Mehmet Akif Ersoy Üniversitesi Uygulamalı Bilimler Dergisi
Journal of Applied Sciences of Mehmet Akif Ersoy University

Thermal Performance of a Tube Equipped with V-nozzle Turbulator Inserts

Sibel Güneş[1*], Orhan Keklikcioğlu[2], Toygur Dağdevir[2], Veysel Özceyhan[3]

[1]Assoc. Prof. Dr., Erciyes University, Faculty of Engineering, Department of Mechanical Engineering

[2]Research Assistant, Erciyes University, Faculty of Engineering, Department of Mechanical Engineering

[3]Prof. Dr., Erciyes University, Faculty of Engineering, Department of Mechanical Engineering

Geliş Tarihi/Received: 07.11.2017
Kabul Tarihi/Accepted: 06.12.2017 Araştırma Makalesi/Research Article

ABSTRACT

This study pertains to the numerical investigation of the influence of V-nozzle turbulators on thermal performance in a circular tube under uniform heat flux condition. The converging-diverging nozzles as a venturi structure were located in the test tube in order to generate turbulance/reverseflow. The computations were carried out by consideration of various pitch ratios, PR = 2, 4, and 6 for Reynolds numbers between 5000 and 32000, using air as a working fluid. The variation of Nusselt number (Nu) and friction factor (f) versus Reynolds number (Re) were introduced for the obtained numerical results. The plain tube results were compared with the works available in literature for confirmation of the used numerical method. The rates of increase in Nusselt number over the smooth tube were presented as 159-218 %, 144-201 % and 132-185 % in this study for PR=2, 4 and 6, respectively. In addition, heat transfer coefficient ratio values for V-nozzle turbulators with PR=2, 4 and 6 were around 1.77, 1.61, and 1.50, respectively. Consequently, the results demonstrated that the usage of V-nozzles at smaller pitch ratio yielded an increase in heat transfer rate and pressure loss.
Keywords: V-nozzle, Turbulent Flow, Thermal Performance.

V-lüle Türbülator Yerleştirilen Bir Borunun Termal Performansı

ÖZET

Bu çalışma da sabit ısı akısı uygulanan silindirik bir boru içerisindeki V-lüle türbülatörlerin ısıl performansa etkisi sayısal olarak incelenmiştir. Ventüri yapısına benzer ıraksak ve

* Sorumlu yazar/Corresponding author
E-mail/e-ileti: sgumus@erciyes.edu.tr

yakınsak lüleler test borusu içerisine türbülans/tersakış oluşturmak amacıyla yerleştirilmiştir. Hesaplamalar PR=2,4 ve 6 Reynolds sayısının ise 5000 ile 32000 aralığında değiştiği farklı değerlerde hava akışkanı kullanılarak gerçekleştirilmiştir. Elde edilen sayısal sonuçlar için Nusselt sayısı (Nu) ve sürtünme faktörü (f) ile Reynolds sayısı (Re) arasındaki değişim ortaya konulmuştur. Boş boru sonuçları, kullanılan sayısal yöntemin doğrulanması için literatürde bulunan diğer çalışmalar karşılaştırılmıştır. Boş boruya oranla Nusselt sayısındaki artış gerçekleştirilen çalışmada PR=2, 4 ve 6 modelleri için sırasıyla % 159-218, %144-201 ve % 132-185 olarak gerçekleşmiştir. Buna ek olarak, V-lüle kullanımı ile birlikte ısı transfer katsayısı PR=2, 4 ve 6 modelleri için sırasıyla 1.77, 1.61, ve 1.50 civarında gerçekleşmiştir. Sonuç olarak, elde edilen sonuçlar daha küçük hatve oranında V-lüle kullanımının ısı transferi miktarında ve basınç kaybında bir artış sağladığını göstermiştir.

Anahtar kelimeler: V-lüle, Türbülanslı Akış, Isıl Performans.

1. INTRODUCTION

Heat transfer is an important matter in thermal engineering and industrial applications, such as solar air/water heater, refrigeration, air-conditioning, heat pump, petroleum, chemical and electricity generation, etc. Several types of inserts such as fin, rib, baffle, coiled wire and twisted tape, have been extensively utilized as the turbulators or passive heat transfer enhancement devices in heat exchangers (Eimsa-ard and Promvonge, 2006; Şahin and Demir, 2008). The inserts can create one or more combinations of the following conditions that are favorable for the increase in heat transfer rate: (1) disruption of the development of thermal/ velocity boundary layer and increase of the turbulence intensity, (2) increase in heat transfer area, and (3) generation of swirling/rotating, vortexing and/or secondary flows. In common, the use of inserts results in an increase flow resistance, thus a rise of the power requirement for pumping the working fluids. The tradeoff between the enhanced heat transfer and increased friction loss is strongly dependent on the insert geometries. The proper design of the inserts is necessary for enhancing heat transfer with a reasonable friction loss.

Reverse/swirl flow devices form an important group of the passive augmentation techniques. The reverse flow, sometimes called ‘‘recirculation flow’’, device or the turbulator is widely employed in heat transfer engineering applications. This is because the convection heat transfer along the tube wall can be improved significantly by introducing the reverse/re-circulation flow to increase the effective axial Reynolds number and decrease the cross-sectional area of flow, leading to an increase in the mean velocity and temperature gradient.

The reverse flow cannot only induce the higher heat fluxes and momentum transfer due to the large effective driving potential force but also the higher pressure drop. The strength of reverse flow and the reattached position are the main interest in many heat transfer applications such as heat exchangers, combustion chambers, gas turbine blades, and electronic devices. The methods of generating swirl can be classified into three main categories. The first is the tangential flow injection to induce a swirling fluid motion along the tube (Dhir et al, 1990; Son and Dhir,1993). The second is the guide vanes swirl generators (Yılmaz et al., 1999; Yılmaz et al., 2003) classified into two types: the radial guide vane and the axial guide vane. The last one is the direct rotation of the tube.

The primal aim of this study to investigate the eff ect of diff erent pitch ratio of V-nozzle turbulators on heat transfer rate and friction factor coefficient at diff erent Reynolds number. In this study, in contrast to the literature, the V-nozzle turbulators are inserted with the constant and proportional pith ratios of 2, 4 and 6. This intervention helps to analyze the variance of pitch ratio is proportional the increment of Nusselt number or pressure drop in the tube with V-nozzle turbulators.

In the present study, the V-nozzles are placed inside the test tube at three pitch ratios (PR), defined as a ratio of pitch length to tube diameter; PR=2.0, 4.0, and 6.0. All of the numerical simulations are performed at the same inlet conditions with the Reynolds number, based on the test tube diameter, in a range of 5000–32000. Consequently, the variation of Nusselt number (Nu) and friction factor (f) versus Reynolds number (Re) were introduced for the obtained numerical results.

2. NUMERICAL SOLUTION PROCEDURE

2.1. Solution domain

The physical properties of numerical study is illustrated in Fig. 1. As depicted in Fig. 1, the test-tube section is consist of copper having inner diameter of 47.5 mm (D_i), outer diameter (D_o) of 50.5 mm, length of 1250 mm (L) and thickness of 1.5 mm. The V-nozzle material is conducted as Aluminum with s=95 mm (2.0D) in length and its end and throat diameters are 46mm (K_2) and 26 mm (K_1), respectively. In order to get fully developed steady flow, the tube is arranged with entrance section having length of 10D. Besides, to prevent the reverse flow error in CFD analyzes, exit section is placed as length of 5D. The V-nozzles are placed with three different pitch lengths, having p=95 mm (PR=2.0), p=190 mm (PR=4.0), and p=285 mm (PR=6.0), for each experiment.

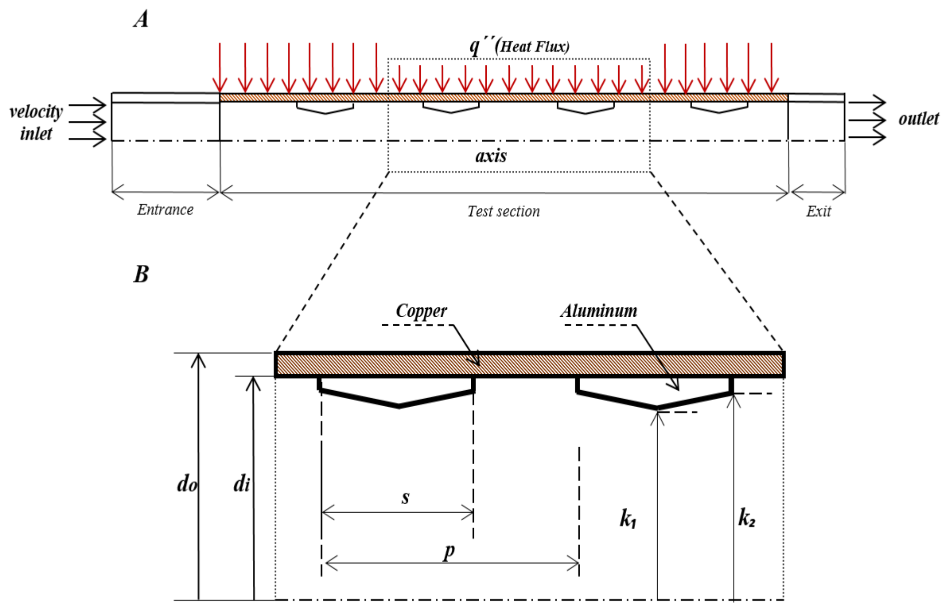

Figure 1. Physical properties of solution domain.

2.2. Analysis method

Numerical calculations are performed to solve the problem depending on physical model which consist three different pitch ratio for ten Re numbers ranging from 5000 to 32000.

A careful check for grid independence is required for ensuring the validity and accuracy of numerical methodology. For this purpose, detailed grid independence tests are conducted and according to the grid models. Mesh models of smooth tube are created for the numbers of cells from 322,423 to 987,113 and are tested for three different (0.25, 0.35, 0.50) cell sizes. After 0.35 cell size as given grid structure in Fig. 2, Nusselt number has increased less than 2.3% conjunction with accrual of cells number.

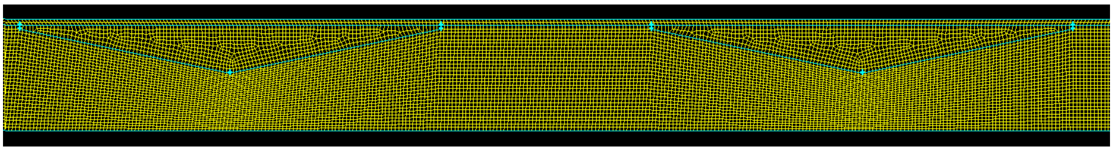

Figure 2. Grid structure for 0.35 cell size.

The two-dimensional(axisymmetric) continuity, momentum and energy of Navier-Stokes equations are solved by using finite volume method and the SIMPLE algorithm scheme is applied to examine the effects of turbulent flow on heat transfer and friction characteristics. Given constant properties in Table 1 for air is used at 300 K inlet temperature.

Table 1. The thermophysical properties of the air at 300 K.

T(K)	*ρ(kg/m³)*	*cp(kJ/kgK)*	$\mu x 10^{-5}$ *(kg/m s)*	*k(W/m K)*
300	1.225	1.0061	789	0.0242

In present study, GAMBIT 4.4.2 is used to plot and mesh the tube and V-type nozzle turbulator. FLUENT 14.5 was preferred to analyze numerical model. Re number calculated from Eq. 11 with respect to thermophysical properties of air and given velocity. The renormalization-group (RNG) which is obtained better agreement with validated model is used. A pressure based, double-precision solver is selected to solve used equations. According to the given assumptions the governing equations for present study are written as follows (Menter,1994).

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

Momentum equation:

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + v\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \quad (2a)$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial y} + v\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) \quad (2b)$$

Energy equation:

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \quad (3)$$

where u and v are the velocities in the x and y directions, respectively, v, ρ, α, and T are the kinematic viscosity, the density, the thermal diffusivity and the temperature of the air, respectively.

The RNG k-ε turbulent model is selected for the analyzing of numerical models. This turbulent model uses two transport equations for turbulence kinetic energy (k) and dissipation rate (ε). The equations can be written as below:

$$\frac{\partial}{\partial \tau}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left(\alpha_k \mu_{eff}\frac{\partial k}{\partial x_j}\right)+G_k-\rho\varepsilon+S_k \quad (4)$$

$$\frac{\partial}{\partial \tau}(\rho \varepsilon)+\frac{\partial}{\partial x_i}(\rho \varepsilon u_i)=\frac{\partial}{\partial x_j}\left(\alpha_\varepsilon \mu_{eff}\frac{\partial \varepsilon}{\partial x_j}\right)+C_{1\varepsilon}G_k\frac{\varepsilon}{k}-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}-R_\varepsilon+S_\varepsilon \quad (5)$$

where $C_{1\varepsilon}=1.42,\ C_{2\varepsilon}=1.68$ are the model constants, $G_k$is the generation of the turbulent kinetic energy, $\alpha_k$and α_ε refer to turbulent Prandtl number in k and ε equations, respectively. Additionally R_ε is given by

$$R_\varepsilon=\frac{C_\mu\rho\eta^3\left(1-\frac{\eta}{\eta_0}\right)\varepsilon^2}{1+\beta\eta^3 k} \quad (6)$$

where $\eta=\frac{S_k}{\varepsilon}, \eta_0=4.38,\ \beta=0.012.$

2.3. Boundary conditions

Determining boundary conditions are so important to be able to get a good agreement with experimental results. In this study, values of thermophysical properties of air have been given in Table 1 for 300 K. The turbulent flow is Reynolds number ranging from 5000 to 32000.At the inlet velocity inlet is normal to boundary and uniform and pressure outlet boundary condition is specified with gauge pressure of 0 Pa at the exit. The turbulence intensity has been estimated tube flow from the Eq. 7 and the hydraulic diameter has been given at numerical model as 47.5 mm. A horizontal plane that is parallel to the x-axis and connected with numerical model was preferred axis as shown in Fig. 1. Uniform heat flux (constant q=1000 W/m^2) is applied outer surface of the tube. The walls in contact with the fluid have no-slip boundary condition.

$$I=0.16(Re)^{-1/8} \quad (7)$$

2.4. Calculation heat transfer and friction factor

The uniform heat flux applied to the tube can be written as;

$$q=\frac{Q}{\pi D L} \quad (8)$$

The convective heat transfer coefficient through the tube is defined as;

$$h = \frac{q}{T_{iw} - T_b} \tag{9}$$

Here, T_{iw} and T_b represent inner wall temperature of the numerical method and bulk temperature of fluid.

The Nusselt and Reynolds numbers can be calculated from;

$$Nu = \frac{hD}{k} \tag{10}$$

where k is the conductive heat transfer coefficient of fluid.

$$Re = \frac{UD}{v} \tag{11}$$

where D is hydraulic diameter, U is velocity, v is kinematic viscosity.

The friction factor is defined as;

$$f = \frac{\Delta P}{\frac{1}{2}\rho . U^2 \frac{L}{D}} \tag{12}$$

The overall enhancement efficiency (η) at constant pumping power is the ratio of the convective heat transfer coefficient of the tube with V-nozzle turbulators to the plain tube which can be written as follows

$$\eta = \left. \frac{h_t}{h_p} \right|_{pp} \tag{13}$$

3. NUMERICAL RESULTS

The effect of V-type nozzle turbulator on heat transfer and pressure drop in a tubular flow numerically investigated in the present study. Three different pitch ratios and ten different Reynolds number are examined and obtained results are presented in this section.

3.1. Calculation heat transfer and friction factor

In this present work, in order to demonstrate the accuracy of the numerical results on Nusselt number and friction factor are compared with a previous study which carried out by Eiamsa-ard and Promvonge (2006) and, as shown in Figs. 3 and 4, respectively.

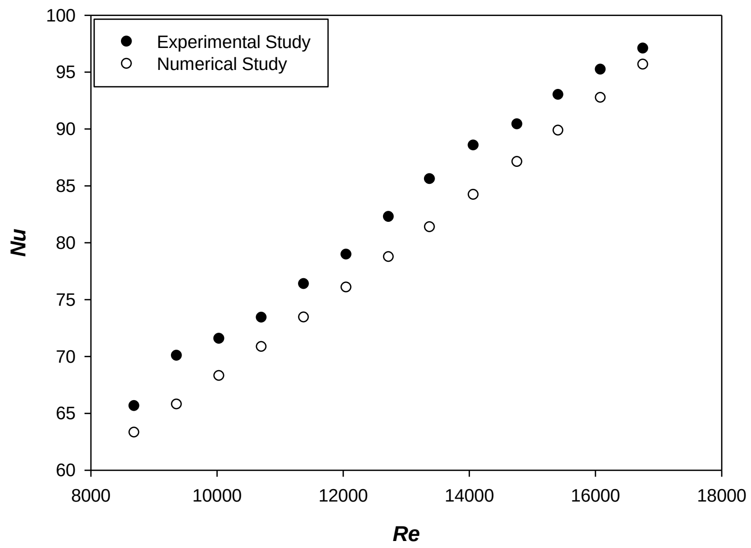

Figure 3. Comparison of Nusselt number results with experimental study for PR=4.

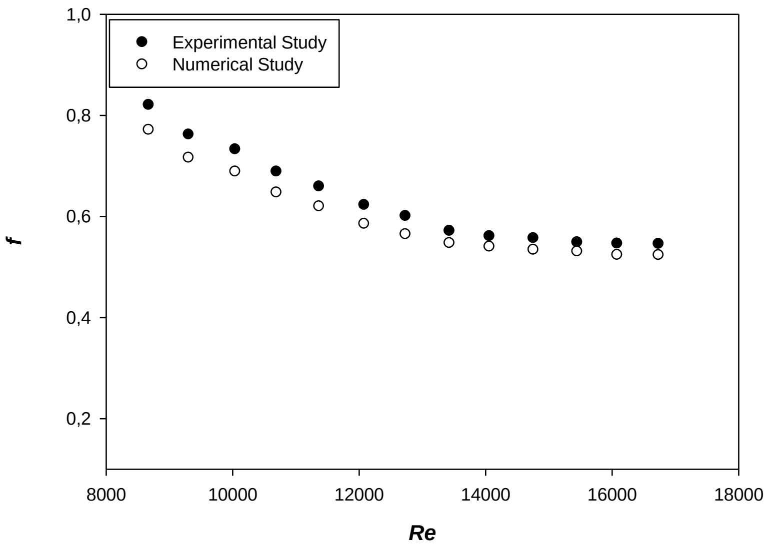

Figure 4. Comparison of friction factor results with experimental study PR=4.

Both Nusselt number and friction factor values were validated in the turbulance flow conditions. As shown in these figures, for all of the Reynolds numbers examined, the results of the numerical study in good agreement with the results obtained from the experimental study. It is found that the maximum deviations between the given results are 9.76% and 4.91% for Nusselt number and friction factor, respectively. This observation demonstrates the accuracy of the numerical procedure.

3.2. Effect of pitch ratio on heat transfer and pressure drop

This experimental study investigated V-type nozzles with three different pitch ratios to determine the heat transfer enhancement and flow friction characteristics of a tube with a V-nozzle type insert. The variations in Nusselt numbers and friction factors related to different pitch-to-tube diameter ratios (PR=2, 4 and 6) are presented in Figs. 5 and 7.

It is clear that the heat transfer rate increases with a decrease in pitch ratio as shown in Fig. 5. It can be seen in from Fig. 5, that an increase in Nusselt number is obtained when the pitch ratio decreases; therefore, for all Re numbers, the greatest heat transfer enhancement occurred for the V-nozzle insert with PR=2.

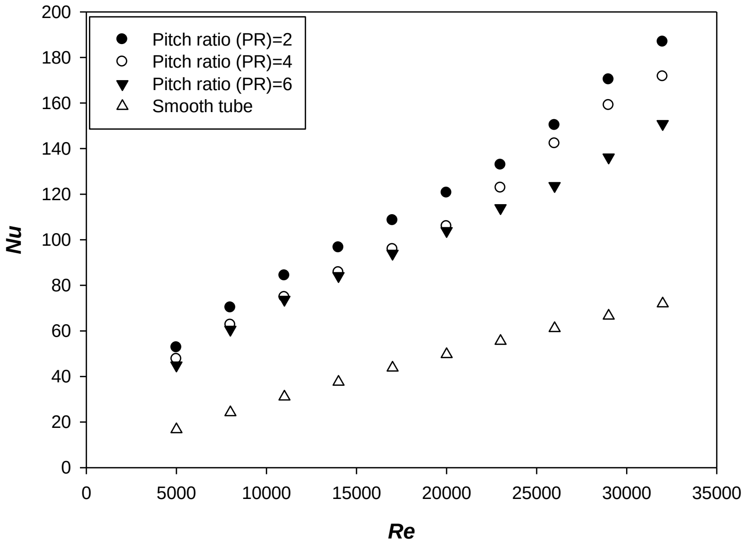

Figure 5. Results of Nusselt number versus Reynolds number for different pitch ratios.

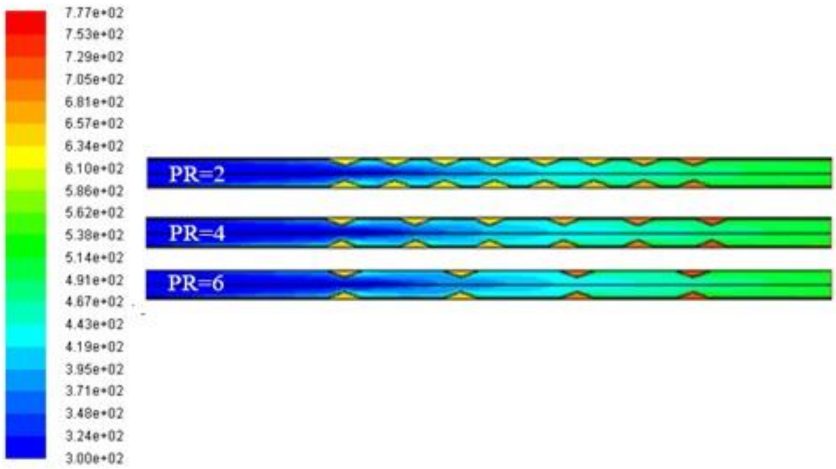

Figure 6. Total temperature contours (K) for all cases at Reynolds number of 20000.

The rates of increase in Nusselt number over the smooth tube are presented as 159-218 %, 144-201 % and 132-185 % in this study for PR=2, 4 and 6, respectively. The highest

Nusselt number is obtained for the configuration of PR=2 model at the highest Reynolds number of 32000.

The friction factors for the tube with the V-nozzle turbulators having the ratios PR=2, 4, and 6 are plotted against the Reynolds numbers in Fig. 7. As expected, the friction factor decreases with increasing pitch ratio. The highest friction factor can be seen for the ratio of PR= 2. This means that as the number of V-nozzle fitted into the test section increases and as the disturbance of the flow field increases, the Nusselt number and the friction factor simultaneously increase.

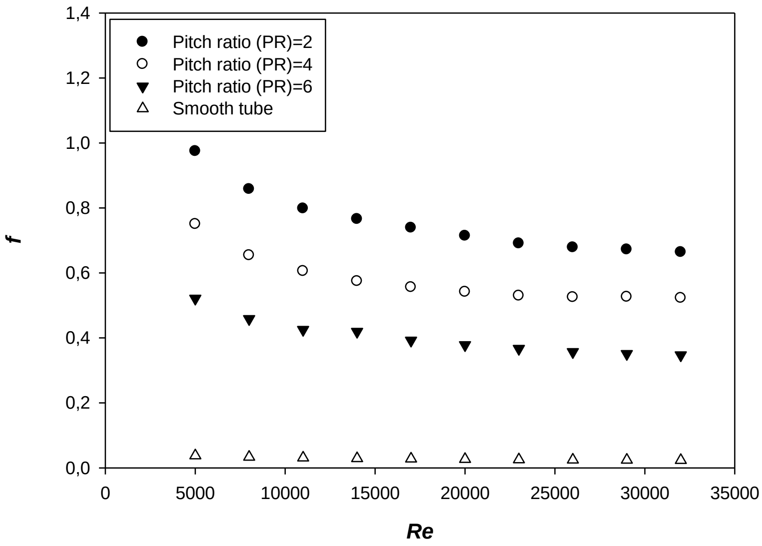

Figure 7. Results of friction factor versus Reynolds number for different pitch ratios.

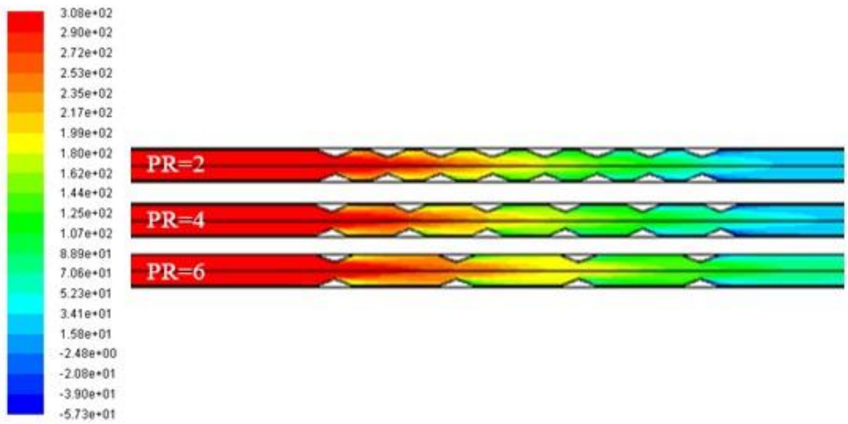

Figure 8. Total pressure contours (Pa) for all cases at Reynolds number of 20000.

3.3. Performance evaluation

Overall enhancement ratios are used to evaluate the performance of the tube fitted with V-nozzle turbulators. As shown in Fig. 9, the convective heat transfer coefficient ratio that describes the ratio of the convective heat transfer coefficient of the tube with V-nozzle turbulator (h_t) to that of the smooth tube (h_{pp}) tends to decrease with increasing Reynolds number for all cases. The variation of the convective heat transfer coefficient ratio with the Reynolds number is presented in Fig. 9. The V-nozzle turbulators in the tube cause not only an increase in Nusselt number but also an increase in the friction factor. For all cases, the convective heat transfer coefficient ratio tends to decrease with increasing Reynolds number.

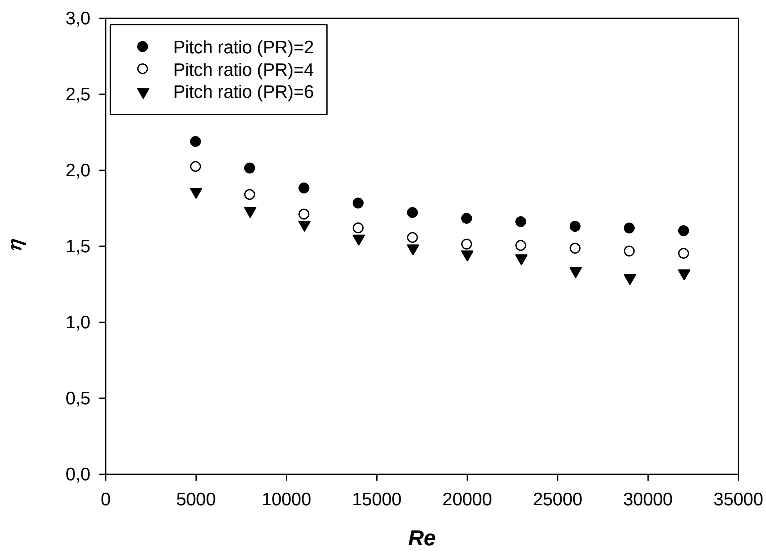

Figure 9. Results of convective heat transfer coefficient ratio values versus Reynolds number for different pitch ratios.

The average convective heat transfer coefficient ratio values for V-nozzle turbulators with PR=2, 4 and 6 are around 1.77, 1.61, and 1.50, respectively.

4. CONCLUSIONS

The primal aim of study is to investigate the heat transfer and pressure drop in a tube fitted with V-nozzle turbulators. The investigations carried out the range of 5000-32000 Reynolds number by a CFD program, after the numerical model had been plotted and meshed in GAMBIT. In this study totally three different pitch ratios were evaluated for the performance evaluation. The following conclusions can be derived as:

1- The V-nozzle turbulators result in a significance increase in both heat transfer rate and friction factor when compared to the smooth tube.
2- For all cases, Nusselt number increases and friction factor decreases with increasing Reynolds number. The highest Nusselt number and friction factor is obtained for PR=2 model. Decreasing pitch length enhance the heat transfer in the tubular flow. The rates of increase in Nusselt number over the smooth tube are presented as 159-218 %, 144-201 % and 132-185 % in this study for PR=2, 4 and 6, respectively.
3- The friction factor is also affected from nozzle pitch length and Reynolds number. Increasing the amount of the pitch length between turbulators, friction factor is getting reduced. The maximum pressure drop introduced for PR=2 model as expected.
4- The results show that not only is an increase in heat transfer obtained, but friction factor is also increased. Especially at low Reynolds numbers, the system achieves higher.
5- The overall enhancement ratio increases with decreasing pitch length of the V-nozzle turbulator. As a consequence of this, the average convective heat transfer coefficient ratio values for V-nozzle turbulators with PR=2, 4 and 6 are around 1.77, 1.61, and 1.50, respectively.

REFERENCES / KAYNAKLAR

Buchlin, J.M.(2002), "Convective Heat Transfer in a Channel with Perforated Ribs", Int J Therm Sci, (41), 332–340.

Chang, S.W., Jan, Y.J. and Liou, J.S.(2007), "Turbulent Heat Transfer and Pressure Drop in Tube Fitted with Serrated Twisted Tape", Int J Therm Sci, (46), 506–518.

Dhir, V.K., Chang, F. and Yu, J.(1990), "Enhancement of Single Phase Forced Convection Heat Transfer in Tubes Using Staged Tangential Flow Injection", Final Report, June 1987–December 1989, GRI-90/0134.

Dhir, V.K. and Chang, F.(1992), "Heat Transfer Enhancement Using Tangential Injection", ASHRAE Transactions (98), BA-92-4-1.

Eimsa-ard, S. and Promvonge, P.(2006), "Experimental Investigation of Heat Transfer and Friction Characteristics in a Circular Tube Fitted with V-nozzle Turbulators", Int Commun Heat Mass Transfer, (33), 591–600.

Eimsa-ard, S., Thianpong, C. and Promvonge, P.(2006), "Experimental Investigation of Heat Transfer and Flow Friction in a Circular Tube Fitted with Regularly Spaced Twisted Tape Elements", Int Commun Heat Mass Transfer, (33), 1225–1233.

Jordan, S.A.(2003), "The Turbulent Character and Pressure Loss Produced by Symmetric Ribs in a Circular Duct", Int J Heat Fluid Flow, (24), 795–806.

Lu, B. and Jiang, P.X.(2006), “Experimental and Numerical Investigation of Convection Heat Transfer in a Rectangular Channel with Angled Ribs”, Exp Therm Fluid Sci, (30),513–521.

Menter, F.R. , (1994), “Two-equation Eddy-Viscosity Turbulence Models For Engineering Applications”, American Institute of Aeronautics and Astronautics, (32), 283–300.

Naphon, P.(2006), “Effect of Coil-Wire Insert on Heat Transfer Enhancement and Pressure Drop of Horizontal Concentric Tubes”, Int Commun Heat Mass Transfer, (33), 753–763.

Sahin, B. and Demir, A.(2008), “Performance Analysis of a Heat Exchanger Having Perforated Square Fins”, Appl Therm Eng, (28), 621–632.

San, J.Y. and Huang, W.C.(2006), “Heat Transfer Enhancement of Transverse Ribs in Circular Tubes with Consideration of Entrance Effect”, Int J Heat Mass Transfer, (49), 2965–2971.

Shaikh, N. and Siddiqui, M.H.K.(2007), “Heat Transfer Augmentation in a Heat Exchanger Tube Using a Baffle”, Int J Heat Fluid Flow, (28), 318–28.

Shoji, Y., Sato, K., and Oliver D.R.(2003), “Heat Transfer Enhancement in Round Tube using Wire Coil Influence of Length and Segmentation”, Heat Transfer-Asian Res, (32), 99-107.

Son, G., Dhir V. K. (1993), Enhancement of heat transfer in an annulus using tangential flow injection, Trans ASME Journal Heat Transfer, (246), 59–66.

Tanda, G.(2001), “Heat Transfer and Pressure Drop in a Rectangular Channel with Diamond-Shaped Elements”, Int J Heat Mass Transfer, (49), 3529–3541.

Tanda, G.(2004), “Heat transfer in Rectangular Channels with Transverse and V-shaped Broken Ribs”, Int J Heat Mass Transfer, (47), 229–43.

Tandiroglu, A., and Ayhan, T.(2006), “Energy Dissipation Analysis of Transient Heat Transfer for Turbulent Flow in a Circular Tube with Baffle Inserts”, Appl Therm Eng, (26),178–185.

Tijing, L.D., Pak, B.C., Baek, B.J. and Lee, D.H.(2006), “A Study on Heat Transfer Enhancement Using Straight and Twisted Internal Fins”, Int Commun Heat Mass Transfer, (33), 719–726.

Yakut, K., Alemdaroglu, N., Sahin, B.,and Celik, C.(2006), “Optimum Design Parameters of a Heat Exchanger Having Hexagonal Fins”, Appl Energy, (83), 82–98.

Yılmaz, M., Comaklı, O. and Yapıcı, S. (1999), “Enhancement of Heat Transfer by Turbulent Decaying Swirl Flow”, Energy Conversion Management, (40), 1365–1376.

Yılmaz, M., Comaklı, O. and Yapıcı, S. and Sara, O.N. (2003), “Heat Transfer and Friction Characteristics in Decaying Swirl Flow Generated by Different Radial Guide Vane Swirl Generators”, Energy Conversion and Management, (44), 283–300.

Zhang, Z., Yu, Z. and Fang, X.(2007), “An Experimental Heat Transfer Study for Helically Flowing Outside Petal-Shaped Finned Tubes with Different Geometrical Parameters”, Appl Therm Eng, (27), 268–272.

Zimparov, V. (2001), “Enhancement of Heat Transfer by a Combination of Three-Start Spirally Corrugated Tubes with a Twisted Tape”, Int J Heat Mass Transfer,(44), 551–574.