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Research Article

PREDICTION OF HEAT AND FLUID FLOW IN A HEAT EXCHANGER TUBE MOUNTED WITH V-CONE BUNDLES

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ABSTRACT:
A numerical prediction of turbulent forced convection is perfromed in order to gain an understanding of turbulence flow field, fluid temperature, local Nusselt number, pressure loss and thermal performance characteristics of a heat exchanger tubes mounted with V-cone bundles. V-cone is equipped at core of test tube and assumed as turbulator. The parameters for the investigation were considered at Reynolds number between 5000 and 15,000 and the distances between the V-cone pitch length to tube diameter were at P/D = 1.0, 2.0, 3.0 and 4.0. From the numerical result, it was found that the heat transfer enhancement increased with increasing Reynolds number and decreasing P/D, which the pressure loss was affected by P/D and Reynolds number, respectively. The thermal performances of the V-cones were also determined and compared with respect to heat transfer from the same tube without turbulators. The numerical results show that V-cone turbulators yield higher Nusselt number and friction factor than those the plain tube. For the V-cone turbulators, the thermal performance increases with increasing pitch ratio (PR) which the thermal performance was up to 1.01 at constant pumping power was archived.

Keywords: *Heat exchanger, heat transfer enhancement, thermal performance, V-cone turbulator*

1. INTRODUCTION

The passive heat transfer augmentation methods which do not need any external power input are used in many engineering applications. Passive heat transfer enhancement methods are implemented with the expense of addition energy consumption due to increase friction loss. There are hundreds of passive methods for example swirl flow device (insertion of twisted or helical tapes), extended surface, rough surface, wavy surface, turbulator device (wire coil, conical-ring, and V-nozzle), and additives for liquids and gases [1-10]. This technique enhances the convective heat transfer by introducing swirl into the bulk flow and disrupting the thermal boundary layer on the tube wall. Acır at al. [11] studied the effects of the circular ring turbulators at different pitch ratios and hole numbers on the heat transfer enhancement in a solar air heater. Their found that heat transfer enhancement was found at the 229% while the thermal performance factor was up to 1.83. Sheikholeslami at al. [12] studied the effect of perforated discontinuous helical turbulators on the heat transfer characteristics at different open area ratios and pitch ratios. Their results found that the heat transfer rate reduce with rise of open area ratio and pitch ratio. Thermal performance was an increasing function of open area ratio while it is a decreasing function of pitch ratio. Rajaseenivasan et al. [13] reported the thermohydraulic and second law efficiency of solar air heater with circular

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and V-shape inserts at different configurations (inline arrangements with 4×4, 5×4, 6×4 and 6×4, and zigzag arrangement of circular inserts. Their results found the thermal performance factor increases with decrease in Reynolds number for all insert devices. Acır and Ata [14] studied the heat transfer in a solar air heater equipped with circular type turbulators at pitch ratios (*PR*) and angle ratios (*AR*). Their results observed that the turbulators arrangement with *PR* = 2 and *AR* = 0.125 provide the maximum heat transfer rate and thermal performance factor up to 416% and 2.9, respectively.

The modification for improving heat transfer is made by introducing V-cone turbulators. The V-cone turbulators are expected to induce strong flow fluctuation and thus turbulence. The modification for reducing friction factor is made by leaving space between each pair of the V-cone turbulators. Four distance between the V-cone pitch length to tube diameter, *P/D* = 1.0, 2.0, 3.0 and 4.0 are examined to find the optimum geometry. The plain tube was also studied for comparison.

2. DETAILS OF V-CONE TURBULATORS

The details of circular tubes fitted with V-cone turbulators are presented in Fig. 1. The distance between the V-cone pitch length to tube diameter (*P/D* = 1.0, 2.0, 3.0 and 4.0) was examined. All V-cone turbulators have the same diameter ratio (*d/D*) of 0.5 and all tubes have the same diameter. The tube wall is subjected to a constant temperature wall heating condition at 310 K. The simulation is carried out for Reynolds number (*Re*) from 5000 to 15,000 using air as the working fluid.

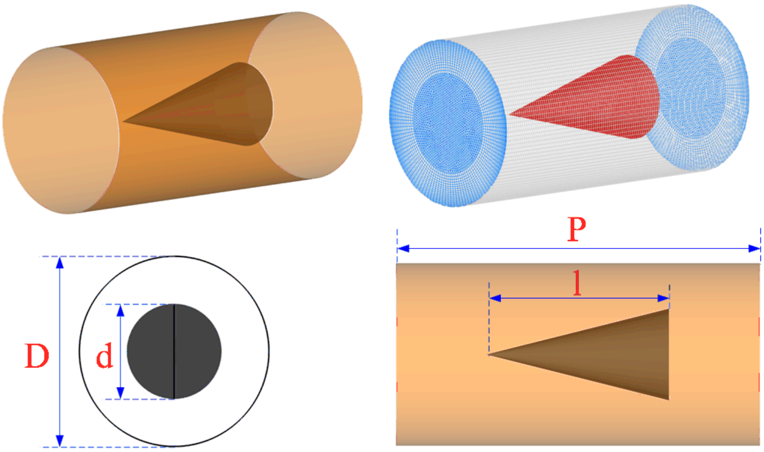


Fig. 1. Grid arrangement for tube with V-cone inserts.

3. GOVERNING EQUATION

The phenomenon under consideration is governed by the steady three-dimensional form of the continuity, the time-averaged incompressible Navier-Stokes equations and the energy equation. In the Cartesian tensor system, these equations can be written in the following form:

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = \mathbf{0} \qquad (1)$$

Momentum equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho\overline{u_i' u_j'}\right) \qquad (2)$$

Energy equation:

$$\frac{\partial}{\partial x_i}\left[u_i(\rho E+p)\right]=\frac{\partial}{\partial x_j}\left(k_{eff}\frac{\partial T}{\partial x_j}\right) \tag{3}$$

$$E=c_pT-\frac{p}{\rho}+\frac{u^2}{2} \tag{4}$$

The Reynolds-averaged approach to turbulence modeling requires that the Reynolds stresses, $-\rho\overline{u_i'u_j'}$ in Eq. (2) be appropriately modeled.

A common method employs the Boussinesq hypothesis to relate the Reynolds stresses to the mean velocity gradients:

$$-\rho\overline{u_i'u_j'}=\mu_t\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\frac{2}{3}\left(\rho k+\mu_t\frac{\partial u_k}{\partial x_k}\right)\delta_{ij} \tag{5}$$

The expression for the turbulent viscosity is given as

$$\mu_t=\rho C_\mu\frac{k^2}{\varepsilon} \tag{6}$$

The time-independent incompressible Navier-Stokes equations are discretized using the finite volume method. QUICK (Quadratic upstream interpolation for convective kinetics differencing scheme) and central differencing flow numerical schemes are applied for convective and diffusive terms, respectively. The discrete nonlinear equations are implemented implicitly. To evaluate the pressure field, the pressure-velocity coupling algorithm SIMPLE (Semi Implicit Method for Pressure-Linked Equations) is selected. At the inlet fully-developed velocity profile is imposed. Impermeable boundary condition is implemented over the wall. The inlet turbulence intensity is kept constant at 10%, unless stated otherwise.

Three parameters of interest are: (1) friction factor, (2) Nusselt number and (3) thermal performance factor. The friction factor, f is computed from the following equation.

$$f=\frac{\Delta P}{\left[\frac{L}{D}\right]\left[\frac{\rho u^2}{2}\right]} \tag{7}$$

Where Δp is pressure drop across the length of the tube (L)

The heat transfer is evaluated in term of Nusselt number which can be expressed as

$$Nu=\frac{hD}{k} \tag{8}$$

The average Nusselt number can be obtained by:

$$Nu_{ave}=\frac{1}{A}\int Nu_x\partial A \tag{9}$$

A useful comparison between reverse and straight flows can be made by comparing heat transfer coefficients at equal pumping power, since this is relevant to the operation cost. For constant pumping power:

$$(Q\Delta P)_p=(Q\Delta P)_t \tag{10}$$

and the relationship between friction and Reynolds number can be expressed as:

$$(f\,Re^3)_p = (f\,Re^3)_t \tag{11}$$

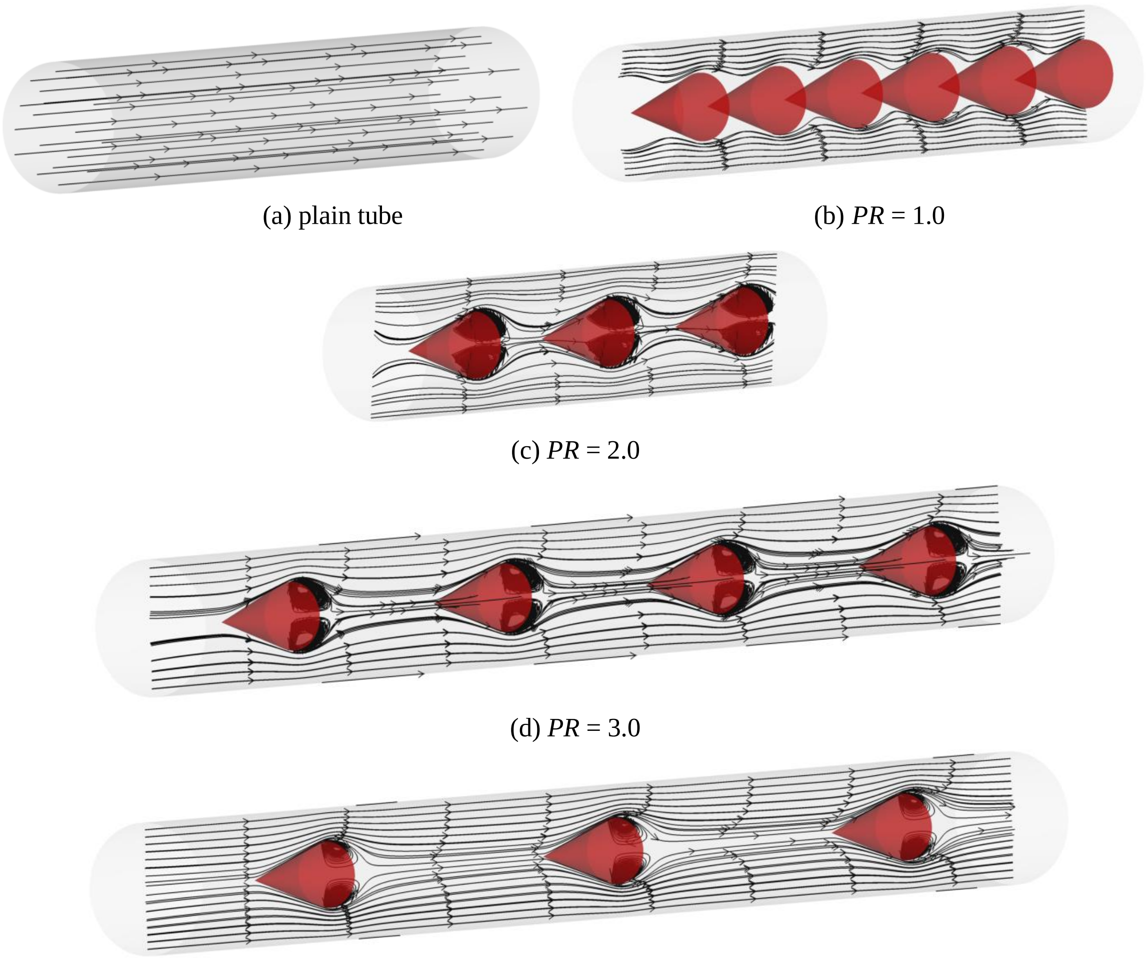
The thermal performance factor (η) at constant pumping power is the ratio of the convective heat transfer coefficient of the tube with V-cone turbulators to the plain tube which can be written as follows:

$$\eta = \left(\frac{Nu_t}{Nu_p}\right)\left(\frac{f_t}{f_p}\right)^{-1/3} \tag{12}$$

4. Numerical results

4.1 Flow and thermal behaviors

Figure 2 demonstrated the contour plots of streamline in circular tube equipped with V-cone turbulators at different pitch ratios (*P/D* = 1.0, 2.0 and 3.0). Turbulence flow stream lines regarding to V-cone geometries are observed in the tubes with all V-cone inserts while only straight stream lines are seen in the plain tube. It is also depicted that the V-cone turbulator gives more consistent turbulence than those the plain tube alone. For V-cone inserts, turbulent consistency become higher and its intensity becomes stranger as pitch ratio (*P/D*) decreases. It can also be found high turbulence due to flow fluctuation is seen around V-cone turbulators especially at the lower pitch ratio (*P/D* = 1.0 and 2.0).



(a) plain tube (b) *PR* = 1.0

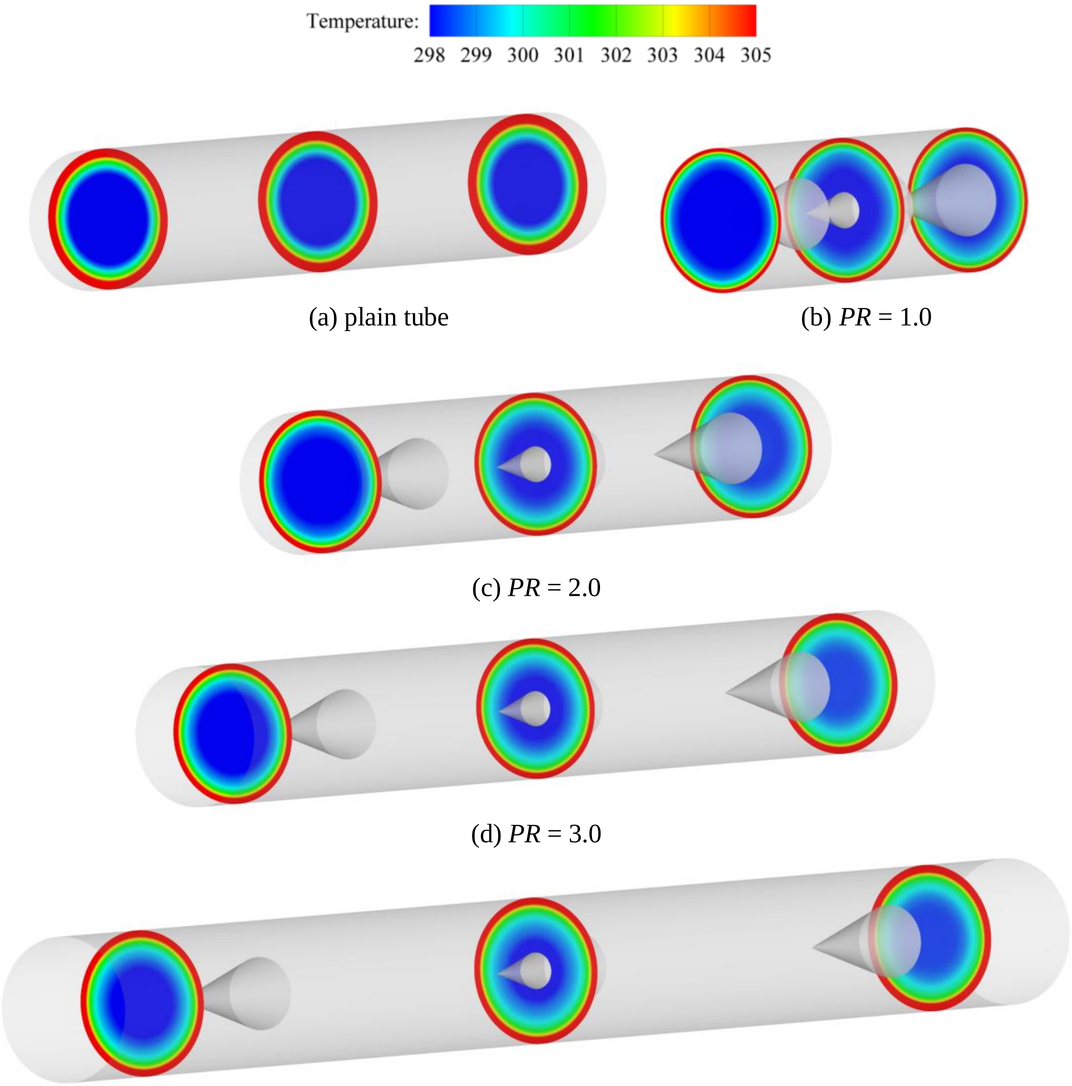
(c) *PR* = 2.0

(d) *PR* = 3.0

(e) *PR* =4.0

Fig. 2. Contour plots of streamline across tubes with V-cone inserts.

Figure 3 depicts the contour plots of fluid temperature in the plain tube with and without V-cone inserts at different pitch ratios (*P/D* = 1.0, 2.0 and 3.0). The numerical results showed that the fluid temperature of the plain tube is unchanged and thick thermal boundary exists throughout the tube. For the tubes with V-cone inserts, the fluid temperatures near the entrances and tube cores are low value. Then, fluid temperatures become higher along the axial direction due to a turbulancing effect that enhances fluid mixing between core and wall regions and thus improves heat transfer from tube walls to fluid especially at the lower pitch ratio (*P/D* = 1.0 and 2.0). In addition, fluid fluctuation in V-cone cases further enhances fluid mixing and heat transfer.



(a) plain tube (b) *PR* = 1.0

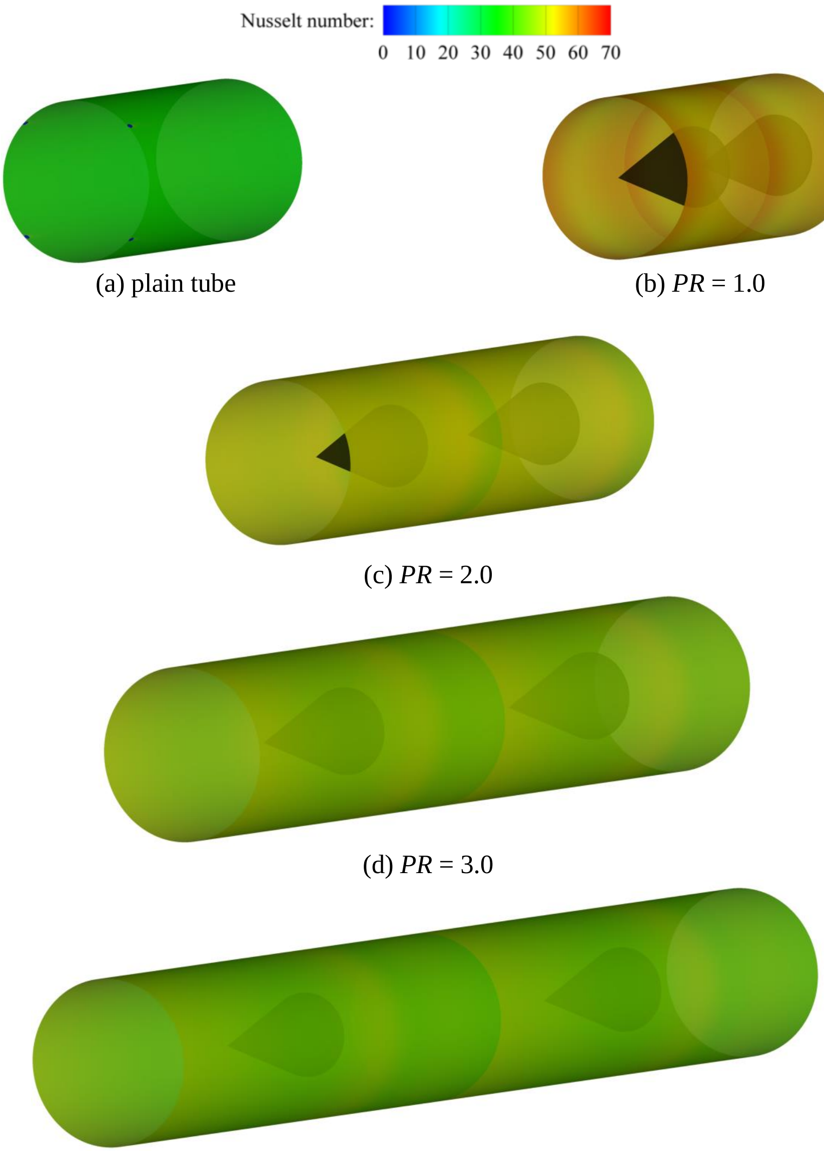
(c) *PR* = 2.0

(d) *PR* = 3.0

(e) *PR* =4.0

Fig. 3. Temperature fields in tubes with V-cone inserts.

Figure 4 also found that the tube with V-cone inserts with the smallest pitch ratio, *P/D* = 1.0, give better fluid mixing than the ones with the larger pitch ratio of *P/D* = 2.0 and 3.0 due to more consistent turbulencing effect. These results accord with those in Fig. 4 which shows that tube with V-cone inserts at the smallest pitch ratios, *P/D* = 1.0, give higher Nusselt number and more uniform Nusselt number distribution than the ones with *P/D* = 2.0 and 3.0.



(a) plain tube
(b) *PR* = 1.0
(c) *PR* = 2.0
(d) *PR* = 3.0
(e) *PR* =4.0

Fig. 4. Nusselt number distributions around tube with V-cone inserts.

4.2 Heat transfer, friction factor and thermal performance

Figure 5(a) presents the heat transfer behaviors in a uniform heat flux tube with V-cone inserts at various pitch ratios (*PR* =1.0, 2.0, 3.0 and 4.0). It can be seen that the heat transfer ratio (Nu/Nu_p) tends to decrease with the rise of pitch ratio (*P/D*) and Reynolds number (*Re*) for all V-cone inserts due to the higher turbulent intensity imparted to the flow between the V-cone turbulators. The use of V-cone inserts with P/D = 1.0, 2.0, 3.0 and 4.0 results in higher heat transfer rate (*Nu*) than the plain tube. According to the obtained results, the heat transfer rate (*Nu*) of the tubes with V-cone turbulators with *P/D* = 1.0, 2.0, 3.0 and 4.0 increase by 88.1%, 72.9%, 60.6% and 52.1% as compared to that of the plain tube, due to the effect of fluid fluctuation induced by alternate axes. This may be attributed to a better and rapid mixing of the fluid between the core and the tube surface regions from turbulent fluctuation flow and the appearance of turbulence flow between the V-cone elements, leading to higher temperature gradients.

Relationship between friction factor (f/f_p) and Reynolds number (*Re*) of all tubes with V-cone turbulators various pitch ratios (*P/D* = 1.0, 2.0, 3.0 and 4.0) is shown in Fig. 5(b). For all cases, friction factor ratio (f/f_p) tends to decrease with the rise of Reynolds number. For V-cone turbulators, friction factor decreases with the rise of pitch ratio (*P/D*) due to the dissipation of dynamic pressure of the fluid due to higher surface area and the act caused by the turbulence flow. The maximum friction factors of the tube with V-cone turbulators with *P/D* = 1.0, 2.0, 3.0 and 4.0 are 8.0, 6.5, 5.0 and 4.1 times of those of the plain tube, respectively.

The variation of thermal performance factor (η) with Reynolds number is shown in Fig. 5(c). For all cases, thermal performance substantially decreases with increasing Reynolds number. All tubes with V-cone turbulators yield thermal performance factors above unity at the lowest Reynolds number (*Re* = 5000) and pitch ratio (*P/D* = 4.0). Among the V-cone turbulators, the one with *P/D* = 4.0 offer the maximum thermal performance factor. Although the V-cone turbulators with *P/D* = 2.0 yields lower heat transfer than the ones with *P/D* = 4.0, its higher thermal performance factor is the result of the dominant influence of the lower friction loss. The mean thermal performance factor at *Re* = 5000 offered by the V-cone turbulators at *P/D* = 4.0 is 1.01% higher than that given by the plain tube.

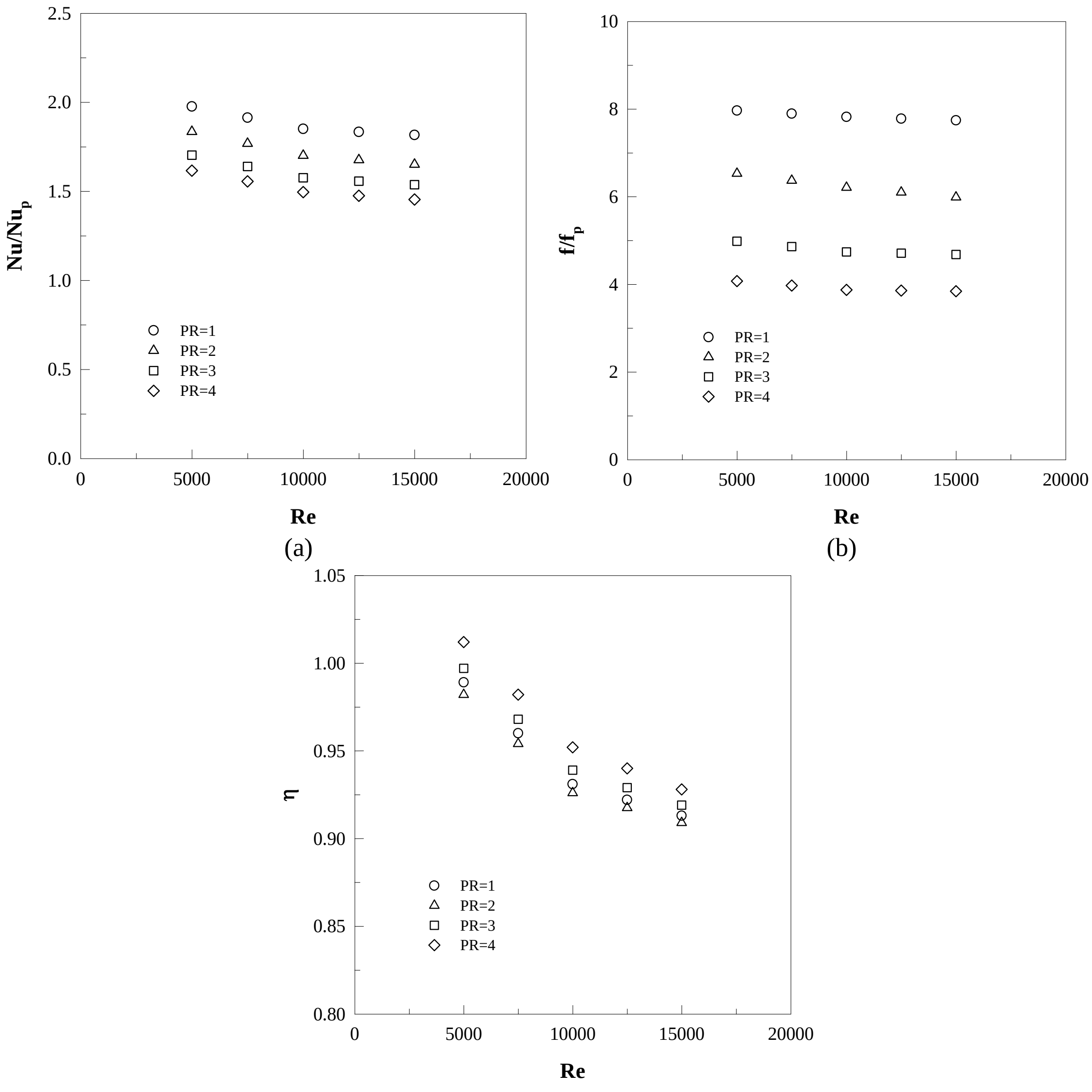


Fig. 5. Effect of the V-cone inserts at various pitch ratios (P/D) on (a) heat transfer rate, (b) friction factor amd (c) thermal performance factor.

5. CONCLUSION

Numerical investigation of the heat transfer enhancement, pressure loss and thermal performance characteristics of constant temperature wall tube equipped with V-cone turbulator at four different pitch ratios (*P/D* = 1.0, 2.0, 3.0 and 4.0) are reported. The tube equipped with V-cone turbulator provides considerable enhancement of convective heat transfer coefficients for heating. It can be observed that there is a significant increase in heat transfer for all V-cone turbulators better than the plain tube which the turbulators is used as a turbulence flow generator. However, since the V-cone turbulator are placed directly to the flow area, they cause pressure losses. V-cone turbulators with pitch ratio of *P/D* = 4.0 gives superior thermal performance factor than those other pitch ratios (*P/D*).

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