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NUMERICAL ANALYSIS OF HEAT TRANSFER ENHANCEMENT IN CONICAL-NOZZLE TURBULATORS INSERTED TUBE

ABSTRACT

This numerical work is associated with the investigation of the effect of turbulator arrangement (diverging and converging) on heat transfer enhancement in conical nozzle turbulators inserted circular tube. Three different pitch ratios (PR=2.5, 5 and 7.5) were considered in the numerical analyses, using air as test fluid. The numerical results of Nusselt number, friction factor and thermal performance factor were introduced for the range of Reynolds numbers of 6000-24000. The obtained results showed that smaller pitch ratios provided more heat transfer rate and pressure drop. As compared to converging nozzles, diverging nozzles generated more effective turbulance/reverse flow, and thus provided higher heat transfer and flow friction. Evidently, obtained results revealed that the usage of nozzle-turbulators as a turbulence/reverse flow generator has an important contribution on heat transfer improvement.

Keywords: Reverse Flow, Nozzle-Turbulator, Heat Transfer, Flow Friction, Nusselt Number

1. INTRODUCTION

High performance thermal systems are of great importance in many engineering applications. Many efforts have been made on heat transfer enhancement according to the progress of thermal systems. The recent researches in heat transfer enhancement lead to the development of currently used heat transfer techniques. These researches focused on finding a technique not only increasing heat transfer, but also achieving higher efficiency. Achieving higher heat transfer rates through various enhancement techniques can result in substantial energy savings, more compact and less expensive equipment with higher thermal efficiency. Ehancement techniques can create one or more combinations of the following conditions that are favorable for the increase in heat transfer rate with an undesirable rise of friction: 1) disruption of the development of boundary layer and increase of the turbulence intensity, 2) increase in heat transfer area, and 3) generation of swirling/rotating and/or secondary flows. Placing different shaped turbulators with different arrangements is one of the extensively used heat transfer enhancement techniques in channel or tube flow due to their advantages of easy fabrication, operation as well as maintenance. There are numerous studies on heat transfer enhancement by inserting various turbulators such as ribs [1 and 2],

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fins [3 and 5], baffles [6 and 8], rings [9 and 10], twisted tapes [11 and 13] and coiled wire inserts [14 and 16].

The convection heat transfer along the tube wall can be improved significantly by introducing the reverse/re-circulation flow with a view to increasing the effective axial Reynolds number, decreasing the cross-sectional area of flow, and increasing the mean velocity and temperature gradient. The reverse flow devices or the turbulators are widely employed in thermal engineering applications. The reverse flow is sometimes called "re-circulation flow". The effect of reverse flow and boundary layer disruption (dissipation) is to enhance the heat transfer coefficient and momentum transfers. The reverse flow with high turbulence can improve convection of the tube wall by increasing the effective axial Reynolds number, decreasing the flow cross-section area, and increasing the mean velocity and temperature gradient. It can help to produce the higher heat fluxes and momentum transfer due to the large effective driving potential force but also 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.

In the last few years, several researchers conducted studies with conical rings. The effects of the conical ring turbulators with different arrangements such as converging conical ring (CR array), diverging conical ring (DR array) and converging-diverging conical ring (CDR array) on the heat transfer, friction factor and overall heat transfer enhancement were studied by Promvonge [17]. The results showed that the rings with DR array provided better thermal performance factor than the rings with the CR and CDR arrays. Another study with conical ring turbulators was performed by Durmus [18]. Four conical angles (5°, 10°, 15° and 20°) were considered in the study and it was reported that both heat transfer rates and friction coefficients increased with increasing turbulator angles. Promvonge and Eiamsa-ard [19] experimentally studied the effects of the conical turbulator combined with the snail entrance on heat transfer rate and flow friction in a heat exchanger. Eiamsa-ard and Promvonge [20] investigated the enhancement efficiency in a tube by usingV-nozzle turbulators at different pitch ratios.

Ayhan et al. [21] studied numerically and experimentally the heat transfer in a tube by means of truncated hollow cone inserts. Yakut et al. [22] experimentally investigated the effect of conical-ring turbulators on the turbulent heat transfer, pressure drop and flow induced vibrations. Their experimental data were analyzed and presented in terms of the thermal performances of the heat transfer promoters with respect to the heat-transfer enhancement efficiencies for a constant pumping power. Yakut and Sahin [23] reported the flow-induced vibration characteristics of conical-ring turbulators used for heat transfer enhancement in heat exchangers. They pointed out that the Nusselt number increased with the rising Reynolds number and the maximum heat transfer was obtained for the smallest pitch arrangement. The combined effects of the conical rings together with twisted-tape to enhance the heat transfer in a circular tube were introduced by Promvonge and Eiamsa-ard [24]. Their experimental results revealed that heat transfer rate in the tube with the conical rings together with twisted-tape was determined to be approximately 10% over that in the tube with the conical-ring alone.

In this numerical study, the conical-nozzle turbulators assumed as a reverse/turbulence flow device, are, therefore, employed with two different turbulator arrangements (diverging and converging). The numerical analyses were performed with three different pitch ratios

(PR=2.5, 5 and 7.5) for Reynolds number ranging from 6000 to 24000. Eventually, the results of Nusselt number, friction factor and thermal enhancement factor were presented for all investigated cases.

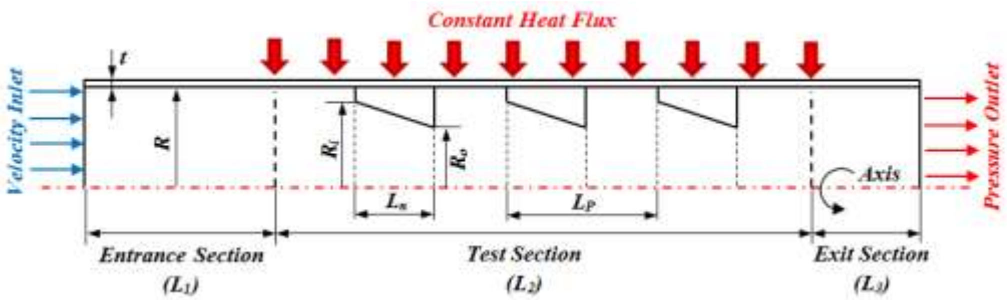
2. RESEARCH SIGNIFICANCE

At last decade, energy costs dramatically rise day by day, since human population increase, and energy sources are consumed. Using the energy efficiently is significantly important for countries that import energy, especially. Heating or cooling systems need so much energy input such as pumping power and electrical resistant for heating and compressor power for cooling. Within this scope, heat transfer enhanced methods are used and investigated methods. Before conducting experimental studies, observing of numerical studies carries importance. In this scope, this study, which is "Numerical Analysis of Heat Transfer Enhancement in Conical-nozzle Turbulators Inserted Tube", contributes to literature about heat transfer enhancement applications. It is observed that placing nozzles with different placements into a horizontal tube affects the heat transfer and hydraulic performance for different working conditions. The results of the study can shed light to researchers who work in energy efficiency area.

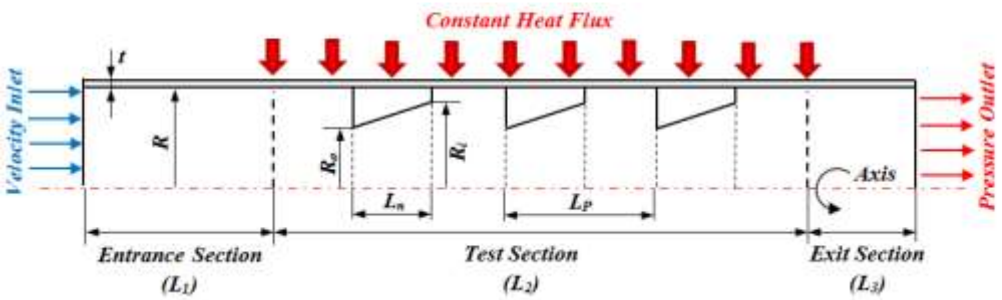
3. NUMERICAL METHODOLOGY

3.1. Solution Domain

In this study, thermal and hydraulic performance of C and D type nozzle inserted tube is numerically investigated by using a CFD program. Solution domain is created as two dimensional with using axis boundary definition. Thus, the governing equations are solved by CFD program as cylindrical coordinates. The solution domain for both C and D type nozzle inserted tube is schematically illustrated with boundary condition definitions in Figure 1.



(a) C type nozzle inserted tube



(b) D type nozzle inserted tube

Figure 1. Solution domains for C (a) and D (b) type nozzle inserted tube

Working fluid is air, and the tube and the nozzle material are copper and aluminum, respectively. In order to gets fully developed steady flow, the tube having a length of 1.5mm thickness (t) is arranged with entrance section (L_1) having length of 10D. Besides, to prevent the revers flow error in CFD analyzes, exit section (L_3) is placed as length of 5D. The nozzle direction to the flow is defined as C and D type. The investigated section is called as test section (L_2) having length of 1250mm. If the flow enters into larger diameter of the nozzle, it is called as C type. In contrast, if the flow enters into smaller diameter of the nozzle, it is called as D type like diffuser. The length of large and small diameter of the nozzle is 46 and 26mm, respectively. In addition to nozzle direction, effect of pitch length (L_P) of the inserted nozzles is investigated to observe their thermal and hydraulic performance. The investigated pitch lengths are selected as 2.5, 5.0 and 7.5 times of nozzle length (L_n).

3.2. Governing Equations and

The CFD programs solve differential equations with finite element model to simulate flow characteristic and to calculate heat transfer. These differential equations are as below:

Continuity equation:

$$\nabla.(\rho u) = 0 \qquad (1)$$

Momentum equation:

$$\frac{\partial}{\partial t}(\rho\vec{v}) + \nabla(\rho\vec{v}\vec{v}) = -\nabla P + \nabla(\bar{\bar{\tau}}) + \rho\vec{g} + \vec{F} \qquad (2)$$

Energy equation:

$$\nabla\big(\vec{v}(\rho E + p)\big) = \nabla\big(k_{eff}\nabla T - \textstyle\sum_j h_j\vec{J}_i + (\bar{\bar{\tau}}_{eff}.\vec{v})\big) + S_h \qquad (3)$$

k-ε standard, enhanced wall treatment, Turbulent model equation:

For turbulent kinetic energy “k”:

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + P_k + P_b - \rho\varepsilon - Y_M + S_k \qquad (4)$$

For dissipation **ε**:

$$\frac{\partial}{\partial x_i}(\rho\varepsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}(P_k + C_{3\varepsilon}P_b) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} + S_e \qquad (5)$$

3.3. Boundary Conditions

The turbulent flow is Reynolds number ranging from 6000 to 24000. In order to simulate the turbulent flow, k-ε standard, enhanced wall treatment model is used for all cases. Velocity inlet magnitude is calculated to provide Reynolds numbers. Gauge pressure outlet is selected 0 Pa, to get atmospheric pressure condition at outlet. Constant heat flux of 1000W/m^2 is applied onto the outer surface of the tube. Properties of the materials are assumed as constant at room temperature.

3.4. Validation of the Numerical Methodology

Validation of the numerical analyzes are dramatically necessary to ensure accurate of the results. For this purpose, experimental results of study by Promvonge and Eimsa-ard [25] are used to validate this present study in terms of both Nusselt number (6) and friction factor (7) for pitch length as 4.0 times of length nozzle.

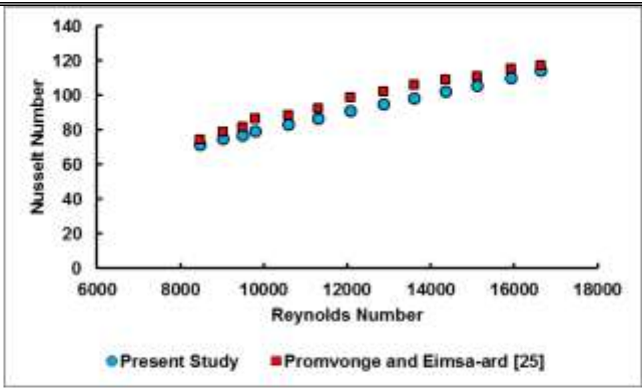


Figure 2. Comparison of the present study with Promvonge and Eimsa-ard [25] results in terms of the Nusselt number versus Reynolds number

As can be seen in Figure 2 and 3, a good agreement is obtained with experimental results [25] for the present study. In order to compare the results of present study with smooth tube, correlation in literature, which are Gnielinski Equation [26] (8) and Blasius Equation [27] (9) are used in terms of the Nusselt number and the friction factor, respectively.

$$\mathrm{Nu} = \frac{\mathrm{hD}}{\mathrm{k}} \tag{6}$$

$$\mathrm{f} = \frac{\Delta \mathrm{P}}{\frac{1}{2}\rho \frac{\mathrm{D}}{\mathrm{L}} \mathrm{V}^2} \tag{7}$$

$$\mathrm{Nu} = \frac{\left(\frac{\mathrm{f}}{8}\right)(\mathrm{Re}-1000)\mathrm{Pr}}{1+\left[12.7\left(\frac{\mathrm{f}}{8}\right)^{0.5}\left(\mathrm{Pr}^{2/3}-1\right)\right]} \tag{6}$$

$$\mathrm{f} = 0.316\mathrm{Re}^{-0.25} \tag{7}$$

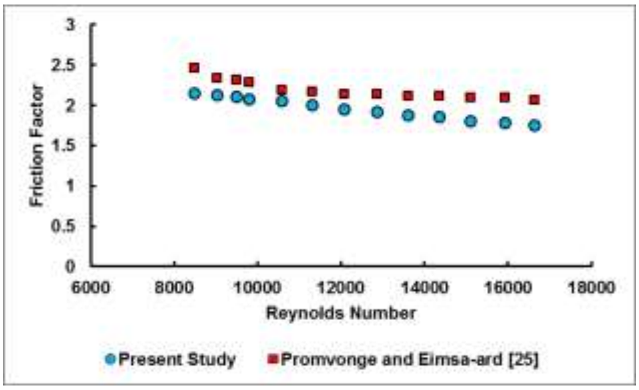


Figure 3. Comparison of the present study with Promvonge and Eimsa-ard [25] results in terms of the Nusselt number versus Reynolds number

4. RESULTS AND DISCUSSIONS

In this study, the effect of C and D type inserted nozzle with various pitch lengths (pitch ratio (PR) x nozzle length (L_n)) into a horizontal straight tube under constant heat flux and at turbulent flow numerically is investigated by using a CFD program. The inserted nozzles have PR of 2.5, 5.0 and 7.5 and the turbulent flow is ranging Reynolds number of 6000 to 2400. The results are separately evaluated as heat transfer and friction factor.

4.1. Heat Transfer

It is commonly known that heat transfer increases with increasing turbulent, in other words Reynolds number. The results of the all cases show like this behavior. Decreasing pitch Ratio (PR) means increasing using nozzle, and it causes to enhance heat transfer. The more using both C and D type nozzle, the more turbulent increases especially near the inner surface of the wall. Distribution of the present study results in terms of the Nusselt number versus Reynolds number is plotted in Figure 4.

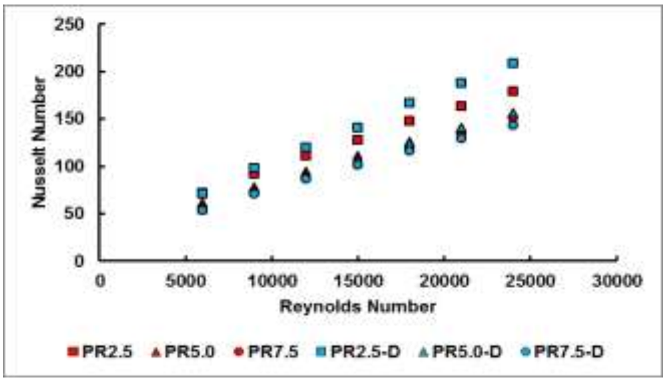


Figure 4. Distribution of the presents study in terms of Nusselt number versus Reynolds number

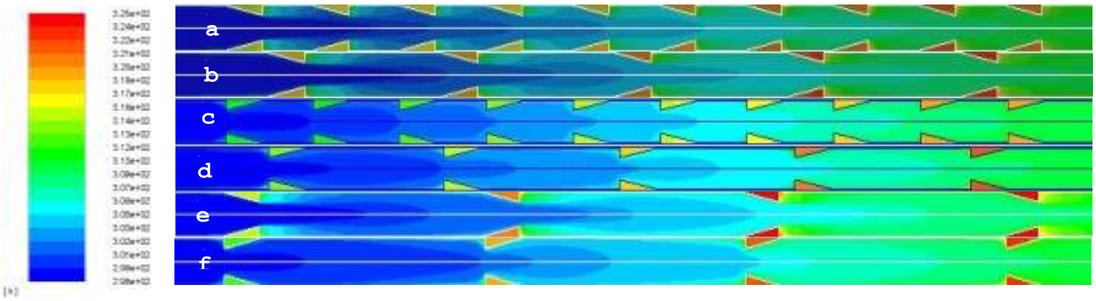


Figure 5. Temperature contours for all cases {a) PR2.5, b)PR2.5-D, c)PR5.0, d)PR5.0-D, e)PR7.5, f)PR7.5-D} at constant Reynolds number of 24.000

Maximum Nusselt number is obtained as 208.59 for 2.5 pitch ratio D type nozzle and at Reynolds number of 24,000. In other words, it means that the value of this configuration is approximately 3.35 times higher than smooth tube at same Reynolds number. As can be seen in Figure 5, the fluid and the nozzle temperatures are the highest and the lowest, for the b model that is PR2.5- D, respectively. This contour demonstrates heat transfer more efficiently occurs from the wall to the fluid thanks to inserted D type nozzles with pitch ratio of 2.5.

4.2. Friction Factor

Friction factor parameter is used to compare hydraulic systems which are used inserted elements, like turbulator.

The friction factor value of the designed system gives an idea for needed pumping power input. Thus, the friction factor results of C and D type nozzle inserted tube having various pitch ratio values analyses are given in a graph in Figure 6. Inserted elements in a tube occurs an obstacle in direction of the flow, so pressure drop decreases and more pumping power input is needed to pump the fluid to anywhere. The highest friction factor value is seen as approximately 4.01 for D type nozzle inserted and pitch ratio of 2.5. This result is found reasonable, because D type nozzles have larger area in direction of the fluid than C type nozzles. It causes more pressure occurs in front of the nozzle, as can be seen in Figure 7. Moreover, with the increment of the inserted nozzles in the tube, this pressure increasing decreases the pressure outlet step by step. In addition to total pressure contours, velocity contours (Figure 8) can explain the flow characteristic and hydraulic performance. As can be seen in this figure, when the fluid entering through the nozzles, D type nozzles show more velocity magnitude than C type nozzle, because the flow suddenly encounters with a larger surface area of the D type nozzle. However, C type nozzle placement helps to smooth over the fluid flow, when the fluid enters through the nozzle.

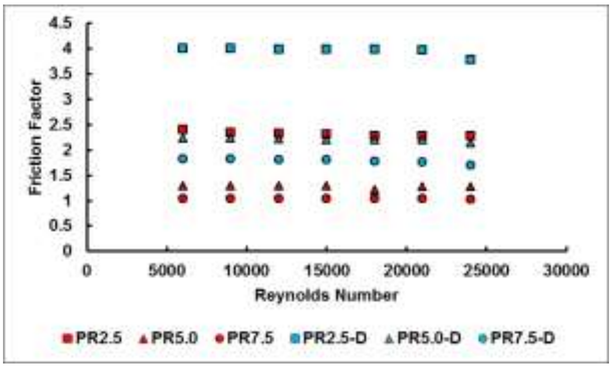


Figure 6. Distribution of the presents study in terms of the friction factor versus Reynolds number

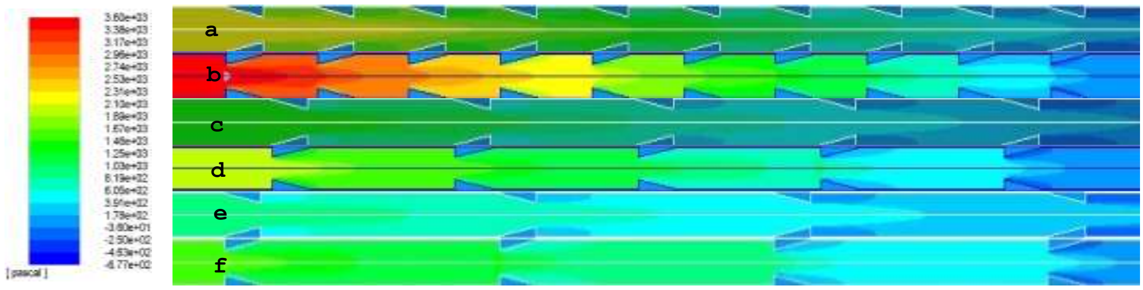


Figure 7. Total Pressure contours for all cases {a) PR2.5, b)PR2.5-D, c)PR5.0, d)PR5.0-D, e)PR7.5, f)PR7.5-D} at constant Reynolds number of 24.000

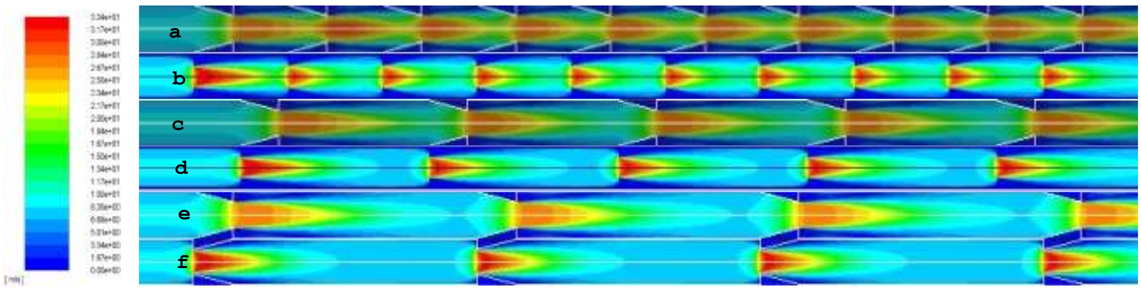


Figure 8. Velocity contours for all cases {a) PR2.5, b)PR2.5-D, c)PR5.0, d)PR5.0-D, e)PR7.5, f)PR7.5-D} at constant Reynolds number of 24.000

4.3. Thermo-Hydraulic Performance

The heating system designs should be determined in terms of both thermal and hydraulic performance together. Thermo-hydraulic performance (THP) criteria shed light to determine these systems. The thermo-hydraulic performance criteria formula is given below:

$$\mathrm{THP} = \frac{\left(\mathrm{Nu_i}/\mathrm{Nu_s}\right)}{\left(\mathrm{f_i}/\mathrm{f_s}\right)^{1/3}} \tag{8}$$

The THP results of the present study for all cases are plotted in a graph (Figure 9) depend on Reynolds number. As can be seen in this figure, with the increment of the Reynolds number, THP magnitudes decrease. The reason of this result, friction factor results are dominated to the Nusselt number, and the friction factor results are much more than smooth tube results. When the results are examined in terms of nozzle and pitch ratio configurations, the highest THP magnitude is obtained as approximately 0.94 for pitch ratio of 5.0 at Reynolds number of 5000.

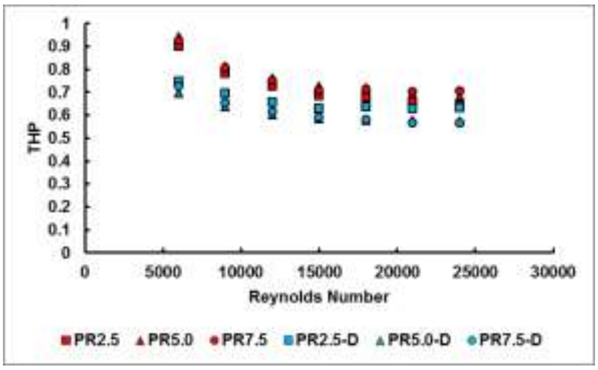


Figure 9. Distribution of the presents study in terms of the thermo-hydraulic performance versus Reynolds number

5. CONCLUSIONS

In this study, the effect of C and D type inserted nozzle with various pitch lengths in a straight horizontal tube under constant heat flux and turbulent flow condition is numerically investigated. The highest Nusselt number, the lowest friction factor and highest THP magnitudes are obtained as 208.59, 1.04 and 0.94 for D type nozzle, pitch ratio of 2.5 and Reynolds number of 24000, C type nozzle, pitch ratio of 7.5 and Reynolds number of 6000, C type nozzle, pitch ratio of 5.0 and Reynolds number of 6000, respectively. The conclusions can be listed as below:

- Inserting both C and D type nozzle in the tube enhances heat transfer.
- Decreasing the pitch ratio means increasing number of inserting nozzle, and it causes to increase heat transfer.
- D type nozzle is more effective to enhance heat transfer than C type nozzle.
- Inserting both C and D type nozzle in the tube increase the pressure drop and friction factor.
- Increasing the pitch ratio decrease the pressure drop and friction factor.
- C type nozzle is more effective to get lower friction factor than D type nozzle.

NOMENCLATURE

- D: diameter of the tube [m]
- R: radius of the tube [m]
- F: friction factor [-]
- H: heat transfer coefficient [W/m^{2}K]
- K: thermal conductivity [W/mK]

- L: length [m]
- Nu: Nusselt number [-]
- ΔP: Pressure drop [Pa]
- Pr: Prandtl number [-]
- q : heat flux [W/m^2]
- Re: Reynolds number [-]
- T : temperature [K]
- THP: thermo-hydraulic performance [-]
- V: velocity [m/s]

Subscripts

- ı: inserted tube
- n: nozzle
- p: pitch
- 1: entrance section
- 2: test section
- 3: exit section
- S: smooth tube
- Greek symbols
- ρ: density [kg/m^3]
- µ: dynamic viscosity [kg/ms]

NOTICE

This study was presented as an oral presentation at the I. International Scientific and Vocational Studies Congress (BILMES 2017) in Nevşehir/Ürgüp between 5-8 October 2017.

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