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A CFD BASED THERMO-HYDRAULIC PERFORMANCE ANALYSIS IN A TUBE FITTED WITH STEPPED CONICAL NOZZLE TURBULATORS

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ABSTRACT

The paper presents a numerical analysis of heat transfer and friction characteristics in a stepped nozzle inserted tube. Nine different configurations are used for numerical analysis. In the first part of study, to certify the Nu (average Nusselt number) and the f (average friction factor), the computational fluid dynamics (CFD) models of tube with conical nozzle turbulators are validated with the experimental results available in the literature. In the present work, the turbulators are thoroughly inserted in the tube three different pitch ratios, with various Reynolds numbers ranging from 6000 to 22000, and also with three step numbers. The investigations were carried out with the RNG k-ε turbulence model in a CFD package program, after some different turbulence models were tried, and grid independency with three different grid models were analyzed, for the average Nusselt number, and the average friction factor. As a consequence, Nusselt number increases and friction factor decreases with increasing Reynolds number. The highest Nusselt number and friction factor is obtained for s/a=4, p/s=2 model. The overall enhancement ratio rises with decreasing spacing of the stepped conical nozzle and increasing step value. The ultimate overall enhancement of 11% was achieved for $Re=6000$ where the spacing and step ratios are 4, 2 respectively.

INTRODUCTION

Enhancement of heat transfer is a process of supplement the thermo-hydraulic performance by using different methods. The process separated into two methods which are passive and active [1]. Passive methods are widely used both experimental and numerical applications when investigating the heat transfer enhancement and friction loss for saving energy and cost. There are many passive methods to increase heat transfer rate using variety of components which located into flow field such as twisted tapes and ribs. Although these kinds of techniques do not necessitate any additional energy to improve heat transfer, they increased friction factor. Therefore they cannot be effectively used in applications where the pressure loss is significant.

Conical turbulators are one of the most commonly used method in heat transfer applications such as power generation, chemical industry, refrigeration and air-conditioning. In addition, many researchers have fulfilled experimental analysis on conical turbulators. Moreover, to save time, energy and cost, there are plenty of numerical analysis are situated in researches.

Promvonge and Eiamsa-ard [2] investigated the heat transfer enhancement in a tube with combined conical-nozzle inserts and swirl generator. They employed swirl generator to provide swirling flow at the inlet of the test tube, and for each run they examined three different pitch ratios (2.0, 4.0, 7.0) in a range of 8.000-18.000 Reynolds number. Ultimately maximum heat transfer rate reached up to 316%. Dagdevir et. al. [3] numerically examined the effect of nozzles formed sinusoidal geometry fitted in a circular tube on heat transfer enhancement. Anvari et. al. [4] carried out in order to find the role of conical rings for the heat transfer enhancement and pressure drop change in a pipe. They placed the conical rings in two different arrangements: converging conical rings, diverging conical rings with Reynolds number which was changed between 1000-10000.Consequently results revealed that conical ring inserts have an unfavorable effect on the enhancement efficiency of the heat transfer in this experiment. Paisarn Naphon [5] presented in his study about heat transfer characteristics and pressure drop in the corrugated channel with two opposite corrugated plates which have three different corrugated tile angles and Reynolds number in the ranges from 500 to 1400. According to experimental results, the corrugated surface has significant effect on heat transfer enhancement and pressure drop. Ozceyhan et. al. [6] investigated the heat transfer enhancement in a tube with circular cross sectional rings numerically. With five different pitch ratios and in the range of Reynols number 4475-43725.At the end of the numerical analysis the best overall enhancement 18% was achieved for Re=15,600. Eiamsa-ard et. al. [7] studied the influences of Perforated-conical ring (PCR) on the turbulent convective heat transfer, friction factor and thermal performance factor characteristics experimentally. The perforated conical-rings used are of three different pitch ratios and three different numbers of perforated holes with Reynolds number between 4000 and 20,000. As a result it is found that the PCR considerably diminishes the development of thermal boundary layer, leading to the heat transfer rate up to about 137% over that in the plain tube. Pongsoi and Wongwises [8] carried out the effect of fin pitches on the performance of L-footed spiral fin and tube heat exchangers. Erdal Cetkin [9] studied the effect of the shape of microvascular channels on the cooling performance. Promvonge and Eiamsa-ard [10] investigated the effect of a free-spacing snail entry together with conical-nozzle turbulators on turbulent heat transfer and friction characteristics in a tube. The insertions of the conical or converging nozzle with different pitch ratios in common with the free-space snail entry are examined in a Reynolds number range from 8000 to 18000. Over the range investigated, the Nusselt numbers for employing both the enhancement devices are found to be higher than that for the plain tube around 315%, 300% and 285% respectively

Validated and geometrical model

Numerical analysis need to be supported experimental results, for the best agreements different turbulence models and grid independency should be investigated. In this present work an experimental investigation was validated with represented work done by Promvonge and Eiamsa-ard. Detailed descriptions of the experimental investigation can be found in Promvonge and Eiamsa-ard [11] . For the validated model, numerical analysis carried out FLUENT 14.5 package program, to ensure methodology of numerical solutions, physics model formed in GAMBIT 4.4.2 using experimental data.

The geometrical parameters of tube fitted with conical nozzle turbulators are depicted in Fig. 1. and Table 1. The effect of different combinations which encapsulate three spacings and steps will be investigated in the present work by using FLUENT 14.5 non-commercial CFD software. As showed in Fig 1. the tube was preferred of copper with 1.5 mm thickness and it had 1860 mm length (L), 25 mm inner-diameter (di). The conical nozzle step diameters are 14 mm(k_1) and 24 mm(k_2) and step lines are also designed parallel that is represent in Fig. 1.b Air is selected as the working fluid and the inlet bulk air at 300 K passed the heat transfer section that had uniform constant heat flux 1000 W/m^2 along the tube. The conical nozzle turbulators which was selected aluminum attached the tube wall without any clearance.

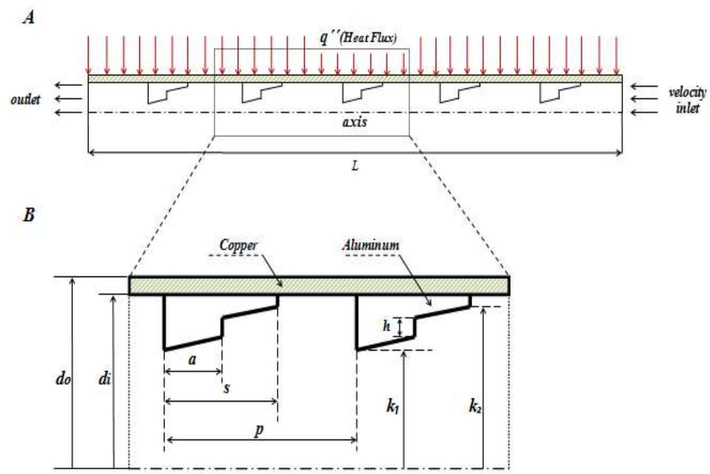
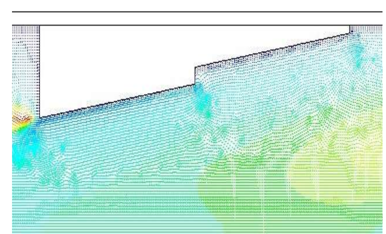
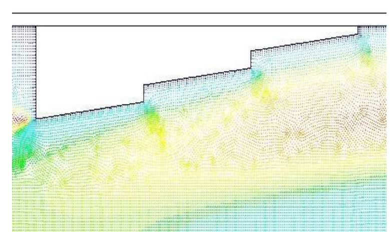


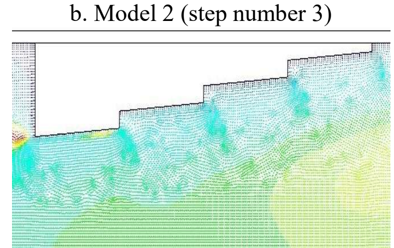
Figure 1 Geometrical properties of nozzle turbulators



a. Model 1 (step number 2)



b. Model 2 (step number 3)



c. Model 3 (step number 4)

Figure 2 Geometrıcal model of nozzle turbulators wıth steps

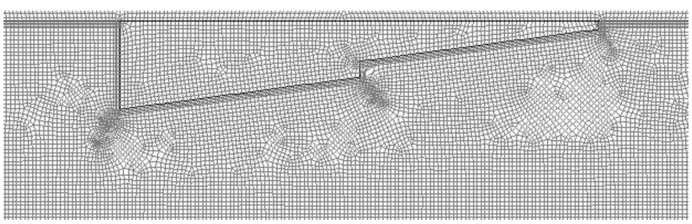
Table 1 Geometrical parameters of tubes

Models	***a(mm)***	***s(mm)***	***p(mm)***	***p/s***	***s/a***	***h(mm)***
Mod.1	*30*	*60*	*120*	*2*	*2*	*2*
Mod.2	*20*	*60*	*120*	*2*	*3*	*2*
Mod.3	*15*	*60*	*120*	*2*	*4*	*2*
Mod.4	*30*	*60*	*180*	*3*	*2*	*2*
Mod.5	*20*	*60*	*180*	*3*	*3*	*2*
Mod.6	*15*	*60*	*180*	*3*	*4*	*2*
Mod.7	*30*	*60*	*240*	*4*	*2*	*2*
Mod.8	*20*	*60*	*240*	*4*	*3*	*2*
Mod.9	*15*	*60*	*240*	*4*	*4*	*2*

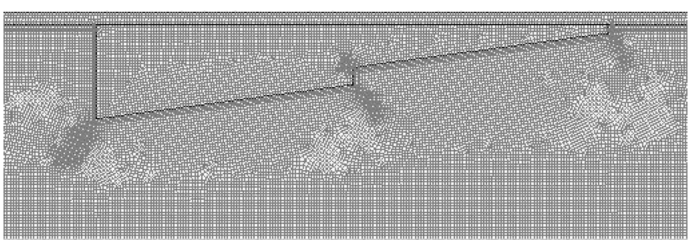
As represented in *Table 1* geometrical models of different configurations are able to seen in Fig.2.

Solution method

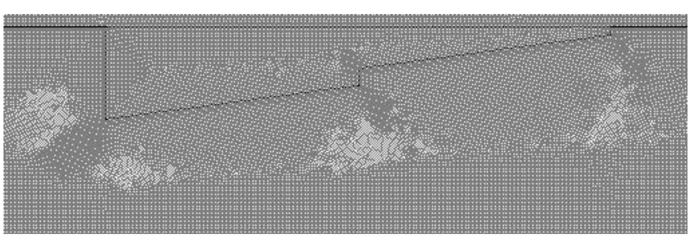
Numerical calculations are performed to solve the problem depending on physical model which consist nine different configurations with steps and spacing at nine Re numbers ranging from 6,000 to 22,000. For grid independence, the numbers of cells varied from 216,813 to 809,783 in three different (0.25, 0.35, 0.50) cell sizes as in Fig. 3.



a. 0.5 cell size and 216,813 cell number



b. 0.35 cell size and 487,253 cell number



c. 0.25 cell size and 809,783 cell number

Figure 3. Grid cell size and cell number for one step conical nozzle turbulators model

According to the grid models, after 0.35 cell size, Nusselt number has increased less than 2% conjunction with accrual of cells number. It is preferred using boundary layer mesh for calculation of viscous effect near wall of the tube and turbulators.

The numerical model under investigation is a steady, two-dimensional (axisymmetric). Given constant properties in Table 2 for air flowing inside assumed at 300 *K* inlet temperature, isotropic. The effect of gravity is negligible and the thermal conductivity of the tube assumed as steady with temperature.

Table 2 The thermophysical properties of
The air at 300 k

T(K)	***ρ(kg/m³)***	***cp(kJ/kgK)***	***$\mu x 10^{-7}$(kg/m- s)***	***k(W/m)***
300	1.1614	1.007	184.6	0.0263

In present study, GAMBIT 4.4.2 was used to plot and mesh the tube and turbulators. After the meshing process, FLUENT 14.5 was preferred to analyze numerical model. Re number calculated from Eq. 13 with respect to thermophysical properties of air and given velocity, because of the calculated Re numbers are in turbulence range *k-ε* model was selected. The renormalization-group (RNG) which was obtained better agreement with validated model was used. A pressure based, double-precision solver is selected to solve used equations. SIMPLE scheme was used in pressure-velocity coupling. The information of pressure drop (ΔP) and temperatures were derived from surface integrals based on area-weighted average.

According to the given assumptions the governing equations for present study are written as follows [12].

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

Momentum equations:

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

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

Energy equation:

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

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 $k-\varepsilon$ turbulent model is selected for the analyzing of numerical models. The k and ε equations in the RNG model described as:

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

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

$$C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}-R_\varepsilon+S_\varepsilon \tag{5}$$

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, α_k and α_ε refer to turbulent Prandtl number in k and ε equations, respectively. Additionally R_ε is given by

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

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

Boundary conditions

It is significant to explain boundary conditions at numerical analysis as results tent to alter least differences. In this study, values of thermophysical properties of air have been given in Table 2 for 300 K. At the tube inlet, as shown in Eq. 7, velocity profile is described in order to keep the hydrodynamic entrance length to be short. Pressure outlet boundary condition is specified with gauge pressure of 0 Pa at the outlet. Temperature is specified as in Eq. 8 Velocity inlet is normal to boundary and uniform. The turbulence intensity has been estimated tube flow from the Eq. 9 and the hydraulic diameter has been given at numerical model as 50 mm. A horizontal plane that is parallel to the x-axis and connected with numerical model was preferred axis as shown in Fig**.** 1. Upper line of copper was defined wall and uniform heat flux applied on the wall. The walls in contact with the fluid have no-slip boundary condition.

$$u=u_c\left(1-\frac{r^2}{R^2}\right) \tag{7}$$

$$T=T_c+\frac{qR}{k}\left[\left(\frac{r}{R}\right)^2-\frac{1}{4}\left(\frac{r}{R}\right)^4\right] \tag{8}$$

$$I=0.16(Re)^{-1/8} \tag{9}$$

Material properties of copper and aluminum are imposed in Table 3.

Table 3 The thermophysical properties of the copper and aluminum at 300 k

Materials	***ρ(kg/m³)***	***cp(J/kgK)***	$\boldsymbol{kx10^3}$ ***(W/m)***
Copper	8978.0	381.0	387.6
Aluminum	2719.0	871.0	202.4

Calculation heat transfer and friction factor

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

$$q=\frac{Q}{\pi D L} \tag{10}$$

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

$$h=\frac{q}{T_{iw}-T_b} \tag{11}$$

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;

$$Nu=\frac{hD}{k} \tag{12}$$

where k is the conductive heat transfer coefficient of fluid.

$$Re=\frac{UD}{v} \tag{13}$$

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

The friction factor is defined as;

$$f=\frac{\Delta P}{\frac{1}{2}\rho.U^2\frac{L}{D}} \tag{14}$$

The overall enhancement efficiency (η) is introduced by Webb[13]:

$$\eta=\left(\frac{Nu_c}{Nu_s}\right)\left(\frac{f_s}{f_c}\right)^{1/3} \tag{15}$$

RESULTS AND DISCUSSION

The numerical investigations of hydraulic performance in a tube fitted with stepped conical nozzle turbulators of different pitch ratios, and stepped number are presented. According to the numerical analysis, obtained results are compared in this section.

Validation of numerical model

First of all, Nusselt number and the friction factor for numerical method are compared with experimental results of Eiamsa-ard and Promvonge [11].

Figs.4 and 5. show comparison between the results of the present study and experimental data which implied above. There is a good agreement between the results for the present numerical method and experimental. The verified that numerical model was accurate.

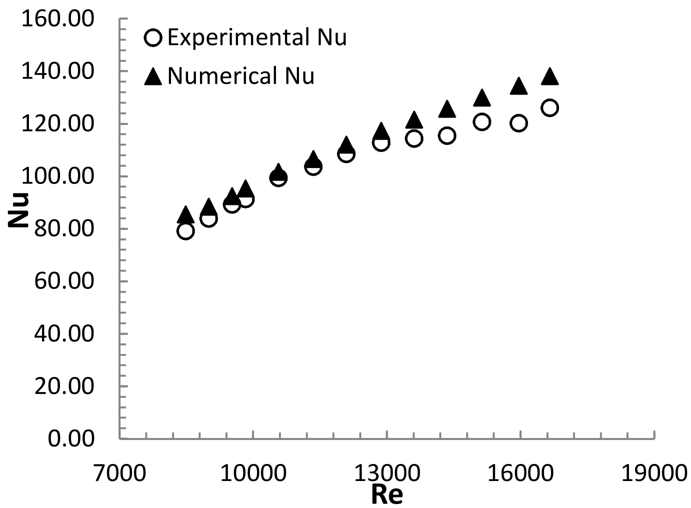


Figure 4 Comparison of experimental and numerical nu number

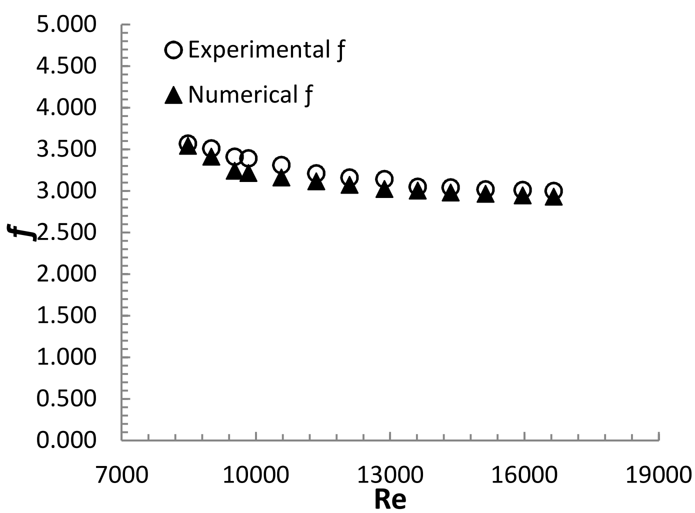


Figure 5 Comparison of experimental and numerical friction factor

Effect of step number

The present numerical study carried out three different spacings with three different steps for investigating the heat transfer enhancement and flow friction chareacteristics of a tube which inserted conical nozzle turbulators inside. The effect of step ratios s/a on heat transfer and friction factor for a constant spacing ratio p/s=2 are represented in Figs.6 and 7.

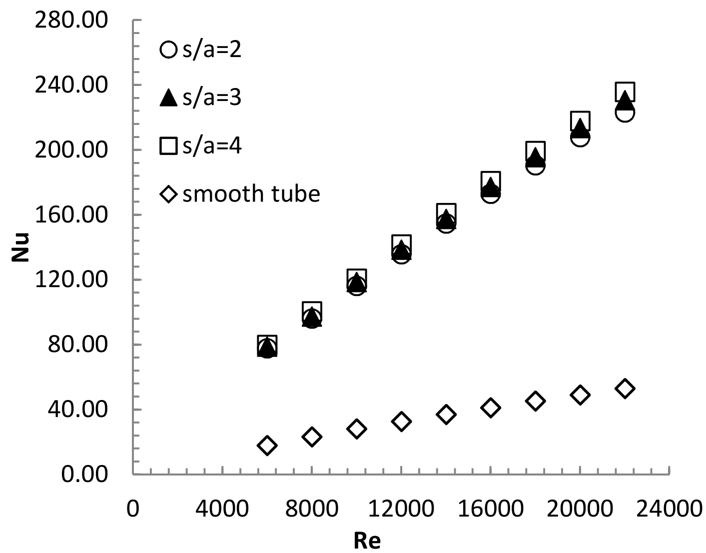


Figure 6 Results of nusselt number with reynolds number for different s/a step ratios at constant p/s=2 spacing ratio

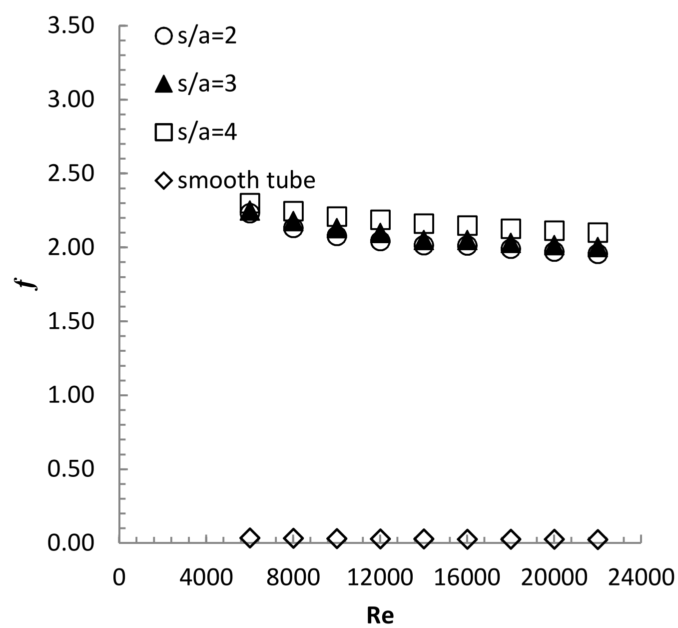


Figure 7 Results of friction factor with reynolds number for different s/a step ratios at constant p/s=2 spacıng ratıo

It can be seen from Figs.6 and 7, both heat transfer rate and friction factor are higher than smooth tube. The Nusselt number increases with the increase of Reynolds number. According to the Fig. 6 while the highest Nusselt number located for s/a=4 ratio,maximum friction factor also seen at same ratio, as expected. This is related with the step number which increase the heat transfer rate and turbulance intensity in the tube.

Effect of spacings

As one can see in Figs.8 and 9 the rise of Nusselt number and friction factor related with different spacing ratios at constant s/a=2 ratio. It is clear that heat transfer rate increases with the reduction of spacing ratio from Fig. 6.

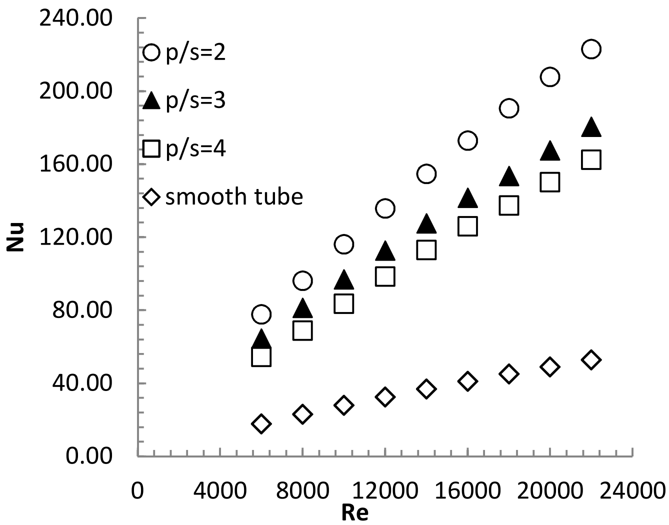


Figure 8 Results of nusselt number with reynolds number for different p/s spacing ratios at constant s/a=2 step ratio

It is clearly seen from Fig.8 The increase of Nusselt number is optimal at the spacing ratio p/s=2 and the smaller spacing ratio, the better Nusselt number is. The increasing range of Nussselt number for p/s=4 that is not the best is 211-220% according to the smooth tube.Though it is not the best, the proportion of increasing Nusselt number is significant.

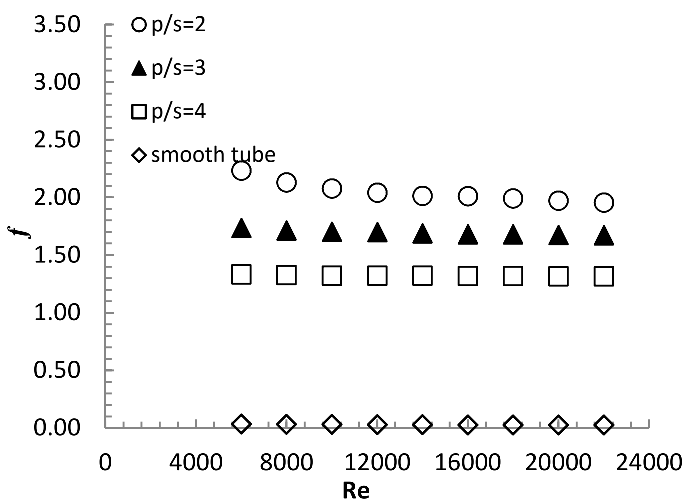


Figure 9 Results of friction factor with reynolds number for different p/s spacing ratios at constant s/a=2 step ratio

From Fig. 9 it can be seen that the friction factor decreases with the increasing Reynolds number. When it compared with the rising spacing ratio, friction factor is started to reduce as shown in Fig. 9. The highest friction factor can be seen at p/s=2 ratio, as expected. That means as the number of the conical nozzles fitted in tube increases Nusselt number and friction factor simultaneously increase.

Performance evaluation

Overall enhancement ratios are used to evaluate the performance of tube fitted with stepped conical nozzle turbulators.

As shown in Fig. 10 optimal enhancement ratio provided for s/a=4, p/s=2 line which means the step number is highest and the spacings are minimum, respectively. At the line enhancement ratio reduce with the increasing Reynolds number as expected. The best enhancement ratio that equals to 1.1137 located on the s/a=4, p/s=2 line for 6000 Reynolds number.

For all models having p/s=2 spacing ratio the enhancement ratio remained above 1 for the range of 6000-22000 Reynolds number

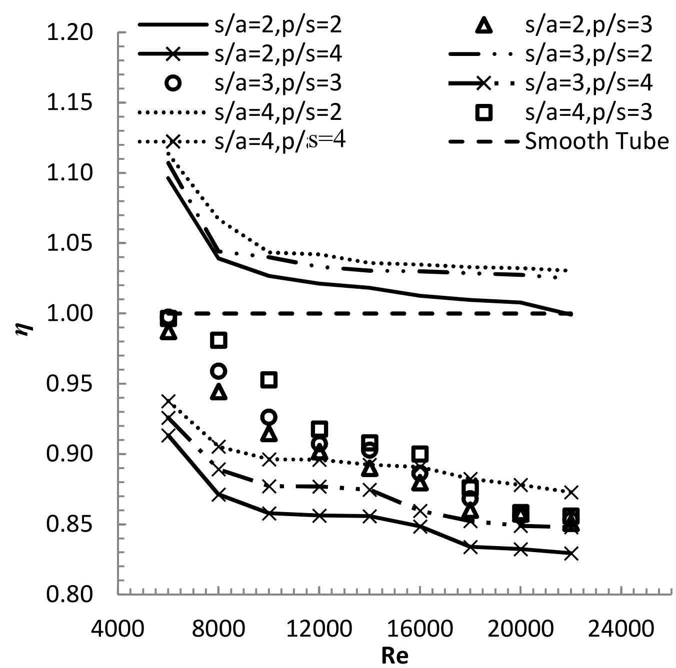


Figure 10 Results of enhancement ratio with reynolds number for both p/s spacing ratios and s/a step ratios

Due to the spacings getting larger and step numbers reduce, the enhancement ratio decrease, as seen in Fig. 10.It can be a good sample for the effect of spacings on the enhancement ratio, comparing between s/a=4,p/s=2 and s/a=4,p/s=4 lines. Besides that comparison of s/a=4,p/s=2 and s/a=2,p/s=2 can be given an example for calculating the effect of step number on enhancement ratio.

CONCLUSION

The primal aim of study is to investigate the heat transfer and pressure drop in a tube fitted with stepped conical nozzle turbulators. The investigations carried out the range of 6000-22000 Reynolds number by using FLUENT 14.5 non-commercial package program, after the numerical model had been plotted and meshed in GAMBIT. In this study totally nine configurations were evaluated for the heat transfer enhancement, the configurations consist three different step number and spacings. The following conclusions can be derived as:

1- For all cases, Nusselt number increases and friction factor decreases with increasing Reynolds number. The highest Nusselt number and friction factor is obtained for s/a=4, p/s=2 model. Increasing step enhance the Nusselt number in same spacing amount, as well.

2- The friction factor is also affected from nozzle spacing and step value. Increasing the amount of the spacings between turbulators and decreasing the step number on the surface of conical nozzles, friction factor is getting reduced. The maximum pressure drop introduced for s/a=4, p/s=2 model as expected.

3-The overall enhancement ratio rises with decreasing spacing of the stepped conical nozzle and increasing step value. As a consequence of this, the ultimate overall enhancement of 11% was achieved for *Re=6000* where the spacing and step ratios are 4,2 respectively.

NOMENCLATURE

Cp	Specific heat of air, j/kgK
D	Hydraulic diameter, mm
h	Heat transfer coefficient, W/m^2K
k	Thermal conductivity of air, W/mK
L	Length of tube, mm
a	Step, mm
h	Height between step, mm
s	Length of a nozzle, mm
q	Heat flux, W/m^2
T	Air temperature, ^0K
T_y	Surface temperature of wall, K
T_b	Mean temperature of fluid, K
T_c	Temperature at the centerline of the tube, K
T_{iw}	Inner wall temperature, K
p	Spacing between two nozzle, mm
d_i	Diameter of fluid area
d_o	Diameter of copper, mm
u	Velocity magnitude of fluid, m/s
u_c	Velocity at the centerline of the tube, K
ΔP	Pressure drop, Pa
k_1	Conical nozzle first step diameter, mm
k_2	Conical nozzle second step diameter, mm
G_k	turbulence kinetic energy generation, m^2/s^2

Dimensionless parameters

I	Turbulence intensity, dimensionless
Nu_D	Average Nusselt number
Nu_c	Nusselt number for tube with conical nozzzle
Nu_s	Nusselt number for smooth tube
f_D	Friction factor
f_s	Friction factor of smooth tube
f_c	Friction factor of tube with conical nozzzle
Re	Reynolds number
Pr	Prandtl number

Greek letters

β	the coefficient of thermal expansion, K^{-1}
ε	turbulence kinetic energy dissipation rate, J/kg.s
k	turbulent kinetic energy, J/kg
μ	Dynamic viscosity, kg/m s
ν	Kinematic viscosity, m^2/s
ρ	Density, kg/m^3
α	Thermal diffusivity, m^2/s

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