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Thermal Characteristics of Rectangular Channels with Inline/Staggered Notched Baffles

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Abstract. Thermal characteristics of baffled rectangular channel has been investigated experimentally over the range of Reynolds numbers of 6000 to 24,000. The aspect ratio of rectangular channels was 3.75. The inner surfaces of the lower plates of channels were installed with the notched baffles in inline and staggered arrangements. The notch height-to-baffle height ratio (a/e), space between adjacent notches (b), and roughness pitch ratio (P/e) were fixed at 0.125, 10, and 8, respectively. Local Nusselt number, average Nusselt number, friction factor, and thermal performance factor results are reported and discussed. The results of a smooth channel and the channel with typical baffles (TB) are also given for assessment. Experimental results suggested that the application of the staggered notched baffles (S-NB), inline notched baffles (I-NB), and typical baffles (TB) resulted in the increases of Nusselt numbers by approximately 48.0-66.6%, 44.3-63.1%, and 43.9 - 62.8%, respectively as compared to those of the smooth channel (SC). At a given Reynolds number, the notched baffles with staggered arrangement caused considerably greater Nusselt numbers accompanied by slightly higher friction losses than the ones with inline arrangement. Accordingly, the staggered notched baffles yielded superior thermal performance factor (*TPF*) to the inline ones. The greatest *TPF* of 1.28 was found by utilizing the staggered notched baffles at Re = 6000.

Keywords: Inline and staggered arrangements, Notched baffles, Thermal performance.

1. Introduction

There are numerous heat transfer enhancement techniques applied in thermal systems such as using inserts to promote turbulence or to generate swirl/vortex flow, modifying heat transfer surface to enlarge heat transfer areas and intensify turbulence, and utilizing nanofluids to enhance fluid conductivity and thus heat transfer by the working fluids. Among the techniques, surface modification through artificial roughness is one of the most effective ways to increase turbulence in air channels employed in heat exchangers, vortex combustors, and solar air heaters. [1]. Yang *et al*. [2] investigated the pressure loss and heat transfer characteristics of a square channel with opposing sides roughened by ribs with symmetric and staggered arrangements. Experiments encompassed rib height to the height of ribbed channel (*e/H*) ratios of 0.2 and 0.33, rib spacing to height ratio (*S/e*) ranged from 5 to 15, and Reynolds numbers from 1400 to 9000. At similar operating conditions, the symmetric ribs gave higher heat transfer coefficients and also caused higher pressure losses than the staggered ones. The maximum heat transfer coefficient was obtained at a moderate rib spacing to height ratio (*S/e* = 10). Habet *et al*. [3] installed inline and staggered baffles having perforation ratios varied from β=0% (solid baffle) to 40% for enhancing heat transfer of a rectangular channel for $12{,}000 \leq Re \leq 32{,}000$. At comparable conditions, the staggered baffles caused a lower friction loss penalty than the inline ones. At larger perforation ratios (β=20-40%), the staggered baffles outperformed the inline ones in heat transfer augmentation. For both inline and staggered baffles, thermal performance increased with decreasing perforation ratio. The largest *TPF* was achieved by the use of the staggered baffles with β of 0% (solid baffles) at *Re*=12,000. On the other hand, the minimum *TPF* was found in the case of the inline baffles with β of 30% at *Re*=32,000. Habet *et al*. [3] also investigated the effect of tilting angles (degree θ= 0º, 30º, 45º, and 60º) and perforation ratios (from 10% to 40%) of baffles heat transfer and flow resistance in a rectangular channel with an aspect ratio of 3:1. Manifestly, baffles with θ = 0º and β=10% yielded the highest heat transfer enhancement and caused the greatest friction loss since the baffles with lower tilting angles and smaller perforation ratios offered better reattachment and recirculating flow on the heating surfaces. In contrast, baffles with θ = 60º and β =40% caused the poorest heat transfer enhancement and the lowest friction factor. The best tradeoff between the enhanced heat transfer and the increased friction loss penalty relating to the maximum TPF was found at tilting angle (θ= 60º), perforation ratio (β=10%), and *Re*=12,000. Promvonge *et al*. [5] numerically investigated flow and heat transfer characteristics in a square channel equipped with 45° baffles. The baffles were installed in tandem and inline on the lower and upper channel walls. The results of the 90° transverse baffles were also reported for comparison. Numerical results revealed that two stream wise twisted vortex (*P*-vortex) flows were generated by the 45° baffles. The P-vortex flows consequently induced impinging flows which significantly promoted heat transfer across the channel. However, the heat transfer augmentation by the inline and staggered baffles are comparable. The optimal baffle height to channel height was 0.2, indicated by the highest *TPF* which was as high as 2.6 (twice as high as that of the 90° transverse baffles). Tanda [6] employed liquid crystal thermography to acquire the detailed distributions of the heat transfer coefficient in the channels installed with transverse continuous, transverse broken and V-shaped broken ribs. Experiments were carried out at attack angles (α) of 45°, 60, and 90°, rib height to channel diameter ratios (*e*/*D*) of 0.09 and 0.15, rib pitch to rib height ratios (*p*/*e*) of 4, 8, and 13.3, rib height to channel height ratios (*e*/*H*) of 0.15 and 0.25. Experimental results suggested that the transverse broken ribs with *p/e* = 4 and 13.3 have the best thermal performance, whereas transverse continuous ribs (again with *p/e* = 4 and 8) provide a little heat transfer augmentation or even a reduction (relative to the reference smooth channel).

The present work aims to extend the scope of the study on the effects of baffle geometry on heat transfer and friction loss characteristics in rectangular channels. The inline and staggered notched baffles were installed on the inner lower surfaces of the channels having an aspect ratio 3.75. The notch height-to-baffle height ratio (*a/e*), a space between adjacent notches (*b*), and a roughness pitch ratio were kept constant at 0.125, 10, and 8, respectively. Typical transverse baffles were also tested as a reference case. The main objective is to determine optimum heat transfer conditions for the Reynolds numbers ranging from 6000 to 24,000.

2. Theoretical Aspects

Nusselt number (Nu) is an important parameter that can contribute to a better heat transfer rate. The dimensionless Nusselt number is defined as

$$Nu = \frac{hD_{\mathrm{h}}}{k} \tag{1}$$

where D_h is the equivalent channel diameter calculated from the cross-sectional area of flow (A) and perimeter (P), and k is the thermal conductivity of the working fluid (Air), h is the convective heat transfer coefficient.

$$D_h = \frac{4A}{2P} = \frac{4(WH)}{2(W+H)} \tag{2}$$

The average heat transfer coefficient (h) under the uniform heat flux condition is calculated from the following equation.

$$h = Q_{conv} / A\left(T_w - T_b\right) \tag{3}$$

The rate of convection heat transfer (Q_{cov}) is evaluated from the experimental data of air as

$$Q_{conv} = Q_{air} = \dot{m}C_p(T_o - T_i) \tag{4}$$

The average bulk air temperature (T_b) can be determined via thermochromic liquid crystal (TLC) sheet colors, whereas the average channel wall temperature (T_w) can be calculated from

$$T_b = \left(T_o - T_i\right)/2 \tag{5}$$

The Reynolds number based on the equivalent diameter (D_h) can be expressed as

$$Re = UD_{\mathrm{h}} / v \tag{6}$$

The friction factor is defined as [8]

$$f = \frac{2}{(L/D_h)} \frac{\Delta P}{\rho U^2)} \tag{7}$$

Finally, the thermal performance factor (TPF) [9] is calculated from the Nusselt number and friction factor ratios as

$$\mathrm{TPF} = \left(\frac{Nu}{Nu_0}\right)\left(\frac{f}{f_0}\right)^{-\frac{1}{3}} \tag{8}$$

3. Experimental Details

3.1 Experimental setup

A rectangular heat exchanger channel with a width of 150 mm, a height of 40 mm (an aspect ratio of 3.75), and a length of 3500 mm was divided into three sections: an entering section (also known as the calm portion), a test section, and an outflow section. The schematic design of an experimental setup is shown in Figure. 1. The channel wall was well-insulated to minimize heat loss. Air was fed into the channel by a 2.2 kW fan. The airflow rate was controlled via an inverter. Airflow rates were measured using an orifice coupled with a digital pressure gauge. A power supply unit supplied heat to the bottom of the channel with a uniform heat flux of 600 W/m^2. Eight RTD Pt100 thermocouples which were located upstream and downstream of the test section were used to collect temperature data to evaluate a bulk temperature. The colors of the TLC sheet, which were captured by a high-resolution digital camera, were used as a guide for evaluating channel wall temperatures. In addition, a digital pressure gauge was used to measure the pressure loss across the test section for calculating the friction factor. All data were recorded at steady state conditions for the Reynolds numbers ranging from 6000 to 24,000.

3.2 Baffle geometries and operating conditions

The configurations of typical transverse baffles and notched baffles with inline and staggered arrays are shown in Fig. 2. The notched baffles were located on the inner surfaces of the lower plates of channels. The notch height-to-baffle height ratio (*a/e*), a space between adjacent notches (*b*), and a roughness pitch ratio were kept constant at 0.125, 10, and 8, respectively.

Figure 3 demonstrates the manner in which the typical transverse baffles are placed. Table 1 presents an overview of the channel, baffle, and operating parameters.

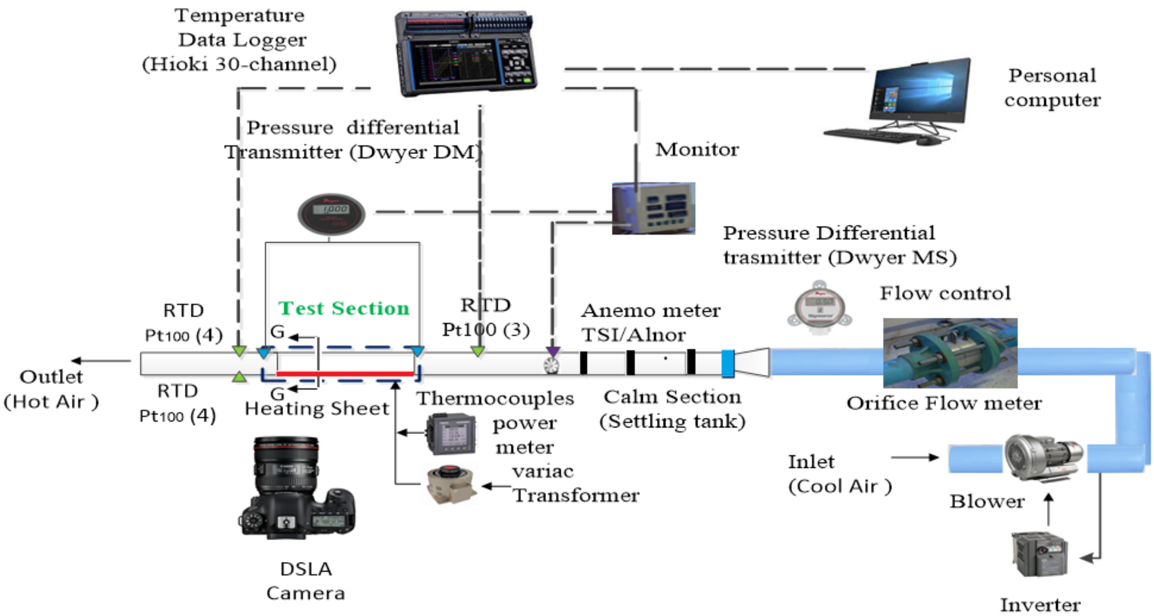


Figure 1. The experimental setup.

Table 1. Geometries of typical transverse baffles and notched baffles with inline and staggered configurations and operating conditions.

Test section	
Channel (Height, Width, Length; $H \times W \times L$)	40 mm ×150 mm ×900 mm
Channel aspect ratio (W/H)	3.75
Baffle material	Polylactic acid plastic (*PLA*)
Space between adjacent notches (b)	10
Roughness pitch ratio (P/e)	8
notch height-to-baffle height ratio (a/e)	0.125
Working fluid	Air
Reynolds number	6,000-24,000
Prandtl number	0.7

(a) transverse baffles (TB) (b) inline notched baffles (c) staggered notched baffles

Figure 2. Configurations of typical transverse baffles and notched baffles with inline and staggered arrays.

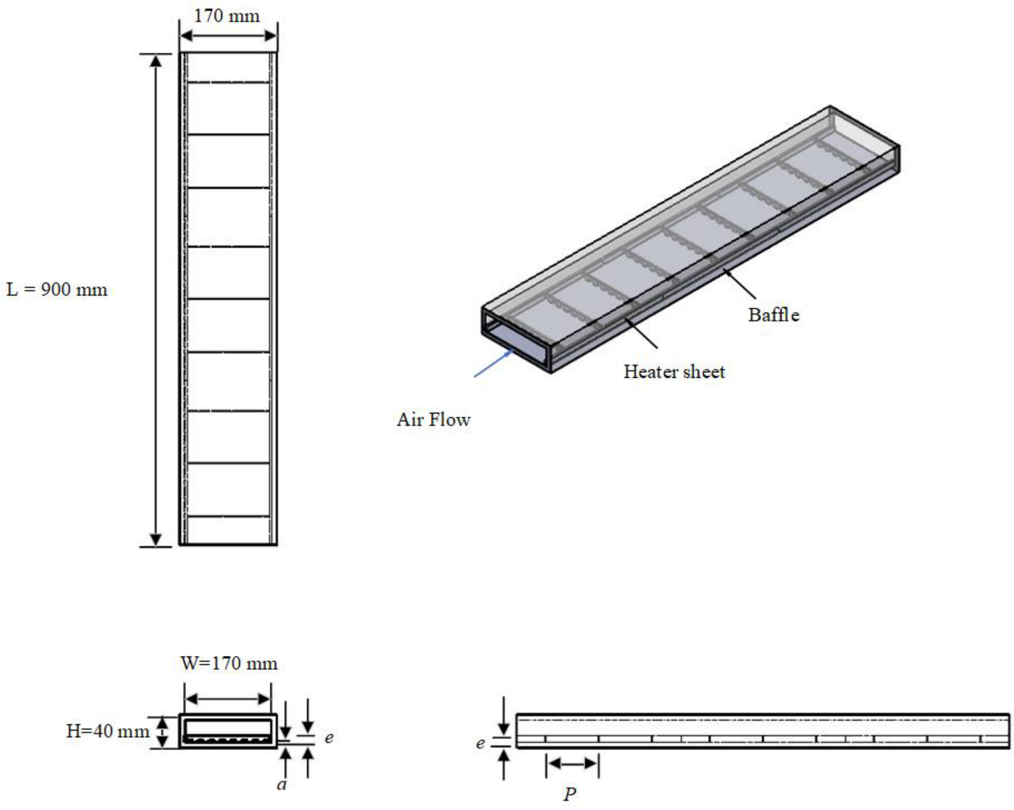


Figure 3. The arrangement of baffles.

4. Experimental Results and Discussion

In order to verify the reliability of the experimental setup and procedure, heat transfer and friction loss results from the current smooth channel are compared to data from well-known correlations which are Dittus-Boelter and Gnielinski correlations (Nusselt number) and Petukhov and Blasius correlations (friction factor). The comparison of the present results and those from the correlations is displayed in Figure 4. Obviously, the data from different sources were in good agreement. The deviations of the current Nusselt numbers and friction factors from the correlations within the ranges of 1.81%, 3.26%, 1.36%, and 9.01%, respectively.

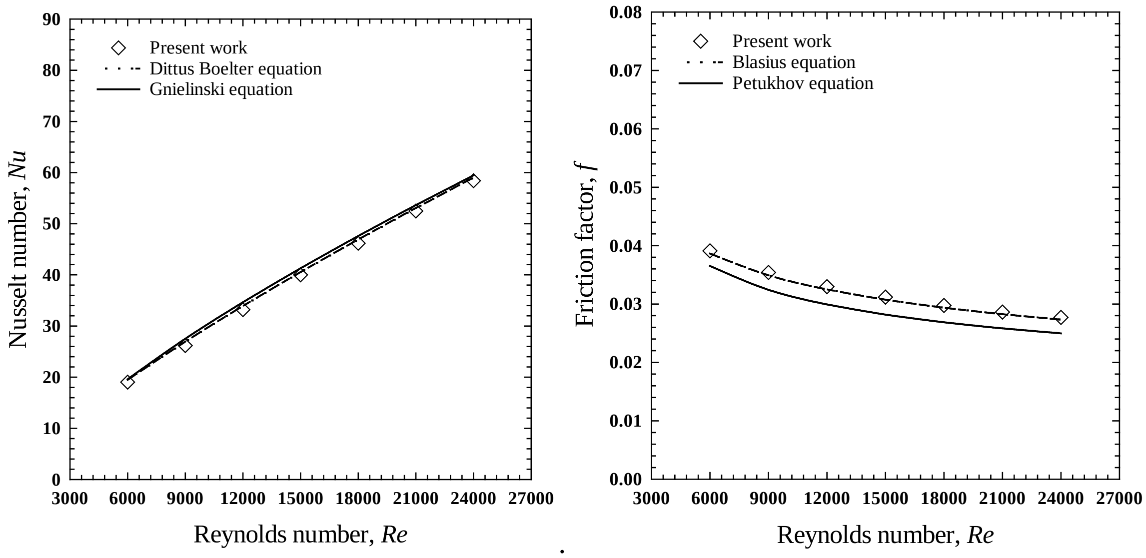


Figure 4. Validation test for the current smooth channel.

4.1 Heat transfer

Figure 5 illustrates the influence of Reynolds number (*Re*) on Nusselt number (*Nu*) and Nusselt number ratio (Nu/Nu_s). The Nusselt numbers of the channel with baffles were consistently greater than those of the smooth channel at a given Reynolds number (all Nu/Nu_s were above unity). Clearly, Nusselt number (*Nu*) increased as Reynolds number rose because turbulence was promoted. However, Nusselt number ratio dropped with increasing Reynolds number. This may be explained by the fact that the thermal boundary layer in the smooth channel is already thin at a high Reynolds number. Consequently, the presence of baffles possessed an insignificant effect on heat transfer enhancement.

Experimental results also showed that the application of the staggered notched baffles (S-NB), inline notched baffles (I-NB), and typical baffles (TB) resulted in the increases of Nusselt numbers by approximately 48.0-66.6%, 44.3-63.1%, and 43.9 - 62.8%, respectively as compared to those of the smooth channel (SC).

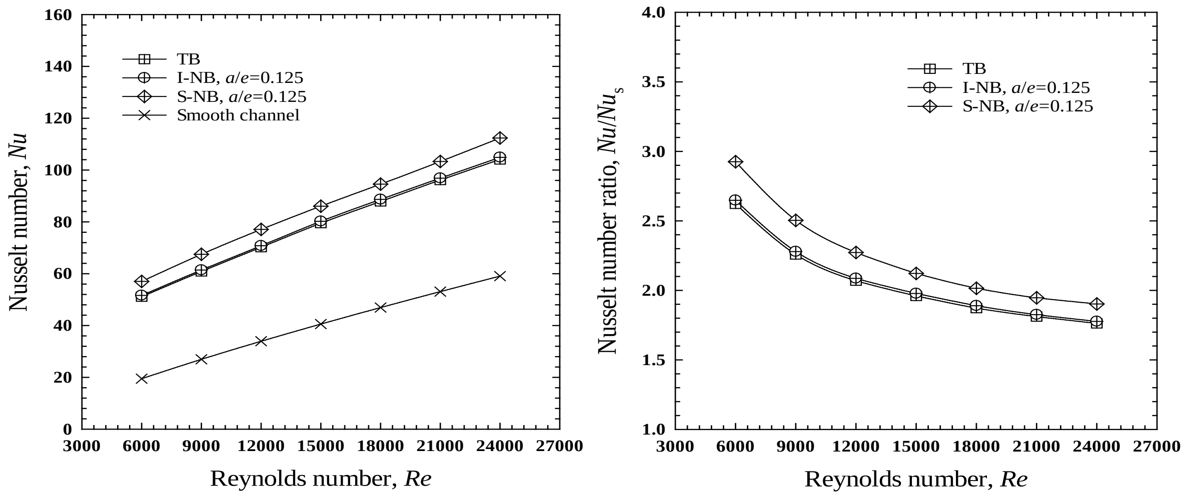
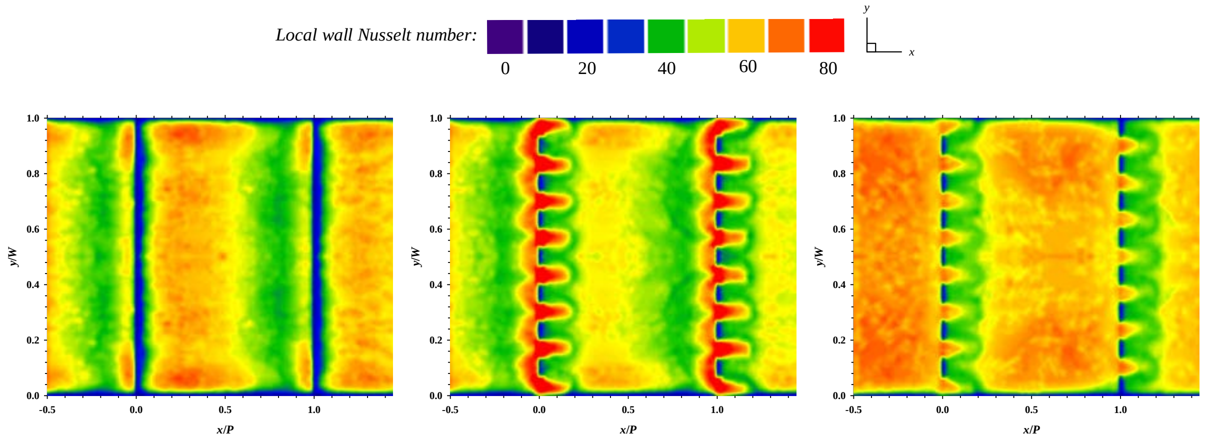


Figure 5. Effects of Reynolds number (*Re*) on Nusselt number (*Nu*) and Nusselt number ratio (Nu/Nu_s).



Typical Baffle (TB), P/e=8 I-NB (a/e)=0.125, P/e=8 S-NB (a/e)=0.125, P/e=8

Figure 6. Nusselt number contours on the surfaces installed with the typical baffle (TB), inline baffle (I-NB) and staggered notched baffles (S-NB) with a/e *of* 0.125, b of 10, P/e of 8, and Reynold number of 6,000.

The details of heat transfer behavior are presented in the form of Nusselt number contours as displayed in Figure 6. Obviously, the staggered notched baffles (S-NB) introduced larger high Nusselt number area than the staggered notched baffles (S-NB) and typical baffles (TB). Relying on the report by Habet *et al.* [3], the staggered perforated baffles showed higher heat transfer augmentation than the inline ones at higher perforation ratios (β=20–40%). Therefore, the presence of the spaces (notches) on the baffles together with the staggered arrangement in the current study may be the synergy factor in heat transfer enhancement. For the examined range, the staggered notched baffles (S-NB) yielded the maximum Nusselt number ratio (Nu/Nu_s) as high as 2.93 at a Reynolds number of 6,000.

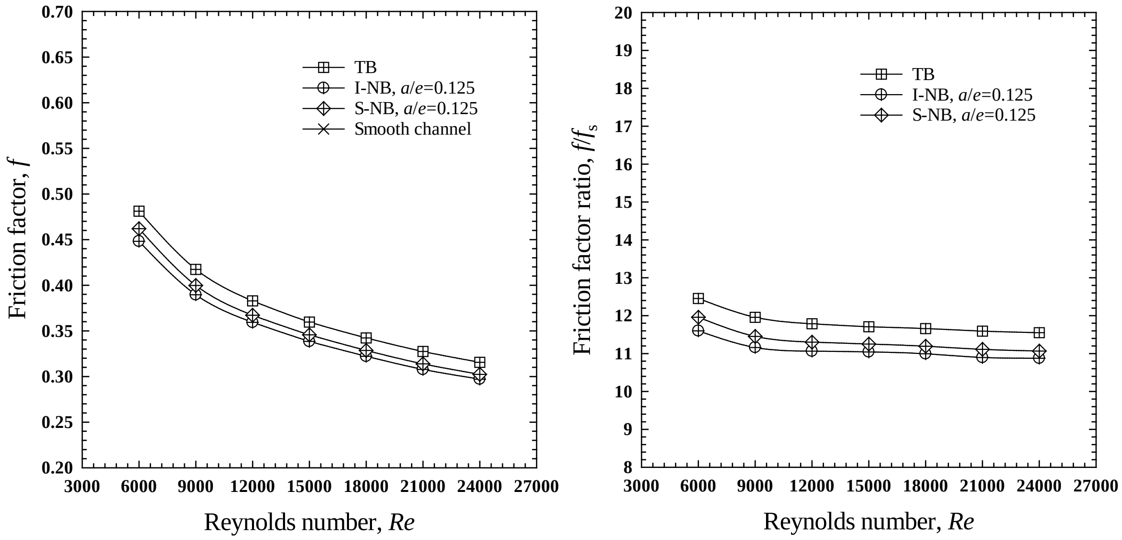


Figure 7. Friction factor & Friction factor ratio VS Reynolds number.

4.2 Friction loss

Figure 7 shows friction factor (f) and friction factor ratio (f/f_s) results at the Reynolds numbers ranging from 6000 to 24,000. Both the average friction factor and friction factor consistently increased as the

Reynolds Number decreased. In general, all channels equipped with baffles showed higher friction factors than the smooth channel. The results can be simply explained that the presence of baffles caused flow disturbance, resulting in higher pressure drop and thus friction loss. At a given Reynolds number, the staggered notched baffles (S-NB) and the inline notched baffle (I-NB) caused lower friction losses than the typical baffles (TB) attributed to the facilitated fluid flow through notches. Friction losses caused by staggered notched baffles (S-NB) were slightly higher than the inline notched baffle (I-NB) by around 1.75-3.09 %.

4.3 Thermal performance factor

Figure 8 shows the relationship between the thermal performance factor (*TPF*) at constant pumping power and Reynolds number. In all cases, thermal performance factors decreased as the Reynolds number rose. The results indicated that the application of the baffles was more favorable for energy saving at lower Reynolds numbers. At an identical Reynolds number, the staggered notched baffles (S-NB) yielded the highest thermal performance factor followed by the inline notched baffle (I-NB) and the typical baffle (TB). The staggered notched baffles (S-NB) outperformed the other baffles primarily attributed to their great heat transfer enhancement with moderate friction losses. In the examined range, the thermal performance factor reached the maximum value of 1.28 times at a Reynolds number of 6000.

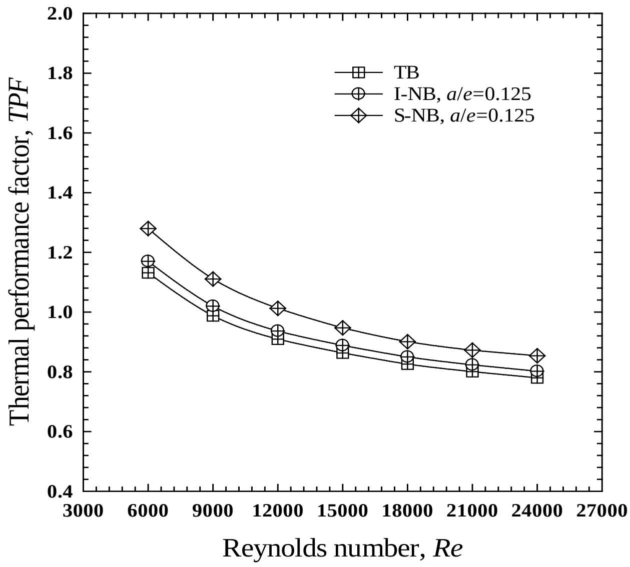


Figure 8. Thermal performance factor versus Reynolds number.

5. Conclusions

The use of the two different type baffles (staggered notched baffles (S-NB), and inline notched baffles (I-NB)) in comparison to the typical baffle (TB) was investigated for Reynolds numbers ranging from 6,000 to 24,000. The major findings are listed below.

- As Reynolds number increased, Nusselt numbers increased while the friction value declined.
- At a given Reynolds number, the staggered notched baffles (S-NB) yielded the highest Nusselt number (up to 2.92 times of the smooth channel) followed by the inline notched baffles (I-NB), and typical baffles (TB).
- The staggered notched baffles (S-NB) caused moderate friction losses which were slightly higher than those caused by the inline notched baffle (I-NB) but lower than those caused by the typical baffles.
- Among the tested baffles, the staggered notched baffles (S-NB) yielded the highest thermal

performance factor of 1.28, primarily attributed to their great heat transfer enhancement with moderate friction losses.

6. References

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