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NUMERICAL HEAT TRANSFER AND TURBULENT FLOW IN A CIRCULAR TUBE FITTED WITH CURVILINEAR CONVERGING RINGS

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Abstract: In this study; heat transfer, friction factor and thermal performance factor characteristics are numerically investigated in a plain tube inserted the curvilinear converging rings (CCRs). Three different pitch ratios are considered as p/D = 1, 2 and 3. The study is conducted in the range of Reynolds number 5000 -20000. Air is accepted as the working fluid (Pr=0.7). The governing equations are solved using the Simple algorithm and Standard k–ε turbulence model with Fluent 6.1.22 programme. To validate the applied numerical method, the results obtained from the plain tube with CCRs are compared with the empirical results presented in literature. The values of Nusselt number, friction factor and thermal performance factor for the plain tube with CCRs are presented depending on Reynolds number and pitch ratios studied. The thermal performance results for the studied cases are also compared with results given in previous studies related to the conical rings. Consequently, the average Nusselt number and friction factor for all the pitch ratios are found to be about 68- 109 % and 412-635 % over the plain tube at the same pumping power, respectively. The best enhancement of 7.5 % was obtained for Re = 5000 and p/D = 1. The cases of p/D = 2 and 3 are thermodynamically advantageous only for Re<10000.
Keywords: Heat transfer, Enhancement, Turbulent flow, Rings, Numerical.

EĞRİSEL YAKINSAYAN HALKALAR YERLEŞTİRİLMİŞ BİR BORU İÇİNDE SAYISAL ISI GEÇİŞİ VE TÜRBÜLANSLI AKIŞ

Özet: Bu çalışmada, akış yönünde yüzeyi eğrisel olarak yakınsayan halkaların yerleştirildiği bir boruda ısı geçişi, sürtünme ve ısıl verim sayısal olarak araştırılmıştır. p/D = 1, 2 and 3 olarak üç farklı adım oranı incelenmiştir. Re sayısı 5000 -20000 aralığındadır. Akışkan havadır (Pr=0.7). Korunum denklemleri Simple algoritması ve Standard k–ε türbülans modeli kullanarak Fluent programıyla çözülmüştür. Sayısal çalışmanın doğruluğunu test etmek için halkalar yerleştirilmiş düz boru için elde edilen sayısal sonuçlar açık kaynaklarda sunulan deneysel sonuçlarla karşılaştırılmıştır. Nusselt sayısı, sürtünme faktörü ve ısıl verim Reynolds sayısı ve adım oranına bağlı olarak sunulmuştur. Çalışılan durumların ısıl verim sonuçları konik halkalarla ilgili sunulmuş deneysel çalışmaların sonuçlarıyla da karşılaştırılmıştır.Sonuç olarak; ortalama Nusselt sayısı ve sürtünme faktörünün tüm adımlar için aynı pompa gücündeki düz borudaki akıştan sırasıyla %68- 109 ve %412-635 daha fazla olduğu bulunmuştur. %7.5'luk en iyi iyileşme Re=5000 ve p/D = 1 için elde edilmiştir. Adım oranlarının p/D = 2 and 3 olduğu durumlar yalnızca Re<10000 için termodinamik olarak avantajlıdır.
Anahtar kelimeler: Isı transferi, İyileşme, Türbülanslı akış, Halkalar; Sayısal.

NOMENCLATURE

A	area [m^2]
C_p	specific heat [J/kg K]
D	diameter of the tube [m]
d	small end diameter of the ring [m]
f	friction factor
k	thermal conductivity [W/m K]
Nu	Nusselt number
P	turbulent kinetic energy product
p	pitch [m]
Pr	Prandtl number
q	heat flux [W/m^2]
Re	Reynolds number
T	temperature [K]
$\vec{V}$	velocity vector
v_x, v_r	velocities in x and r directions [m/s]
$\bar{V}$	mean velocity [m/s]

Greek letters

η	thermal performance factor
ε	turbulent dissipation rate
ν_t	turbulent viscosity
ρ	density [kg/m^3]
$\sigma_\varepsilon, \sigma_k, C_1, C_2$	model coefficients
μ	dinamic viscosity [kg/ms]
$\vec{\tau}$	stress tensor

Subscripts

b	bulk
c	cross- sectional
CCR	curvilinear converging ring
w	wall
p	plain

INTRODUCTION

In recent years, various heat transfer enhancement techniques have been researched to improve the efficiency of heat exchangers and so to reduce their size and cost. These techniques are classified into two main categories, active and passive techniques. The use of the turbulence promoters or roughness elements, such as ribs, grooves or wires on the surface, is a common passive technique to enhance the rate of heat transfer (Bilen et al., 2009). Turbulator devices inserted in tubes have been commonly used so as to improve the heat transfer in heat exchangers. Turbulators create a secondary flow and so the boundary layers of fluid occured near the surface of tube are distributed. To enhance heat transfer in heat exchangers, the several shaped turbulators have been investigated such V-nozzle, conical-nozzle, combined conical-ring and coiled wire, circular cross sectional ring, baffles and so on (Eiamsa-ard and Promvonge, 2006; Yakut and Sahin, 2004; Promvonge and Eiamsa-ard, 2007; Ozceyhan et al., 2008; Onur et al., 2007).

The performance characteristics of the conical-ring turbulators were determined by means of the entropy-generation minimization method based on the second law and enhancement efficiency based on the first law of thermodynamics (Yakut et al., 2004). It was found that the conical rings are efficient for Re < 8000 compared to generation of entropy and the enhancement in the range of 0.86-1.16 increases as the pitch (10, 20 and 30 mm) decreases. Conical rings with three different diameter ratios of the ring to tube diameter (0.5, 0.6 and 0.7) were introduced experimentally and the rings were placed with three different arrangements as converging, diverging and converging–diverging conical rings for each ratio (Promvonge, 2008). The experiment was conducted with cold air at ambient condition for 2000< *Re* <26000 under uniform heat flux condition. The maximum values of enhancement were found to be 1.8 using the diverging conical rings at d/D = 0.5 for the lowest *Re*. In addition, it was found that the enhancement increased with decreasing *Re* and diameter ratio. In other experimental work (Promvonge and Eiamsa-ard, 2007), the diverging and converging nozzles with various pitch ratios (2, 4 and 7) were investigated for 8000 <*Re*< 18000. It was found that the heat transfer rates obtained from using both nozzles were found to be higher than that from the plain tube at a range of 236 - 344%, depending on *Re*. The effect of perforated conical rings (PCRs) on the turbulent convective heat transfer, friction factor and thermal performance factor were investigated experimentally by Kongkaitpaiboon et al. (2010). The PCRs were taken with three different pitch ratios (PR = 4, 6 and 12) and three different numbers of perforated holes (N = 4, 6 and 8). The experiments were performed for 4000 < *Re* < 20000 under uniform heat flux condition using air as the working fluid. It was found that the maximum thermal performance factor of around 0.92 is found at PR=4 and N=8 holes for *Re* = 4000. The conical converging nozzle turbulators with different pitch ratios (PR= 2, 4 and 7) in common with the free-space snail entry were examined by Promvonge and Eiamsa-ard (2007) for 8000< *Re* <18000. It was found that the best efficiency occured at the lowest Reynolds number and pitch ratio values. The effects of cut out conical turbulators placed in a tube at constant outer surface temperature on the heat transfer rate, friction factor and energy were experimentally investigated by Durmus (2004) for turbulent flow.

The various numerical studies on enhancement of heat transfer using turbulence promoters were also reported in literature. Ozceyhan et al. (2008) investigated the effect of the circular cross sectional rings inserted in a plain tube on heat transfer and flow using Standart k- ε model and Fluent program. The investigation of three dimensional heat transfer enhancement was investigated numerically by using turbulator which inside the pipe as a turbulence generator produced of stainless steel and two different winglet distances and with three different winglet angles by Kahraman et al. (2008). The flow and temperature fields are computed numerically using Simple algoritm with the Fluent computational fluid dynamics (CFD) code. Similarly, Fan et al. (2011) numerically investigated turbulent flow and heat transfer in a circular tube fitted with conical strip inserts using k-ε model. In literature, many numerical studies using k–ε turbulence model related to turbulators inserted tubes are reported (Deb et al., 1995, Hiseh et al., 2001, Hung et al., 1997, Mon and Gross, 2004).

As declared in the literature review, all of the studies related to the conical ring turbulators were conducted experimentally. Furthermore, the surfaces of those converge linearly. Thus, the curvilinear converging ring turbulator (CCR) had not been investigated previously experimentally or numerically. In this study, the surface of the investigated ring converges nonlinearly along the flow. The present work proposes numerically the heat transfer enhancement ratio of a plain tube inserted the curvilinear converging rings (CCRs) with three different pitch ratios (*p/D=1, 2* and *3*) using the commercial code Fluent, for 5000 < *Re* < 20000. The uniform heat flux and fully-developed turbulent flow with air (*Pr* = 0.7) are assumed on the outer surface and at the inlet of the tube, respectively. For all the arrangements, the first CCR is placed at *24D* downstream of the tube inlet to provide fully developed boundary layers and the last one is placed at *10D* at the back of the outlet to avoid reverse pressure effects (Ozceyhan et al., 2008).The results of Nusselt number, friction factor and thermal performance factor are presented for all the studied cases. The results of thermal performance are also compared with other similar studies related to the conical rings.

NUMERICAL METHOD

In this study, the flow is accepted as two dimensional due to the symmetry. The continuity equation is given as

$$\nabla . \vec{V} = 0 \tag{1}$$

The momentum equation is

$$\nabla\left(\rho\vec{V}\,\vec{V}\right) = -\nabla P + \nabla . \vec{\tau} \tag{2}$$

where, $\vec{V}$, ρ, P and $\vec{\tau}$ are the velocity vector, density, static pressure and the stress tensor, respectively.

In this study, Simple algorithm is applied with finite volume method approach (Patankar, 1980). Standart k-ε model (Launder and Spalding, 1972) is used for the solution of turbulent flow. In open literature, this model has been widely used for fully turbulent flows in tubes including turbulator (Launder and Spalding, 1972). k and ε are turbulent kinetic energy and its dissipation rate, respectively and expressed as

$$\frac{\partial}{\partial x}(\rho k v_x) = \frac{\partial}{\partial r}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\right]\frac{\partial k}{+\partial r} G_k + G_b - \rho\varepsilon \tag{3}$$

$$\frac{\partial}{\partial x}(\rho \varepsilon v_x) = \frac{\partial}{\partial r}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial r}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{4}$$

where, the turbulent viscosity , μ_t , is defined as

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{5}$$

G_k and G_b are the products of turbulence kinetic energy due to the mean velocity gradients and buoyancy. $C_{1\varepsilon}, C_{2\varepsilon}, C_\mu$ and $C_{3\varepsilon}$ are model constants, σ_k and σ_ε are the turbulent Prandtl numbers for *k* and *ε*, respectively (Launder and Spalding, 1972). Fluent 6. 1. 22 is used for the numerical solutions. The convergence criteria for all variables are accepted about 10^{-5}. To discretize the governing equations, the second order upwind scheme is used.

Boundary Conditions

In this study, the mathematical model is solved for the following conditions:

- The flow is steady, turbulent, fully developed and two-dimensional due to the axial symmetry
- Uniform heat flux on the tube wall
- The constant thermophysical properties of the air
- No- slip conditions ($v_x = 0$, $v_r = 0$) on the CCRs and tube walls
- At the inlet, the temperature of the air is 300 K
- At the outlet, the relative pressure is 0 Pa.

At the inlet, the velocity profile is accepted as fully developed. Thus, the inlet velocity profile for turbulent flow is given as

$$v(r) = \bar{V}\left[1 - \left(\frac{r}{R}\right)\right]^{1/7} \tag{6}$$

Eq. (6) is used in Fluent code as a user defined function to calculate velocity at the inlet. The constant thermophysical properties of air are given in Table 1.

Table 1. Thermophysical properties of the air at 300 K.

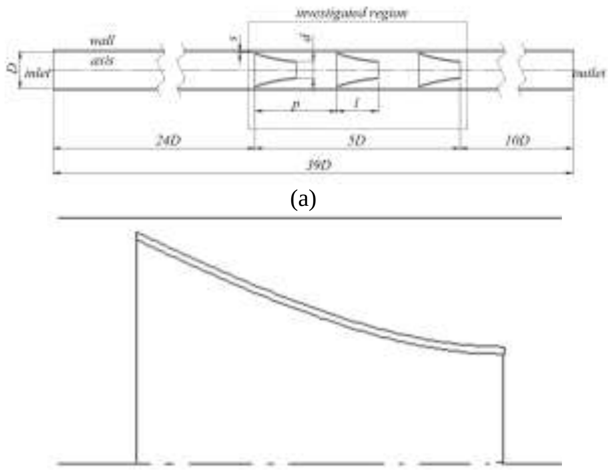
ρ (kg/m^3)	μ (N.s/m^2)	C_p (J/kgK)	k (W/mK)	Pr
1.1614	184.6x10^{-7}	1007	26.3x10^{-3}	0.707

Figure 1 shows the physical model of the geometry. The diameter (*D*) and length (*L*) of the plain tube are 66 mm and 2574 mm, respectively. The clearance between the CCR and tube wall (*s*) is 2 mm (0.03*D*). The length (*l*) of CCRs made of aluminum with 1 mm uniform thickness and its small end diameter are 50 mm (0.8*D*) and 30 mm (0.5*D*), respectively. The equation of the surface of the ring is given by

$$r(x) = [(0.1 + x^2)/\pi]^{0.5}, \quad -0.5 \le x \le 0 \tag{7}$$

where, r(x) is the radius of the ring at *x*. Three different pitch ratios are considered as *p/D* = 1, 2 and 3. The number of the CCRs for the studied pitch ratios (*p=D*, *2D* and *3D*) are 5, 3 and 2, respectively.

For all arrangements, the first CCR is placed at *24D* downstream of the inlet to provide the fully developed thermal boundary layer and the last one is placed at *10D* at the back of the outlet to avoid reverse effects of the pressure.



(a)

(b)

Figure 1. (a) Physical model of the plain tube with curvilinear converging nozzle rings for *p=2D* (b) detail of CCR.

Grid Independency

Fig. 2 shows the grid structure of the computational solution region. For grid independence, four different cases varied the number of cells from 54178 to 74943 are compared in Table 2. The relative error between the fine (g_f) and coarse (g_c) grid solutions can be written as

$$error\ (\%) = 100 \times \left|\frac{g_f - g_c}{g_f}\right| \quad (8)$$

It is seen from Table 2 that the variations of Nusselt number (*Nu*) and friction factor (*f*), after 72990 cells (Case 2), is obtained about 0.5% and 0.075%, respectively.

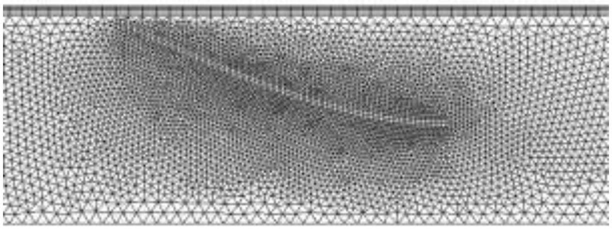


Figure 2. Grid structure.

Nevertheless, the y + value on the wall for the Case 4 has closer unity than those obtained from the others, where the space between the first row of the boundary layer and the tube wall is 0.2 mm.

Thus, Case 4 is accepted for the grid structure of the solution region (Fig. 3). For the accepted case, the variations in *Nu* and f are about 1.4% and 2.2% in comparison with Case 2.

Validation Study

In this study, representing calculated results the following dimensional and nondimensional quantities have been used

$$\mathrm{Re} = \frac{\rho \bar{V} D}{\mu} \quad (9)$$

The friction factor

$$f = \frac{\Delta P D/L}{\rho \bar{V}^2/2} \quad (10)$$

and Nusselt number

$$Nu = \frac{D q_w}{k(T_w - T_b)} \quad (11)$$

where, the bulk temperature of fluid, T_b, is calculated

$$T_b = \frac{1}{\bar{V} A_c} \iint \bar{V} T dA_c \quad (12)$$

with mass flow averaging and T_w

$$T_w = \frac{1}{A_w} \iint T d A_w \quad (13)$$

is the wall temperature with area averaging.

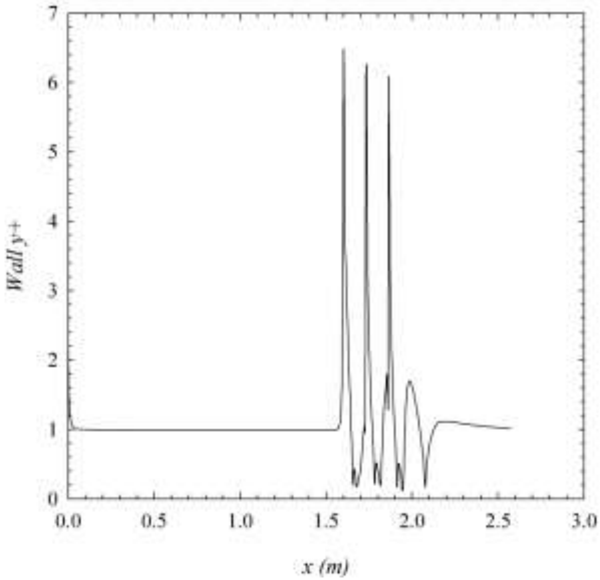


Figure 3. The y + values on the wall for *Re* =10000 and *p* = 2*D*.

In order to validate the numerical method applied in this study, the numerical results obtained from the plain tube with CCRs are compared with Eqs. (14) and (15) presented by Promwonge (2008). In ref. (Promwonge, 2008), heat transfer and flow in the circular tubes fitted with converging (CR), diverging (DR) and converging-diverging rings (CDR) were experimentally investigated and the empirical correlations of Nusselt number and friction factor for the converging rings (CRs) are obtained as

$$Nu = 0.09155 Re^{0.65} Pr^{0.4} (d/D)^{-1.31} \quad (14)$$

$$f = 1.12 Re^{-0.258} (d/D)^{-4.4} \quad (15)$$

where (*d/D*) is the ratio of the smal end diameter of converging ring to tube diameter. The value of (*d/D*) is selected as 0.7 to compare with the present study.

The comparisons between numerical results obtained from this study and empirical correlations in ref. (Promwonge, 2008) are shown in Figs. 4 and 5. As seen from these figures, in comparison with the empirical

Table 2. The study of grid independence.

Case	Boundary layer			İnterval count -*x*	İnterval count -*r*	Number of cells	*f*	*Nu*	*y* +	Variation of *f* %	Variation of *Nu* %
	First row (mm)	Growth factor	Number of rows								
1	0.3	1.2	4	1000	30	54178	0.187	45.820	1.610		
2	0.3	1.2	4	1257	40	72990	0.188	53.070	1.600	0.532	13.661
3	0.3	1.2	4	1257	50	73542	0.189	53.030	1.600	0.529	0.075
4	0.2	1.2	5	1257	40	74943	0.185	54.250	1.070	1.402	2.175

results, the numerical method has quite different results for friction factor as it has a good agreement for Nusselt number. The discrepancies between the numerical and empirical results for *Nu* and *f* are about 9% and 54 %, respectively. However; it should be taken consideration that the empirical correlations in ref. (Promwonge, 2008) agree with experimental data within ±10 % for the Nusselt number and friction factor. Therefore, it can be concluded that the present results agree with experimental data in ref. (Promwonge, 2008).

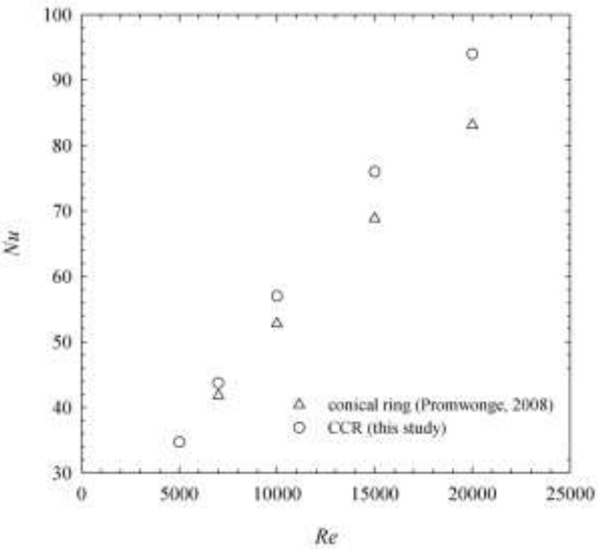


Figure 4. Comparison between experimental and numerical results for Nusselt number.

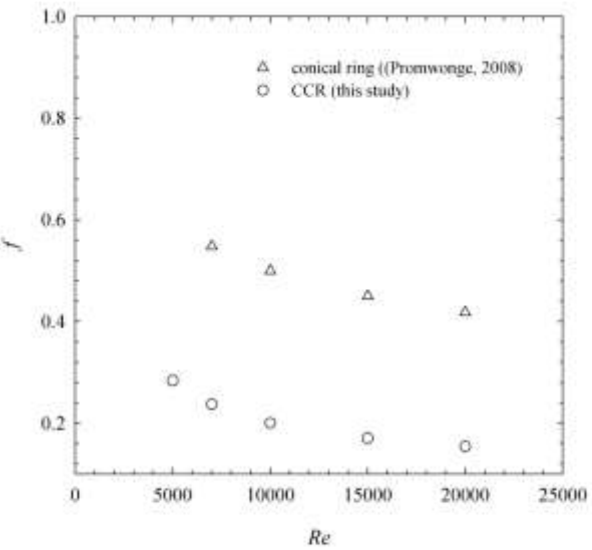


Figure 5. Comparison between experimental and numerical results for friction factor.

However, the discrepancy of the numerical and empirical results is quite major for friction factor. It may be various reason of this difference. Firstly, the number of rings studied in ref. (Promwonge, 2008) is much more than that of the present study. Secondly, this discrepancy may have occured due to the effect of the clearance between the rings and tube wall caused the leakage flow (Guo et al., 2010). In the present work, differently from the experimental work presented in ref. (Promwonge, 2008), it was created a clearance of 2 mm between the curvilinear conical rings and the tube wall. Besides, the first and last rings are inserted at the inlet and outlet in ref. (Promwonge, 2008) whereas they are placed *24D* downstream at the inlet to provide fully developed thermal boundary layer and *10D* at the back of the outlet to avoid reverse effects of the pressure in this study. Therefore, it can be estimated that the friction factor in the present study is smaller than that presented in ref. (Promwonge, 2008). In this regard, it can be said that the numerical method used in the present study is acceptable.

RESULTS AND DISCUSSION

This paper presents the numerical results of Nusselt number, friction factor and thermal performance factor in a plain tube inserted curvilinear converging rings with different pitch ratios (*p/D* = 1, 2 and 3) for steady state and single phase turbulent flow. The values of Nusselt number depending on Reynolds number for all the studied pitch ratios are shown in Fig. 6. As expected, it can be seen from Fig. 6 that Nusselt number increases with Reynolds number. The use of curvilinear converging rings causes higher heat transfer rate about 68-109 % than that of the plain tube for all the studied pitch ratios. The first reason of this case may be the flow through both the CCRs and the clearance with *s* = 2 mm which causes to disturb of the boundary layer. Secondly, the mean velocity increases due to decreasing cross sectional area of the flow. Decreasing in the distance between the CCRs also causes an increase of Nusselt number. Thus, the highest Nusselt number is obtained for *p/D* = 1.

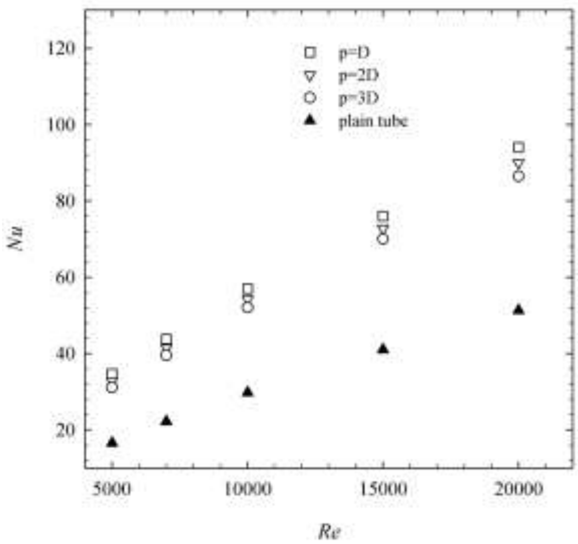


Figure 6. Distribution of Nusselt number with Reynolds number for different pitch ratios.

Nevertheless, it can be seen from the Figure 7 that the CCRs also cause more friction about 412-635 % in comparison with the plain tube. The friction factor increases with decreasing pitch ratio due to the number of CCRs caused a resistance counter to flow. The highest friction factor is obtained for *p/D* = 1 (5 CCRs).

As expected, the friction factor also decreases with increasing Reynolds number.

It is well known that the improvement in heat transfer accompanies increased friction factor caused by CCRs. Therefore, it is useful to estimate both of them simultaneously. For this purpose, a thermal performance analysis is performed for this study. Thermal performance factor, η, can be written as (Ozceyhan et al., 2008).

$$\eta = \left(\frac{Nu_{CCR}}{Nu_p}\right)\left(\frac{f_{CCR}}{f_p}\right)^{-1/3} \qquad (16)$$

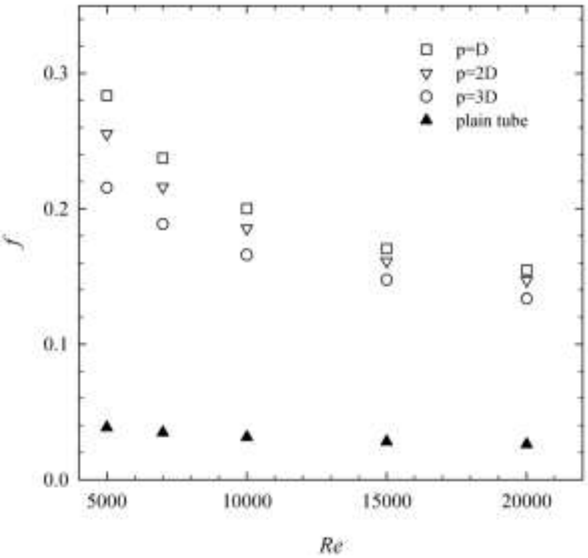


Figure 7. Distribution of friction factor with Reynolds number for different pitch ratios.

It is evident from Fig. 8 that for all *Re* numbers the thermal performance factor values of the case $p/D = 1$ are more than unity. For this case, the thermal performance factor varies from 1.075 to 1.012 for the studied *Re* and the highest performance factor is obtained for $Re = 5000$. For $p/D = 2$ and 3, the thermal performance factor values are about 1.059-0.99 and 1.056-0.98 for the studied *Re*, respectively. It is seen from this figure that the thermal performance factor for the studied cases increases with the decrease of ring spacing, but it is very small. The cases of $p=2D$ and $p=3D$ are advantageous thermodynamically only for $Re < 10000$.

The thermal performance results for the studied cases are also compared with results given in previous studies related to the conical rings (Yakut et al., 2004; Promvonge, 2008; Kongkaitpaiboon et al., 2010; Promvonge and Eiamsa-ard, 2007) as shown in Fig. 9.

It is considered the cases obtained maximum thermal performance in the previous studies. According to this figure, the thermal performance factor values obtained from the studies presented by Kongkaitpaiboon et al. (2010), Promvonge and Eiamsa-ard (2007) and Yakut et al. (2004) (only for $Re > 10000$) are less by about 15-20%, 15-24% and 2-9% than that of this study, respectively. The reason for these differences may have occured due to the clearance of 2 mm between the rings and tube wall in this study. This clearance may cause less friction than that of the studies compared (Yakut et al., 2004, Kongkaitpaiboon et al., 2010; Promvonge and Eiamsa-ard, 2007). Thus, the rise of *Nu* in this study may be important in view of thermal performance.

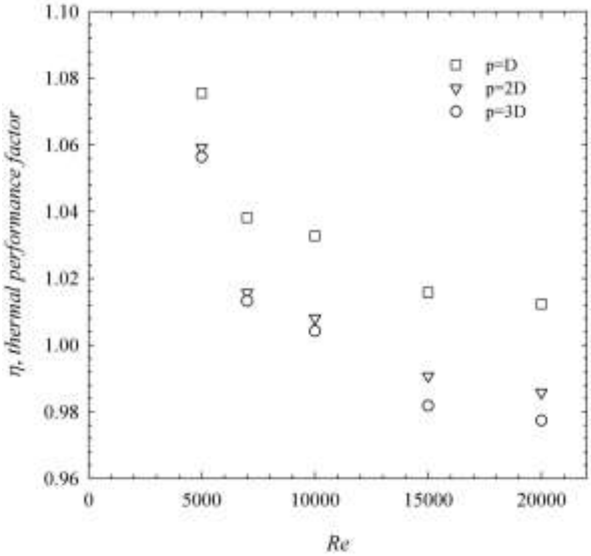


Figure 8. The variation of thermal performance factor with the studied *Re* for different pitch ratios.

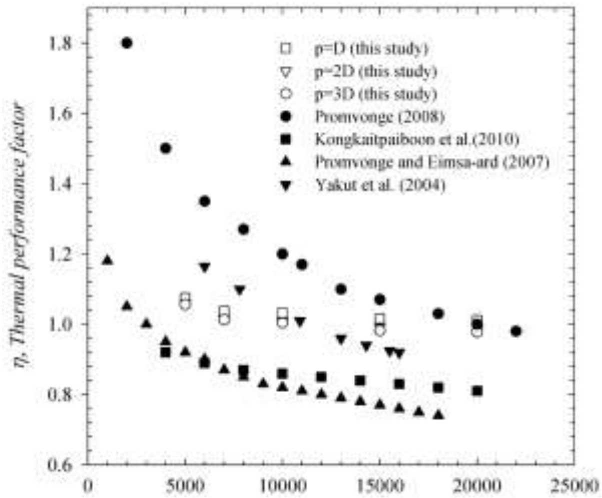


Figure 9. Comparison of numerical and experimental results of the thermal performance.

However, the results presented by Promvonge (2008) and Yakut et al. (2004) are bigger by about 5-40% (except $Re = 20000$) and 6-8% (except $Re > 10000$) than that of this study, respectively. The reason for this discrepancy may have occured due to the use of the diverging conical rings demonstrated to have obtained higher thermal performance than the use of the converging conical rings in refs. (Yakut et al., 2004; Promvonge, 2008).

CONCLUSIONS

In this study, characteristics of heat transfer and turbulent flow of single phase air in a plain tube inserted into curvilinear converging rings (CCRs) with different pitch ratios are presented numerically under uniform heat flux condition. The Navier- Stokes and energy equations have been solved using Simple algorithm and Standard k-ε formulation for the range of Reynolds number from 5000 to 20000. The governing equations are solved using the commercial code Fluent 6. 1. 22. The thermal performance factor is also presented for three different cases (*p/D* = 1, 2 and 3). Based on this study; following conclusions have been drawn:

1. For all the cases, as expected, Nusselt number increases with increasing Reynolds number. The increase of Nusselt number is obtained about 68-109 % compared to the plain tube for 5000≤ *Re* ≤ 20000.
2. For all the studied cases, friction factor decreases with increasing Reynolds number. The increase of friction factor is about 412-635 % compared to the plain tube for the studied *Re*.
3. The maximum values of Nusselt number and friction factor is found for the case of *p = D*.
4. The Nusselt number and also friction factor increase as the spacing between the CCRs decreases. As expected, the case of *p = D* creates more friction than the others because of the number of CCRs.
5. The thermal performance factor values for *p/D* = 1, 2 and 3 are about 1.012-1.075, 0.99-1.059 and 0.98- 1.056 for the studied *Re*, respectively.
6. The value of thermal performance factor increases as the pitch ratio decreases. However; the rate of increase is very small.
7. The highest enhancement of 7.5 % was achieved for *Re* = 5000 in which pitch ratio is *p/D* = 1. In view of the thermal performance, the cases of *p/D* = 2 and 3 are efficient only for *Re* < 10000.

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