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Theoretical analysis and experimental confirmation of the uniformity principle of temperature difference field in heat exchanger

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Abstract

Theoretical analysis and experimental confirmation for the principle to improve the thermal performance of heat exchangers is performed in this paper. The more uniform the temperature difference field (TDF), the higher the effectiveness of heat exchanger for the fixed N_{tu} and C_r. The uniformity of the TDF and the effectiveness of 13 types of heat exchangers are studied analytically and numerically, and the results support the uniformity principle of TDF. Further verification is given by the asymptotic solution for TDF in terms of a recurrence formula of heat transfer area distribution for the same kind of heat exchanger. The analysis of entropy generation caused by the heat transfer indicates that the uniformity principle of TDF satisfies the second law of thermodynamics. The results of the experimental setup, governed by the uniformity principle of TDF, show that the effectiveness increases with the increase of the uniformity of TDF. Heat exchanger effectiveness for the best flow distribution was found to be 11.2% greater than that of the conventional flow distribution without associated increase in pressure drop. Two ways, redistributing heat transfer areas and varying the connection between tubes, are presented for improving the uniformity of TDF and increasing effectiveness for the crossflow heat exchangers.

1. Introduction

Heat exchangers are widely used in most engineering fields, such as the process, power, automotive, cryogenics, refrigeration, air conditioning, heat recovery, manufacturing industries, etc. To develop and to improve heat exchanger effectiveness has been the main attempts of many heat exchanger studies [1–11]. Although the study of heat exchangers has a history for approximate 100 years, the development of modern science and technology and the appearance of emerging fields raise new requirements to improve the performances of heat exchangers. Tremendous efforts have been made in the study of heat exchanger surface enhancement techniques. Bergles [12] classified augmentation techniques as passive methods, which require no direct application of external power, and as active schemes, which require external power. Indeed such techniques have dramatically helped in improving heat exchanger thermal performance, but they have raised problems of complexity in structure, manufacturing, fouling characteristics, and increased pressure drops in heat exchangers.

It is well known that the effectiveness of counterflow heat exchangers is the highest among different flow arrangements [3,13–15]. The inherent reason is that the log-mean temperature difference of counterflow heat exchanger is the largest. The schematic of temperature distributions of the hot and cold fluids along the flow path for parallel-flow and couterflow heat exchangers is shown in Fig. 1. It is easy to see that the temperature difference between hot and cold fluids along their flow path in couterflow heat exchangers is more uniform than that in parallel-flow heat exchangers. Guo et al. [16] provided an idea that for counterflow heat exchanger high effectiveness could be owing to most uniform local

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Nomenclature

A	heat transfer area
C_a	dimensionless distribution of heat transfer area
c_p	constant pressure specific heat
C_r	heat capacity rate ratio
d_i	inside tube diameter
d_o	outside tube diameter
h	heat transfer coefficient
L	dimension of heat exchanger in length direction
M	number of subelement in length direction
$\dot{m}$	mass flow rate
N	number of subelement in width direction
N_{tu}	number of heat transfer units
Q	heat transfer rate ratio
R''_f	fouling factor
R_w	conduction resistance
S	entropy generation rate
T	temperature of hot fluid
T_{dr}	relative temperature difference
t	temperature of cold fluid
U	overall heat transfer coefficient
W	dimension of heat exchanger in width direction

Greek symbols

ε	heat exchanger effectiveness
η	fin efficiency
τ	increasing percentage of effectiveness
Φ	uniformity factor of TDF

Subscripts

c	cold fluid
h	hot fluid
i	inlet section
max	maximum
min	minimum
o	outlet section

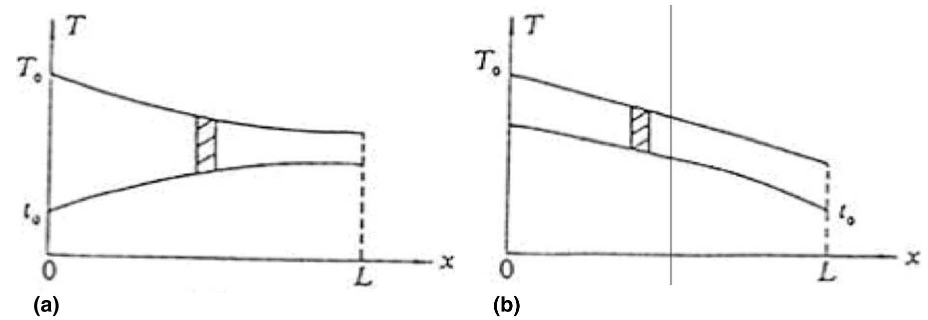


Fig. 1. Variation of fluid temperature along flow path in heat exchangers: (a) parallel-flow; (b) counterflow.

temperature differences between the two flowing fluids compared to other heat exchangers, and hence it could be a new way to upgrade the heat transfer by improving the uniformity of the temperature difference field (TDF) in heat exchangers. In this paper, the impact of the uniformity of TDF on the effectiveness of heat exchanger is investigated. The uniformity principle of temperature difference field in heat exchangers is presented, and verified analytically, numerically and experimentally. Two ways, redistributing heat transfer areas and varying the connection between tubes, are presented for improving the uniformity of the TDF and consequently, for increasing effectiveness of a crossflow heat exchanger.

2. Impact of uniformity of TDF on effectiveness

2.1. TDF and its uniformity factor

A concept of TDF has been proposed to study the effect of the temperature difference distribution between the hot and cold fluids on the performance of heat exchangers. For convenience, in discussing the effects of the uniformity of TDF on effectiveness, a parameter describing the degree of uniformity of TDF is defined as the uniformity factor of TDF, Φ. For the two-dimensional problem, shown in Fig. 2, Φ is defined in the following ways:

$$\Phi = \frac{\int_0^W \int_0^L [T(x,y) - t(x,y)]\,\mathrm{d}x\,\mathrm{d}y}{\sqrt{WL \int_0^W \int_0^L [T(x,y) - t(x,y)]^2\,\mathrm{d}x\,\mathrm{d}y}} \tag{1}$$

or in finite difference terms,

$$\Phi = \frac{\sum_{i=1}^{M} \sum_{j=1}^{N} [T(i,j) - t(i,j)]}{\sqrt{MN \sum_{i=1}^{M} \sum_{j=1}^{N} [T(i,j) - t(i,j)]^2}} \tag{2}$$

where $T(x,y)$ and $t(x,y)$ are the hot and cold fluid temperature distributions, respectively; L and W are the dimensions of the heat exchanger; M and N are the numbers of subelements in the length and width direc-

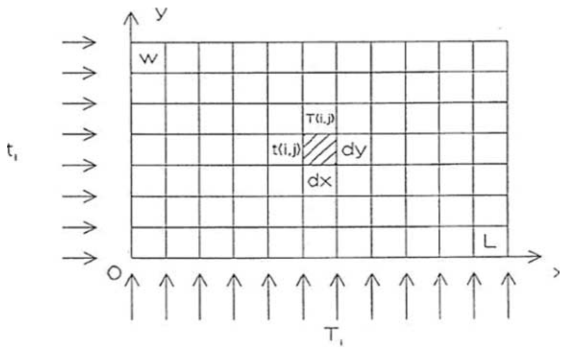


Fig. 2. 2-D grid partition in the crossflow heat exchanger.

tions. The uniformity factor of TDF in reality is always smaller than unity, and the more non-uniform the TDF, the smaller the uniformity factor of TDF.

2.2. Relationship between uniformity factor of TDF and effectiveness

In order to investigate the relationship between the uniformity factor of TDF, Φ, and the effectiveness of heat exchangers, ε, the uniformity factors of TDF for typical heat exchanger designs (counterflow, parallel-flow and crossflow) were calculated. Guo et al. [16,17] developed the analytical formulas of uniformity factors for parallel-flow, counterflow, and crossflow heat exchangers with one mixed stream ($C_{\max}$ or $C_{\min}$).

For parallel-flow heat exchangers:

$$\Phi = \frac{[1-\exp(-2(1+C_r)N_{tu})]/[(1+C_r)N_{tu}]}{\sqrt{[1-\exp(-2(1+C_r)N_{tu})]/[2(N_{tu}(1+C_r)]}}. \tag{3}$$

For counterflow heat exchangers:

$$\Phi = \frac{\exp((1-C_r)N_{tu})-1}{(1-C_r)N_{tu}}\sqrt{\frac{2(1-C_r)N_{tu}}{\exp(2(1-C_r)N_{tu})-1}}. \tag{4}$$

For crossflow heat exchangers when the fluid of the larger heat capacity rate is mixed, and the fluid of the smaller heat capacity rate is unmixed:

$$\Phi = 2\sqrt{\frac{1-\exp[-(1-\exp(-N_{tu}))C_r]}{C_r N_{tu}[1+\exp(-N_{tu})][1+\exp(1-\exp(-N_{tu})C_r)]}}. \tag{5}$$

For crossflow heat exchangers when the fluid of the smaller heat capacity rate is mixed, and the fluid of the larger heat capacity rate is unmixed:

$$\begin{aligned}\Phi = {} & 2[1-\exp(-(1-\exp(-C_r N_{tu}))/C_r)] \\ & /\{N_{tu}[1+\exp(-C_r N_{tu})] \\ & \times [1-\exp(-2(1-\exp(-C_r N_{tu}))/C_r)]\}^{1/2}.\end{aligned} \tag{6}$$

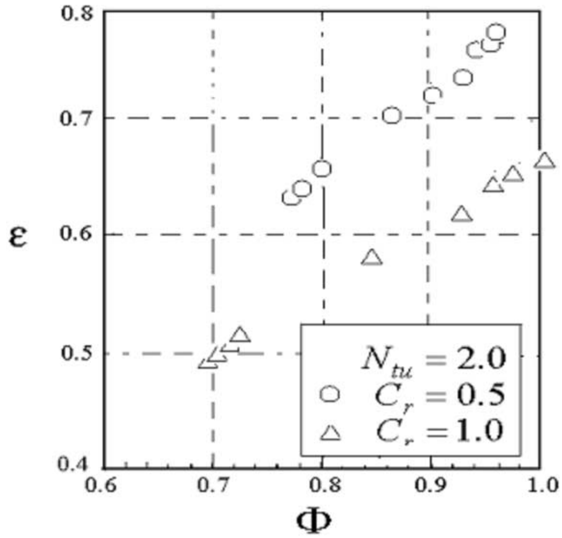


Fig. 3. Effectiveness versus uniformity factor in several counterflow, parallel-flow and crossflow heat exchangers.

Here $C_r = C_{\min}/C_{\max}$ is the heat capacity rate ratio. From the above four equations, it is obvious that uniformity factor of TDF, Φ, is only a function of N_{tu} and C_r. We also know that the effectiveness (ε) of heat exchangers is also only the function N_{tu} and C_r, and therefore, the map of Φ to ε is one-to-one correspondence.

In general, it is difficult to obtain an analytical expression for the uniformity factor of the TDF of a heat exchanger except for several simple heat exchangers discussed above. The uniformity factor of TDF for another nine kinds of heat exchangers which are widely used in engineering have been numerically analyzed, and the results of the effectiveness, ε, versus uniformity factor, Φ, for all 13 kinds of heat exchangers are shown in Fig. 3 with $N_{tu} = 2.0$ [16,17]. It can be seen from Fig. 3 that the more uniform the TDF in the heat exchanger, the higher the effectiveness in different kinds of heat exchangers for given N_{tu} and C_r. The highest/lowest effectiveness corresponds to the largest/smallest uniformity factors of TDF. This indicates that the difference of effectiveness between various heat exchanger configurations with constant N_{tu} and C_r results from the different uniformity of TDF.

2.3. Uniformity principle of TDF in heat exchanger

Based on the study of the relationship between the effectiveness, ε, and the uniformity factor of TDF, Φ, the uniformity principle of temperature difference field in heat exchanger should be: the more uniform the temperature difference field, the higher the effectiveness of the heat exchanger for a given number of heat transfer units, N_{tu}, and a fixed heat capacity rate ratio, C_r.

3. Verification of the uniformity principle of TDF

Actually, the uniformity principle of TDF in heat exchangers is difficult to be proved using a general mathematical approach because it involves to the optimization of partial differential equations with some constraints, which is still an unsolved problem. Asymptotic and experimental methods are used, here, to check the uniformity principle of TDF.

3.1. Asymptotic verification of uniformity principle of TDF

In the analysis and design of a heat exchanger, one usually assumes that the overall heat transfer coefficient is constant, and treats the number of heat transfer units N_{tu} as a bulk parameter, and therefore, then the effectiveness of the same kind of heat exchanger would be fixed for given N_{tu} and C_r. For a two-dimensional crossflow heat exchanger, the numerical results of TDF are plotted in Fig. 4, from which the temperature difference on the diagonal and in the inlet region is larger than those on both its sides and in the outlet region. The uniformity of TDF in the heat exchanger, therefore, can be changed through the rearrangement of the heat transfer area to improve the performance of heat exchangers. The log-mean temperature difference, ΔT_m, can be evaluated as usual from the discrete temperature difference by the expression [17]:

$$\Delta T_m = \frac{1}{MN}\sum_{i=1}^{M}\sum_{j=1}^{N} C_a(i,j)\Delta T(i,j), \tag{7}$$

where $\Delta T(i,j)$ is the local temperature difference between the hot and cold fluids in the subelement; M and N are the numbers of subelements in the length and width directions; and $C_a(i,j)$ is the dimensionless distribution of heat transfer area. The log-mean temperature difference equals to the arithmetic mean local temperature difference only when the heat transfer area is uniformly distributed, and $C_a(i,j)$ equals to unity. The log-mean temperature difference, however, equals to the arithmetic mean local temperature difference with a weight function of $C_a(i,j)$ when the heat transfer area is non-uniformly distributed. Based on Eq. (7), the recurrence formula is derived for improving the uniformity of TDF in the heat exchanger through the rearrangement of the heat transfer area as [17]:

$$C_{a,k+1}(i,j) = C_{a,k}(i,j)\frac{\Delta T_k(i,j)}{\Delta T_{m,k}}, \tag{8}$$

where $C_{a,k}(i,j)$ and $C_{a,k+1}(i,j)$ are the distributions of heat transfer area at step k and step $(k+1)$; and $\Delta T_{m,k}$ and $\Delta T_k(i,j)$ are the log-mean temperature difference and the local temperature difference at step k, respectively. The distribution of heat transfer area in the heat exchanger, $C_{a,0}(i,j)$, should be unity when the heat transfer area is uniformly distributed, and therefore,

$$C_{a,l}(i,j) = \frac{\Delta T_0(i,j)}{\Delta T_{m,0}}. \tag{9}$$

It is obvious that the log-mean temperature difference should be between the maximum local temperature difference and minimum local temperature difference, hence the physical essence of Eq. (8) indicates that more heat transfer area is distributed in the region of the larger local temperature difference while less heat transfer area is distributed in the region of the smaller one. One can see from Fig. 4 that more heat transfer area should be distributed in the diagonal region and in the inlet region of a crossflow heat exchanger, and less heat transfer area in the remaining region.

Combining Eq. (8) with the equations of energy conservation and known heat transfer rate, the uniformity factor of TDF, Φ, and the effectiveness, ε, for every recursion can be obtained numerically, and are shown in Fig. 5, where τ is the percentage increase in effectiveness compared to the case of uniform area distribution. Fig. 5 indicates that: (a) the TDF could be uniform and the effectiveness is consequently increased by rearranging heat transfer area, (b) when C_r is unity the TDF of the

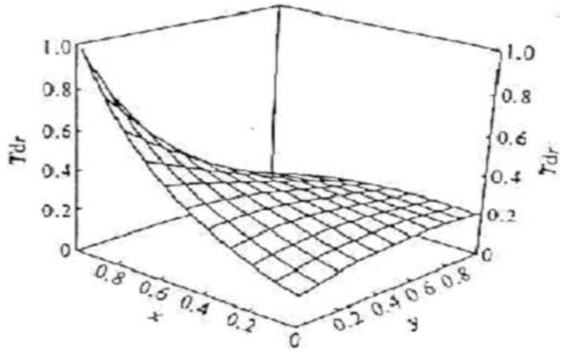


Fig. 4. TDF in crossflow heat exchanger for the uniform distribution of heat exchanger area.

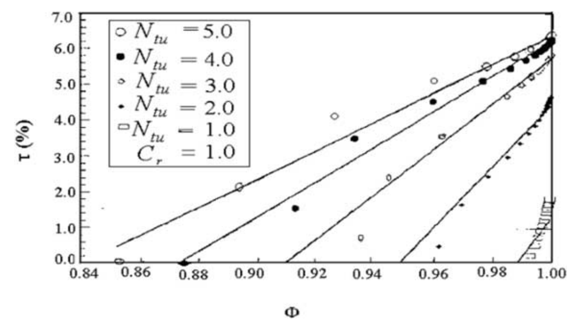


Fig. 5. Percentage increase in effectiveness, τ, versus uniformity factor of TDF, Φ, for the redistribution of heat transfer area in crossflow heat exchanger ($C_r = 1.0$).

crossflow heat exchanger could be fully uniform in an ideal distribution of heat transfer area, and the effectiveness of the crossflow heat exchanger becomes identical to that of the counterflow heat exchanger.

3.2. *Consideration of minimum irreversibility of heat transfer*

The effectiveness of a heat exchanger, ε, is well known as

$$\varepsilon = \frac{Q_{\mathrm{real}}}{Q_{\max}} = \frac{\max((T_{\mathrm{i}} - T_{\mathrm{o}}), (t_{\mathrm{o}} - t_{\mathrm{i}}))}{T_{\mathrm{i}} - t_{\mathrm{i}}}, \tag{10}$$

where Q_{real} is the actual heat transfer rate in the heat exchanger, and $Q_{\max}$ is the maximum possible heat transfer rate.

From the second law of thermodynamics, the irreversible process results in the decrease of the quality of energy. In a heat exchanger, the entropy generation through heat transfer leads to a decrease in the quality of energy. Hence it can be followed that, the more the loss of the quality of energy in a heat exchanger, the smaller the effectiveness.

The parallel-flow or counterflow heat exchanger was regarded as an open system in [2], and the entropy generation in these heat exchangers can be evaluated from the inlet and outlet temperatures of fluids by:

$$S_{\mathrm{gen}} = (\dot{m}c_{\mathrm{p}})_{\mathrm{c}} \ln\frac{t_{\mathrm{o}}}{t_{\mathrm{i}}} + (\dot{m}c_{\mathrm{p}})_{\mathrm{h}} \ln\frac{T_{\mathrm{o}}}{T_{\mathrm{i}}}, \tag{11}$$

where S_{gen} is the entropy generation and $\dot{m}c_{\mathrm{p}}$ is the heat capacity rate. Combining Eqs. (10) and (11), the entropy generation can be expressed by the inlet parameters and effectiveness as

$$S_{\mathrm{gen}} = (\dot{m}c_{\mathrm{p}})_{\mathrm{c}} \ln\left(1 + \varepsilon\left(\frac{t_{\mathrm{i}}}{t_{\mathrm{i}}} - 1\right)\right) + (\dot{m}c_{\mathrm{p}})_{\mathrm{h}} \ln\left(1 - C_{\mathrm{r}}\varepsilon\left(1 - \frac{t_{\mathrm{i}}}{T_{\mathrm{i}}}\right)\right). \tag{12}$$

Since it is more reasonable to evaluate the irreversibility due to heat transfer rate in a heat exchanger using the entropy generation per unit heat transfer rate, a dimensionless parameter, called the number of entropy generation, is defined [12] as

$$N_{\mathrm{s}} = \frac{S_{\mathrm{gen}}}{(\dot{m}c_{\mathrm{p}})_{\mathrm{h}}\varepsilon}. \tag{13}$$

Thus, N_{s} is the entropy generation per unit heat transfer rate in the heat exchanger.

The numerical results of entropy generation for different heat exchangers are plotted in Fig. 6, where $R_{\mathrm{t}} = t_{\mathrm{i}}/T_{\mathrm{i}}$. It can be seen from Fig. 6 that the entropy generation of the counterflow heat exchanger is the smallest as compared to those of crossflow and parallel-flow heat exchangers for a given N_{tu} and C_{r}. For crossflow heat exchanger with varying heat transfer area distribution, a smaller N_{s} corresponds to a higher effectiveness. The numerical results for the dimensionless entropy generation and the uniformity factor of TDF are plotted in Fig. 7 for the crossflow heat exchanger with different area distributions. This shows that the more uniform the TDF, the smaller the dimensionless entropy generation. In the extreme case of the TDF being entirely uniform ($\Phi = 1.0$), the dimensionless en-

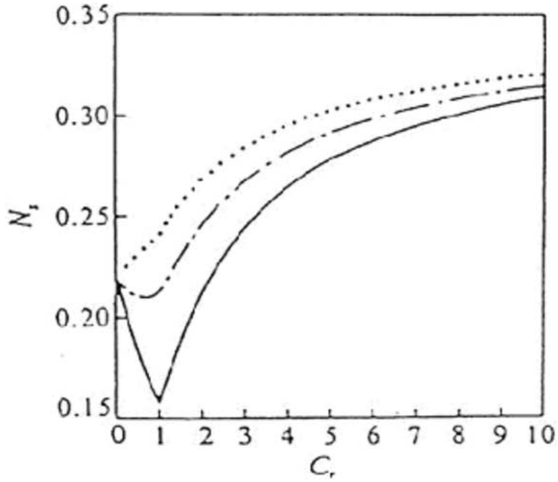


Fig. 6. Entropy generation of several kinds of heat exchanger $R_{\mathrm{t}} = 0.5$, $N_{\mathrm{tu}} = 2.0$: (——) counterflow, (–·–·) two-side crossflow, (- - - -) parallel-flow.

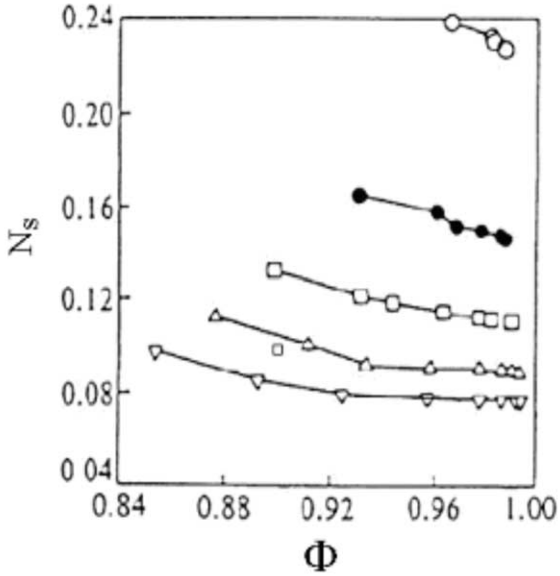


Fig. 7. Entropy generation versus the uniformity of the TDF in the heat exchanger: (○) $N_{\mathrm{tu}} = 1.0$, (●) $N_{\mathrm{tu}} = 2.0$, (□) $N_{\mathrm{tu}} = 3.0$, (△) $N_{\mathrm{tu}} = 4.0$, (▽) $N_{\mathrm{tu}} = 5.0$, $C_{\mathrm{r}} = 1.0$.

tropy generation is the smallest, which agrees with that of a counterflow heat exchanger. That is, the uniformity principle of TDF for a heat exchanger design, which is theoretically based on the second law of thermodynamics, is valid and effective.

3.3. Experimental check

So far we already demonstrated through the mathematical and numerical analyses that the more uniform the TDF, the higher the effectiveness of heat exchanger will be. However, it is necessary to investigate this principle experimentally [17].

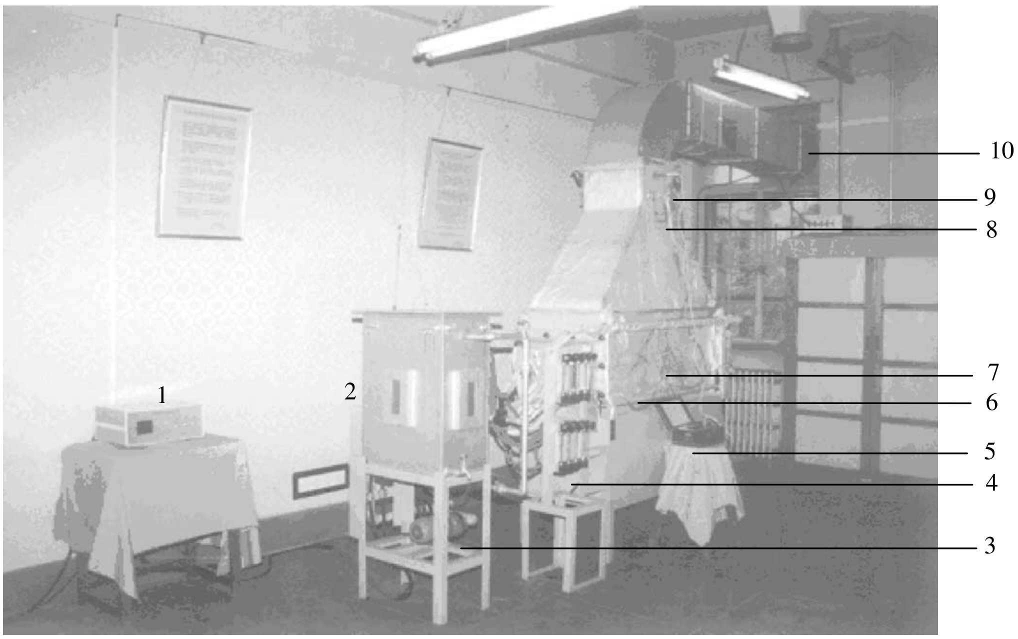
It is important for our experiment to operate with a high number of heat transfer units while using a small heat transfer area. The definition of number of heat transfer units is

$$N_{\mathrm{tu}} = \frac{UA}{(\dot{m}c_{\mathrm{p}})_{\min}}, \tag{14}$$

where U is the overall heat transfer coefficient, A is the overall heat transfer area, $\dot{m}$ is the mass flow rate, c_{p} is the specific heat at constant pressure. The heat transfer coefficient could not be very large because of the structure of the heat exchanger; the flow velocity and physical properties of the fluids typically limit this value. In this study, a small mass flow rate was used to get a high number of transfer units.

3.3.1. Experimental apparatus

An air-cooled module heat exchanger is selected as the basic configuration for our experimental study. A photograph of the heat exchanger experimental setup is shown in Fig. 8. This includes mainly the water supply system, the wind tunnel facility, the operation parameter control and measurements system, and the heat exchanger module. The heat exchanger module consisted of 28 aluminum finned tubes with length of 1.2 m and 433 fins per meter. Plastic tubes were used to connect the ends of the heat exchanger fined tubes to enable the number of passes and the flow distribution to be changed easily. Forty eight calibrated copper–constantan thermocouples were used to measure the temperatures of the air within the heat exchanger and of the water in the inlet and outlet sections of every finned tube. The inlet and outlet temperatures of air were measured by two thermopiles in order to get more accurate air temperatures in the big wind tunnel sections. The water supplied to the heat exchanger was heated in a tank, and its temperature was controlled by a computer to maintain constant inlet water temperature for the heat exchanger. All temperatures were measured with accuracy of ± 0.1 °C.



1— heating controller; 2—water tank; 3—pump; 4—flowmeter;
5—hot-bulb anemometer; 6—finned tube heat exchangers; 7—thermocouple;
8—thermoelectric pile(outlet of induced-air system); 9—probe of hot-bulb anemometer; 10—fan.

Fig. 8. Photograph of heat exchanger experimental setup.

The water flow rate was measured by a calibrated flow meter with accuracy of 0.4 g/s, and the air-flow rate was measured by a calibrated hot-bulb anemometer.

The heat exchanger setup, which is shown in Fig. 9, was tested to obtain the relationship between the effectiveness and the uniformity factor of TDF. The conventional configuration of heat exchanger used in industry is shown in Figs. 9(a). This configuration is expected to exhibit the least uniform TDF and effectiveness. The configurations shown in Figs. 9(b)–(d), which are improved by paying attention to the uniform principle of TDF, were expected to increase the uniformity of TDF and effectiveness gradually.

3.3.2. Experimental results and discussion

The definition of N_{tu} was given by Eq. (14) and the overall heat transfer coefficient U is expressed by the following equation:

$$\frac{1}{UA}=\frac{1}{(\eta_c hA)_c}+\frac{R''_{f,c}}{(\eta_o A)_c}+R_w+\frac{R''_{f,h}}{(\eta_o A)_h}+\frac{1}{(\eta_o hA)_h}. \qquad (15)$$

The definition of uniformity factor of TDF was given by Eq. (2). The air temperatures were measured directly by thermocouples, and the water temperatures at the corresponding measured points of the air were calculated by an energy balance equation from the inlet and outlet temperatures of water in every end of the finned tubes. The energy balance was evaluated and was found to be $100\% \pm 5\%$.

Fig. 10 shows the experimental results for four arrangements of heat exchanger with $N_{tu}=2.0$ and $C_r=1.0$, $C_r=0.75$, $C_r=0.5$. In Fig. 10, hollow points are the experimental data and solid points are the theoretical data. From Fig. 10, it is seen that the experimental results have the same tendency as the theoretical data. The experimental results of Fig. 10 clearly demonstrate that flow distribution can be used to control the TDF of a heat exchanger, and the more uniform the TDF, the higher the effectiveness of the heat exchanger for given N_{tu} and C_r.

4. Applications of uniformity principle of TDF

The uniformity principle of TDF in heat exchanger design as developed in this paper presents a new path of heat exchanger enhancement. The effectiveness of crossflow heat exchangers can be upgraded via improvement of the uniformity of TDF by the following two ways.

Redistribution of heat transfer area: As shown in Section 3, redistribution of heat transfer area can improve the uniformity of TDF and consequently upgrade the heat transfer performance of crossflow heat exchangers. It could be accomplished by changing the fin

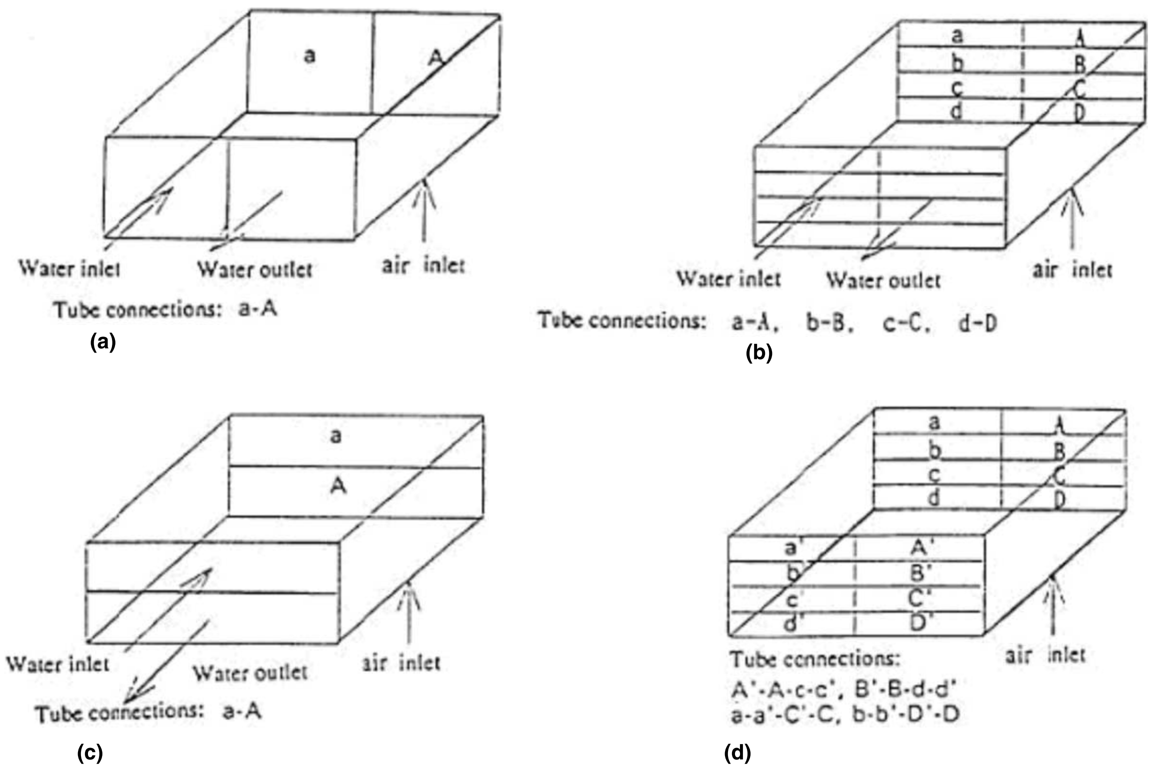


Fig. 9. 3-D views of flow arrangement.

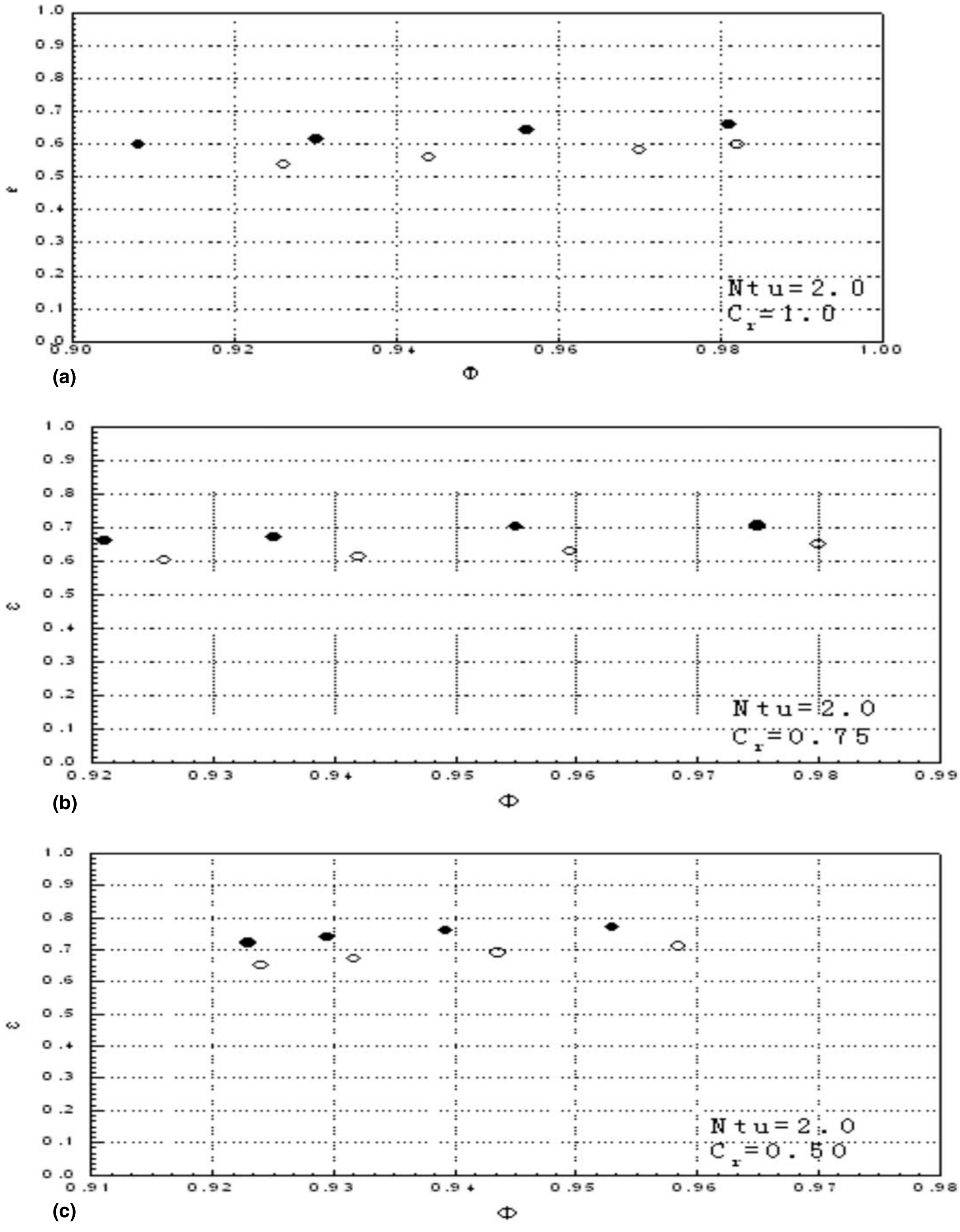


Fig. 10. Top and side views of flow arrangement.

pitch continuously, but it is easy to divide an exchanger into four regions with two fin pitches [17,18].

Rearrangement of the connections between tubes: For multi-path crossflow heat exchanger, the uniformity of TDF can be improved through the rearrangement of the connections between tubes, see Fig. 9 in this paper and [8,11].

5. Conclusions

The importance of the uniformity principle of TDF in heat exchanger was demonstrated in this paper. It states that the more uniform the TDF in a heat exchanger, the higher the heat transfer effectiveness will be for a given N_{tu} and C_r. Analytical and experimental results confirm this statement. A second law thermodynamic analysis was also used in this evaluation. It is clear that the uniformity principle of TDF provides a new way to design crossflow heat exchangers to enhance their performance. It has advantage over the conventional methods for performance enhancement because there is no associated increase in pressure drop.

The rearrangement of heat transfer area and flow distribution can improve the uniformity of TDF in a heat exchanger, and consequently upgrade the thermal performance of crossflow heat exchangers. In essence, the log-mean temperature difference is increased by improving the uniformity of TDF of crossflow heat exchangers. The heat transfer enhancement via improvement of the uniformity of TDF is more effective for crossflow heat exchangers with large N_{tu} and C_r around unity.

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