entransy dissipation always exists in practical heat transfer. Based on the concept of entransy dissipation, the concept of generalized thermal resistance was defined as [2,13,17]

$$R = G_{dis}/Q^2, \tag{3}$$

where G_{dis} is the entransy dissipation rate, and Q is the heat transfer rate. For two-stream heat exchangers as shown in Fig. 1 [2,13,17], their expressions are

$$G_{dis} = (C_H T_{H\text{-}in}^2 + C_L T_{L\text{-}in}^2)/2 - (C_H T_{H\text{-}out}^2 + C_L T_{L\text{-}out}^2)/2, \tag{4}$$

$$Q = C_H(T_{H\text{-}in} - T_{H\text{-}out}) = C_L(T_{L\text{-}out} - T_{L\text{-}in}), \tag{5}$$

where C_H, $T_{H\text{-}in}$ and $T_{H\text{-}out}$ are the heat capacity flow rate, inlet and outlet temperatures of the hot stream, while C_L, $T_{L\text{-}in}$ and $T_{L\text{-}out}$ are those of the cold stream, respectively.

3. Discussions on the analyses of heat exchangers

As shown in Table 1, we discuss six two-stream heat exchangers below. The relations between ε and NTU are also listed in Table 1 [25,26], where ε is the heat exchanger effectiveness, ε_t is the effectiveness of the TEMA E-type shell-and-tube heat exchanger, and C^* is

$$C^* = C_{min}/C_{max}, \tag{6}$$

where C_{min} and C_{max} are the minimum and maximum heat capacity flow rates of the streams, respectively, and NTU can be calculated by

$$NTU = U/C_{min}. \tag{7}$$

Therefore, if the inlet temperatures and heat capacity flow rates of the streams are given, the heat transfer rate can be obtained,

$$Q = \varepsilon Q_{max} = \varepsilon C_{min}(T_{H\text{-}in} - T_{L\text{-}in}), \tag{8}$$

where Q_{max} is the maximum possible heat transfer rate. Then, the outlet temperatures of the streams can be obtained with Eqs. (5) and (8), the entropy generation rate can be got [2,17],

$$S_g = C_H \ln(T_{H\text{-}out}/T_{H\text{-}in}) + C_L \ln(T_{L\text{-}out}/T_{L\text{-}in}), \tag{9}$$

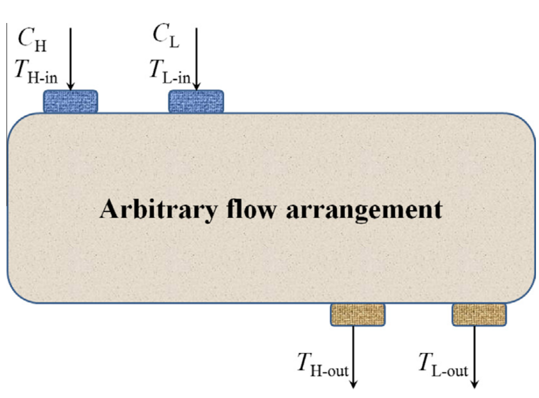

Fig. 1. Sketch of a two-stream heat exchanger [2,13].

and the entransy dissipation rate can be calculated by Eq. (4). Then, the entropy generation in terms of thermal conductance and the generalized thermal resistance can be obtained with Eqs. (1) and (3), respectively.

As below, we discuss some numerical examples. First, assume that C_H = 5 W/K, C_L = 3 W/K, $T_{H\text{-}in}$ = 360 K and $T_{L\text{-}in}$ = 300 K. For the parallel flow heat exchanger, the variations of ε, Γ and R with NTU are shown in Fig. 2. The numerical results show that the variation tendencies of ε, Γ and R for the other five heat exchangers in Table 1 are the same as those in Fig. 2, so the figures for the other five heat exchangers are not presented. It can be seen that both Γ and R decrease with increasing ε. Therefore, both Γ and R are appropriate for the analyses of this case.

Furthermore, we can discuss the influence from the fluid arrangement. The values of ε, Γ and R for different flow arrangements are shown in Table 2. When NTU = 1, R decreases with increasing ε, while Γ does not. The best arrangement is the sixth arrangement in Table 1, but Γ gets to an intermediate value for this arrangement. When NTU = 2, Γ and R both decrease with increasing ε. The best arrangement is the fourth one, and Γ gets to its minimum value. Γ does not always decrease with the performance improvement of the heat exchanger, while R does. Therefore, compared with Γ, R is more appropriate for the analyses of the fluid arrangement.

As below, the relation between R and ε can be derived. With Eqs. (3)–(5), we can get that [13]

$$R = -(1/C_H + 1/C_L)/2 + (T_{H\text{-}in} - T_{L\text{-}in})/Q. \tag{10}$$

Considering Eq. (8) leads to

$$R = -(1/C_H + 1/C_L)/2 + 1/(C_{min}\varepsilon). \tag{11}$$

When the heat capacity flow rates of the streams are given, Eq. (11) shows that smaller R always corresponds to larger ε. In the numerical cases above, the heat capacity flow rates are fixed, so R decreases monotonically with increasing ε. Eq. (11) explains the variation tendencies of R and ε for the results in Fig. 2.

Furthermore, let us discuss the relation between Γ and ε. When the heat capacity flow rates of the stream are given, considering Eqs. (1) and (7) leads to

$$\frac{d\Gamma}{d\varepsilon} = \frac{d}{d\varepsilon}\left(\frac{S_g}{C_{min}NTU}\right) = \frac{1}{C_{min}NTU}\left[\frac{dS_g}{d\varepsilon} - \frac{S_g}{NTU}\frac{d(NTU)}{d\varepsilon}\right], \tag{12}$$

which shows that $\frac{d\Gamma}{d\varepsilon}$ is determined by $\frac{dS_g}{d\varepsilon}$ and $\frac{d(NTU)}{d\varepsilon}$ because C_{min}, NTU and S_g are always positive for any heat exchanger. In Eq. (12), we have [27]

$$\frac{dS_g}{d\varepsilon} = -C_H \frac{C_{min}(T_{H\text{-}in} - T_{L\text{-}in})}{C_H T_{H\text{-}in} - C_{min}(T_{H\text{-}in} - T_{L\text{-}in})\varepsilon} + C_L \times \frac{C_{min}(T_{H\text{-}in} - T_{L\text{-}in})}{C_L T_{L\text{-}in} + C_{min}(T_{H\text{-}in} - T_{L\text{-}in})\varepsilon}. \tag{13}$$

Cheng and Liang [17] proved that Eq. (13) equals zero when $\varepsilon = \varepsilon_0 = 1/(1 + C^*)$. If $\varepsilon < \varepsilon_0$, Eq. (13) is positive, while it is negative for $\varepsilon > \varepsilon_0$. Therefore, sometimes Eq. (13) is positive, while it is negative in the other cases. On the other hand, for the case in which the flow arrangement of the heat exchanger is fixed, the relations in Table 1 show that ε increases with increasing NTU for the discussed six heat exchangers, which means that $\frac{d(NTU)}{d\varepsilon}$ is always positive. Therefore, the second term (with the negative sign) in the square bracket of Eq. (12) should be negative, which may make the whole equation be negative. This is the mathematical reason why Γ decreases with increasing ε for the results in Fig. 2 and the other five heat exchangers in Table 1.

For the numerical case with which we discuss the influence from the flow arrangement, NTU does not change, so $\frac{d(NTU)}{d\varepsilon}$ is zero and the value of $\frac{d\Gamma}{d\varepsilon}$ is determined by $\frac{dS_g}{d\varepsilon}$. For the discussed case,