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- EEI = $k/\nabla p^n$ is proposed to evaluate the energy efficiency of plate heat exchangers.
- A normal distribution function of EEI of plate heat exchangers is obtained.
- EEI can be used to grade plate heat exchangers in terms of energy efficiency.
- The principles of energy efficiency indexes of heat exchangers grading are proposed.

A Quantitative Energy Efficiency Evaluation and Grading of Plate Heat Exchangers

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ABSTRACT

Energy efficiency evaluation of heat exchangers are crucial for energy saving in industrial applications. Currently, no energy efficiency evaluation is appropriate for the grading of energy efficiency of heat exchangers. In this paper, an energy efficiency index (EEI) is proposed to evaluate the energy efficiency of single-phase flow and heat transfer of plate heat exchangers. EEI is defined by $k/\nabla p^n$, where k is overall heat transfer coefficient, ∇p is flow pressure gradient and exponent n is the indicator of the relative importance between heat transfer and flow resistance. EEI indicates the overall heat transfer coefficient per unit equivalent pressure drop. The recommended n is 0.31 for plate heat exchangers based on the data of 281 heat exchangers. We conclude a normal distribution function for the evaluation of energy efficiency of plate heat exchangers in Chinese industrial applications. The present method can quantitative evaluate the performance of plate heat exchangers and hence can grade them in terms of energy efficiency.

Keywords: plate heat exchanger; Energy Efficiency Index; overall heat transfer coefficient; flow pressure gradient.

1. Introduction

Heat exchangers are commonly used in energy conversion and transport processes in which heat transfers from hot fluid to cold fluid. Nowadays, the use of compact heat exchangers, for example, the plate heat exchangers, has been increasing owing to their advantages of compactness and high heat transfer efficiency [1]. Although heat exchangers play an important role in the industrial secondary energy utilization, waste heat recovery, energy saving and emission reduction, they have to consume much pumping or blowing power inevitably because of the viscous fluid flowing. How to evaluate quantitatively and grade their energy efficiency is becoming a key issue.

Energy efficiency concerns the technical ratio between the quantity of energy consumed and the quantity of obtainable services [2]. Improving energy efficiency is viewed to be one of the most useful and cost-effective solutions of energy and environmental problems. Many specified mandatory energy efficiency standards are established especially for some equipments such as air conditioner and boiler [3]. However, to the best of our knowledge, the energy efficiency standards of heat exchangers is absence, mainly due to the scarcity of an appropriate heat exchanger energy efficiency index. The indexes in the evaluations of heat exchangers based on the First Law of Thermodynamics include [4, 5] specific pressure drop $\Delta p/NTU$, energy coefficient Φ/N, and heat exchanger energy efficiency ε, etc. The evaluations are clearly specified, but may present significant disparity for different heat transfer areas and under different working conditions. Webb [6, 7] and Bergles [8] proposed a series of evaluation criteria for heat transfer corresponding to practical requirements including saving materials, increasing heat transfer, and reducing pump power consumption, but they did not provide a uniform evaluation or index. Yilmaz et al. [9] proposed that practical requirements and working conditions should be taken into consideration in deciding an appropriate evaluation index. Tao's research team [10] suggested an energy-saving-oriented evaluation in terms of the ratio of enhancement surface to reference surface, i.e. the relationship of Nu_a/Nu_0 and f_a/f_0 for the purpose of heat transfer enhancement. This method is applied to evaluate the enhancement for the heat transfer element [11], and the evaluation results are dependent on the characteristics of the selected reference surface. However, it doesn’t work for the assessment of heat exchangers for lack of the analysis of wall heat conduction, flow arrangements and other decisive parameters.

Some evaluation methods based on the Second Law of Thermodynamics are the Entropy Evaluation Method and the Exergy Evaluation Method. The Entropy Evaluation Method [12-115] concerns entropy generation analysis, entropy production efficiency, augmentation entropy generation number, and so on. Exergy efficiency evaluation method [16-18] is typical of the Exergy Evaluation Method which concerns exergy efficiency, rational efficiency and exergy recovery index. Exergy Evaluation Method also provides an approach to evaluate the heat exchanger in terms of exergy loss [19-21] including specific irreversibility, exergy loss rate, and irreversible number. Recently, Ebrahimzadeh et al. proposed a heat exchanger efficiency index affected by the actual exergy loss and the largest achievable exergy loss [22]. The above-mentioned evaluations have been widely applied in the study on the characteristics of flow and heat transfer and the optimization of the overall structure and the operational conditions of heat exchangers [23]. But the fluid entropy and exergy are dependent on the pressure and temperature. Generally, the temperatures under testing conditions and working conditions of heat exchangers are significantly different. The heat capacity ratio, the number of heat transfer units, and flow arrangements also impact on the entropy increase and exergy loss in the process of heat transfer. In addition, entropy increase and exergy loss caused by viscous flow are one to two magnitude orders less than the entropy increase and exergy loss caused by temperature-difference heat transfer. Guo [24] worked out a new physical concept, entransy, to describe the heat transfer capacity, and pointed out that the entropy-production-based evaluation was applicable for the heat-work conversion, but the irreversibility of the heat transfer for the purposes of heating or cooling should be evaluated in terms of entransy [25].

Although extensive works were done and numbers of indexes were proposed for the performance evaluation of enhanced heat transfer technology, no index was obtained to evaluate the energy efficiency quantitatively and grade the heat exchangers. In this paper, we propose an energy efficiency evaluation for plate heat exchangers which proves to work well in terms of stability on different practical conditions such as flow velocities, heat transfer areas and fluid properties. Hence, the energy efficiency for a specific plate heat exchanger can be determined, which is the precondition for the energy efficiency grading. In addition, the energy efficiency for the plate heat exchangers can be quantitatively evaluated with the statistical distribution function. Based on the principle above, we propose the energy efficiency index (EEI) for plate heat exchanger evaluation and grading [26]. The distribution of EEI is obtained based on the single-phase flow and heat transfer data of 281 plate heat exchangers. And the effects of plate heat exchanger structure, thermal conductive resistance, and fluid thermo-physical properties on EEI are analyzed. Furthermore, the effectiveness of EEI is validated, and some applications are presented for the sake of the practical significance of our principle.

2. Energy efficiency evaluation of plate heat exchangers

2.1. Theoretical derivation

To characterize the single-phase flow in a plate heat exchanger, pressure gradient ∇p is defined as Eq. (1), which equals the mechanical energy loss for a unit volume of the fluid through the unit length.

$$\nabla p = \omega_c \frac{\Delta p_c}{l_c} + \omega_h \frac{\Delta p_h}{l_h} \tag{1}$$

where Δp, l and ω are flow pressure drop, flow length and weight coefficient, respectively. The fluid flow length is the characteristic length of the plate heat exchanger, i.e. the distance between the centers of the angle-holes in the flow direction. The subscripts c and h are indicators of cold fluid and hot fluid, respectively. The weight coefficients satisfy the condition of $\omega_c + \omega_h = 1$.

For the liquid-liquid turbulent flow and heat transfer in the plate heat exchanger, flow pressure drop and weight coefficient at the cold and hot sides can be assumed to be equal, i.e. $\omega_c = \omega_h = 0.5$. Pressure gradient ∇p and convective heat transfer coefficient h of the plate heat exchanger are expressed as follows:

$$\nabla p = \frac{\left(\Delta p_c + \Delta p_h\right)}{2l} = \frac{c_{\Delta p}}{l}\left(\frac{d_e}{\nu}\right)^{n_{\Delta p}} \rho u^{2+n_{\Delta p}} \tag{2}$$

$$h = c_k \frac{\lambda_f}{d_e}\left(\frac{d_e}{\nu}\right)^{n_k} \mathrm{Pr}^{n_1} u^{n_k} \tag{3}$$

where h is convective heat transfer coefficient, d_e hydraulic diameter of the plate heat exchanger, and ρ, ν, u, λ_f and Pr represent density, kinematic viscosity, velocity, thermal conductivity and Prandtl number of the fluid, respectively. The exponent n_1 is commonly taken as 0.4 for cold fluid and 0.3 for hot fluid. c_k, n_k, $c_{\Delta p}$ and $n_{\Delta p}$ are constants, and measured experimentally.

In oil-oil or oil-water flow and heat transfer, the overall heat transfer coefficient of the plate heat exchanger is generally within the range of 400-1350 $W \cdot m^{-2} \cdot K^{-1}$. In water-water flow and hear transfer the overall heat transfer coefficient is comparatively higher, which ranges between 2900 and 9400 $W \cdot m^{-2} \cdot K^{-1}$; thus, the overall heat resistance to liquid-liquid flow and heat transfer is in the range of about 1.1×10^{-4}-$2.5\times10^{-3} \cdot m^2 \cdot K \cdot W^{-1}$, and it tends to increase with gas present in the heat transfer. Thermal conductive resistance consisting of plate thermal conductive resistance and fouling resistance generally ranges from about 2.4×10^{-5} - $6.8\times10^{-5} \cdot m^2 \cdot K \cdot W^{-1}$ and it is neglected in the derivation. Assume coefficient c_k and exponent n_k, in the correlation and the fluid thermal properties are equal and the overall heat transfer coefficient k of the plate heat exchanger is reduced as Eq. (4).

$$k = \frac{1}{\frac{1}{h_c} + \frac{1}{h_h} + \frac{\delta}{\lambda_w} + R} \approx \frac{1}{\frac{1}{h_c} + \frac{1}{h_h}} \tag{4}$$

where k is the overall heat transfer coefficient of the plate heat exchanger, λ_w and δ are thermal conductivity and thickness of the plate, respectively, and R is fouling resistance.

The heat transfer characteristics of the plate heat exchanger can be represented by overall heat transfer coefficient k. The higher k is helpful to reduce the terminal temperature difference, and heat transfer area. Similarly, the flow characteristics can be represented by pressure gradient ∇p. The higher ∇p is, the more pumping power is consumed for the same volume rate and length of the flow. In other words, k and ∇p represent the gains and losses of the process. As we know that exergy loss caused by temperature-difference heat transfer and viscous flow have a difference on the order of 1 to 2 magnitudes. To address the relative importance of heat transfer and flow for energy efficiency, $k/\nabla p^n$ is introduced. By combining Eq. (2) and Eq. (4), EEI is expressed as Eq. (5).

$$EEI = \frac{k}{\nabla p^n} = z_s z_p u^{n_k - n\left(2 + n_{\Delta p}\right)} \tag{5}$$

where $z_s = d_e^{\,n_k - n n_{\Delta p} - 1} \dfrac{c_k}{\left(c_{\Delta p}/l\right)^n}$, $z_p = \dfrac{\lambda_f}{\rho^n \nu^{n_k - n n_{\Delta p}}} \dfrac{\Pr^{0.3}\Pr^{0.4}}{\left(\Pr^{0.3} + \Pr^{0.4}\right)}$.

Exponent n is determined by statistical analysis and hypothesis testing on the performance test data of the plate heat exchangers, which minimizes the EEI dispersion degree at different flow velocities.

$$n_k - n\left(2 + n_{\Delta p}\right) \approx 0 \tag{6}$$

i.e.,

$$EEI = \frac{k}{\nabla p^n} \approx z_s z_p \tag{7}$$

$$z_s \approx d_e^{\,2n-1} \frac{c_k}{\left(c_{\Delta p}/l\right)^n} \tag{8}$$

$$z_p \approx \frac{\lambda_f}{\rho^n \nu^{2n}} \frac{\Pr^{0.3}\Pr^{0.4}}{\left(\Pr^{0.3} + \Pr^{0.4}\right)} \tag{9}$$

Thus, EEI indicates the overall heat transfer coefficient k per unit equivalent pressure drop in single-phase flow and heat transfer of a plate heat exchanger, which reflects the inherent energy efficiency property of the plate heat exchanger. z_s is related to the plate heat exchanger structure which is predetermined by the design and production technology of its manufacturer, and z_p is related to the thermo-physical properties of the fluid in the plate heat exchanger which represent the influences of fluid temperature, pressure and type.

2.2. Physical meaning of energy efficiency index

To investigate the physical meaning of EEI, Φ/N^n is introduced and analyzed as follows.

$$\frac{\Phi}{N^n} = \frac{k \cdot A \cdot \Delta t}{\nabla p^n \left(l \cdot q_v\right)^n} = EEI \cdot \frac{A \cdot \Delta t}{\left(l \cdot q_v\right)^n} \tag{10}$$

where Φ, N, A, Δt and q_v are heat transfer rate, pumping power, actual heat transfer area, temperature difference between cold and hot fluid and volume flow, respectively. According to Eq. (10), when Δt and q_v are equal and plate heat exchangers have the same A and l, a higher EEI results in a larger Φ/N^n. Therefore, the physical mechanism of EEI is a measure of the heat transfer rate from hot fluid to cold fluid with the cost of per unit pumping power.

3 Energy efficiency index of plate heat exchangers

3.1. Flow and heat transfer database of plate heat exchangers

The flow and heat transfer of plate heat exchangers is measured in National Center of Oil Drill and Refining Equipment Supervision and Inspection in Shanghai, China. The flow and heat transfer database in present study consists of 281 gasketed plate heat exchangers produced by 82 companies (see Table 1 for specifications). The performance test follows the prescribed procedures of *Plate Heat Exchanger GB16409-1996* and *Test method for the performance of heat exchangers and heat exchange element GB/T27698-2011*. The experiment setup (see Ref. 28) is mainly composed of hot and cold loops, and water is used as working fluid. Plate heat exchangers are tightened to achieve the close metal contact between the plates, i.e., compressed pitch per plate equaling plate thickness plus plate gap in industrial applications. Volume flux, pressure drop between the inlet and the outlet, temperature of the inlet and outlet of both hot and cold fluid are measured, respectively, from which plus some structure parameters of plate heat exchangers, velocities of fluid and overall heat transfer coefficient k can be deduced. Notably, the impact of thermal conductive resistance on heat transfer is contained in overall heat transfer coefficient.

Experimental data are obtained for 0.1 m/s ≤ u ≤ 1.0 m/s [27]. Firstly, the velocities of the cold and hot fluid are equally changed from 0.1 m/s to 1.0 m/s with the variation of 0.1 m/s. Then keep the hot fluid velocity at 0.5 m/s, and change the velocity of the cold one from 0.1 m/s to 1.0 m/s with the variation of 0.1 m/s. Finally, exchange the velocity conditions of the hot and cold fluid. Thus, we have 30 testing cases. To improve the accuracy, we test 3 times and average them for each case. And for each case, data will be recorded after a stabilization of 5 minute with absolute heat balance error no more than 5%. The temperature of the hot and cold fluid at the inlets are set as 50 ℃ and 30 ℃, respectively. The experimental uncertainty is analyzed according to Ref. 28. The accuracies of all volumetric flow meters and differential pressure transducers are ±0.5% full scale, and the maximum deviation of thermocouples is ±0.2 ℃. The measurement uncertainties of fluid volume flux, pressure drop and temperature are ±5%, ±5% and ±2%, respectively. k is calculated through four temperatures and one flow flux. Owing to the independence of those parameters, an uncertainty of ±6.5% exists in the calculated values of k. And the uncertainty of EEI is ±8.5%.

3.2. Recommended value of exponent n

Exponent n in EEI can be determined from the above-mentioned database of plate heat exchangers. Figure 1 shows overall EEI with the variation of flow velocity and with different values of exponent n for plate heat exchangers, of which the overall EEI is the average of EEI for heat exchangers in the database. Meanwhile, Figure 1 also provides the variance σ^2 of the overall energy efficiency for different exponents to characterize the fluctuation of overall EEI. When n equals 0.31, overall EEI has minimal fluctuation and optimal stability; it follows that EEI is under little influence of flow velocity, and it can precisely reflect the plate heat exchanger's energy efficiency for different flow velocities. Therefore, we recommend that exponent n in EEI for plate heat exchangers is 0.31.

3.3. Validation

EEI can be validated by the exergy efficiency of the 281 plate heat exchangers in our database. Exergy loss in flow and heat transfer includes the loss caused by temperature-difference heat transfer ΔE_T and by viscous flow $\Delta E_{\Delta p}$. According to the exergo-economic criteria, the latter is mechanical energy loss, and has a higher energy grade. Therefore, a conversion relationship exists between $\Delta E_{\Delta p}$ and ΔE_T, and the conversion coefficient in this paper is set as 4 [29].

$$\eta = 1 - \frac{\Delta E_T + 4\Delta E_{\Delta p}}{Q} \tag{11}$$

where η and Q are exergy efficiency and heat transfer quantity in flow and heat transfer, respectively, and ΔE_T and $\Delta E_{\Delta p}$ are exergy loss caused by temperature-difference heat transfer and by viscous flow, respectively.

The exergy efficiency of the 281 plate heat exchangers under the standard testing conditions is calculated according to Eq. (11). As EEI increases, it gradually increases; i.e. these two indexes display an identical tendency of change. This finding is in good agreement with the Second Law of Thermodynamics. Figure 2 also shows that plate heat exchangers with triple corrugated plates have relatively low exergy efficiency.

4 Influencing factors of energy efficiency index

4.1. Plate heat exchanger structure

As shown in Eq. (5), the parameters in EEI relevant to the structure of the plate heat exchanger are c_k, n_k, $c_{\Delta p}$, $n_{\Delta p}$, d_e and l which are predetermined by the design and production technology of the manufacturers. According to Eq. (6), the exponent n is also influenced by the structure of heat exchangers. Coefficient $c_{\Delta p}$ is proportional to flow length l, thus the plate heat exchangers with identical structures have the same EEI with different characteristic length. The plate width w is incorporated in the mean flow velocity u. As h and Δp are expressed as the functions of u in this paper, w doesn't appear directly in EEI. Moreover, the ratio, l/w, also impacts h and Δp. Its influences on EEI are considered in c_k, n_k, $c_{\Delta p}$ and $n_{\Delta p}$. Thus, the EEI can evaluate the energy efficiency of plate heat exchangers with different structures.

4.2. Thermal conductive resistance

Thermal conductive resistance of plate heat exchangers is mainly affected by the plate thickness; thus to investigate its effects on the magnitude and stability of the energy efficiency index EEI, the variation of EEI with fluid velocity u for different plate thickness δ is plotted in Figure 3. In our database, the plate heat exchangers with δ = 0.6 mm are in the majority. Thus, EEI of the plate heat exchangers with δ = 0.6 mm is obtained from the experimental data. Then, assume that the flow and convection heat transfer characteristics remain constant, EEI of the plate heat exchangers with δ = 0.4 mm and 0.8 mm can be calculated through changing the plate thickness and accordingly changing the thermal conductive resistance. Notice that EEI decreases with δ increasing, which indicates that reducing the plate thickness can significantly improve the energy efficiency of plate heat exchangers in water-water flow and heat transfer. Additionally, as u increases, EEI at δ = 0.6 mm at first increases and then decreases, while EEI at δ = 0.4 mm increases uniformly and EEI at δ = 0.8 mm uniformly decreases. The possible reason is that the relative magnitude of thermal conductive resistance and thermal convective resistance influence the exponent relationship of overall heat transfer coefficient with flow velocity and thereby influence the stability of EEI in terms of velocity (Eq. 4). The increase in the plate thickness which enlarges the thermal conductive resistance leads to the decrease in the exponent of the flow velocity in the overall heat transfer coefficient, thus the energy efficiency index decreases with fluid velocity for the thicker plate (e.g., δ = 0.8 mm).

4.3. Fluid Thermal properties

As shown in Eq. (9), the parameters in energy efficiency index EEI concerning fluid thermal properties are λ_f, ρ, v and Pr; EEI is influenced by the fluid type, temperature and pressure. According to Eqs. (7)− (9), when the exponent difference of kinematic viscosity between various plate heat exchangers is negligible, the changes in fluid temperature, pressure and type exert no effect on the relative magnitude of EEI. Hence, the present energy efficiency evaluation can be used for fluids with different thermal properties.

5 Applications of energy efficiency index

5.1. Distribution of energy efficiency index for plate heat exchangers

The overall distribution of EEI in plate heat exchanger industry can be predicted based on the EEI of the plate heat exchangers in the present database. The EEI of the 281 plate heat exchangers under standard testing conditions is calculated and the distribution histogram is plotted in Figure 4. According to parameter estimation and hypothesis testing, the probabilistic density distribution of overall EEI for plate heat exchangers agrees with the normal distribution function with mean μ = 191.2 and variance σ^2 = 22.1^2 at 95% confidence. Furthermore, the overall EEI agrees with the normal distribution and has approximate probabilistic density for different flow velocities (u = 0.3~0.8 $m \cdot s^{-1}$) at 95% confidence (data uncovered in the figure).

5.2. Grading of plate heat exchangers

According to the probabilistic density function $f(x) = N(191.2, 22.1^2)$ for overall EEI of plate heat exchangers , the range of energy efficiency index for each grade can be determined. Table 2 demonstrates the grading of plate heat exchangers based on our principle. Thereby, the plate heat exchanger can be graded in one of the high, middle, transient or low energy efficiency. The average overall EEI will be continuously improved with the technological advances. Thus, the database needs to be updated periodically and accordingly the range of EEI needs to be also updated.

5.3. Quantification of energy saving in plate heat exchangers

EEI can be used to quantify the energy-saving by replacing low-energy-efficiency plate heat exchangers with high- or middle-energy-efficiency plate heat exchangers under same heat transfer conditions and for same practical demands. Refer to the energy efficiency grading in Table 2, and the average energy efficiency index of the low-, middle- and high-energy-efficiency plate heat exchangers can be calculated with the weighted average algorithm. Take the high-energy-efficiency plate heat exchanges for example, the average EEI, EEI_h, is calculated through Eq. (12).

$$EEI_h = \frac{\sum_{206.12}^{+\infty} [f(EEI) \cdot EEI]}{\sum_{206.12}^{+\infty} f(EEI)} = 219.24 \tag{12}$$

Similarly, it can be concluded that $\mathrm{EEI}_m = 194.09$ and $\mathrm{EEI}_l = 163.14$. According to the principle of $EEI = k/\nabla p^{0.31}$, when demanding the same overall heat transfer coefficient, the pressure drop ratio between low- and high-energy-efficiency plate heat exchangers can be determined by

$$\frac{\nabla p_l}{\nabla p_h} = \left(\frac{EEI_h}{EEI_l}\right)^{1/0.31} = 2.59 \tag{13}$$

$$\frac{\nabla p_l\text{-}\nabla p_h}{\nabla p_l} = \frac{2.65-1}{2.65} = 61.46\% \tag{14}$$

Replacing low-energy-efficiency plate heat exchangers with high-energy-efficiency ones can reduce 61.46% of flow pressure loss on average, or achieve a higher overall heat transfer coefficient with the same pressure loss. Similarly, replacing low-energy-efficiency plate heat exchangers with middle-energy-efficiency ones can reduce an average of 42.91% of flow pressure loss. Therefore, in practice substituting plate heat exchangers of low grade for those of high or middle grade will significantly reduce energy consumption.

5.4. Feature curves of plate heat exchangers

Based on our principle of EEI, the working feature curves of a plate heat exchanger can be pictured. The curves in Figure 5 indicates the working feature of a plate heat exchanger for different flow velocities in water-water flow and heat transfer. This knowledge is of great help for both the manufacturers and the users to design the working conditions for their plate heat exchangers.

6 Conclusions

In the present study, we use k and ∇p to indicate the gains and costs of the flow and heat transfer, exponent n to describe the relative importance between heat transfer and pressure drop, and EEI defined by $k/\nabla p^n$ is proposed to evaluate and grade the single-phase flow and heat transfer energy efficiency of heat exchangers. EEI is an index of the overall heat transfer coefficient per unit equivalent pressure drop. According to our study on the flow and heat transfer database, exponent n is selected as 0.31 Principle validation of EEI shows a good agreement with the Second Law of Thermodynamics. The impacts of plate heat exchanger structure, thermal conductive resistance, and fluid thermo-physical properties on EEI are analyzed, which concludes that reducing the plate thickness can effectively improve the energy efficiency of plate heat exchangers in water-water flow and heat transfer. The overall distribution $f(\mathrm{x}) = N(191.2, 22.1^2)$ of EEI for plate heat exchangers are concluded, and EEI can be used to quantitatively evaluate and grade the energy efficiency of plate heat exchangers. We propose EEI = $k/\nabla p^n$ and this evaluation method can be a reference for other types of heat exchangers.

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Nomenclature

A actual heat transfer area, m^2
c_k constant, experimentally measured
$c_{\Delta p}$ constant, experimentally measured
d_e hydraulic diameter, m
EEI Energy Efficiency Index
f friction factor
h convective heat transfer coefficient, $W \cdot m^{-2} \cdot K^{-1}$
k overall heat transfer coefficient, $W \cdot m^{-2} \cdot K^{-1}$
l characteristic flow length, m
N pumping power, W
Nu Nusselt number
n_1 constant, 0.4 for cold fluid and 0.3 for hot fluid
n_k constant, experimentally measured
$n_{\Delta p}$ constant, experimentally measured
∇p pressure gradient, $Pa \cdot m^{-1}$
Pr Prandtl number
Q heat transfer, J
q_v volume flow, $m^3 \cdot s^{-1}$
R fouling resistance, $m^2 \cdot K \cdot W^{-1}$
u flow velocity of fluid, $m \cdot s^{-1}$
w plate width, m

Greek alphabet

ΔE_T exergy loss caused by temperature-difference heat transfer, J
$\Delta E_{\Delta p}$ exergy loss caused by viscous flow, J
Δp pressure drop, Pa
Δt temperature difference, K
δ plate thickness, m
η exergy efficiency

λ_f thermal conductivity of fluid, $W \cdot m^{-1} \cdot K^{-1}$

λ_w thermal conductivity of plate, $W \cdot m^{-1} \cdot K^{-1}$

ν fluid kinematic viscosity, $m^2 \cdot s^{-1}$

ρ fluid density, $kg \cdot m^{-3}$

Φ heat transfer rate, W

ω weight coefficient

Subscripts

a augment

c cold fluid

h hot fluid，high

l low

m middle

0 original

Figure Caption

Figure 1. Overall EEI with the variation of flow velocity and with different values of exponent *n* for plate heat exchangers.

Figure 2. Distribution of exergy efficiency and EEI for plate heat exchangers under standard testing conditions.

Figure 3. Variation of EEI with u under different δ.

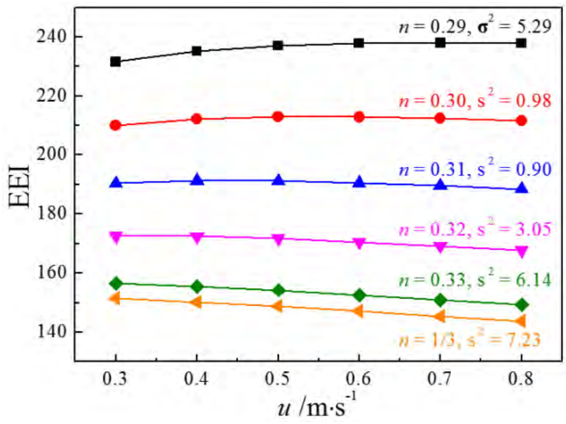
Figure 4. Distribution of EEI of plate heat exchangers in present database and fitting curve.

Figure 5. Working feature curves of plate heat exchangers. Variation of k, Δp, and EEI with different u in water-water flow and heat transfer.

Table Caption

Table 1. Specifications of plate heat exchangers in present database

Table 2. Grading of plate heat exchangers



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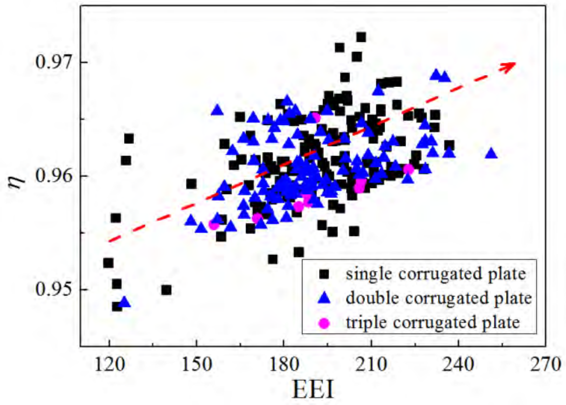

Figure 2. Distribution of exergy efficiency and EEI for plate heat exchangers under standard testing conditions.

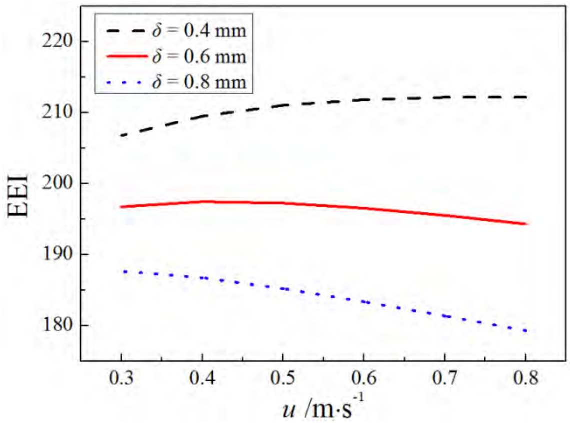

Figure 3. Variation of EEI with u for different δ.

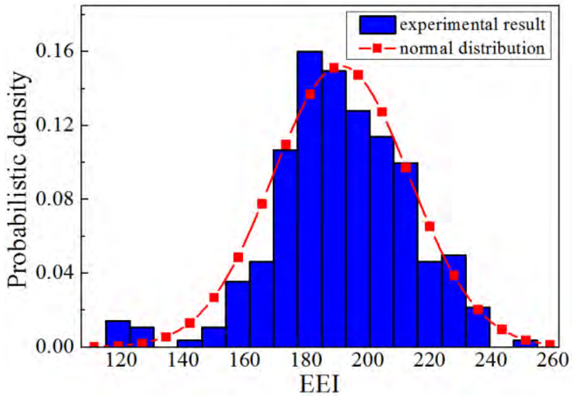

Figure 4. Distribution of EEI of plate heat exchangers in present database and fitting curve.

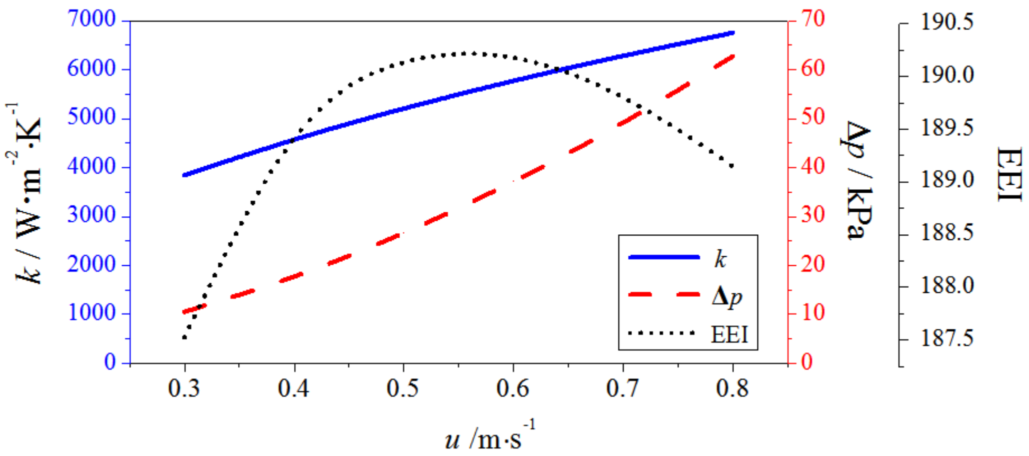

Figure 5. Working feature curves of a plate heat exchanger. Variation of k, Δp, and EEI for different u in water-water flow and heat transfer.

Table 1. Specifications of plate heat exchangers in present database

Structural parameters	Range (Quantity)
Corrugated plate	Single (162), double (110) and triple (9) chevron
Wave form	Sine (186), trapezoidal (70), triangle (25)
Plate thickness	0.4–0.8 mm
Wave angle	22.5–63°
Corrugated pitch	6–20 mm
Wave density	2.75–5.00
Ratio of length to width	0.35–5.60
Plate number for each heat exchanger	No less than 7

Table 2. Grading of plate heat exchangers

Grade	EEI range	Percentage
High energy efficiency	$k/\nabla p^{0.31} \geqslant 206.12$	25%
Middle energy efficiency	$206.12 > k/\nabla p^{0.31} \geqslant 182.73$	40%
Transition	$182.73 > k/\nabla p^{0.31} \geqslant 176.35$	10%
Low energy efficiency	$k/\nabla p^{0.31} < 176.35$	25%