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Performance evaluation of heat transfer enhancement in plate-fin heat exchangers with offset strip fins

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Abstract

Generally, the Offset Strip Fin (OSF) in a plate-fin heat exchanger provides a greater heat transfer coefficient than plain plate-fin, but it also leads to an increase in flow friction. A new parameter, called relative entropy generation distribution factor, Ψ^*, is proposed to evaluate the thermodynamic advantages of OSFs. This parameter presents a ratio of relative changes of entropy generation. The relative effects of the geometrical parameters α, γ and δ are discussed. The results show that there exist the optimum values of α and γ at a certain flow condition, which obviously maximize the degree of the heat transfer enhancement of OSFs.

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Keywords: plate-fin heat exchanger; offset strip fin; heat transfer enhancement; entropy generation, constant wall temperature

1. Introduction

Plate-fin heat exchangers, which are widely used in aerospace, automobile and gases industries, are characterized by plate-fin surfaces with large ratio of total heat transfer area to volume. Offset strip fin (OSF) is one of the enhanced plate-fin surfaces. Owning to the fin offset that induces the laminar boundary layer restarting as well as the from-drag, the heat transfer enhancement of OSF fin is substantial, and the pressure drop along the flow direction is also considerable. Therefore, the performance of the heat transfer enhancement of OSF fins should be evaluated in

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conjunction with the increased flow friction, which is important to the thermo-hydraulic design of a plate-fin heat exchanger with OSF fins.

To evaluate the heat transfer enhancement of OSFs, the second-law analysis that makes it possible to combine heat transfer and hydraulic characteristics appears to be more attractive. There have been two types of comparison methods for the second-law performance evaluation. One is the augmentation entropy generation number, $N_{s,a}$, proposed by Bejan and Pfister [1]. This method is characterized by the comparison of total entropy generation in an augmented channel and that in a reference channel, and has been extensively applied to the plate fin-channel by Tagliafico and Tanda[2] and various other enhanced devices by Chen and Huang [3], Zimparov [4] and Chakraborty [5]. Another is the entropy generation distribution factor, ψ, presented by Manglik and Fang [6]. It involves the comparison between the reduction of the entropy generation due to the heat transfer enhancement and the increase of the irreversibility due to the increase in pressure drop. Using this method, Manglik and Fang [6] has been evaluate the performance of the heat transfer enhancement of different OSFs with plain plate-fin as a reference fin-channel.

The present study aims at providing new experience and reference for the use of second-law performance evolution of the heat transfer enhancement in plate-fin heat exchangers with OSF fins. The characteristics of the evaluations for OSF fins given by the two types of second law methods are carefully investigated. Moreover, a new performance parameter denoted by Ψ^* is developed for the performance evaluation under different operating conditions. Finally, the geometrical effects on the performance evaluation are also delineated.

2. Entropy generation analysis

2.1. Physical model

Fig. 1 schematically shows the geometries of an OSF fin and a plain plate-fin with the same total length L. Their configurations are described by the fin height h, fin space s and fin thickness t. For OSF fins, the structure parameters also include the interrupted length l due to the fin offset. In addition, the fin thickness-to-height ratio α ($=t/h$), fin density γ ($=t/s$) and the fin thickness-to-length ratio δ ($=t/l$) are adopted here considering the comprehensive effect of the fin thickness.

2.2. Entropy generation

2.2.1. Equations

From Bejan [7], the rate of entropy generation per unit channel length for internal flow with constant wall temperature and thermo-physical properties is:

$$\frac{dS_{gen}}{dx} = -mC_p \frac{T-T_w}{TT_w}\frac{dT}{dx} + \frac{m}{\rho T}\left(-\frac{dp}{dx}\right), \tag{1}$$

where m, T, T_w, C_p, ρ and p represent mass flow rate, fluid temperature, wall temperature, specific heat at a constant pressure, density of fluid and local pressure, respectively.

The distribution of fluid temperature along the length of the fin channel is represented by [8]:

$$T = T_w + \left(T_0 - T_w\right)\exp\left(-\frac{j}{\mathrm{Pr}^{2/3}}\frac{4x}{D_h}\right), \tag{2}$$

where T_0, j, Pr and D_h are the inlet temperature, Colburn heat transfer factor, Prandtl number and hydraulic diameter of plate-fin channel, respectively.

With Eqs. (2) and (5), the entropy generation is integrated along the fin channel as:

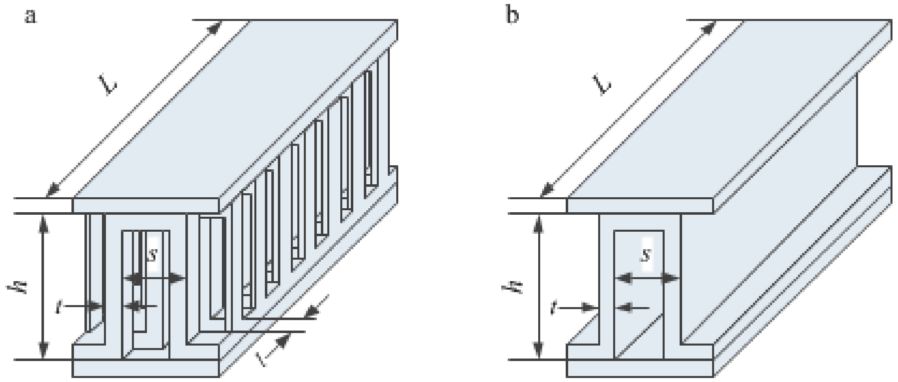

Fig. 1. (a) OSF fin; (b) plain plate-fin.

$$S_{gen} = mC_p \left\{ \tau_0 \varepsilon + \ln\left[\frac{1+\tau_0(1-\varepsilon)}{1+\tau_0}\right]\right\} + \frac{f}{2j}\frac{m^3 \mathrm{Pr}^{2/3}}{\rho^2 A_c^2 T_w}\ln\left[\frac{1+\tau_0(1-\varepsilon)}{(1+\tau_0)(1-\varepsilon)}\right], \tag{3}$$

where f is the Fanning friction factor, A_c is the free flow area. The relative temperature difference on the inlet, $\tau_0=(T_0- T_w)/T_w$, and the heat transfer effectiveness, ε=1-exp(-4·L·j/D_h/Pr$^{2/3}$).

The first term on the right-hand side of the Eq. (3) represents the irreversibility due to heat transfer across a finite temperature difference ($S_{gen,\Delta T}$), and the second term accounts for the entropy generation caused by friction loss ($S_{gen,\Delta p}$). Following Hesselgreaves [9], a non-dimensionless entropy generation can be given by:

$$N_{s1} = \frac{T_w S_{gen}}{Q} = \left[1 + \frac{1}{\tau_0\varepsilon}\ln\left(\frac{1+\tau_0(1-\varepsilon)}{1+\tau_0}\right)\right] + \frac{1}{\tau_0\varepsilon}\frac{f\,\mathrm{Re}^2}{2j}\frac{\mathrm{Pr}^{2/3}\mu^2}{\rho^2 D_h^2 C_p T_w}\ln\left(\frac{1+\tau_0(1-\varepsilon)}{(1+\tau_0)(1-\varepsilon)}\right), \tag{4}$$

where Q denotes the total heat transfer rate, μ is the dynamic viscosity. In addition, it is logical to denote the first and second terms on the right-hand side of the Eq. (4) by $N_{s1,\Delta T}$ and $N_{s1,\Delta p}$ respectively.

2.2.2. Thermal hydraulic performance

Different predictive equations for thermal hydraulic performance of OSFs in both laminar and turbulent flow have been reported by Manglik and Bergles [10], Min-Soo Kim et al. [11] and Yang and Li [12]. The correlations proposed by Yang and Li [12] is adopted here since they have been validated by comparing with the experimental data from Literatures [13-15] and are shown to provide more reliable predictions in a wider range.

As for plain plate-fin, the j and f factor can be obtained from Kays and London [13] or by CFD technique.

2.3. The conventional evaluation methods

To evaluate the merit of an augmentation technique, Bejan and Pfister [1] have introduced augmentation entropy generation number ($N_{s,a}$) expressed by:

$$N_{s,a} = \frac{S_{gen,a}}{S_{gen,o}} = \frac{S_{gen,a,\Delta T} + S_{gen,a,\Delta p}}{S_{gen,o,\Delta T} + S_{gen,o,\Delta p}}, \tag{5}$$

and Manglik and Fang [6] have developed entropy generation distribution factor (ψ) as:

$$\psi = \frac{S_{gen,o,\Delta T} - S_{gen,a,\Delta T}}{S_{gen,a,\Delta p} - S_{gen,o,\Delta p}}, \tag{6}$$

where the subscripts 'a' and 'o' refer to the augmented (OSF fin) and reference (plain plate-fin) channel, respectively. For the criteria, $N_{s,a}<1$ and $\psi>1$ are desirable.

Both the two conventional evaluation methods actually make sense only under the constraints of keeping heat transfer rate fixed since they are directly relevant to the absolute entropy generation. To investigate the characteristics of the evaluation methods more appropriately, the $N_{s,a}$ and ψ should be revised respectively as:

$$N_{s1}^{*} = \frac{N_{s1,a}}{N_{s1,o}} = \frac{N_{s1,a,\Delta T} + N_{s1,a,\Delta p}}{N_{s1,o,\Delta T} + N_{s1,o,\Delta p}} = \frac{N_{s1,\Delta T}^{*} + \phi_0 N_{s1,\Delta p}^{*}}{1+\phi_0}, \tag{7}$$

$$\Psi = \frac{N_{s1,o,\Delta T} - N_{s1,a,\Delta T}}{N_{s1,a,\Delta p} - N_{s1,o,\Delta p}} = \frac{1}{\phi_0}\frac{1-N_{s1,\Delta T}^{*}}{N_{s1,\Delta p}^{*}-1}, \tag{8}$$

where the irreversibility distribution ratio $\phi_0=N_{s1,o,\Delta p}/N_{s1,o,\Delta T}$, $N^{*}_{s1,\Delta T}=N_{s1,a,\Delta T}/N_{s1,o,\Delta T}$ and $N^{*}_{s1,\Delta p}=N_{s1,a,\Delta p}/N_{s1,o,\Delta p}$. Here, the criterions of $N^{*}_{s1}<1$ and $\Psi>1$ are still applied to the performance evaluation.

3. Results and discussion

3.1. Performance Evaluation with the Conventional Methods

The OSF fin with α=0.017, γ=0.089, δ=0.034, which shares the same cross section as the plain plate-fin 11.11(a) [13], is studied here. From Fig. 2, it is seen that the small N^{*}_{s1} and large Ψ results are obtained when the heat flux is large. The variations of N^{*}_{s1} and Ψ for relative temperature difference and Reynolds number shown in Fig. 3 further indicate that it is more advantageous to use OSF fins with larger temperature difference. In Fig. 4, moreover, small ϕ_0 contributes to small N^{*}_{s1} and large Ψ. This suggests that the evaluation criteria advocate the operating conditions with high irreversibility of heat transfer for OSF fins, since small ϕ_0 is the result of relatively large $N_{s1,o,\Delta T}$ from its definition.

From the above analysis, it is concluded that the results of the performance evaluation based on either the augmentation entropy generation number or the entropy generation distribution factor suggest promoting the use of augmented channel in the condition of heat transfer with a large temperature difference, which is evidentially preposterous in thermodynamics. According to these evaluation methods, the irreversibility of heat transfer which should have been reduced as much as possible has to be increased when conducting the trade-off between the thermodynamic benefits and penalties for augmented techniques.

3.2. A new evaluation method

To weigh the thermodynamic advantage and disadvantage of augmented passage structures in plate-fin heat exchanger more comprehensively, a relative comparison method is employed, and a new parameter, called relative entropy generation distribution factor (Ψ^{*}), is developed as:

$$\Psi^{*} = \frac{\left(N_{s1,o,\Delta T} - N_{s1,a,\Delta T}\right)/N_{s1,o,\Delta T}}{\left(N_{s1,a,\Delta p} - N_{s1,o,\Delta p}\right)/N_{s1,o,\Delta p}} = \frac{1-N_{s1,\Delta T}^{*}}{N_{s1,\Delta p}^{*}-1}. \tag{9}$$

It is physically a ratio of relative reduction of entropy generation number due to the enhanced heat transfer and the relative increase in irreversibility due to the higher friction loss. The high magnitude of Ψ^{*} represents the high degree of the heat transfer enhancement of OSF fin.

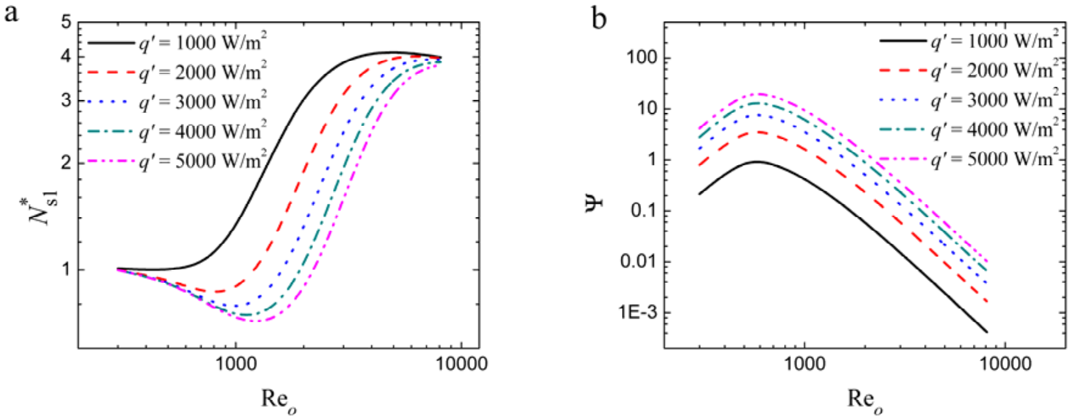

Fig. 2. (a) N^*_{s1} vs. Re$_o$ for q'; (b) Ψ vs. Re$_o$ for q'.

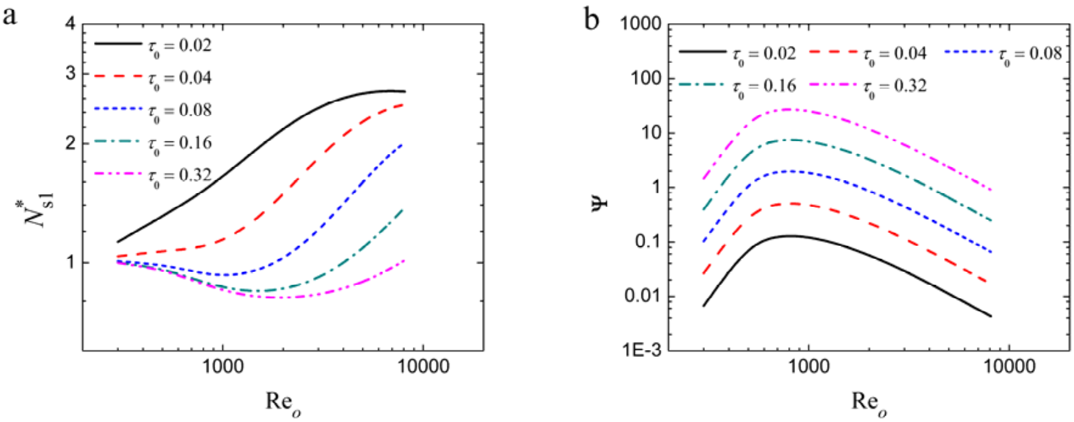

Fig. 3. (a) N^*_{s1} vs. Re$_o$ for τ_0; (b) Ψ vs. Re$_o$ for τ_0.

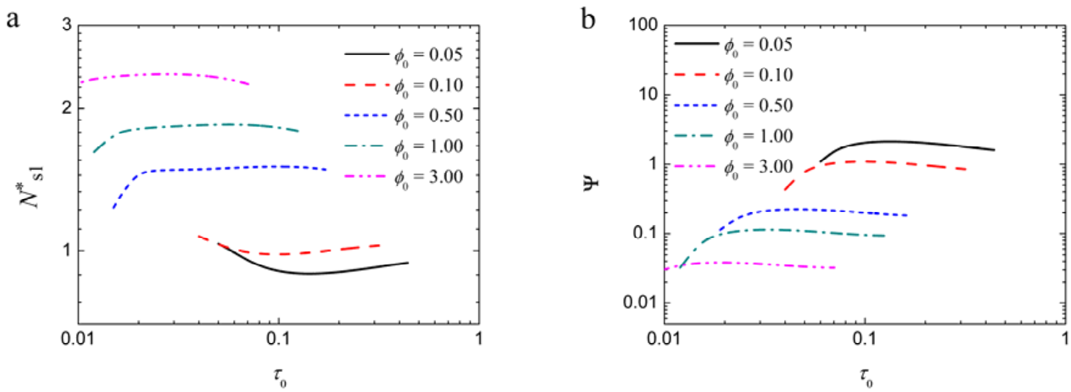

Fig. 4. (a) N^*_{s1} vs. τ_0 for ϕ_0; (b) Ψ vs. τ_0 for ϕ_0.

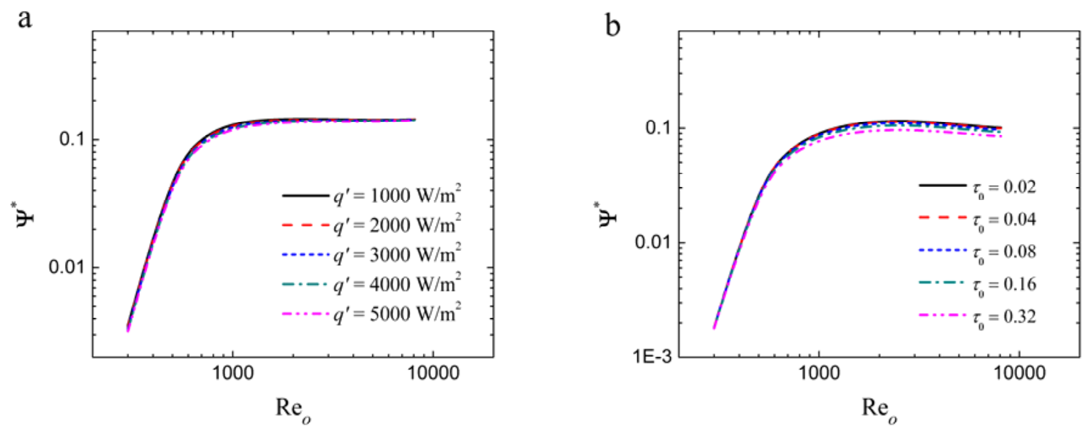

Fig. 5. (a) Ψ^* vs. Re_o for q'; (b) Ψ^* vs. Re_o for τ_0.

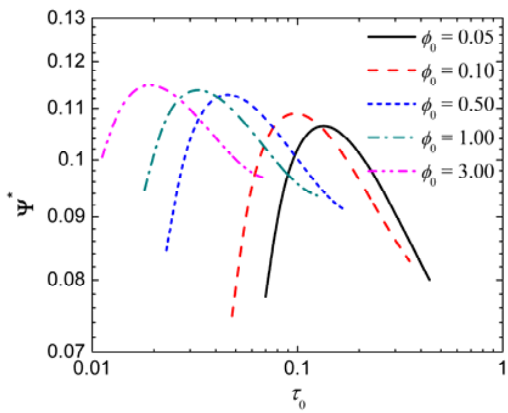

Fig. 6. Ψ^* vs. τ_0 for ϕ_0.

The values of Ψ^* are plotted against Re_o in Fig. 5 and 6. It is shown from Fig. 5(a) that the heat flux has little influence on the Ψ^* results, and Ψ^* is a strong function of Re_o. From Fig. 5(b), the effect of relative temperature difference is showed up at Re_o>800. The performance of the heat transfer enhancement of OSF fin degrades as τ_0 is increased, which indicates that the evaluation based on the parameter Ψ^* complies with the thermodynamic principle. From Fig. 6, it is also shown that for a particular irreversibility distribution ratio, there is an optimum condition of τ_0 and Re_o (or m) which corresponds to the maximum Ψ^*.

3.3. Effects of geometric parameters

The variations of Ψ^* vs. α are shown in Fig. 7(a). It is seen that Ψ^* increases initially and then decreases with an increasing of α. The influence of relative temperature difference on the behaviour of Ψ^* is pronounced in the region of α=0.04-0.06. Moreover, the optimum point of α=0.051 is stable against relative temperature difference.

The effects of fin density γ for OSF fins on the behaviours of Ψ^* are presented in Fig. 7(b). Similar with the trend in terms of α, Ψ^* also passes through a maximum value as γ is increased. From Fig. 7(b) the maximum values of Ψ^* are stably obtained when γ=0.138.

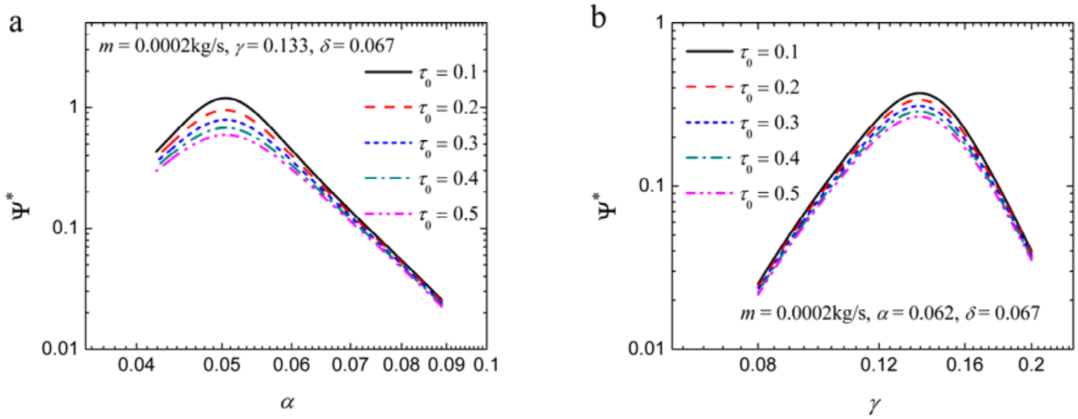

Fig. 7. (a) Ψ^* vs. α; (b) Ψ^* vs. γ.

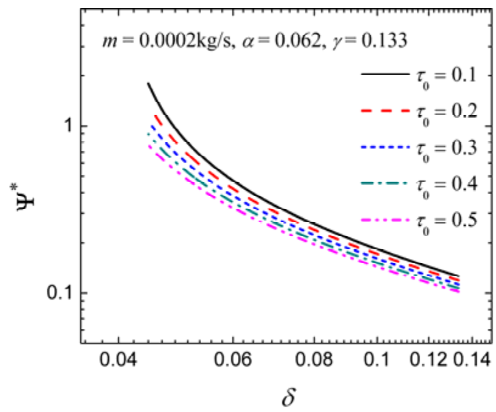

Fig. 8. Ψ^* vs. δ.

The fin thickness-to-length ratio δ has a monotonic influence on the performance of OSF fins, this is illustrated in Fig. 8. The studied range of δ is from 0.04 to 0.133. According to the evaluation curves, smaller δ at lower m would be more available to improve the performance of the heat transfer enhancement for OSF fin, which is consistent with the conclusion in literature [6].

4. Conclusion

(1) The conventional evaluations, which suggest that the larger temperature difference of heat transfer, the more advantageous using the augmented channels, contradict the thermodynamic principle.

(2) The present method not only presents evaluations that complies with the thermodynamic principle, but applies to the situation in which the reference for comparison is changing. For a particular ϕ_0, there is an optimum operating condition which corresponds to the highest degree of the heat transfer enhancement for OSF fin.

(3) There are optimum parameters α and γ that contribute to the maximum Ψ^* for a specific m and τ_0, and smaller δ at lower m yields an improvement in performance of the heat transfer enhancement for OSF fin.

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