Improving the thermal efficiency in heat exchangers of transport vehicles

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Improving the thermal efficiency in heat exchangers of transport vehicles

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Abstract. The heat exchangers are widely used in the cooling and heating system of transport vehicles. Therefore, enhancement of thermal efficiency of heat exchangers can lead to energy savings of transport vehicles. To increase the efficiency of heat transfer equipment, different methods of enhancement of heat transfer are widely used. However, when methods of enhancement of heat transfer employed, hydraulic resistance also increases. Thus, to evaluate the effectiveness of the method of enhancement of heat transfer, thermal efficiency is used. In this paper, we consider the effect of tube pitch with a different configuration of tube bundles on the thermal efficiency of enhancement of heat transfer by pulsating flows. Results obtained for four typical tube array arrangements (triangular staggered, rotated triangular staggered, rotated square staggered, square inline). The tube pitch ratio of the tube bundle was 1.25, 1.5, and 1.75. Results presented the effect of pulsations in tube bundles on the thermal performance ratio at the same Reynolds numbers and thermal performance ratio at the same power required for pumping the coolant in steady and pulsating flow. It is found that the change in tube pitch has a different effect when the thermal performance ratio was evaluated with the same Reynolds number and at the same power required for pumping. With an increase in tube pitch, the thermal performance ratio at the same Reynolds numbers can either decrease or increase depending on the tube array arrangements and parameters of the pulsating flow. Higher values on the thermal performance ratio at the same Reynolds numbers and at the same power are mainly observed for the square inline array, which may change depending on the tube pitch and parameters of the pulsating flow. The maximum value of the thermal performance ratio at the same Reynolds numbers was 0.623 at the tube pitch of 1.25 and square inline array. The maximum value of the thermal performance ratio at the same power was 1.92 for tube pitch of 1.75 and square inline array.

1. Introduction

Shell and tube heat exchangers are widely used in the cooling and heating system of transport vehicles. Improving the energy efficiency of heat exchangers depends on the methods used for the enhancement of heat transfer [1-2]. One of these methods is externally forced flow pulsation. [3]. Pulsating flow for enhancement of the heat transfer in various heat exchangers has been investigated by many authors. There are a large number of works in which heat transfer and hydrodynamics are studied for pulsating flows in circular pipes [4–7], in channels with grooves [8], single square cylinder [9,10], semi-cylinder [11], tandem cylinders [12], backward-facing step [13,14], rod bundle channel [15] blocks and other obstacles [16]. However, among these works, there are only a few in which pulsating flows in tube bundles are studied [17–20], studies are more often devoted to a single cylinder

[21–26]. Studies of tube bundles are complicated by a large number of their configurations used in heat exchangers [27].

Heat transfer enhancement in tube bundles might be increase to proportional to the amplitude and frequency of the pulsations [28]. However, with the enhancement of heat transfer, hydraulic losses also increase [1,2]. Therefore, any enhancement of heat transfer due to pulsations should be compared to an increase in the capacities required for pumping coolants through a heat exchanger. In this case, it is necessary to find optimal parameters of pulsation with maximum efficiency, which depends not only on the parameters of pulsation but also on the heat transfer geometry.

The effectiveness of the chosen method of enhancement of heat transfer can be estimated using the thermal performance ratio [29] when the increase of heat transfer compared with the increase of hydraulic losses. In previous works [30,31], the thermal-hydraulic efficiency of the inline and staggered tube bundle by the pulsations was evaluated. In [30], the numbers Re <1000, the Prandtl number Pr = 5.5, the product of the dimensionless amplitude, and the Strouhal numbers βSh were in the range from 0.026 to 2.5. In another work [31], the Pr numbers had higher values and ranged from 214 to 363, Re = 100, the ratio of the dimensionless amplitude to the Fourier number β/Fo was inside the range from $1 \cdot 10^{-4}$ to $6 \cdot 10^{-4}$. In both works, it was found that thermo hydraulic efficiency significantly depends on the parameters of pulsation. This paper presents the results of the thermal efficiency of three configurations of staggered tube bundles and one inline tube bundle with different tube pitch in pulsating flows.

2. Thermal efficiency

Heat transfer efficiency evaluated with thermal efficiency [32]

$$E = q / N \tag{1}$$

where q – heat flux, W/m^2; N – pumping power, W/m^2

Thermal performance ratio η_{Re} at equal Reynolds numbers, $\mathrm{Re}_p=\mathrm{Re}_{st}$

$$\eta_{\mathrm{Re}} = \frac{E_p}{E_{st}} = \frac{\mathrm{Nu}_p / f_p}{\mathrm{Nu}_{st} / f_{st}} \tag{2}$$

where E_p, Nu_p, f_p, Re_p, E_{st}, Nu_{st}, f_{st}, Re_{st} – thermal efficiency, Nusselt number, friction factor, and Reynolds number at a pulsating and steady flow.

Thermal performance ratio η_N at equal pumping powers [33] in a pulsating and steady flow, $N_p=N_{st}$

$$\eta_N = \frac{E_p}{E_{st}} = \frac{\mathrm{Nu}_p}{\mathrm{Nu}_{st}} \tag{3}$$

$$\mathrm{Re}_p = \frac{\mathrm{Re}_{st}}{\left(f_p / f_{st}\right)^{1/3}}; \tag{4}$$

$$\mathrm{Nu}_p = A\mathrm{Re}_p^m \cdot \mathrm{Pr}^n \cdot \beta^b \cdot \mathrm{Fo}^c \cdot \psi^d = \tag{5}$$

$$= A \cdot \left(\frac{\mathrm{Re}_{st}}{\left(f_p / f_{st}\right)^{1/3}} \right)^m \cdot \mathrm{Pr}^n \cdot \beta^b \cdot \mathrm{Fo}^c \cdot \psi^d =$$

$$= A \cdot \mathrm{Re}_{st}^m \cdot \mathrm{Pr}^n \cdot \beta^b \cdot \mathrm{Fo}^c \cdot \psi^d \cdot \frac{1}{\left(f_p / f_{st}\right)^{m/3}} = \frac{\mathrm{Nu}_p\left(\mathrm{Re}_{st}\right)}{\left(f_p / f_{st}\right)^{m/3}};$$

$$\eta_N = \frac{\mathrm{Nu}_p(\mathrm{Re}_{st}) / \mathrm{Nu}_{st}(\mathrm{Re}_{st})}{\left(f_p / f_{st}\right)^{m/3}} . \tag{6}$$

Where Pr Prandtl number; relative dimensionless pulsating amplitude $\beta = A/D$, where A is the distance that the fluid particle travels in the opposite direction in the tube bundle by forced pulsating flow; D the diameter of the tube of the bundle; $\mathrm{Fo} = a/(f \cdot D^2)$ Fourier number, where a thermal diffusivity, $f_p = 1/T$ pulsating frequency, where $T=T_1+T_2$ – pulsating period s; $\psi = T_1/T$ duty of pulsating flow (figure 1). An empirical correlation for pulsating flow in equation (5) was obtained in [34]. The parameters of equation (5) are given in table 2.

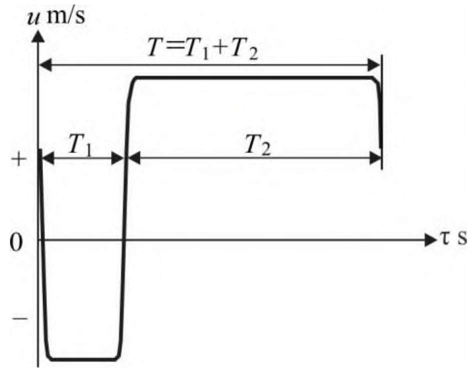

Figure 1. Pulsating velocity.

3. Results

The thermal performance ratio for pulsating flows in tube bundles of various configurations was calculated according to equation (2)–(6) for the data obtained in works [28,34]. The range of considered Reynolds numbers was inside $100 \le \mathrm{Re} \le 1000$, Prandtl number $215 \le \mathrm{Pr} \le 363$, frequency and dimensionless amplitude of pulsations $0.2 \le f \le 0.5$, Hz, $15 \le \beta \le 35$, duty cycle of pulsations $0.25 \le \psi \le 0.5$. The configurations of the considered tube bundles are presented in Figure 2, the geometric parameters of the tube bundles are given in table 1.

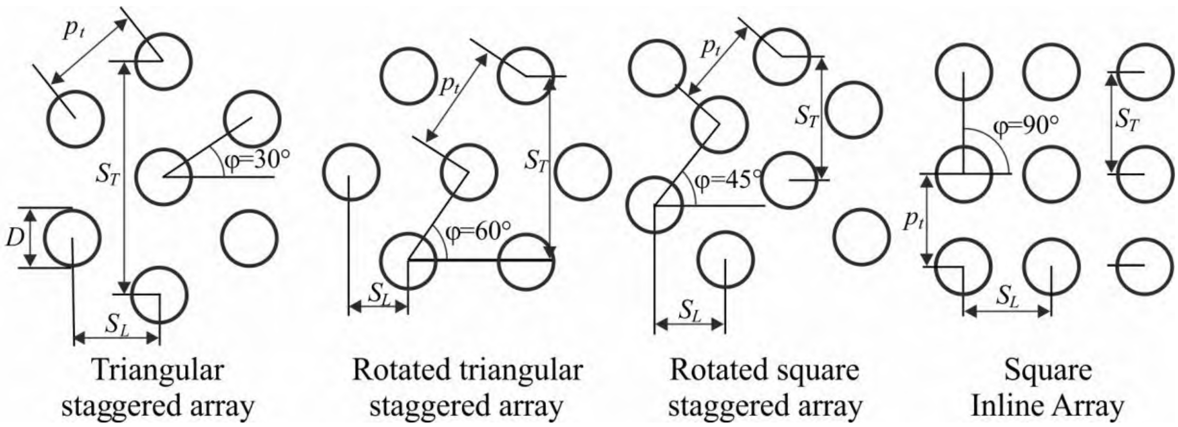

Figure 2. Configuration of tube bundle.

Table 1. Parameters of tube bundles.

φ	30°			45°			60°			90°		
p_t/D	1.25	1.50	1.75	1.25	1.50	1.75	1.25	1.50	1.75	1.25	1.50	1.75
D, m	0.016											

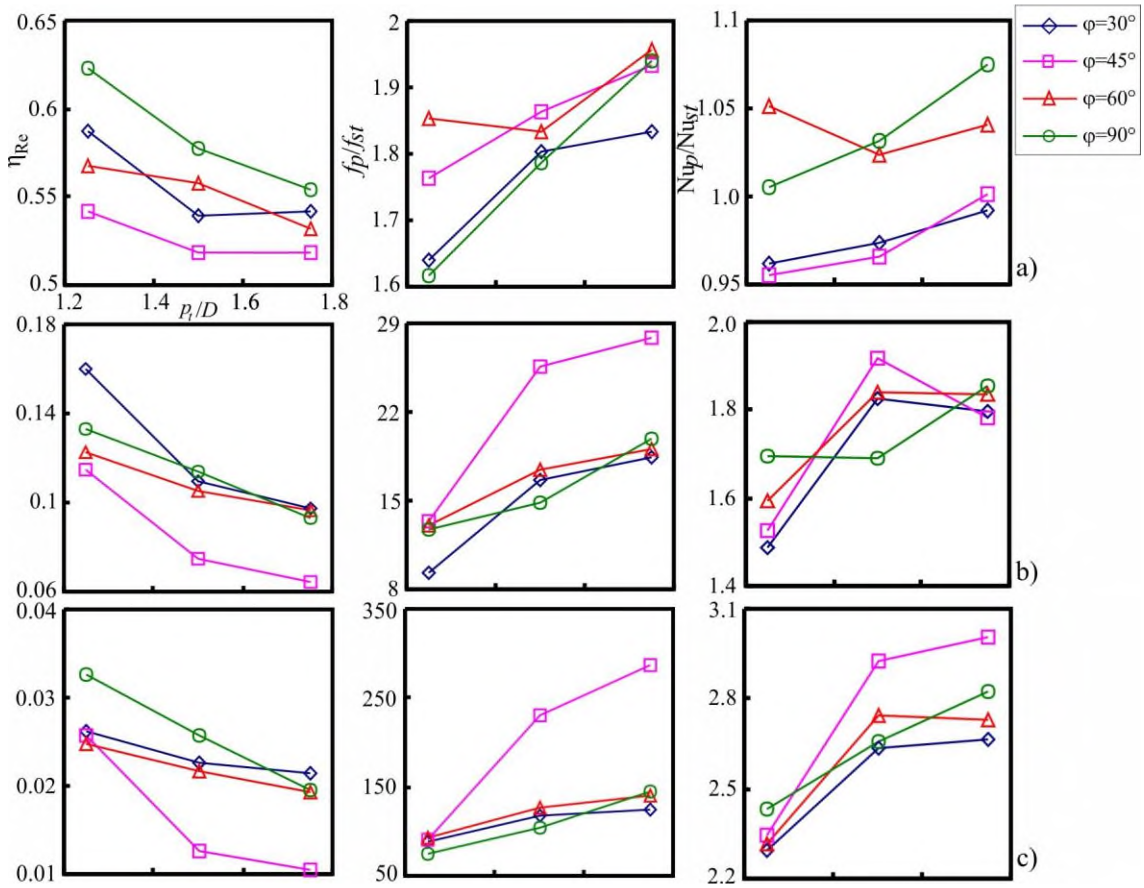

Figure 3. Variation of η_{Re}, f_p/f_{st} and $\mathrm{Nu}_p/\mathrm{Nu}_{st}$ with p_t/D: (a) Re = 1000, Pr = 363, f = 0,2 Hz β = 15, ψ = 0.25; β/(Fo · Re· Pr) = 0.028; (b) Re = 100, Pr = 363, f = 0,312 Hz, β = 35, ψ = 0.25, β/(Fo · Re· Pr) = 1.012; (c) Re = 100, Pr = 214, f = 0,5 Hz, β = 35, ψ = 0.40, β/(Fo · Re· Pr) = 2.741.

Variation the performance ratio η_{Re} at equal Reynolds numbers, $\mathrm{Re}_p=\mathrm{Re}_s$, friction enhancement factor f_p/f_{st}, Nusselt number enhancement factor $\mathrm{Nu}_p/\mathrm{Nu}_{st}$ with tube pinch p_1/D shown respectively in Figure 3. Higher p_1/D results in higher f_p/f_{st} and $\mathrm{Nu}_p/\mathrm{Nu}_{st}$, but the growth friction enhancement factor is ahead of the increase Nusselt number enhancement factor, and therefore the thermal performance ratio η_{Re} decreases. With an increase in β/(Fo · Re· Pr) a significant increase in f_p/f_{st} occurs, which leads to a decrease in η_{Re} (figure 3, c). Figure 3 also shows the effect of tube array arrangements φ on η_{Re}. For the case of β/(Fo · Re· Pr) = 0.028 (figure 3, a) the performance ratio η_{Re} shows a maximum for φ = 90°, which similar for cases of β/(Fo · Re· Pr) = 1.012 (figure 3, b), 2.741 (figure 3, c). When φ = 45°, the η_{Re} shows a minimum for all cases of β/(Fo · Re· Pr).

The thermal performance ratio η_N at equal pumping powers in pulsating and steady flow, $N_p=N_{st}$ with tube pinch p_1/D shown in figure 4. Here tube array arrangements φ are presented for φ of 30°, 45°, 60°, 90°. For the case of β/(Fo · Re· Pr) = 0.028 (figure 4, a) with an increase in p_1/D an increase in η_N occurs for all φ, except φ = 60°. When the dimensionless number β/(Fo · Re· Pr) was 1.012 and

2.741 (figure 4, b, c), with an increase in p_1/D to 1.50 an increase in η_N with a further increase to 1.75 the value of η_N can either increase or decrease depending on φ and parameters of pulsation.

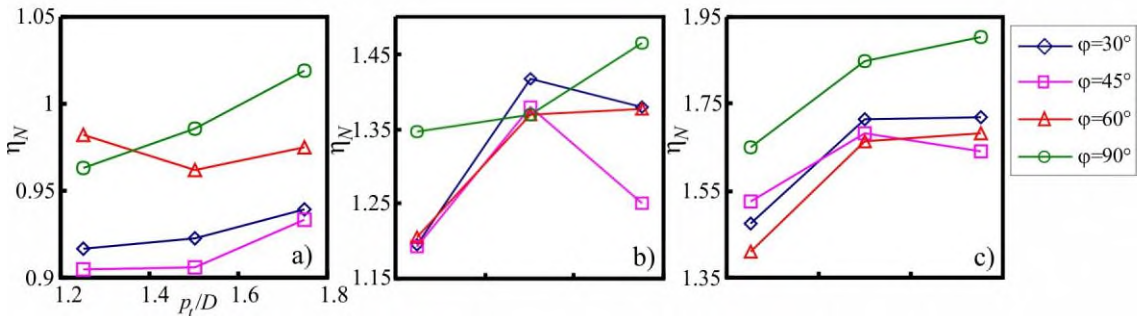

Figure 4. Variation of η_N with p_1/D: (a) Re = 1000, Pr = 363, f = 0,2, β = 15; ψ = 0.25, β/(Fo · Re· Pr) = 0.028; (b) Re = 100, Pr = 363, f = 0,312, β = 35; ψ = 0.25,(β/(Fo · Re· Pr) = 1.012; (c) Re = 100, Pr = 214, f = 0,5, β = 35; ψ = 0.40, β/(Fo · Re· Pr) = 2.741.

Table 2. The parameters of the equation (5) [34].

φ	p_t/D	A	m	n	b	c	d
	1.25	0.901	0.296	0.144	0.198	–0.213	–0.092
30°	1.50	1.050	0.271	0.124	0.200	–0.228	–0.081
	1.75	1.112	0.272	0.132	0.182	–0.226	–0.068
	1.25	1.020	0.285	0.143	0.196	–0.211	–0.097
45°	1.5	0.833	0.305	0.110	0.232	–0.252	–0.079
	1.75	0.656	0.321	0.146	0.170	–0.237	–0.037
	1.25	0.865	0.327	0.146	0.194	–0.214	–0.016
60°	1.50	0.924	0.310	0.133	0.192	–0.238	–0.022
	1.75	1.109	0.294	0.162	0.142	–0.226	–0.006
	1.25	0.885	0.270	0.144	0.186	–0.228	–0.063
90°	1.50	1.218	0.236	0.133	0.167	–0.234	–0.046
	1.75	1.194	0.238	0.128	0.154	–0.255	–0.033

Figure 5 shows the effect of η_{Re} on β/(Fo · Re· Pr) for different p_1/D for ψ = 0.25 and φ of 30°, 45°, 60°, 90 Figure 5 shows that with an increase in the intensity of pulsations η_{Re} decreases for all p_1/D and φ. Tube bundles with a lower p_1/D value are most effective in almost the entire range of the dimensionless number β/(Fo · Re· Pr).

Figure 6 shows the effect of friction enhancement factor f_p/f_{st} and Nusselt number enhancement factor Nu_p/Nu_{st} with the dimensionless number β/(Fo · Re· Pr), which shows that the growth of f_p/f_{st} is more significant than Nu_p/Nu_{st} for all p_1/D.

Figure 7 shows the effect of thermal performance ratio η_N with the dimensionless number β/(Fo · Re· Pr). If the thermal performance ratio η_{Re} decreases with increasing β/(Fo · Re· Pr), then η_N increases. A close spaced tube bundles less effective for enhancement of heat transfer at the equal pumping powers η_N.

Figure 8 shows the effect of η_N and η_{Re} on the duty cycle of pulsations ψ at $p_1/D = 1.5$ и $\varphi = 30°$. It was found that with an increase in the intensity of pulsations, the influence of ψ decreases, it is also seen that asymmetric pulsations are more effective for the enhancement of heat transfer.

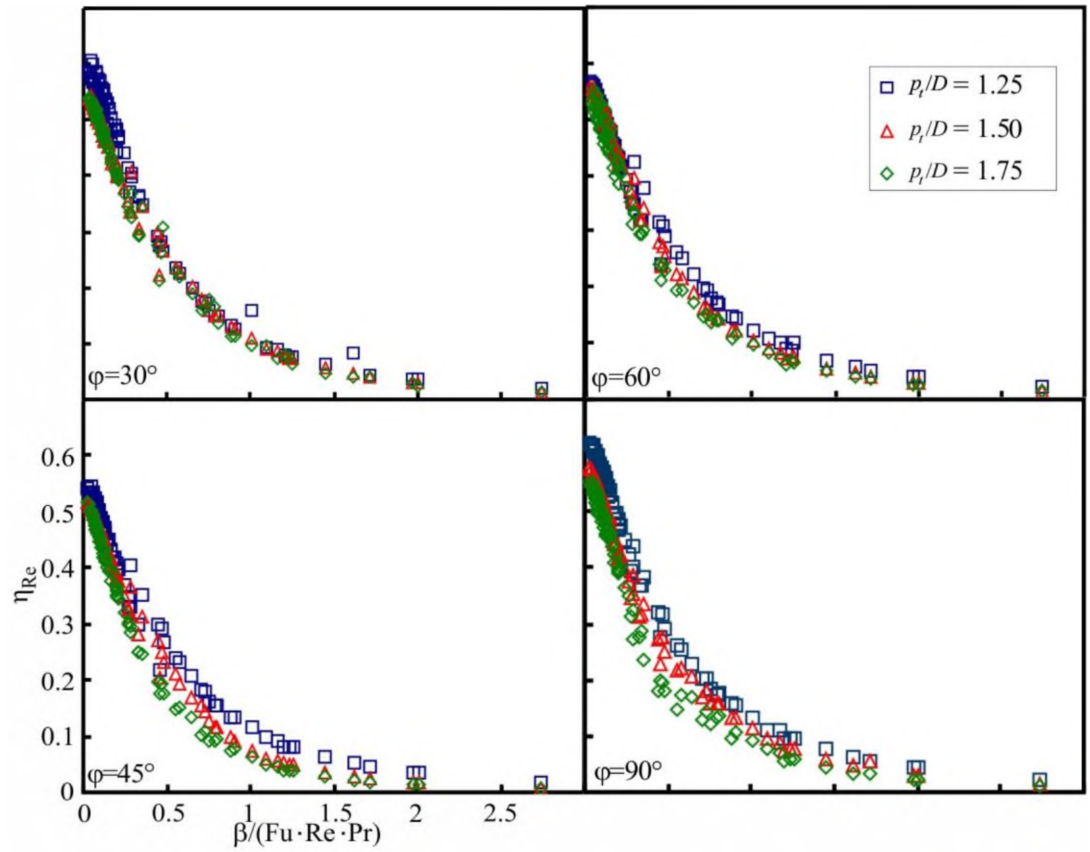

Figure 5. Variation of η_{Re} with β/(Fo · Re· Pr) for ψ = 0.25.

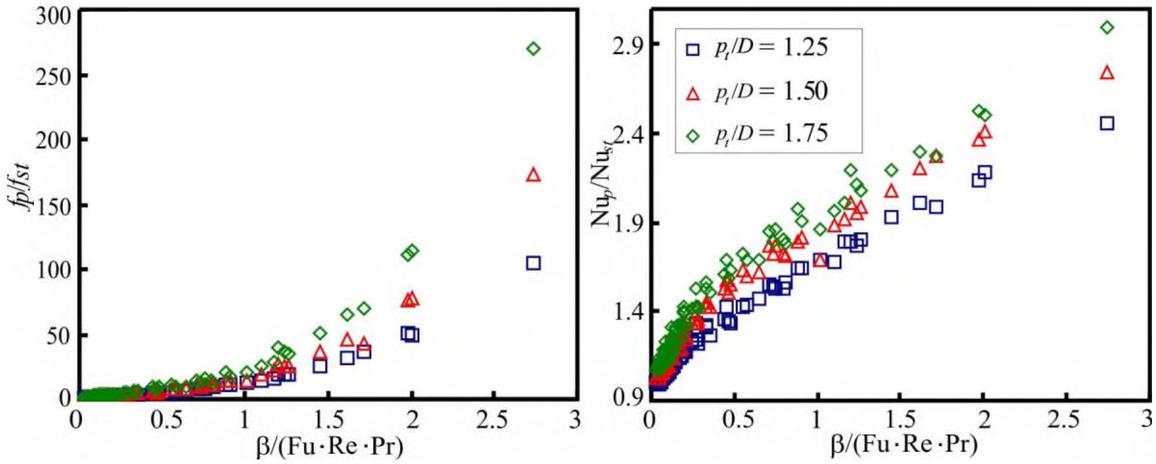

Figure 6. Variation of f_p/f_{st} and Nu_p/Nu_{st} with β/(Fo · Re· Pr) for ψ = 0.25; φ=90°.

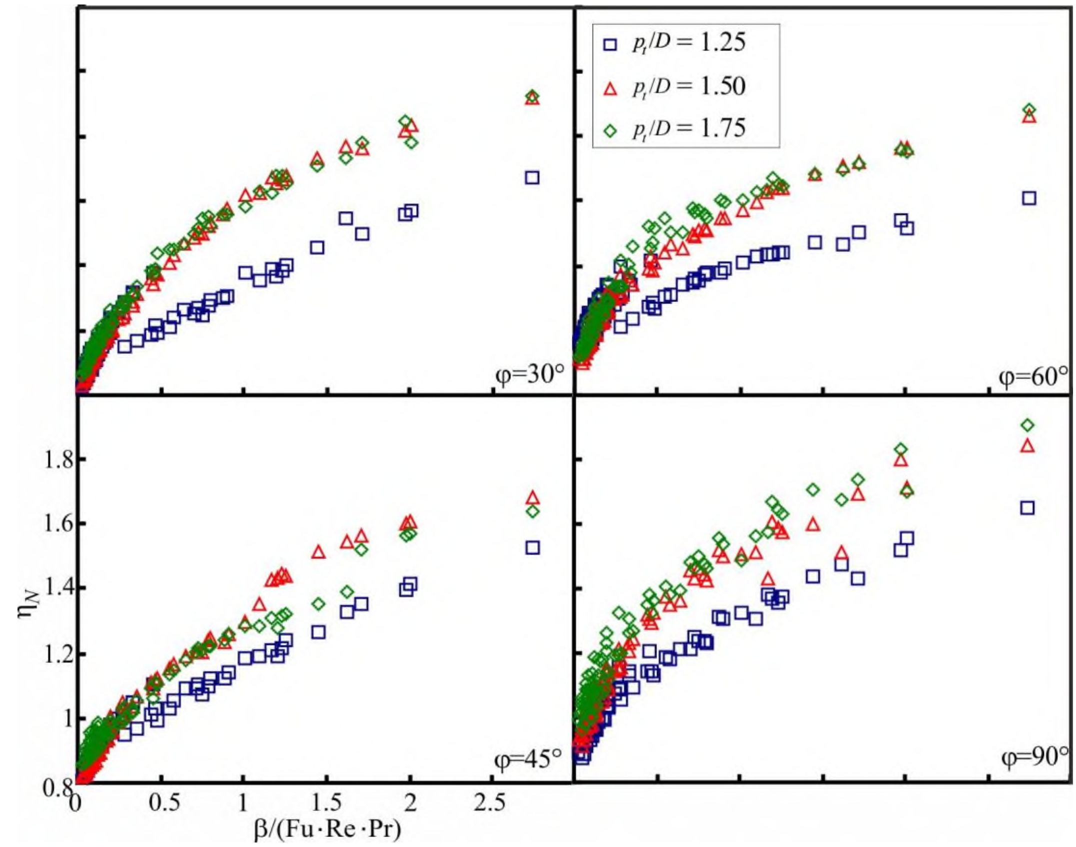

Figure 7. Variation of η_N with $\beta/(Fo \cdot Re \cdot Pr)$ for $\psi = 0.25$.

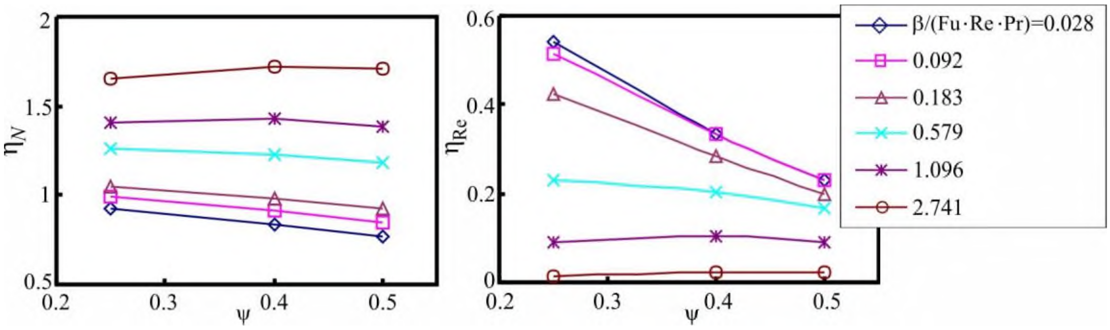

Figure 8. Variation of η_N and η_{Re} with ψ.

4. Conclusion

The thermal efficiency for four typical tube array arrangements with different tube pitch by pulsating flows has been studied.

It is shown that p_t/D affects the thermal performance ratio at the same Reynolds numbers η_{Re}. When tube pitch p_t/D is increasing the thermal performance ratio η_{Re} is decreasing. Configuration of tube bundles also has an effect on the η_{Re}. When the array arrangement was $\varphi = 45°$ the thermal performance ratio η_{Re} has the minimum value. The higher thermal performance ratio η_{Re} occurs at the square inline array arrangement $\varphi = 90°$. In the entire studied range, the maximum value of η_{Re} was 0.623 at the ($\varphi = 90°$, $p_t/D = 1.25$, Re = 1000, Pr = 363, f = 0,2 Hz, $\beta = 15$; $\psi = 0.25$), minimal value of η_{Re} was 0.0062 at the ($\varphi = 45$ p_t/D = 1.75 Re = 100, Pr = 215, f = 0,5 Hz, $\beta = 35$; $\psi = 0.25$).

When p_t/D increases from 1.25 to 1.5 the thermal performance ratio η_N increases, with a further increases in p_t/D thermal performance ratio η_N can either increase or decrease. The higher thermal performance ratio η_N at equal pumping powers occurs at the square inline array arrangement φ = 90°. In the entire studied range, the maximum value of η_N was 1.92 at the (φ = 90°, p_t/D = 1.75 Re = 100, Pr = 215, f = 0,5 Hz, β = 35; ψ = 0.25), minimal value of η_N was 0.74 at the (φ = 45°, p_t/D = 1.25, Re = 1000, Pr = 363, f = 0,2 Hz, β = 15; ψ = 0.5).

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