



## Numerical Investigation on Heat Enhancement Method with Using Circular Dimpled Tube

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### ABSTRACT

In this study, the effect of dimpled tube which is heated with constant heat flux on heat transfer enhancement is numerically investigated. Using physically enhanced tube is one of the passive heat transfer enhancement methods. The reason of using dimpled tube is increase turbulence through the tube and destruct the thermal boundary layer. The numerical study is validated with an experimental study and configurations of cases are expanded with changing pitch length. The considered cases are conducted in Reynolds number range of 3000 to 8000. Used fluid through the flow and its material copper are selected as water and copper, respectively. The k- $\epsilon$  RNG turbulence model is used to simulate turbulent flow adjacent the inner wall surface. The analyses are determined with Nusselt number (Nu), friction factor (f) and performance evaluation criteria (PEC). The highest Nusselt number and the minimum friction factor is obtained when the Reynolds number of 8000, as 135.54 and 0.0422, respectively, for pitch length of 10 mm and 80 mm.

**Keywords:** Dimpled Tube, Numerical Study, Heat Transfer, Pressure Drop.

## **Dairesel Oyuntu ile Isı Transferi İyileştirilmesi Üzerine Sayısal Araştırma**

### **ÖZET**

Bu çalışmada sabit ısı akısı ile ısıtılmış bir boruya oyuntu yerleştirilmesinin ısı transferi iyileştirilmesi üzerindeki etkisi sayısal olarak araştırılmıştır. Fiziksel olarak boru üzerinde yapılan iyileştirme ısı transferi iyileştirilmesinin pasif metotlarından birisidir. Oyuntulaştırılmış boru kullanımının sebebi akış boyunca türbülansı arttırmak ve termal sınır tabakayı parçalamaktır. Sayısal çalışma, deneysel başka bir çalışma ile doğrulanmış ve incelenen değişen hatve uzunlukları genişletilmiştir. İncelenen analizler Reynolds sayısının 3000'den 8000' e kadar gerçekleştirilmiştir. Akış boyunca kullanılan akışkan ve boru malzemesi sırasıyla su ve bakır olarak seçilmiştir. İç duvar etrafındaki türbülanslı akışın simülasyonu için k-ε RNG türbülans modeli kullanılmıştır. Analizlerin sonuçları Nusselt sayısı (Nu), sürtünme katsayısı (f) ve performans değerlendirme kriteri (PEC) ile değerlendirilmiştir. En yüksek Nu ve en düşük f Re sayısı 8000 iken, sırasıyla 130.54 ve 0.0453 olarak, hatve uzunluğunun sırasıyla 10 mm ve 50 mm olduğu durumlarda elde edilmiştir.

**Anahtar kelimeler:** Oyuntulaştırılmış Boru, Sayısal Çalışma, Isı transferi, Performans Değerlendirme Kriteri.

### **1. INTRODUCTION**

Thermal systems carry crucial importance in many engineering application due to provide stability and occur a sustainable cycle. Many investigation are focused on heat transfer enhancement of the thermal systems. These investigation are not only enhance the heat transfer performance but also decrease the energy consumption. In short, various heat tranwsfer enhancement techniques are conducted to

- accomplish a high heat transfer performance with minimum pumping power
- optimize the cost of energy, material and time
- increase efficiency of process and system
- design optimal heat exchanger dimensions
- reduce the weight and volume of the heat exchangers

The heat transfer enhancement methods are considered under two heading: active and passive methods. Active methods are required external power input, however; passive methods contain surface or geometrical modification.

**Table 1.** Kinds of heat transfer enhancement methods

Active methods	Passive methods
Magnetic field	Flow distribution
Jets	Surface roughness
Spray	Secondary flow
Mechanic aids	Chanel curvature
Surface vibration	Fluid additives
Fluid vibration	Re-entrant obstructions
	Out of plane mixing

Numerous studies are available in literature on heat transfer enhancement by inserting various turbulators such as ribs (Eimsa-ard and Promvonge., 2010: 49), (Lu and Jiang., 2006:30) , fins (Yakut et al., 2006: 83), (Sapali and Patil., 2010: 34), (Dong, et al., 2010: 30), baffles (Shaikh and Siddiqui., 2007: 28), (Tandiroglu and Ayhan., 2006: 26), (Promvonge, 2010: 37), rings (Ozceyhan et al., 2008: 85), (Akansu., 2006: 83), twisted tapes (Chang et al., 2007: 46), (Eimsa-ard., 2010: 30), (Eimsa-ard, 2010: 34) and coiled wire inserts (Guens et al., 2010: 34), (Gunes et al., 2010: 30), (Promvonge., 2008: 49).

The convective heat transfer can be enhanced with the occurrence of re-circulation / reverse flow, occurrence secondary flow in cross-sectional area of the flow, by disrupting the thermal boundary layer and increasing mean velocity and temperature gradient. These effects along a tube are occurred by using a turbulator.

The strength of especially reverse flow and disrupting the thermal boundary layer are the main interest in many heat transfer applications.

Several researchers conducted studies with inserting conical ring turbulators inside to the tube. Promvonge (Promvonge., 2008: 49) investigated the heat transfer enhancement and flow characteristics by using conical ring turbulators with different arrangements in a tube. The results showed that the diverging conical ring provided better heat transfer performance than the other ring combinations. Another investigations with conical ring turbulators was conducted by Durmus (Durmus., 2004: 45). Various conical angles from 5° to 20° were investigated in the study and it was concluded that with the increment of turbulator angles heat transfer performance and friction coefficients increased. Promvonge and Eiamsa-

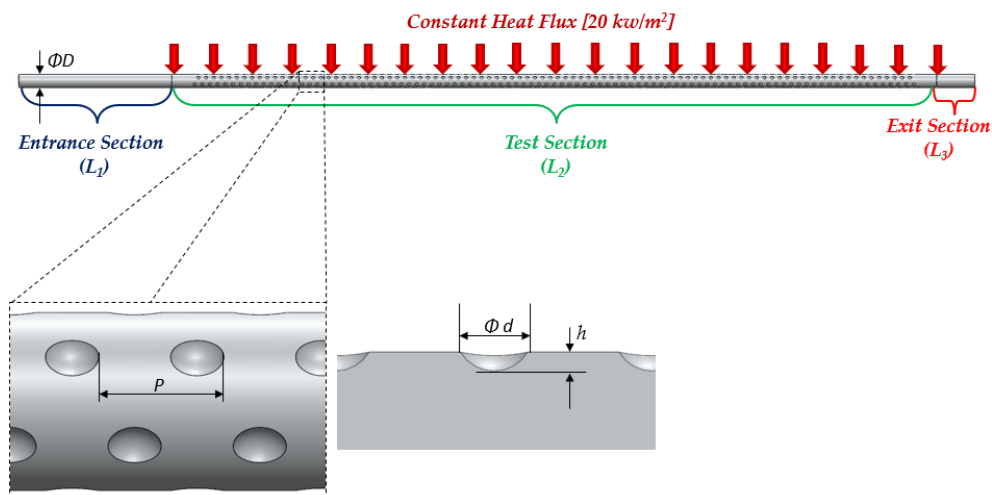
ard (Promvonge and Eiamsa-ard., 2006: 47) experimentally investigated especially the effect of snail entrance by using concail turbulator performnasc according to heat transfer and pressure drop.

Eiamsa-ard and Promvonge (Eiamsa-ard and Promvonge., 2006: 33) investigated the effect of using V-nozzle turbulators at different pitch ratios on enhancement efficiency in a tube. Ayhan et al. (Ayhan et al., 1999) studied numerically and experimentally the heat transfer performance in a tube by using truncated hollow cone inserts. Yakut et al. (Yakut et al., 2004: 79) experimentally investigated the heat transfer performance, pressure drop and flow induced vibrations by using conical-ring turbulators along the tube. Li et al (Li et al., 2016: 101) investigated dimpled effect on heat transfer performance and flow characteristic on a horizontal tube experimentally and numerically. Their investigation contains 5mm dimple d diameter, 1.2 mm dimpled height and pitch length of 10 mm. The working fluid is in this study water, glycol and water/glycol mixture. They concluded that it obtained that the highest PEC in the Reynolds number range 150-2000 with glycol/water fluid and the best performance ( $PEC=1.55$ ) was obtained with water flow at Reynolds number 3500-4500.

## 2. NUMERICAL METHODOLOGY

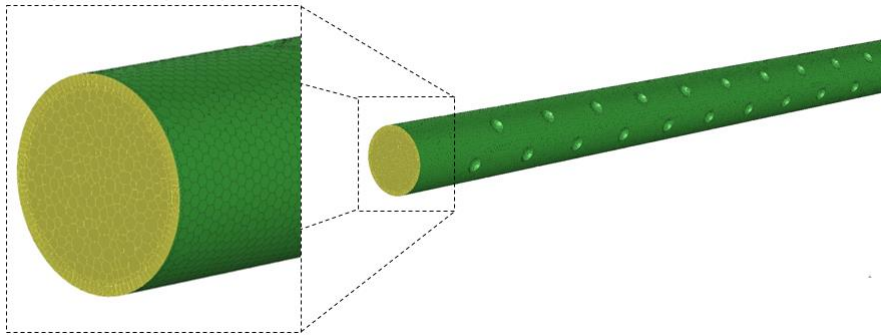
### 2.1. Solution Domain

In this study, thermal and hydraulic performance of dimpled tube is numerically investigated by using a CFD program. Solution domain is created as three dimensional. Solution domain for dimpled tube is schematically illustrated with boundary condition definitions in Fig. 1 and Table 2.



**Figure 1.** Physical details and boundary types of solution domain

Water is used for working fluid with assumption of in compressed, steady and continuous. Thermal and physical property of the water is considered as independent of temperature. In order to get fully developed steady flow, the tube is arranged with entrance section ( $L_1$ ) having length of more than 10D. Besides, to prevent the revers flow error in CFD analyzes, exit section ( $L_3$ ) is placed as length of 50 mm. The investigated section is called as test section ( $L_2$ ) having length of 1000 mm.



**Figure 2.** Mesh structure of considered dimpled tube

**Table 2.** Definition and values of the considered model

Symbol	Definition	Value [mm]
D	Tube Diameter	17.272
d	Dimple diameter	5
h	Dimple height	1.2
P	Dimple Pitch	from 10 to 50
$L_1$	Entrance Section	200
$L_2$	Test Section	1000
$L_3$	Exit Section	50

## 2.2. Governing Equations

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:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \quad (1)$$

Momentum equation:

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

Energy equation:

$$\frac{\partial}{\partial t} (\rho E) + \nabla (\vec{v} (\rho E + p)) = \nabla (k_{eff} \nabla T - \sum_j h_j \vec{j}_i + (\bar{\tau}_{eff} \cdot \vec{v})) + S_h \quad (3)$$

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

For turbulent kinetic energy “k”:

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

for dissipation ε:

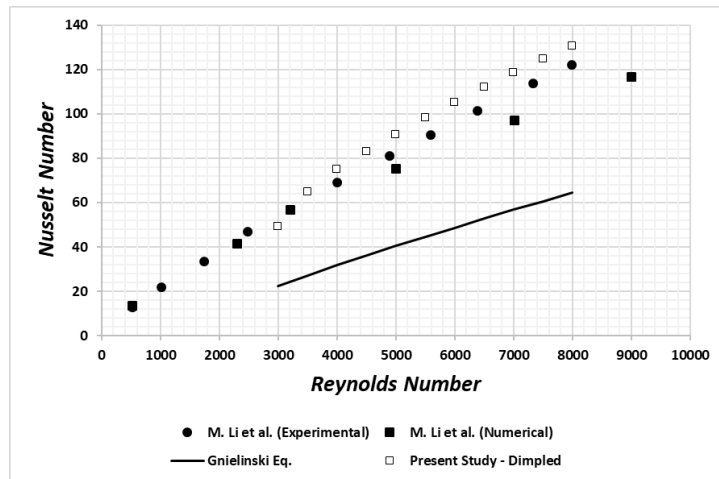
$$\frac{\partial}{\partial x_i} (\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} (P_k + C_{3\varepsilon} P_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k}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

### 2.3. Boundary Conditions

The turbulent flow is Reynolds number ranging from 3000 to 8000. To simulate the turbulent flow, k-ε RNG, enhanced wall treatment model is used for all cases. Velocity inlet magnitude is calculated by derived from Reynolds number. Gauge pressure outlet is selected 0 Pa, to get atmospheric pressure condition at outlet. Constant heat flux of 20 kW/m<sup>2</sup> is applied onto the outer surface of the tube. Properties of the materials are assumed as constant at room temperature.

### 2.4. 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 Li et al. (Li et al., 2016: 101) 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 3.** Comparison of the present study with Li et al (Li et al., 2016: 101) results in terms of the Nusselt number versus Reynolds number

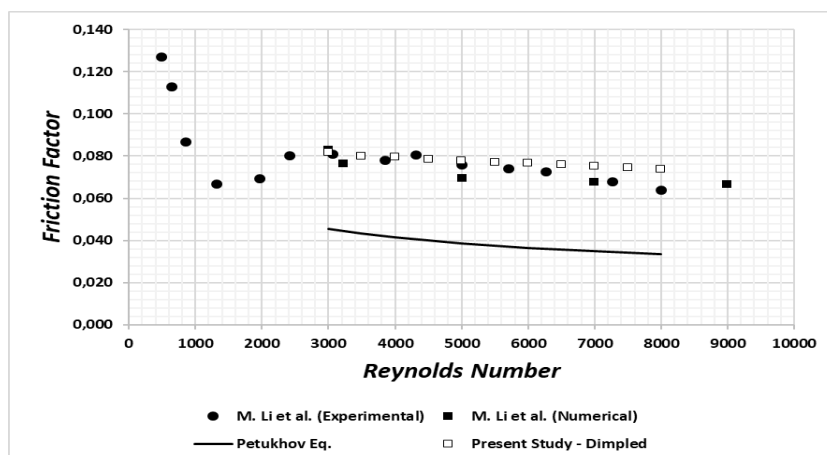
As can be seen in Fig. 3 and 4, a good agreement is obtained with experimental and numerical results (Li et al., 2016: 101) for the present study. To compare the results of dimpled tube, correlation in literature, which are Gnielinski Eq. (Incropera et al., 2006) (8) and Blasius Eq. (Petukhov, 1970) (9) are used in terms of the Nusselt number and the friction factor, respectively.

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

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

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

$$f = 0.316Re^{-0.25} \tag{9}$$



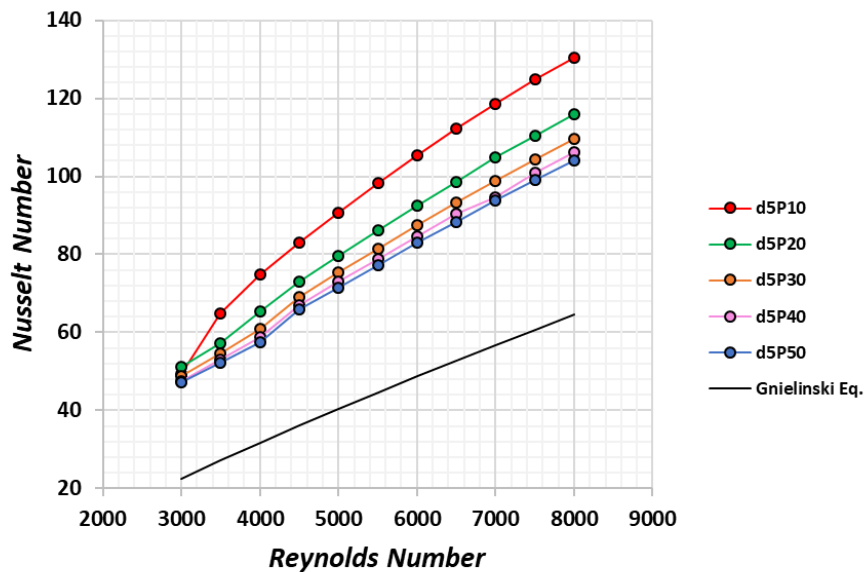
**Figure 4.** Comparison of the present study with Li et al (Li et al., 2016: 101) results in terms of the Nusselt number versus Reynolds number

### 3. RESULTS AND DISCUSSIONS

In this study, the effect of various pitch lengths of the dimpled tube under constant heat flux and at turbulent flow numerically is investigated by using a CFD program. The dimples are placed with length of 10 - 50 mm onto the tube and analyses are conducted under turbulent condition which the flow is Reynolds number of 3000 to 8000. The results are separately evaluated as heat transfer and friction factor.

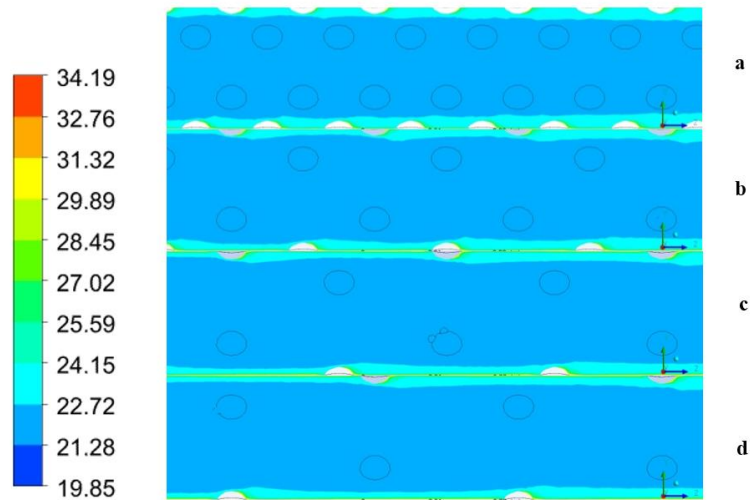
#### 3.1. Heat Transfer

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 length of the dimpled means increasing using dimples, and it causes to enhance heat transfer. The more using dimples, 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 Fig. 5.



**Figure 5.** Distribution of the presents study in terms of Nusselt number versus Reynolds number



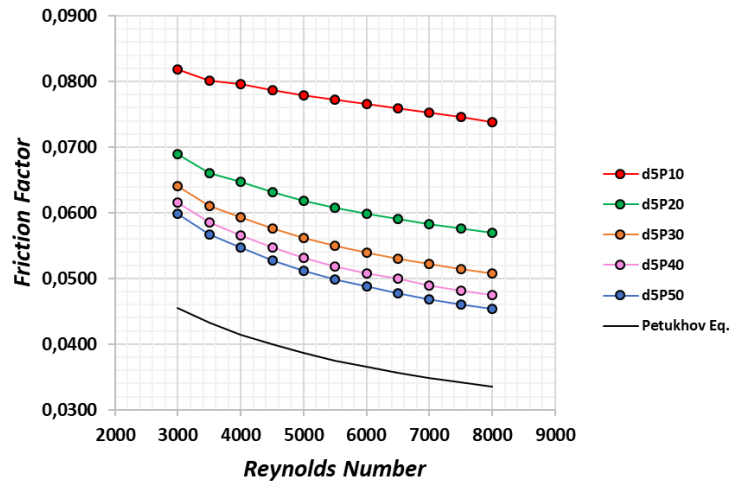


**Figure 6.** Temperature [ $^{\circ}\text{C}$ ] contours for the case of (a) P10, (b) P20, (c) P30, (d) P40 constant Reynolds number of 5000

Maximum Nusselt number is obtained as 130.54 for pitch length of 10 mm at Reynolds number of 8000. In other words, it means that the value of this configuration is approximately 2.02 times higher than smooth tube at same Reynolds number. As can be seen in Fig. 6, the dimples are destroy the thermal boundary layer at the heated inner surface. At this result, the given heat energy more effective reach to the fluid. The closer dimples on to the tube, the more destroy the thermal boundary layer.

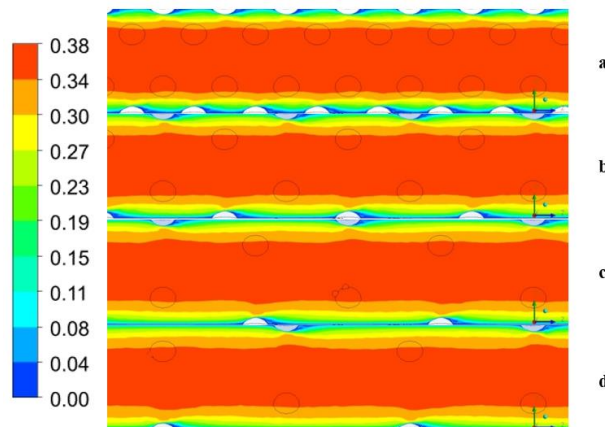
### 3.2. Friction factor

Friction factor parameter is used to compare hydraulic systems which are used inserted elements, like turbulators. The friction factor value of the designed system gives an idea for needed pumping power input. Thus, the friction factor results of considered dimpled tubes are given in a graph in Fig. 7. Placed dimples onto the 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.



**Figure 7.** Distribution of the presents study in terms of the friction factor versus Reynolds number

The highest friction factor value is seen as approximately 0.0819 for pitch length of 10 mm case at Reynolds number of 3000. This result can be found reasonable, because the closer dimples destroy the flow behind the other dimpled, as can be seen in Fig. 8.

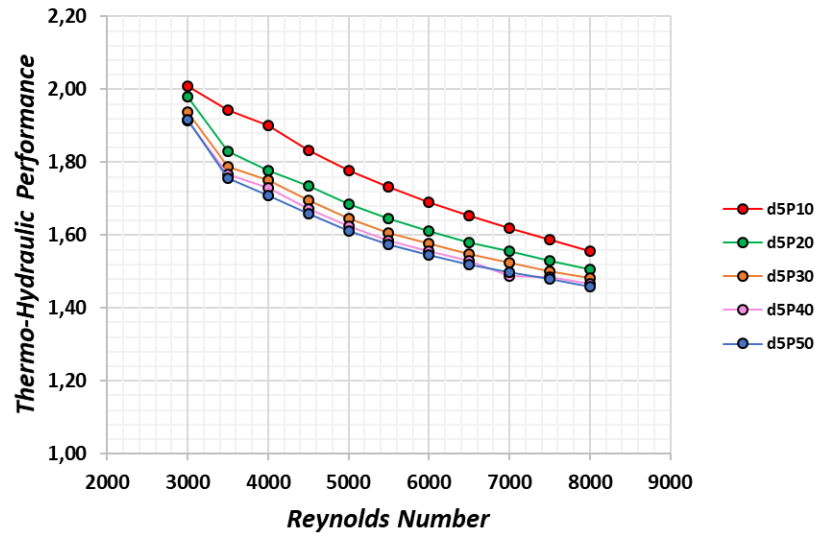


**Figure 8.** Velocity [m/s] contours for the case of a) P10, (b) P20, (c) P30, (d) P40 constant Reynolds number of 5000.

### 3.3. Thermo-Hydraulic Performance

The heating system designs should be determined in terms of both thermal and hydraulic performance together. Thermo-hydraulic performance (THP) (10) criteria shed light to determine these systems. The thermo-hydraulic performance criteria formula is given below:

$$THP = \frac{(Nu_d/Nu_s)}{(f_d/f_s)^{1/3}} \quad (10)$$



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

The THP results of the present study for all cases are plotted in a graph (Fig. 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 2.01 for pitch ratio of 5.0 at Reynolds number of 3000.

#### 4. CONCLUSIONS

In this study, the effect dimples with various pitch length on the horizontal tube under constant heat flux and turbulent flow condition is numerically investigated. The highest Nusselt number, the lowest friction factor and the highest THP magnitudes are obtained as 130.54, 0.0453 and 2.01 for pitch length of 10 mm at Reynolds number of 8000, for pitch ratio of 50 mm at Reynolds number of 8000 and for pitch length of 10 mm at Reynolds number 3000; respectively.

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