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Characteristics of Heat Transfer and Turbulent Flow in a Baffled Pipe with Different Arrangements

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Abstract — **In the heat transfer, fluid flow and energy fields, baffles are an advanced enhancer to improve heat transfer and fluid mixing by working as an obstacle to the flow particles and then increasing the turbulence. The present paper numerically investigates the thermal performance of a circular pipe with a centralized baffle in two arrangements, with a Reynolds number (Re) (ranging from 10,000-50,000) under constant wall heat flux boundary conditions. Ansys Fluent software is used to solve the flow field considering six conical baffles with different Pitch ratios (PR) (from 1 to 5). Results show that baffles shape, arrangement, and PR have a significant impact on the properties of flow and heat transfer. The obtained results show an effective role for the baffles to promote thermal performance when it is used in heated pipes. Heat transfer rate is increased for the baffled pipe by 1-2 compared with the smooth pipe. Moreover, the best value of friction factor, thermal performance, and Nusselt number is recorded at PR= 5 at Re=30000 in the baffled pipe for the second arrangement.**

Key words — **Baffled Circular Pipe, Fraction Factor, Heat Transfers, Pitch Ratio, Turbulent Flow.**

I. Introduction

The improvement of the heat transfer rate of a surface could be attained by either extending the coefficient of heat transfer or the heat transfer surface area by conducting some extended surfaces onto the duct walls such as fins, baffles, and ribs. In circular pipes, the turbulent flows are encountered in various applications of engineering for example nuclear reactors, cooling of gas turbines, combustion chambers, solar thermal energy storage, shell and tube heat exchangers, and electronic devices Paul *et al.* [1].

In these findings, baffles were discovered to enhance the turbulence flow and increase the heat transfer rate by convection. Consequently, the existence of baffles increases the turbulence and leads the flow to swirl. Increment in pressure drop, on the other hand, has always been a major source of concern. To deal with the issue, more investigations are required to find an efficient design of the baffle that enhances the coefficient of heat transfer while reducing pressure drop. The following is a summary of the most important literature on baffles.

Smulsky *et al.* [2] conducted experiments to examine the impact of baffle slopes which ranging from 50° to 90° and found that reducing recirculation areas enhances heat transfer. The local heat transfer for an angle of 50° was observed to be around 40% more than that of an angle of 90°.

The characteristics of heat transfer of a tube provided with four various forms of zigzag-cut baffles were investigated by Nuntadusit *et al.* [3]. It was discovered that the rectangular zigzag cut baffle provided the best thermal efficiency, leading to improve the transfer of thermal energy throughout the upstream and downstream sides of the baffle.

Lu *et al.* [4] examined the impact of inclination baffle in a channel with a rectangular shape experimentally and numerically and found that the model of the K-ω SST turbulence leads to better consequences for forced convection calculations.

Sripattanapipat *et al.* [5] stated that the baffle with a diamond shape and various angles ranging from 5 to 35°, could improve the heat transfer for Re ranging from (100 to 600), by (200 to 680%). Conversely, this enhancement is accompanied by a 20 to 220-times growth in friction loss over the smooth tube.

In the direction of increasing the cooling rate in the manufacturing of gas turbines, Kamali *et al.* [6]suggested numerous baffle shapes. They determined that the baffle with a trapezoidal shape is the optimal choice for improving heat transfer and reducing pressure drop as compared to other shapes. Turgut and Kizilirmak [7] performed a numerical simulation to examine the steady state of 3D turbulent forced convection flow in a circular pipe with a baffle. A numerical analysis was conducted by Selvaraj *et al.* [8] to examine the flow and heat transfer in a circular pipe. Wang *et al.* [9] studied numerical heat transfer and turbulent flow in tubes with longitudinal internal finned pipes. Guo *et al.* [10] proposed a CFD simulation to enhance the heat transfer rate in a circular pipe configured with helical screw-tape supplements from the principle of the viewpoint of field synergy. They discovered that helical insertions with alternating left and right twists functioned considerably better than the helical inserts with uniformly right twists. Zhang *et al.* [11] achieved a study of numerical analysis on the friction factor and heat transfer characteristics of a pipe that is configured by four diverse widths of helical screw-tape without inserts of core-rod. They revealed that the output thermal of the helical screw-tape inserts of altered widths diverges from 1.58 to 2.35.

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Using fluent software, Shahdad and Fazelpour [12] investigated the heat transfer in the flow around an array of plain and perforated fins installed on a plate in the 20,000–50,000 Reynolds range. According to the findings, perforated fins increase both the heat transfer coefficient and the Nusselt number. Furthermore, it was discovered that increasing the Reynolds number significantly increases the heat transfer coefficient and Nusselt number. Alfarawi *et al.* [13] studied fully developed turbulent flow via a duct with a rectangular shape with the ribbed bottom wall for various parameters and geometries of ribs such as the rib pitch to height ratio, and Reynolds number.

Kwankaomeng *et al.* [14] performed a numerical investigation for the performance of the coefficient of Thermal energy transport on a duct with a square shape that was fitted with 30 cutting angle baffles. They examined the problem for Re ranging from 100 to 2000 and established that as the Reynolds number increased, the thermal energy transport coefficient improved. Salhi *et al.* [15]adopted a new technique to improve the heat transfer efficiency of a cooling system. They inserted vertical and rotating baffles in a rectangular-section channel to create discontinuities in the flow. By completing the numerical investigation, the results show that the new technique is more efficient with the behavior of thermohydrodynamic and is affected significantly by the orientation of baffles.

Sriromreun *et al.* [16] performed, experimentally and numerically, an examination of the influence of baffle tabulators on heat transfer enhancement in a rectangular duct. The results of the experiment revealed that the Z-type of the baffle significantly affects the frictional loss and the coefficient of thermal energy transport as compared to a smooth channel without a baffle.

The utilization of porous baffles to improve the coefficient of thermal energy transport in a rectangular duct was investigated by Ko and Anand [17]. They discovered that using porous baffles increased heat transfer as up as 300% compared to smooth straight channels. They developed the equations of correlation for heat transfer enhancement ratio and heat transfer enhancement per unit increase in pumping power as a function Re.

A numerical study on heat transfer and laminar periodic flow has been done by Eiamsa and Promvonge [18]. The model of study is a rectangular channel with triangular wavy baffles fitted on the upper and lower surface of the channel and it is inclined with an angle of 45°. The obtained results regarding the Nusselt number, pressure drop, and friction factor showed that the baffled channel is more effective than the channel without baffles.

Another experimental study was carried out by Mohsen *et al.* [19], they used different baffles geometries including interrupted rectangular, circular, and helical ribs for a variety of Reynolds numbers of cold and hot fluid. The results showed that using rectangular baffles gives the best heat transfer enhancement compared with other used geometries.

Hasan *et al.* [20] carried out experimental work to study the influence of installing a helical wire on the inner and outer surface of the tube for different values of Re. They found an improvement in heat transfer by 86–150% by using the helical wire in the double-pipe heat exchanger while there is no enhancement in the smooth surface.

Majidi *et al.* [21] studied the influence of installing a copper-wire fin on the outer area of the internal tube in a double-pipe heat exchanger on the overall heat transfer coefficient. The results showed that the flow rates of hot and cold-water streams significantly affect the heat transfer enhancement and increase it up to 6%.

Using numerical modelling, Sheikholeslami *et al.* [22] examined the impact of helical twisted tapes on the enhancement of heat transfer in heat exchangers. They discovered that the stronger secondary flow results in a thinner thermal barrier layer, which enhances heat transfer.

When consider using both fins and nanoparticles to improve heat transfer in the heat exchanger, Sheikholeslami *et al.* [23] found that the rate of heat transfer increased noticeably.

Another work has been done numerically by Al-Juhaishi, *et al.* [24] to study the thermal-hydraulic act in a channel with curved shape of a journal bearing equipped with oblique horseshoe baffles. Different parameters of baffles and their impacts such as attack angle and number of baffles, have been considered. The results show a significant improvement in heat transfer in the baffled channel about 2.5 to 3.8 times compared to the soft channel.

It follows that underflow circumstances, the fin design and dimensions have a significant impact on the rate of heat transfer and the pressure drop. The performance of a heat exchanger can be significantly affected by small adjustments to the fin design or dimensions. The surface temperature of the fin and the effectiveness of heat transfer are both impacted by the flow direction over the fin. Additionally, the fin's placement—internal or external, in the hot or cold fluid-affects the flow and heat transfer rate. Also, from above study there were not a study the effects of the baffle with conical shape fitted in the central of circular pipe with different arrangements. Therefore, the present study aims to numerically investigate the hydrodynamic and heat transfer characteristic of turbulent flow inside the pipe with conical central baffles under different values of Reynolds number. The effects of the arrangement of baffles and pitch ratio are studied.

II. Model Description

A. *Baffle Geometry and Its Arrangement*

Fig. 1. shows the geometry and parameters of the problem that was covered in this paper. It consists of a circular pipe with internal central baffles. Air is the used fluid with inlet velocity U_{in} and inlet temperature T_{in}. The flow is considered 3-D, turbulent, steady-state, Newtonian, incompressible, and constant properties of the fluid. The pipe is subjected to constant heat flux $\bar{\bar{q}}$ from the side walls neglecting the radiation from the walls and the buoyancy effect. The baffles are conical with a base diameter (D) of 8 mm and length of 4 mm and the ratio between the baffles is PR (S/d).

The distance between each baffle starts from 8 mm for the first PR. The pitch ratio is altered gradually (from 1 to 5) by altering the distance between each baffle equally.

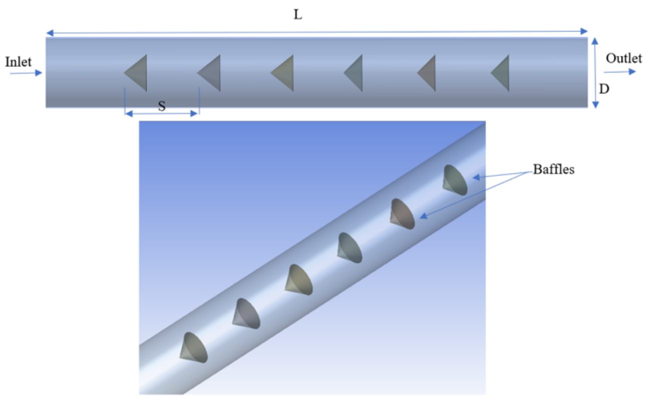

Fig. 1. 3D and side view of baffles inside the pipe.

The governing equations are given below.

1) Continuity equation

Continuity equation is given in (1).

$$\frac{\partial \overline{u_i}}{\partial x_i} = 0 \quad (1)$$

where i = (1, 2, and 3) which presents three dimensions x, y, and z with mean velocities u, v, and w, respectively.

2) Momentum equation

The momentum equation in Cartesian coordinates is given as (2).

$$\rho \overline{u_j} \frac{\partial}{\partial x_j}(\overline{u_i}) = -\frac{\partial \bar{P}}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu\left(\frac{\partial \overline{u_j}}{\partial x_i} + \frac{\partial \overline{u_i}}{\partial x_j}\right) - \rho \overline{u_i' u_j'}\right) \quad (2)$$

where

ρ: the density of a fluid;

P: pressure;

μ: viscosity of a fluid.

Equation (2) is called Reynolds-averaged Navier-Stokes where $\rho\overline{u_i' u_j'}$: is the turbulent stresses tensor or Reynolds stresses. The Reynolds stresses are computed in (3), using the familiar Boussinesq relationship [25].

$$\rho \overline{u_i' u_j'} = \frac{2}{3}\mathrm{k}\delta_{ij} - \mu_t\left(\frac{\partial \overline{u_j}}{\partial x_i} + \frac{\partial \overline{u_i}}{\partial x_j}\right) \quad (3)$$

where δ_{ij} represents the Kronecker delta.

$\delta_{ij} = 1$ if $i = j$ and $\delta_{ij} = 0$ if $i \neq j$.

3) Energy equation

Energy equation as given in (4).

$$\bar{u}\frac{\partial \bar{T}}{\partial x} + \bar{v}\frac{\partial \bar{T}}{\partial y} + \bar{w}\frac{\partial \bar{T}}{\partial z} = \alpha\left(\frac{\partial^2 \bar{T}}{\partial x^2} + \frac{\partial^2 \bar{T}}{\partial y^2} + \frac{\partial^2 \bar{T}}{\partial z^2}\right) \quad (4)$$

where $\bar{T}$ is temperature and α is thermal diffusivity of fluid.

4) Turbulence Models (Stander $k - \epsilon$ model)

The standard $\mathrm{k} - \epsilon$ model is used in the present study, which tackles two independent transport equations. These equations are used over a wide area in industrial applications because it supplies robustness, economy, and reasonable accuracy.

The standard $\mathrm{k} - \epsilon$ model is k, and is shown in (5) and (6), according to [7].

$$\rho u_j \frac{\partial \mathrm{k}}{\partial x_j} = \frac{\partial \mathrm{k}}{\partial x_j}\left((\mu + \frac{\mu_t}{\sigma_k})\frac{\partial \mathrm{k}}{\partial x_j}\right) + 2\mu_t S_{ij}.S_{ij} - \rho\epsilon \quad (5)$$

and ϵ is given in (6).

$$\rho u_j \frac{\partial \epsilon}{\partial x_j} = \frac{\partial \epsilon}{\partial x_j}\left((\mu + \frac{\mu_t}{\sigma_\epsilon})\frac{\partial \epsilon}{\partial x_j}\right) + C_{1\epsilon}\frac{\epsilon}{k}\mu_t S_{ij}.S_{ij} - C_{2\epsilon}\rho\frac{\epsilon^2}{\mathrm{k}} \quad (6)$$

where σ_k and σ_ϵ are the Prandtl numbers linking the eddy viscosity μ_t, with diffusivities of k and ϵ, the strain rate tensor as a function of velocity can be written as (7).

$$S_{ji} = \frac{1}{2}\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right) \quad (7)$$

It is obvious that the transport equations comprise five adjustable constants: σ_k, σ_ϵ, C_μ, $C_{1\epsilon}$ and $C_{2\epsilon}$. The values for these constants have been attained by comprehensive data fitting to the standard $\mathrm{k} - \varepsilon$ model for a wide domain of turbulent flow. These values are as follows; $\sigma_k = 1.00$, $\sigma_\epsilon = 1.30$, $C_\mu = 0.09$, $C_{1\epsilon} = 1.44$ and $C_{2\epsilon} = 1.92$.

The boundary conditions are:

1. The flow with constant inlet velocity $u=u_{in}$ and temperature $T=T_{in}$.
2. The top and bottom walls of the pipe are subjected to constant heat flux = q.
3. No slip condition at the walls u=v=w.
4. Out pressure at the outlet of the pipe P=0.

5) Reynolds Number

The Reynolds number Re is defined as (8).

$$Re = \frac{u_{in}\,\rho\, D}{\mu} \quad (8)$$

6) Nusselt Number

The local Nusselt number (Nu_x) of fluid on the walls can be calculated using (9).

$$Nu_x = \frac{h_x D}{k} \quad (9)$$

where

k: thermal conductivity of the fluid.

h_x is the local heat transfer coefficient can be estimated from the relation ($h_x = \frac{q''}{T_x - T_{in}}$).

While the average Nusselt number (Nu_{avg}) can be evaluated from (10).

$$N_{avg} = \frac{1}{L}\int_0^L Nu_x dx \quad (10)$$

7) Numerical Simulation

The governing equations 1-6 are solved numerically using CFD. To check the validity of gird, 6 different numbers of mesh elements ranging from 19273 to 2066283 are considered at one value of Re and PR.

Fig. 2 shows the grid independence checking which shows that the value of outlet temperature is independent after the number of elements 604630, where the number of nodes is 147430 with average skewness quality of 0.32 which is good [26]. The error is approximately 0% up to 2-decimal places.

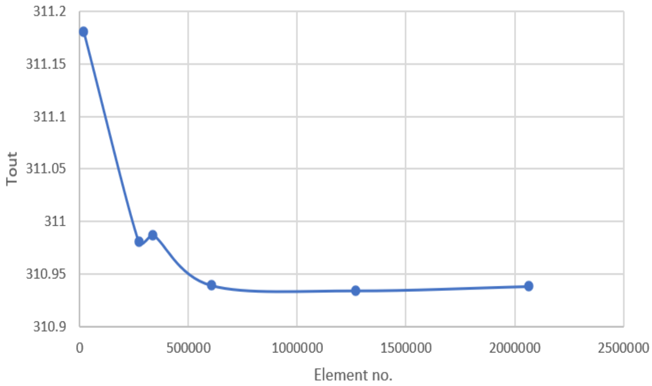

Fig. 2. outlet temperature VS mesh element number.

III. Results and Discussion

The following sections will discuss the investigation results of the impacts of different shapes and configurations of baffles on heat transfer characteristics in a circular pipe, which was done using the Ansys Fluent software CFD model.

A. Verification of the Smooth Pipe

In direction of confirming the accuracy of the numerical method undertaken and the reliability of the calculation code, the turbulence and computational model are verified by calculating friction factors for smooth pipe without baffles and then compared with friction factor calculated using the Petukhov equation as shown in Fig. 3. The findings showed that the current numerical results of the smooth pipe have extremely good agreement with the exact Petukhov equation with ± 1.4% accuracy.

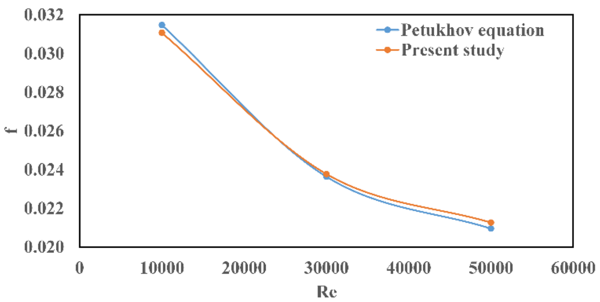

Fig. 3. The frication factor calculated in the present with Petukhov equation.

B. Nusselt Number

Since the heat exchanger enhancement is affected by the structure of a flow, the existence of different baffles in a pipe can enhance the heat transfer of this pipe. The reason is that the baffles increase the turbulence and lead the flow to swirl relying on the intensity of velocity [27]. Fig. 4 and Fig. 5 show the velocity distribution along the pipe at the same value of Re =10000 and PR (1, 3, 5).

The maximum value of velocity is at the region between the pipe and baffles wall for both arrangements. This is due to a change in flow direction as well as a reduction in cross-section area. Moreover, Fig. 4 and Fig. 5 indicate that the baffled pipe with the second arrangement has higher recirculation rates than the pipe with the first arrangement and then the velocity values are higher. For example, at Re=10000 and PR=5, the maximum value of velocity for the second arrangement is recorded as 20.8 m/s whereas it is 17.6 for the first arrangement.

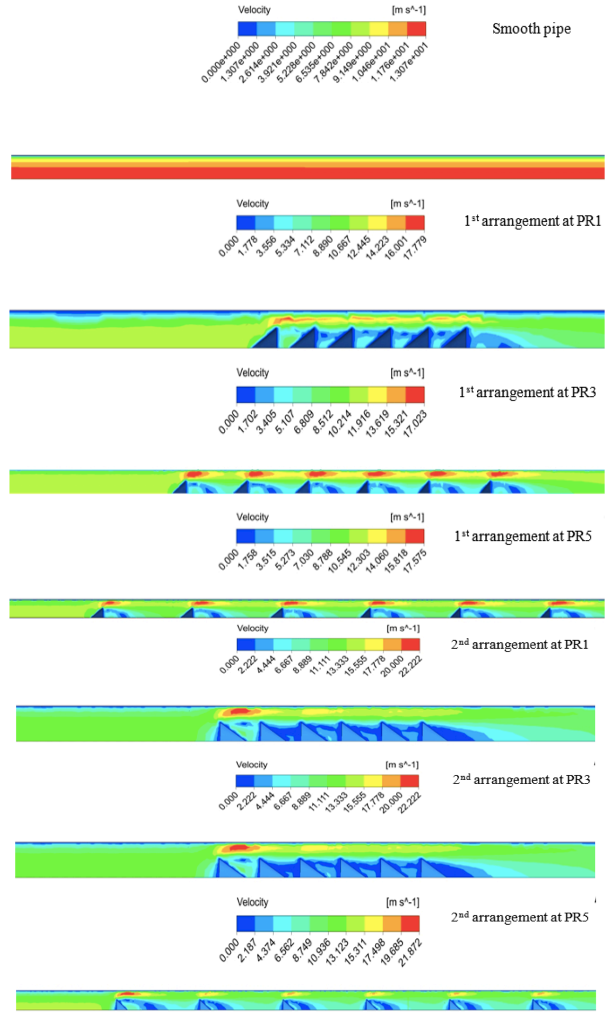

Fig 4. Velocity distribution contour at (Re=10000) at PR (1,3,5) for both arrangements.

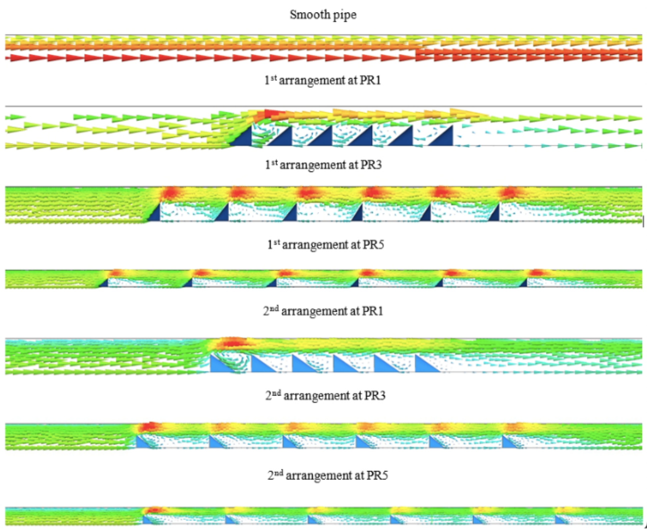

Fig. 5. Variation of swirl flow (Re=10000) at PR (1,3,5) at both arrangements.

The variation of the average Nu over the whole range of Re values is shown in Fig. 6 for the smooth and baffled pipe for the same PR. From Fig. 6, it can be seen that Nu continues increasing with raising the values of Re for both types of pipes because of increasing the heat transfer coefficient with

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increasing velocity (Reynolds number). The values of Nu for the baffled pipe are higher than the same values of smooth pipes. The optimum value of Nu of the pipe with baffles in the second arrangement is 9.2% higher than smooth pipe at Re=50000.

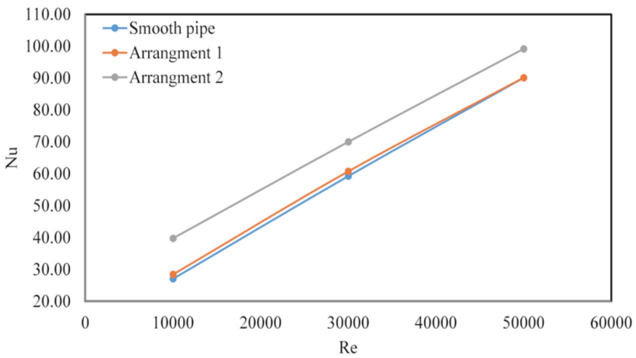

Fig. 6. The variation of the average Nu with Re for Smooth and baffled pipe.

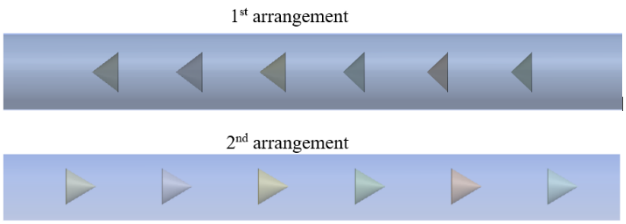

Fig. 7. The arrangements of baffles.

The simulation has been done for different pitch ratios from 1 to 5 for both arrangements (Fig. 8 and Fig. 9) to show the effect of PR on Nu. As it is clear that the heat exchanger can be enhanced by increasing the PR and as it has been found in Fig. 8 and Fig. 9 which show that Nu continues increasing as the baffles distance is raised as a reason for increasing turbulence intensity between baffles and increase the bath of flow.

The heat transfer rate is increased by about 1-2 higher than the heat rate of smooth pipe at PR=5 at Re=10000. Also, it is revealed that *Nu* is changed slightly at a higher value of *Re* because of the increase in the intensity of turbulence at a high Reynolds number. Also, it can be seen from Fig. 8 and Fig. 9 that the value of Nu for arrangement 2 is greater than arrangement 1 at the same *Re*.

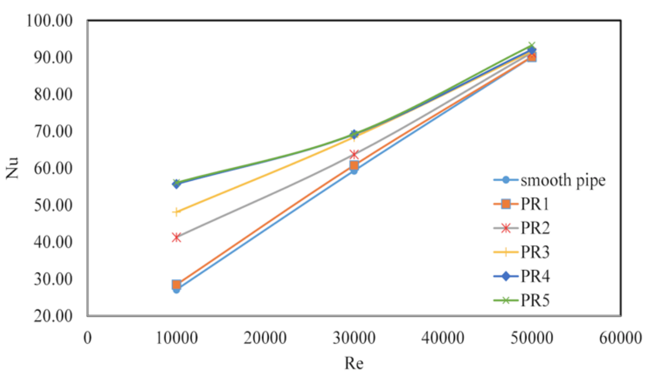

Fig. 8. Average Nu versus Re values for Smooth and baffled pipe (arrangement 1) for PR 1-5.

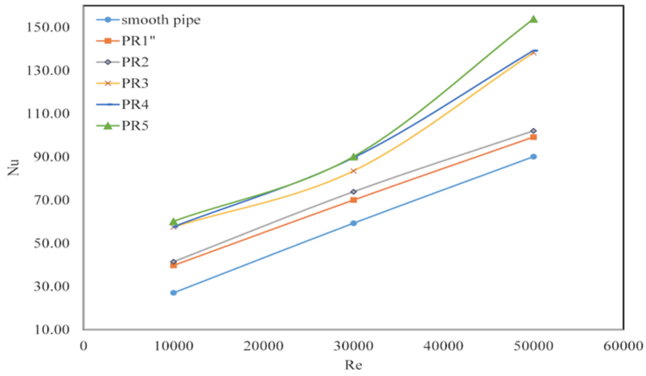

Fig. 9. Average Nu versus Re values for Smooth and baffled pipe (arrangement 2) for PR 1-5.

C. *Friction Factor*

The value of the friction factor is affected by a variety of factors, including flow direction and velocity fluctuations near the wall. It has an inverse proportion relationship with *Re*. These effects are depicted in Fig. 10 and Fig. 11. When the values of *Re* are increased, the friction factor goes down. Also, it can be seen in Fig. 10 and Fig. 11, a raising in friction factor values when using baffles inside the pipe compared with the smooth pipe. The reason returns to the dissipation of dynamic pressure as a reason for the act of the reversing flow and higher surface area. Moreover, it is clear that friction factor values are affected by increasing the distance between baffles, whereas PR increased from 1 to 5, the friction factor is increased by 60% and 49.5 for the first and second arrangement respectively at Re=10000. It has also been revealed in the figures that values of friction factor tend to have slightly changed at higher pitch ratios such as in PR 3, 4, and 5.

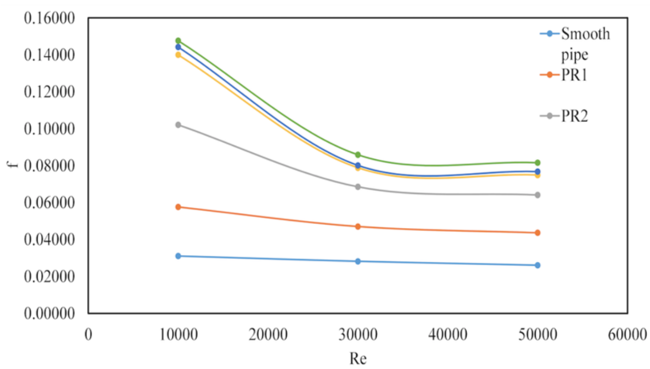

Fig. 10. Variations of friction factor vs. Reynolds Numbers for the first arrangement at PR 1,2,3,4 and 5.

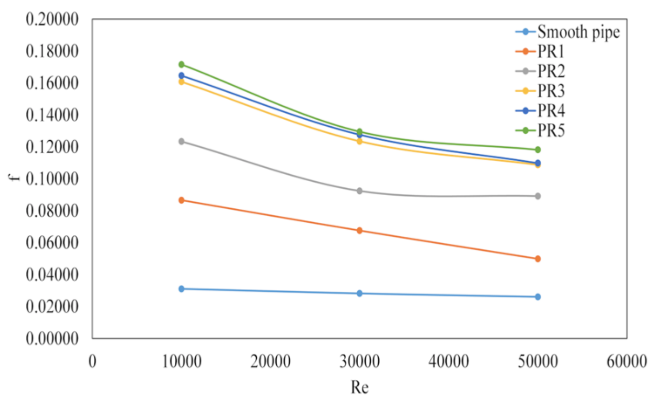

Fig. 11. Variations of friction factor vs. Reynolds Numbers for the second arrangement at PR 1, 2, 3, 4 and 5.

D. Evaluation of Thermal Performance

The thermal enhancement factor (ɳ) can be defined as the ratio of the heat transfer coefficient of a baffled pipe h to a smooth pipe h_o. Both Fig. 12 and Fig. 13, show the variation of (ɳ) versus *Re* for different arrangements. At the same value of *Re*, the thermal enhancement factor changes with the variation of PR, in which it increases with a higher distance between the baffles for both arrangements. For example, the maximum enhancement factor is the record between (2 and 2.25) for PR=5 at Re=10000. On the other hand, it can be seen that the values of the thermal enhancement factor are always higher than one which proves more advantages can be utilized by using a baffled pipe than a smooth pipe. It is also noted that (ɳ) tends to decrease with higher values of *Re* for the first arrangement Fig. 12, while for the second arrangement it decreases at Re=30000 and then goes up at Re=50000 for PR (3, 4 and 5), Fig. 13.

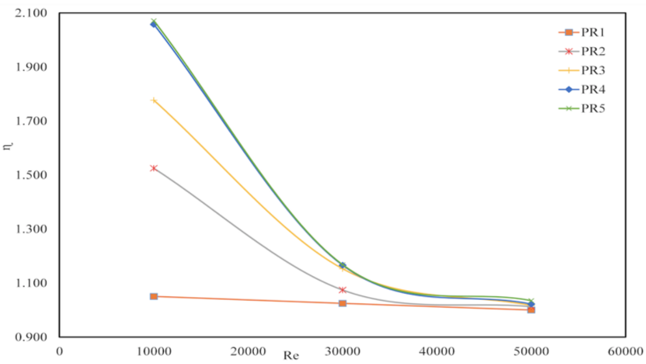

Fig. 12. Variations of Thermal Performance vs. Reynolds Numbers for the first arrangement at PR 1, 2, 3, 4 and 5.

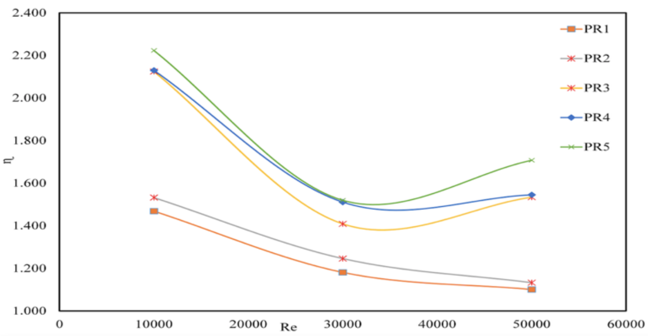

Fig. 13. Variations of Thermal Performance vs. Reynolds Numbers for the second arrangement at PR 1, 2, 3, 4 and 5.

Fig. 14 shows the contour of the temperature of the smooth pipe and baffled pipe (arrangement 1 and 2) at the same Re=10000 and PR (1,3,5). Fig. 14 indicates that the thermal growth boundary layer of arrangement (1) is greater than arrangement (2) which gives more heat transfer rate and then Nusselt number Nu.

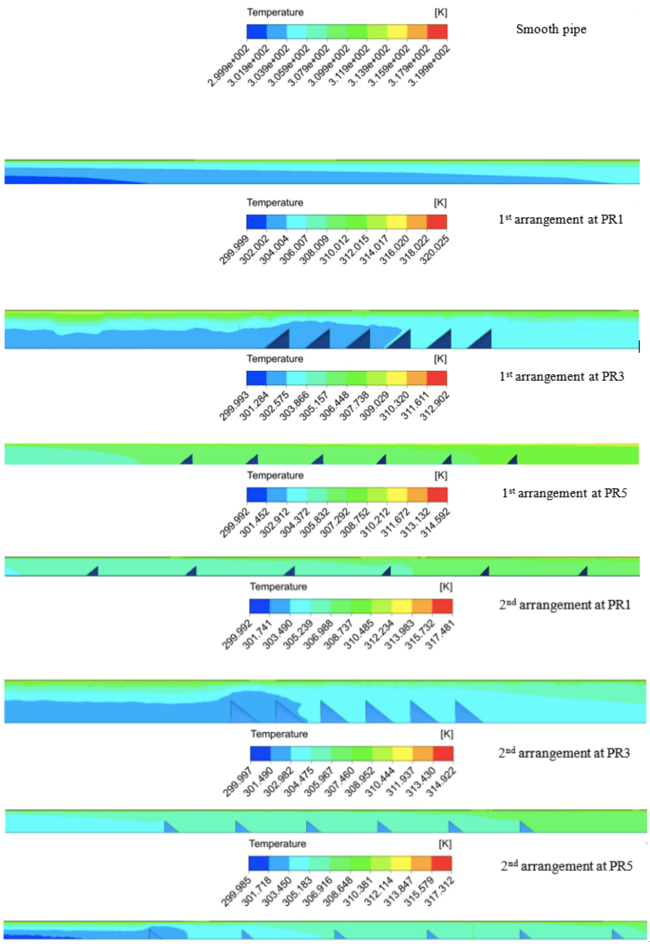

Fig. 14. Temperature distribution contour comparison of smooth pipe with baffled pipe at (Re=10000).

fluid is flowing through the pipe at Reynolds number which variates from 10000 to 50000. For all Re numbers used, the heat transfer rate of baffled case was higher than that in no-baffle case, which demonstrate the efficiency of the baffled pipe. Considerable advantages on heat transfer characteristics have been shown for using the baffles which guarantee a great mixing of flow and then increase the swirling of flow inside the pipe. Moreover, the arrangement and distance between the baffles have a significant impact on the separation of boundary layer and recirculation in the area behind the baffles. This leads to higher value of Nusselt number. Also, it increased with increasing PR, in which the best values are recorded using the second arrangement at PR= 5 at Re=30000. Moreover, Friction value and thermal performance show better values at PR=5 with increasing Re. As a result, the employment of a conical baffled pipe in a very high turbulent flow is shown to be more appropriate and successful in improving the thermal performance of heated pipes.

IV. Conclusion

It has been claimed that inserting a baffle in a pipe, can improve heat transfer. In the present study, a 3D model of circular pipe with six conical baffles fitted in has been considered to run a CFD analysis of a characteristic of heat transfer and turbulent flow. The new design was studied under different pitch ratios and different Re. The working

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