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Numerical Investigations and Optimization of Thermo-hydraulic Performance in Channels with Variable Number and arrangement of Dimples

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Abstract

Heat transfer enhancement in double heat exchangers contains to be a significant challenge in thermal engineering. This paper investigates how varying the number of dimples (4, 6, and 8) in cross-sectional dimpled channels influences forced convection phenomena and the friction factor. A numerical study was conducted featuring a three-dimensional analysis of the friction factor, enhanced heat transfer, and thermal performance criteria (PEC) within a dimpled channel with water flow. To simulate the flow within a circular channel, a commercial software application, ANSYS FLUENT, was utilized. The simulation employed governing equations, include continuity, momentum, and energy equations, as well as the RNG k-ε turbulence model. This model was used to assess the impact of varying the number of dimples on turbulent flow and heat transfer enhancement. The research focused on Reynolds number (Re) range of 2500 to 12000, especially targeting turbulent flow. The findings indicate that the existence of dimples on the channel wall significantly influences both heat transfer and friction factor when compared to a smooth channel. Numerical analysis notified that the Nu for the three cases of dimples in cross-sectional area (4,6, and 8) was 30.2% ,41.3% and 49.06% larger respectively, than that of the conventical channel. Additionally, the channel with 8-dimples exhibited the highest (*f*) in comparison to the other configurations. The design featuring 4-dimples in cross sectional area achieved the highest value of the performance evaluation criteria (PEC) across all Re values. These findings offer significant scientific insights for the optimizing thermo-hydraulic performance through varying dimple numbers, providing practical design guidelines for advanced and energy-efficient heat exchange systems.

Keywords: *Heat transfer enhancement, dimpled channel, turbulent flow, pressure drop, thermal performance criteria.*

1. Introduction

In varies locations around the globe, improving heat transfer through by modifying channel walls is a commonly utilizing technique. To enhance performance of internal surfaces in channels and dimpled channels, features such as corrugations, perforations, grooves, fins, and flutes are added both internally and externally [1]. Dimples enhance heat transfer effectiveness functioning similarly to eddy current generators by promoting flow regions adjacent to surfaces and reducing pressure drop. Enhancing the heat transfer can lead to significant energy savings and system optimization [2]. Dimples are highly effective in enhancing heat transfer in industrial applications, as they increase the heat transfer coefficient and the available surface area for exchange relative to flow-to-volume ratio [3],[4]. Numerous research studies have investigated the effects of varies dimple shapes, including circular, elliptical, and triangular forms [5]. Recent numerical and experimental investigations have indicated that the geometry, depth, and arrangement of dimples significantly influence the thermo-hydraulic performance of channels [6]. Increasing the depth of the dimples and optimizing the spacing between them can enhance heat transfer; however, this may also lead to a higher-pressure drop [7],[8]. Furthermore, advanced designs, such as hybrid rib-dimple structures, have demonstrated greater enhancements in heat transfer compared to traditional smooth or single-structure channels [9], [10].

Investigations have showed that the enhancement in heat transfer rate for dimple channels exceeds that of smooth channels. Dimples serve as passive interruptions that increase turbulence strength within the fluid near the wall, thereby enhancing the convective heat transfer coefficient without requiring an external energy source [11],[12]. This characteristic makes dimples equipped devices an attractive and suitable option for modern heating systems [13]. The hydrothermal efficiency of a dimpled tube was numerically investigated using CFD to address with suboptimal heat transfer by utilizing CFD in turbulent flow conditions, concentrating on single-phase flow. The results indicated that an increase in the Re led to a high average Nusselt number (Nu) , increased pressure drops, and improved performance criteria [14]. Many researchers have examined how dimples influence flow behavior and velocity distribution in a double heat exchanger [15]. They studied Reynolds numbers ranging from 500 and 8000 and conducted experiments using a water/glycol solution. The results showed that dimples disturb boundary layers and elevate turbulence, resulting in a heat transfer enhancement exceeding 200% compared to smooth channels [16].

Various efforts were examined to enhance heat transfer by increasing the surface area of channels in heat exchanger. The studies show that improvements in heat transfer and friction within a channel wall, which utilizes a shaped sheet metal surface and features Reynolds numbers ranging from 6,000 to 35,000 demonstrate significant enhancements compared to conventional designs [17]. Additionally, an analysis is conducted on the heat transfer and pressure drop of R-134a during the condensation phase in a dimpled channel. While the input is generally clear, it could benefit from minor adjustments to enhance clarity and flow [18].

Two concentric horizontal channels were utilized as a test section, with water circulating in the annulus and R-134a flowing through the inner channel. Both dimpled and smooth channels were employed for the test runs in the inner channel. The experimental results indicated that dimpled channels exhibited a higher heat transfer coefficient and increased pressure drop compared to smooth channels, with an enhancement of the Nu by approximately 1.3 to 1.4 relative to the Re [19]. A numerical analysis was performed to examine the enhancement of heat transfer and fluid flow behavior in channels with surface protrusions and dimples [20]. Turbulence models enable engineers and researchers to analyze complex fluid flows with reasonable computational cost while maintaining acceptable accuracy [21].

The K-$\boldsymbol{\varepsilon}$ turbulence model is one of the most widely used due to its simplicity and robustness, as it solves two transport equations for single-phase flow [22]. To enhance heat transfer, the procedures were illustrated using local streamlines, temperature contours, Nusselt number (Nu), friction factor, and velocity contours. Increasing the depth of the dimple enhances heat transfer and the friction factor; however, the thermal performance criteria diminish. Additionally, it was observed that when the pitch between two dimples increases, the friction factor initially decreases before increasing again in a dimpled channel [23]. A numerical study was conducted using a semi-dimple pair fixed to the fins and wall of a channel heat exchange system. The characteristics examined include the angle of attack, dimple diameter, and semi-dimple position. The Re varied between 813 and 4019, and the performance criteria of the dimpled fin were compared to other types of dimples. The results indicate that the semi-dimple design achieves the highest heat transfer coefficient and pressure loss. The thermal performance criterion of the semi-dimple pair is approximately 15-20% higher than that of the smooth channel and 33-37% higher than that of traditional dimples [24]. Other researchers examined the heat transfer characteristics in a channel with drilled dimples, comparing these findings to those of cylindrical and circular dimples. The enhanced channel featuring slot dimples exhibited a higher heat transfer rate than circular or spherical/dimpled channels. This improvement is attributed to the slot dimples, which promote increased eddy flow, enhance turbulence, and raise flow resistance [25]. The effect of ratio pitch to dimple diameter, and the flow rate vary through channel by reducing the hydraulic diameter and restricting fluid flow, the dimples in the channel increase the heat transfer rate. Additionally, a maximum increase of 2.5 in the Nu was calculated for the channel with circular dimples [26]. Additionally, the slot dimples facilitate a breakdown of the boundary layer and further enhance flow turbulence, significantly improving thermal-hydraulic performance. The optimal thermal enhancement was achieved at approximately 2.02 with an enhanced channel featuring dimples of about 1.5 mm in diameter, a Reynolds number of 5000, and a pitch of 30 mm. To enhance heat transfer in solar water heating systems, dimpled channels are utilized through numerical analysis [27]. Moreover, the various dimple diameters in a turbulent three-dimensional channel of single-phase flow were examined. The k-ε turbulence model employed in three configurations of channel dimpling over a Re range of 1500 to 24000. The numerical findings indicated that the central cross-sectional orientation of both the dimpled and smooth channels exhibited symmetric flow fields. Additionally, low turbulent flow was observed adjacent to the wall dimples, leading to chaotic behavior. The temperature fluctuations in the 2 mm diameter dimple channel, at a lower flow rate of approximately 0.56 L/min, were 26.8%, 10.57%, and 3.68% greater, respectively, compared to smooth channels [28]. Several designs investigations of dimple layouts incorporating a combination of corrugated channels and twisted tape were also investigated numerically. The fluid flowing through the channel is wastewater. In this study, dimensionless diameters of 0.09, 0.18, 0.27, and 0.36 are used, with a mean diameter of 1 mm. Reynolds numbers ranging from 1500 to 14,000 are applied in the numerical calculations. The findings indicate that (Nu) increases with the dimple diameter. A numerical investigation of the flow dynamics and thermal transfer within the cross-combined dimple channel was performed using the k-ε model. The cross-combined dimple is created on the channel surface through the merging of two ellipsoidal dimples, which are uniformly distributed along the axial direction [29].

The influence of dimple parameters such as pitch, diameter, quantity, and spacing, on thermo-hydraulic efficiency is analyzed using computational fluid dynamics (CFD) to examine thermal flow and heat performance in dimpled pipes. The investigations reveal that dimples improve heat transfer by 35.8% to 36.2% compared to smooth pipes by altering flow patterns and inducing swirling. Optimizing the shape of the dimples is crucial for enhancing heat transfer, with Nusselt number (Nu) and friction factor (f) showing variations of approximately 7.5% and 6.5%, respectively [30].

Despite these efforts, most previous researchers have focused on the size, shape or depth of dimples or limited configurations of dimpled channels. Additionally, many studies were conducted within moderate Reynolds number ranges or relied on flow assumptions. As a result, the combined effects of varying the number of dimples in the channel cross-section and operating at higher Reynolds number ranges have not been thoroughly explored.

Although there is limited research on the simultaneous increase in heat transfer and decrease in pressure, this study seeks to investigate how varying dimple numbers affect performance through simulations for industrial cooling designs. Specifically, it will numerically investigate the heat transfer and water flow characteristics in a channel with different dimple counts in the cross-sectional area, covering a wider Reynolds number range (2500-12000) than previous studies. The study will also examine the impact of dimple density and flow intensity on thermo-hydraulic performance, contributing to the optimization of enhanced heat transfer channels. Also, this research will analysis the effects of fluid flow in both smooth and dimpled channels, focusing on pressure distribution, temperature distribution, and friction factor.

2. Methodology

2.1. Physical Model

This study investigates the thermal-hydraulic performance of a circular channel featuring dimples on its inner surface. The channel is constructed from copper, known for its superior thermal conductivity, which enhances heat transfer within it. It has a length of approximately 600 mm, an inner diameter of 25 mm and an outer diameter of 28 mm [26]. The geometrical dimensions of the channel are illustrated in Fig. 1. The dimples, each with a diameter of 2 mm, are affixed to the inner wall of the channel and are arranged periodically along its length with a pitch of 25 mm.

Three different configurations of dimple arrangement in the cross-section were considered to evaluate their influence on flow and heat transfer enhancement:

Configuration A: 4 dimples per cross-section

Configuration B: 6 dimples per cross-section

Configuration C: 8 dimples per cross-section

Fig. 2 and Fig. 3 illustrate the three geometric configurations of dimple arrangements in the cross-section of the channel. In this study, water was utilized as the working fluid. The investigation covered a range of Reynolds number from 2000 to 12000, encompassing the transition and turbulent flow regimes typically observed in practical heat exchanger applications. The thermophysical properties of water were assumed to be constant at the reference temperature.

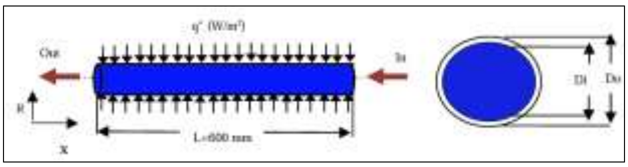


Fig. 1. Details and dimensions with boundary conditions along channel.

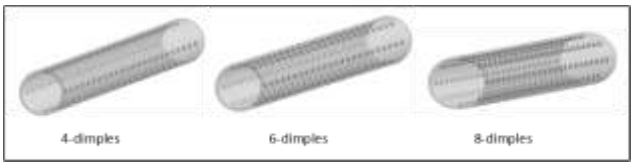


Fig. 2. Circular dimples of three cases of dimple number in cross sectional area.

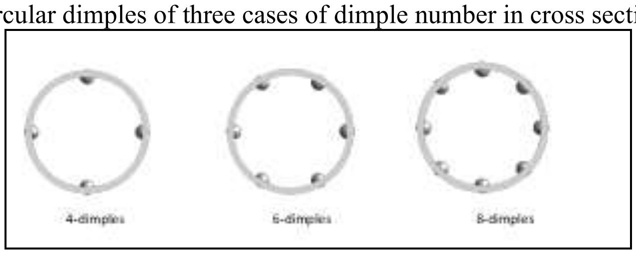


Fig.3. Cross section area of circular dimple channels.

2.2 Governing Equations

The flow is characterized as three-dimensional, turbulent, incompressible, maintains a consistent property through the channel. The simulation employs the RNG (K− ε) turbulence model [20]:

Continuity equation

$$\frac{\partial}{\partial x}(\rho ui)=0 \tag{1}$$

Momentum equation

$$\frac{\partial}{\partial x_i}(\rho u_i u_j)=-\frac{\partial P}{\partial x_i}\pm\rho g+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}-\frac{2}{3}\delta ij\frac{\partial u_i}{\partial x_i}\right]+\frac{\partial}{\partial x_j}\left(-\rho\acute{u}_i\acute{u}_j\right) \tag{2}$$

Energy equation

$$\frac{\partial}{\partial x_i}(\rho Tu_i)=\frac{\partial}{\partial x_j}\left[\left[Cp\left(\frac{\mu}{Pr}+\frac{\mu_t}{\sigma_k}\right)\frac{\partial T}{\partial x_j}\right]\right] \tag{3}$$

Where Cp =specific heat ,μ =viscosity of fluid ,ρ=density of fluid .

2.3 Turbulence Modelling

Turbulent kinetic energy

$$\frac{\partial}{\partial x_i}(\rho Ku_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial K}{\partial x_j}\right]+(G_K-\rho\varepsilon) \tag{4}$$

Turbulent dissipation rate .

$$\frac{\partial}{\partial x_i}(\rho Kiu_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_j}\right]+\left(\frac{C_{\varepsilon1}G_K\varepsilon}{K}-\frac{C_{\varepsilon2}\varepsilon^2\rho}{K}\right) \tag{5}$$

$$G_K=-\rho\overline{\acute{u}_i\acute{u}_j}\frac{\partial u_j}{\partial x_i} \tag{6}$$

$$\mu_t=G_K\rho f_\mu\frac{K^2}{\varepsilon} \tag{7}$$

The model constants used are as follows: $C_\mu = 0.09$ which represents the turbulent viscosity constant, $\sigma_k = 1.0$ turbulent Prandtl number for turbulent kinetic energy (k) , $\sigma_\varepsilon = 1.30$ the turbulent Prandtl number for turbulent dissipation rate ($\boldsymbol{\varepsilon}$), $C_{\varepsilon1} = 1.44$, the production constant of turbulent dissipation rate ($\boldsymbol{\varepsilon}$) and $C_{\varepsilon2} = 1.92$ the dissipation constant of turbulent dissipation rate ($\boldsymbol{\varepsilon}$).

2.4 Boundary Conditions

The numerical simulations adhered to rigorous protocols to accurately depict the flow and temperature within the dimpled channel. In the entrance region, the inlet velocity was defined to achieve the specified Reynolds number, while the fluid temperature was maintained at 303 K. The outlet was designated as a pressure outlet at atmospheric pressure, permitting the flow to exit freely. We ensured that the walls of the channel maintained a no-slip condition, meaning that the fluid and the wall surface did not move relative to one another. Additionally, we also applied a uniform heat flux of 1000 W/m² to the exterior of the copper channel to simulate constant heating. These conditions enabled an accurate prediction of flow behavior, turbulence formation, and heat transfer through the channel, particularly in areas were dimples.

3. Data Reduction

The entrance of circular channel is assumed to have uniform profile for both temperature and velocity, as described in equations below:

$v=0$,$w=0$,$u=u_{in}$,$T=T_{in}$

The turbulent dissipation rate (K) and the entrance profile of turbulent kinetic energy (ε_{in})are determined by [20] :

$$K_{in}=0.03u_{in}^2 \tag{8}$$

$$\varepsilon_{in}=C_\mu\frac{K_{in}^{3/2}}{0.03R} \tag{9}$$

At exit, the boundary conditions:

$$\frac{\partial u}{\partial x}=\frac{\partial v}{\partial x}=\frac{\partial w}{\partial x}=0, \qquad \frac{\partial p}{\partial x}=0$$

Also, the (K−ε) model have been a value:

$$\frac{\partial K}{\partial x}=\frac{\partial\varepsilon}{\partial x}=0$$

The walls (no-slip condition), and velocity components are zero:

u=v=w=0

The Reynolds number (Re) is computed as equation (10), where D_i is inner diameter:[14] ,[30]:

$$Re = \frac{\rho\, u_{in}\, Di}{\mu} \tag{10}$$

A coefficient of heat transfer (h_x)is defined as [22],[30]:

$$h_x = \frac{q''}{(T_{wall(x)} - T_{bulk(x)})} \tag{11}$$

$$T_{bulk} = \frac{\int_0^R uT(2\pi r)dr}{\int_0^R u(2\pi r)dr} \tag{12}$$

Where q'' =heat flux ,T_{wall} =wall temperature ,T_{bulk}=bulk temperature

The amount of enhancing of heat transfer is [26] [30]:

$$Nu_x = \frac{1}{L}\int_0^L Nu_x dx \tag{13}$$

The friction factor (f) is defined as follows [30]:

$$f = \frac{\Delta P}{(\frac{L}{D_i})(\frac{\rho u_{in}^2}{\lambda})} \tag{14}$$

The test section has L as the length of channel, ΔP is pressure drop along channel, and u_{in} is the mean inlet velocity.

Thermal performance criterion (PEC) represented as ratio of the friction loss to enhancement in heat transfer as.[26].

$$PEC = \frac{Nu_{with\ dimple} / Nu_{without\ dimple}}{(f_{with\ dimple} / f_{without\ dimple})^{1/3}} \tag{15}$$

4. Mesh Generation

The computational domain was discretized using a three–dimensional mesh generated with ANSYS Meshing. Due to the geometric complexity introduced by the dimples on the inner surface of the channel, a refined mesh was applied in regions near the wall and inside the dimple cavities to accurately capture the velocity gradients and thermal boundary layers. Inflation layers were implemented along the channel wall to properly resolve the near-wall flow behavior and to improve the prediction of wall shear stress and heat transfer characteristics. Fig. 4 and Fig. 5 show the mesh in the dimple channel.

To ensure that the numerical results were independent of the mesh size, a grid independence test was performed using several mesh densities. Three different mesh sizes were examined with approximately 0.8 million, 1.4 million, and 2.1 million elements. The comparison was conducted based on the predicted average Nu and friction factor, as illustrated in Fig. 6. The results showed that the variation between the medium and fine meshes was less than 3.2%, indicating that the medium mesh provided sufficient accuracy with reasonable computational cost. Therefore, the mesh containing approximately 1.4 million elements was selected for the present simulations.

The numerical solution was considered converged when the residuals of the governing equations reached 10^{-6} for the energy equation and 10^{-4} for the continuity and momentum equations.

The quality of the near-wall mesh was evaluated using the dimensionless wall distance (y^+) parameter. The values of y^+ were maintained within an acceptable range to ensure accurate turbulence modeling and proper resolution of the boundary layer near the channel wall.

Moreover, in the simulations to utilize the improving wall functions, the grids domain needs to be repeatedly debugged to get the wall y-plus value is about equal to or below 1.0, satisfying the enhanced wall function requirement and ensuring accurate resolution of the viscous sublayer. This confirms the reliability of the numerical predictions for both heat transfer and flow characteristics. Table 1 presents the grid refinement levels along with the corresponding wall y+ values used to validate the applicability of the enhanced wall treatment. It can be observed that progressive mesh refinement significantly reduces the y+ values.

Table 1: Grid refinement and corresponding y+ values.

Elements (million)	Minimum y+	Maximum y+	Status
0.8	1.35	2.10	Not acceptable
1.4	0.92	1.40	Acceptable (marginal)
2	0.65	0.98	Fully acceptable

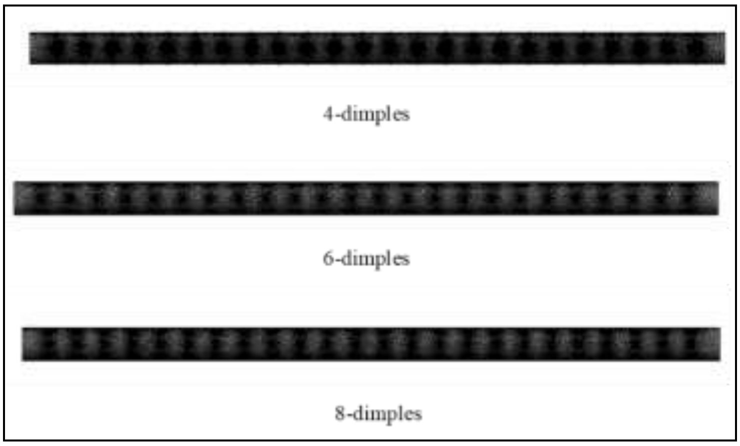
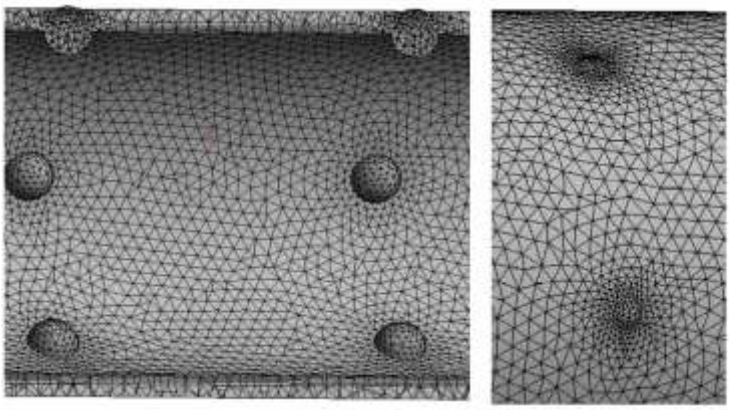
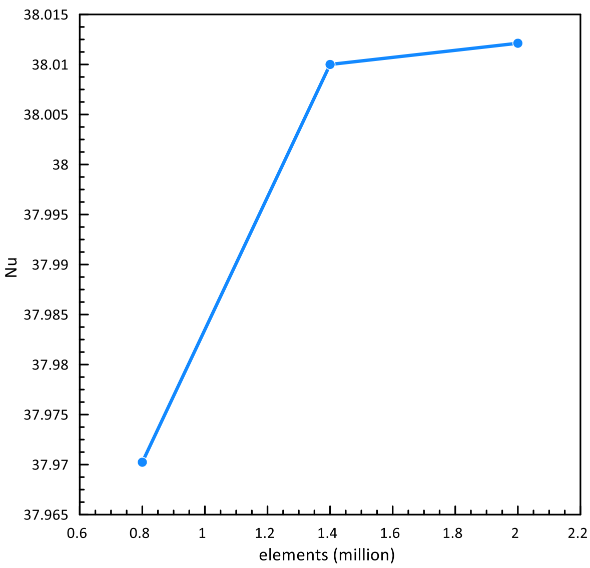


Fig. 4. Mesh along the dimple channels.

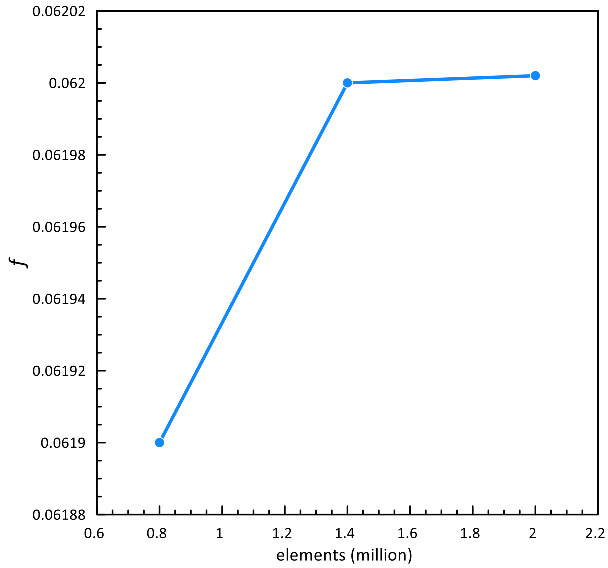


(a) Inside **(b) O**utside

Fig. 5. Mesh configuration of dimple channels.



(a)Average Nu



(b)friction factor

Fig. 6. Mesh independence test

5. Validation

To validate the current model and verification utilized the numerical data of currently available was compared by using the numerical data of [26]. At turbulent flow, the following equations were also used to get a second validation for the *f* and Nu in relation in the smooth channel [10].

Dittus- Boelter equation:

$$Nu=0.023 \times Re0.8 \times Pr0.4 \qquad (16)$$

Blasius equation:

$$f=0.3164 \times Re-0.25 \qquad (17)$$

Gnielinski equation:

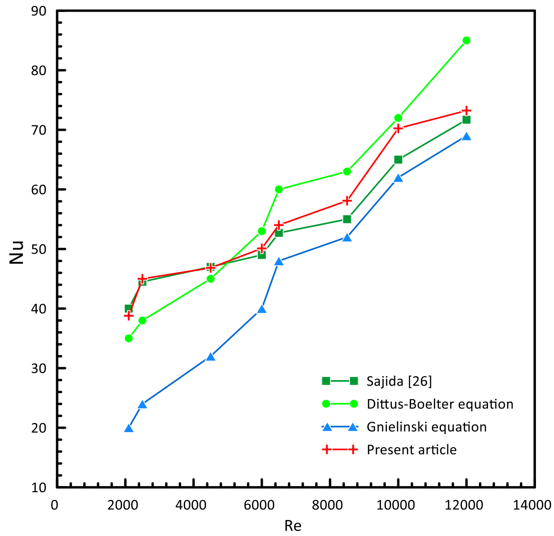
$$Nu=0.03277 \times Re0.742 \times Pr0.4 \qquad (18)$$

The Petukhov equation:

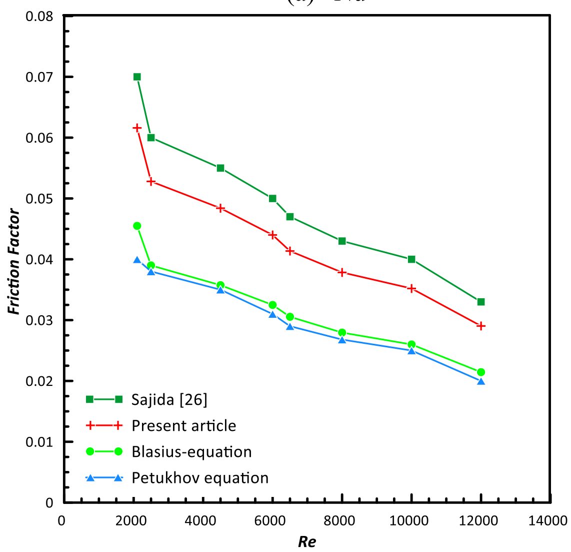
$$f=0.6165\times Re-0.317 \qquad (19)$$

A comparison of smooth channel is illustrated in Figure 7. The empirical equations are compared to the numerical results of the Nu in Fig. 7a. The numerical results for a smooth channel appear as being in good agreement with the correlation data. The investigations show that the current data and the Dittus-Boelter equation deviate from each other by a maximum of 9.3% at Re = 2500 and a minimum of 4.2% at Re = 12000. Additionally, the results of the current study are in excellent agreement with the Sajida [26], with a minimum deviation of 5% and a maximum of 7.69%. Also, the current work's results, which have a minimum variation of 5% and a maximum variance of 9.1%, are in good accord with the Gnielinski equation.

Figure 7b illustrates the comparison between the current study and empirical data from equations friction behavior. The variation in data of the current results and the Petukhov equation is very high about 9.12 % at a low Re of 2500, decreasing to 7.4% as the Re value increases to 12000. Results show that the Blasius-equation and the current results differ by a maximum of 5.7% at Re = 2500 and a minimum of 10.2% at Re = 12000. It is found that correlation data is satisfactorily fitted by the numerical outflow for a smooth channel. It is clear that there are significant differences between the current findings and the Sajida [26]] deviate from each other by a maximum of 6.3% at low Reynolds numbers (at 2500), and that these differences drop to 4.3% when Reynolds numbers are raised to Re = 12000.



(a) Nu



(b) Friction factor

Fig. 7. Behavior of the empirical equations and simulation results

Fig.8 illustrates the divergence in behavior of the empirical equations and simulation results between the current data and the results experimental data [16] and [26] regarding the pressure though smooth channel during operational conditions. The observed greatest variation in pressure across the inflow and outflow ranges from 4.7% to 8.7%. All deviation percentages incorporate certain simplifications, including the assumption of perfectly smooth dimples, different range of Re, and ideal boundary conditions.

However, manufacturing defects, surface irregularities, and variations in fluid properties can significantly affect the actual characteristics of heat transfer and pressure loss specially in experimental data.

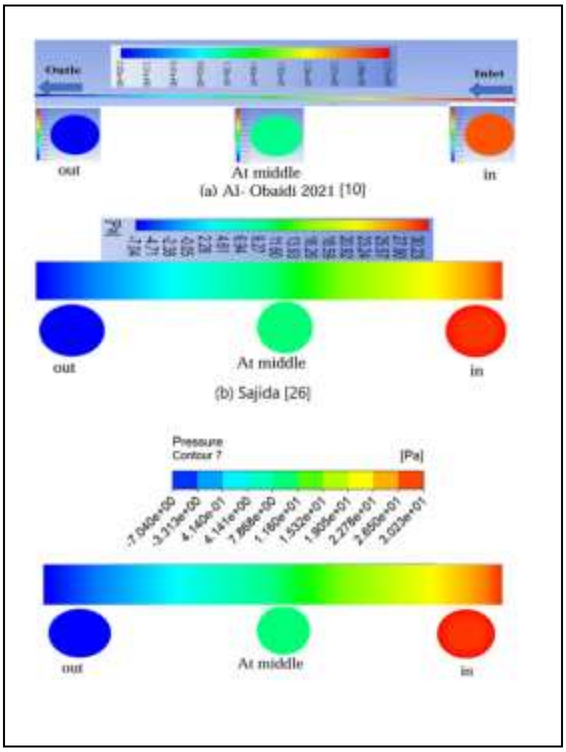


Fig. 8. Validation of the pressure with Al- Obaidi 2021[16] and Sajida [26].

6. Results and Discussion

Figure. 9 presents the effect of dimples on temperature distribution on smooth and dimple channel at conditions of Re about 2500 by comparing the temperature contours of dimpled and smooth channel with 1000W as heat flux. The temperature contour distribution along the channel shows a gradual change in color from green at the inlet to red near the outlet of the dimpled channel. This change in color shows that the fluid's temperature is slowly rising as it flows through the heated channel and absorbs heat from the wall, which is always getting hotter. The temperature contours at the inlet are mostly green, which shows that the fluid coming in is cooler. The temperature of the fluid rises slowly as it moves downstream because the heated wall keeps transferring heat to it. The middle of the channel starts to show yellow and orange colors as the temperature rises. The contours near the outlet region change to red, which means that the channel is the hottest place [26]. The presence of dimples in the channel considerably modifies the temperature distribution compared to a smooth wall configuration. The dimples generate localized vortices and recirculation zones that improve mixing between the core flow and the fluid adjacent to the heated wall. This mechanism interferes with the thermal boundary layer, leading to a more efficient heat transfer mechanism [12],[30].

Because of this, the temperature in the dimpled channel becomes more even and disturbed, due to flow separation and reattachment, which create noticeable local differences near the dimples. The smooth channel, on the other hand, has a temperature field that changes more slowly and in layers. The thermal boundary layer grows steadily along the length of the channel. This makes mixing less effective and heat transfer less efficient. The extra dimples help with flow reattachment and mixing, which keeps the boundary layer from settling down and makes the temperature distribution more even [17]. In summary, the contour plots illustrate that both configurations exhibit a temperature increase along dimple channel, the dimpled channel exhibits stronger thermal mixing and improved heat transfer characteristics compared to the smooth channel, due to the disturbance of the boundary layer caused by the dimples.

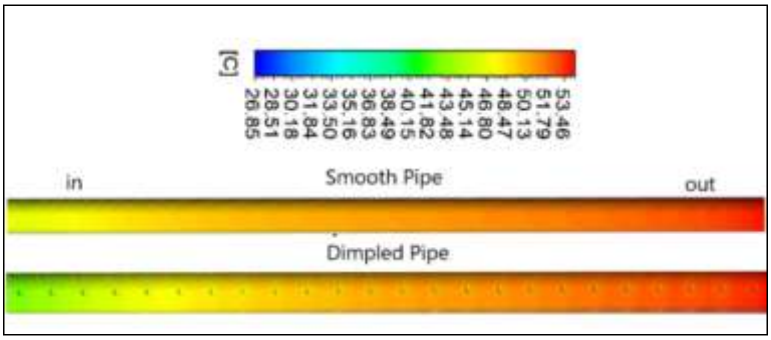


Fig. 9. The effect of dimples on temperature in smooth and dimple channel at Re = 2500

Figure 10 and 11 represent pressure distribution along the channel for case of 4-dimples in cross sectional area with a Reynolds number of 2500 and heat flux of 1000 W/m2, the pressure fluctuations reduced as the channel length increased. The pressure strength grows in the inflow region and reduced in the outflow region. In this figure, many observations may be noted, the maximum pressure value is observed at the region that between the near-wall and center of the channel. Furthermore, due to the low velocity value in this region, the pressure is low near the channel wall.

Moreover, it is important to analysis that the increasing the dimple diameter leads to increase in the pressure through the channel. The results of increased obstruction due to grow resistance within the channel.

Although, the pressure drops increases in 8-dimples in cross sectional area in comparison to a 4-dimpled ,6-dimpled and smooth channel because the resistance flow increase caused by a dimple, and pressure drop increasing [30].

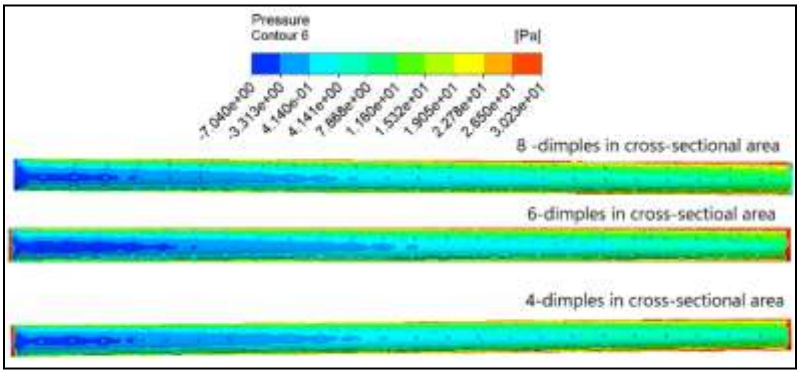


Fig. 10. Contour of pressure along channel with different number of circle dimple.

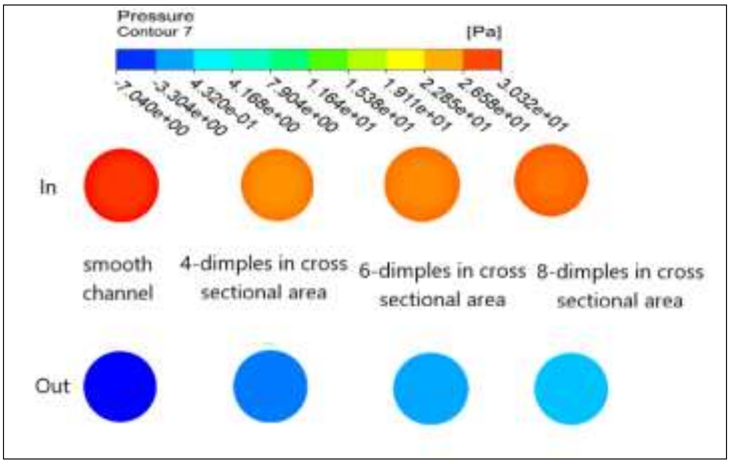
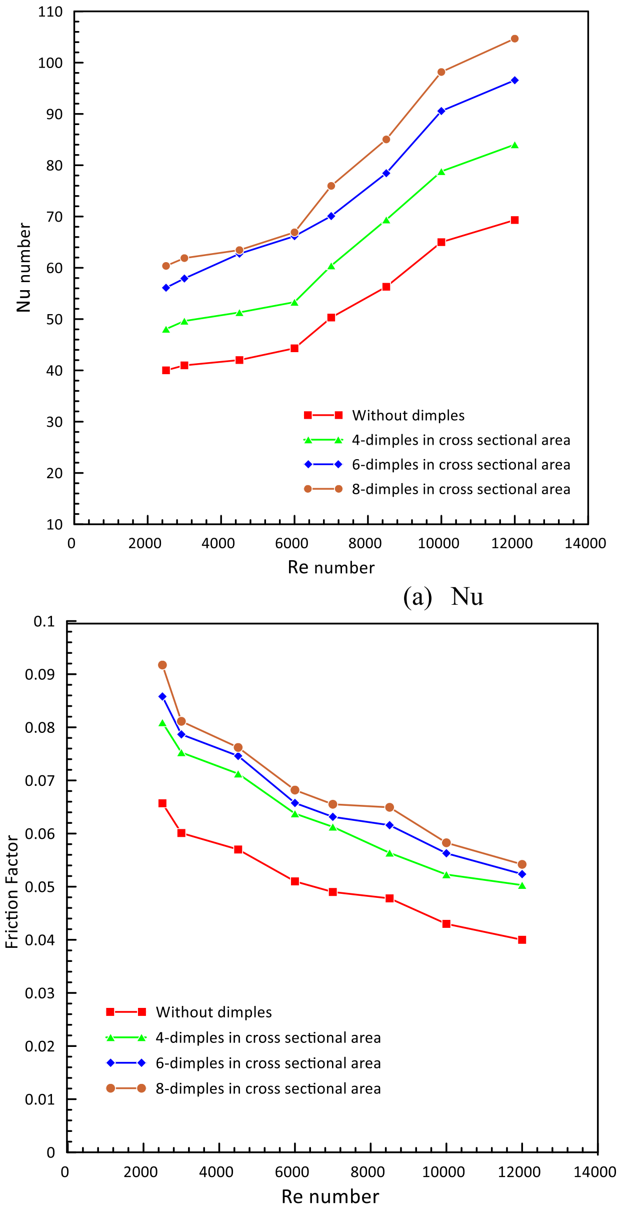


Fig. 11. Contour of pressure in cross section area of channel with different regions.

Figure 12a shows the influence of various number in cross section area on Nu and Re. The behavior of Nu decreases as increasing Re and decreasing in numbers of dimple in cross section area. Obviously, the case of 4-dimples in cross section area has the smallest Nu while the case 6- dimples in cross section area has a rather high Nu. The case 8-dimples in the cross-section area provide the highest Nu due to their significant divergence and an obstruction.

Additionally, the dimple in cross section area increased, and the Nu in the dimple channel over the Reynolds number range of 2500-12000 by 28.8-30.2% for case 4-dimples in cross sectional area, 40.4- 41.3%, for case 6-dimples in cross sectional area, and 47.6-49.06% for case 8-dimples in cross sectional area over the, compared to the smooth channel. This is related to the impact of turbulence and a secondary vortex that develops near the dimple surfaces.

The friction factor rises as the number of dimples decreases. Figure 12b shows the behavior of Re with respect to friction factor. It should be analyzed that the friction factor enhances as the number of dimples grows. Also, high flow impingement and loss, which resulted in a larger pressure drop, are the primary causes of the case 8-dimple's quickly rising friction factor in cross section area.



(a) Nu

(b) Friction factor

Fig.12. The influence of Nu and friction factor by dimple numbers.

The numerical results indicate that the number of dimples has a significant influence on the thermal performance of the channel. Figure 13 represents the behavior of PEC with Re for various dimples in cross section area. The PEC declines with increasing Re because of increasing in turbulence so enhancement in Nu. Increasing the number of dimples enhances heat transfer by generation of vortices and promoting stronger mixing between the core flow and the fluid near the heated wall. This mechanism function reduces the thickness of thermal boundary layer and consequently improves convective heat transfer compared to the smooth channel.

However, increasing the number of dimples also leads to a rise in the friction factor and pressure drop because of the additional flow resistance created by the surface disturbances. Therefore, an excessive number of dimples may negatively affect the overall thermo-hydraulic performance.

Among the investigated configurations, the channel with four dimples (4 dimples) demonstrated the best thermal performance. This configuration provided a noticeable enhancement in the Nu while maintaining a moderate increase in the friction factor, which resulted in the highest performance evaluation criterion (PEC) compared with the other tested configurations and the smooth channel. The 4-dimple in cross section area achieves the highest PEC in all Re tests, with values varying from 1.17 to 1.45, due to the significantly reduced intensity and extent of the recirculation flow.

These results find that 4-dimples in cross section area represent an optimal configuration for improving heat transfer performance while maintaining acceptable hydraulic losses.

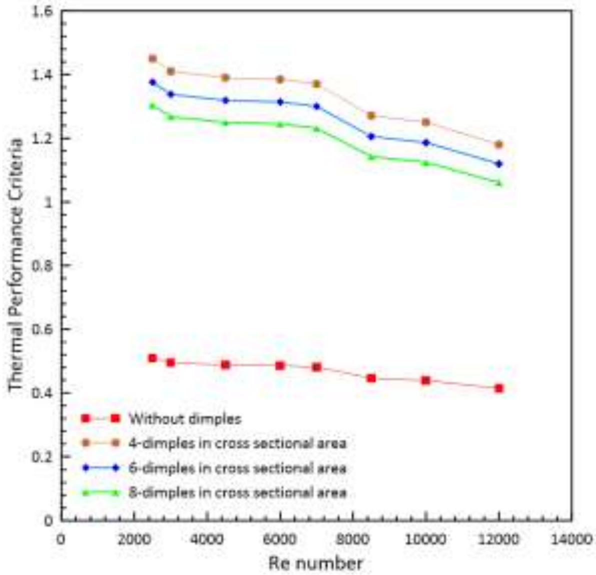


Fig.13. The influence of thermal performance criterion by dimple numbers.

7. Conclusions

A 3-dimensional, steady-state numerical model and turbulent flow were used to investigate the influence of the dimple number in cross-sectional area on flow structure and heat transfer. in the channel. The results can be described below:

1. The dimpled configurations produced a more uniform temperature distribution along the channel compared with the smooth wall case, indicating improved thermal transfer characteristics due to enhanced near wall flow interaction.
2. The introduction of dimples significantly improved the convective heat transfer performance with thermal enhancement increasing progressively as the number of dimple s increased over the investigated Reynolds number range.
3. Although the dimples caused an increase in friction losses because of flow disturbance and vortex generation, the pressure -drop penalty remained moderate relative to the achieved heat transfer enhancement.
4. The overall thermal-hydraulic performance evaluation showed that the heat transfer enhancement exceeded the corresponding hydraulic resistance increase, confirming the effectiveness of the dimpled configuration.
5. Among the investigated cases, the four-dimple configuration achieved the best overall performance evaluation criterion, indicating an optimal balance between heat transfer enhancement and frictional losses.
6. In real-world systems, dimpled pipes and surfaces provide a practical, long-lasting, and economical means to enhance heat transfer. Someone has already shown promise in heat exchangers, cooling devices, and thermal management systems, and with more enhancements, these should soon be common in high-efficiency thermal systems.
7. The present work is limited to numerical simulations under specific geometric and operating conditions. The absence of independent experimental validation represents a limitation of current study. Future work should include experimental verification and investigation over wider Reynolds number ranges to further confirm and generalize the obtained results. this.

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