

Research Article

CFD Analysis of the Thermal Hydraulic Performance of a Heat Exchanger Tube Utilizing a Combination of Different Inserts

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Received 22 December 2024

Revised 24 February 2025

Accepted 5 May 2025

Abstract:

This inserts embedded heat exchanger tube is numerically investigated at a wide range of Reynolds number (Re) i.e. 4000-22000. The study examines the geometrical parameters of inserts, such as pitch, peg form length, and circular shaped inserts diameter. The smooth tube has a length of $L = 1500$ mm, an inner diameter of $D_{in} = 37$ mm, and an outer diameter of $D_{out} = 42$ mm. This model has been designed with multiple variations in ring diameters ($d = 10, 12$ and 14 mm) and peg leg lengths ($l_p = 5, 6$ and 7 mm), maintaining a constant arc, resulting in a seamless design of a smooth tube, ultimately referred to as the round peg-shaped HET. The width of the combined shaped inserts is represented as $w_p = 37$ mm, and the hydraulic diameter of the tube is $D_h = 0.037$ m, with pitch variations of round peg-shaped inserts $P = 70, 90, 110$ and 130 mm. The study displays the results of all modifications in these parameters. The CFD analysis of the RNG $k-\epsilon$ model, shows that heat transfer rate increases with Reynolds number. Nusselt number increases with pitch to hydraulic diameter ratio ($P = 70$ and 90 mm) and roughness variation. The highest Nusselt number is 209.3 for round peg leg HET. The friction factor is decreased with increasing Nusselt number and the lowest friction factor is achieved at 130 mm and $d = 14$ mm. These variations are pretended to enhance the thermal performance, and observed 329.53% increment of heat transfer as compared to smooth tube, and found maximum thermal performance of 2.2 at $Re = 22000$, $d = 12$ mm, $l_p = 6$ mm) and $p = 90$ mm.

Keywords: CFD, Heat transfer, Nusselt Number, Friction factor, THP

1. Introduction

The effect of thermal conduction and convection phase shifts on heat flow capacity and thermal performance in cooling and heating applications is investigated in this work. It describes a thermal system heat exchanger employing several insert geometries and characteristics and reviews a number of research theories. Different results were obtained by adjusting insert geometries and settings in the experimental work. The goal of the study is to increase thermal performance in cooling and heating applications and prolong the system's lifespan. The study employs CFD to resolve the governing equation, namely the energy equation, within the geometric model of a circular tube featuring peg-shaped ring inserts, and also serves to validate experimental and numerical research studies. CFD managed both laminar and turbulent flows utilising a turbulent model with the reference of Sheikholeslami and Ganji [1] utilized a double pipe heat exchanger with perforated tabulators, observing a smaller pressure loss compared to the typical tabulator due to decrement in temperature increases in Re and PR cause a drop in thermal performance, which is closely correlated with open area ratio in perforated turbulators. These turbulators also have lower temperature gradients.

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Mohsen et al. [2], used Swirl flow devices were used in the study as a passive method, and the results showed that these devices improve by altering the tube surface's boundary layer and creating a swirl in the bulk flow, convective heat transfer is achieved. Sheikholeslami. et al. [3], The study discovered that as nanofluid concentration increases, the Nusselt number—which is dependent on nanoparticle volume fraction, Rayleigh number, and Reynolds number—decreases. Sheikholeslami et al. [4], an investigation into a discontinuous helical turbulator water-to-air heat exchanger revealed that while the open area ratio improved thermal performance, the pitch ratio and friction factor decreased.

Table 1: List of Reviewers.

Citations	Geometry of the inserts	Observations
[5]	Double pipe inserts with twisted tape.	The data indicates the rate of transfer of heat in counter flow twisted tapes increases with Re in the range 250 - 1000. The maximum performance is noted at Re = 1000, where there is a 3% fraction at the inner and outer tubes.
[6]	Twisted quad channel	The findings indicated a $\pm 2\%$, has been noted that both friction factors and the rate of heat transmission have increased. Varied in ranges of $\pm 5\%$ with respect to practical usage.
[7]	Twisted tape	The findings showed a 42% of heat transfer and 235% reduction in pressure from the modification of twisted tape inserts.
[8]	V-cut on twisted tape	The results are found between smooth tube and twisted tape tube and showed the maximum variation of 10%. The paper further displays the fluctuations in the friction factors and heat transfer rate.
[9]	Twisted tape in double tube	In the results found the Nusselt number increased by 22% when alumina particles were added to water, which raised the inserts and friction factors, heat transfer rate, and pressure drop.
[10]	Conical HET	The findings demonstrate that as nanoparticle diameter decreases, Nusselt number and friction factors increase at Reynolds number.
[11]	Perforated conical inserts	The investigation shows that the greatest thermal efficiency is 1.241 at Re = 4000, d/D = 0.1, D2/D1 = 0.6, and N = 10, with the heat transfer rate decreasing from N = 4 to 10.
[12]	Perforated conical ring	Under certain conditions, the 1.10 model performs optimally hydraulically, outperforming conventional tubes by 278.2% in heat transfer rate at PCR N = 4. With varying N values, the Nusselt number also rises by 278.2%, 231.5%, 196.1% and 159.7%.
[13]	Twisted tape with rectangular cut.	According to the study, cut ratio has a major effect on pressure drop and heat transfer, with single-cut twisted tape showing thermal performance values of 1.24 to 1.64.
[14]	Single tube	Smooth tube is working as wire coil at low Reynolds number and the smooth tube with a wire coil installed, the heat transmission rate increased by 200%. The findings indicate that at smaller construction ratios, the maximum fluctuation in heat flux led to an increase in the heat transfer rate.
[15]	Perforated votex tube	The higher heat transfer rate is obtained 1.65 for TEF with PCR at PI = 16 %, P/P _a = 2 and Re = 3000.
[16]	Plate-fin heat exchangers with serrated fin	The entire heat flow rate, total annual cost, and number of entropy producing units are determined by optimizing PFHEs by taking mass flow rate and available space into account.

Table 1: List of Reviewers. (continuous)

Citations	Geometry of the inserts	Observations
[17]	Triangular ribbed channel with longitudinal vortex generator	According to the study, the heat transfer rate and friction loss along a smooth wall channel are significantly impacted by the combination of rib turbulators and WVGs.
[18]	Microchannels enhanced by porous materials	The study shows that rarefied porous inserts can effectively raise the HTP in flow passages, especially when $0.001 < Kn < 0.1$ in the slip-flow regime.
[19]	Helical baffled heat exchangers	The analysis demonstrates that when each scheme's mass flow rate increases, so do the shell side heat transfer coefficient and pressure drop.
[20]	Twisted tape	The twisted tape heat exchanger tube improved $\pm 8\%$ heat transfer, $\pm 5\%$ friction factor, $\pm 8\%$ Thermal efficiency with respect to smooth tube.
[21]	Twisted HET tube	When compared to smooth tubes, twisted tape tubes have a greater maximum heat transmission rate (27.3 - 90.5%).
[22]	Conical tube	The maximum THPP is observed 0.92 at $PR = 4$, $N = 8$ and $Re = 4000$ and heat transfer rate increased 137% as compared to smooth tube.

S.K. Gugulothu (2022) [23] calculated Nu, TPE, and fr of ribbed tubes with twisted tape inserts using CFD technology. Full-length TT showed better performance, higher heat transfer, and larger fr compared to half-length TT. Lal Kundan and M. B. Darshan (2022) [24] investigated TiO₂-H₂O nanofluid in a horizontal DTCF heat exchanger with or without TT inserts, finding higher HTC in TiO₂-H₂O nanofluid (0.05% by vol.) than in the base fluid (4-6%). According to Kumar and Sarviya's 2023 study, the Nusselt number and friction factor increased by 1.48 when rectangular incisions were removed from turbulent zones in a perforated rectangular cut twisted tape heat exchanger tube [25]. According to the reviewers, there are many methods like passive, active, and compound methods are used for increasing the thermal hydraulic performance of HET tube. Using Ansys as a tool for modelling and simulation, to create a heat exchanger tube with or without inserts according to specifications. There are the main motives of the study are validating the model of the heat exchanger tube that was built by comparing it to the standard correlations of the smooth tube. To investigate the altered behaviour of fluid flow observed within the heat exchanger tube as a consequence of peg and ball-shaped inserts having been inserted. In order to investigate the impact that various geometrical. Flow characteristics of peg and ball-shaped inserts have on the heat transfer, friction factor, and thermo hydraulic performance of heat exchangers, the fourth target is to conduct research. The study explores the role of Ansys in designing circulation tubes, analyzes thermal performance of Peg and circular shaped inserts-shaped heat exchanger tubes, and investigates the impact of Reynolds number and pitch space on design models.

2. Methodology of the Design Model

This section expresses the geometry of the heat exchanger tube with inserts of round peg shaped inserts and after that this design is analysis employing computational fluid dynamics (CFD) software for the numerical, thermal performance study and simulation of the heat exchanger tube. The geometric modelling of the circular heat exchanger tube with different shape of combination of round peg inserted inserts was generated in Ansys software in Fig. 3 (a and b) and the geometric model contains a single tube with combination of peg and ring inserts which have constant double arc in different length, diameters and constant thickness $t_p = 2.7$ mm which are complete designed. The length of the smooth tube $L = 1500$ mm, the inner diameter $D_{in} = 37$ mm and the Outer Diameter $D_{out} = 42$ mm. We have designed this model in such a manner that it has number of variations in the circular shaped inserts diameters ($d = 10, 12$ and 14 mm) and length of Peg leg ($l_p = 5, 6$ and 7 mm) with constant arc and all shapes are attached together to make a perfect design of smooth tube which is finally known as round Peg shaped HET. With copper inserts that are quantitatively examined to identify their impact, the Reynold number Re impacts tube working fluid with insert pitches ranging from 70 to 130 mm and hydraulic diameters $P/D_h = 1.89, 2.43, 2.97$ and 3.51 .

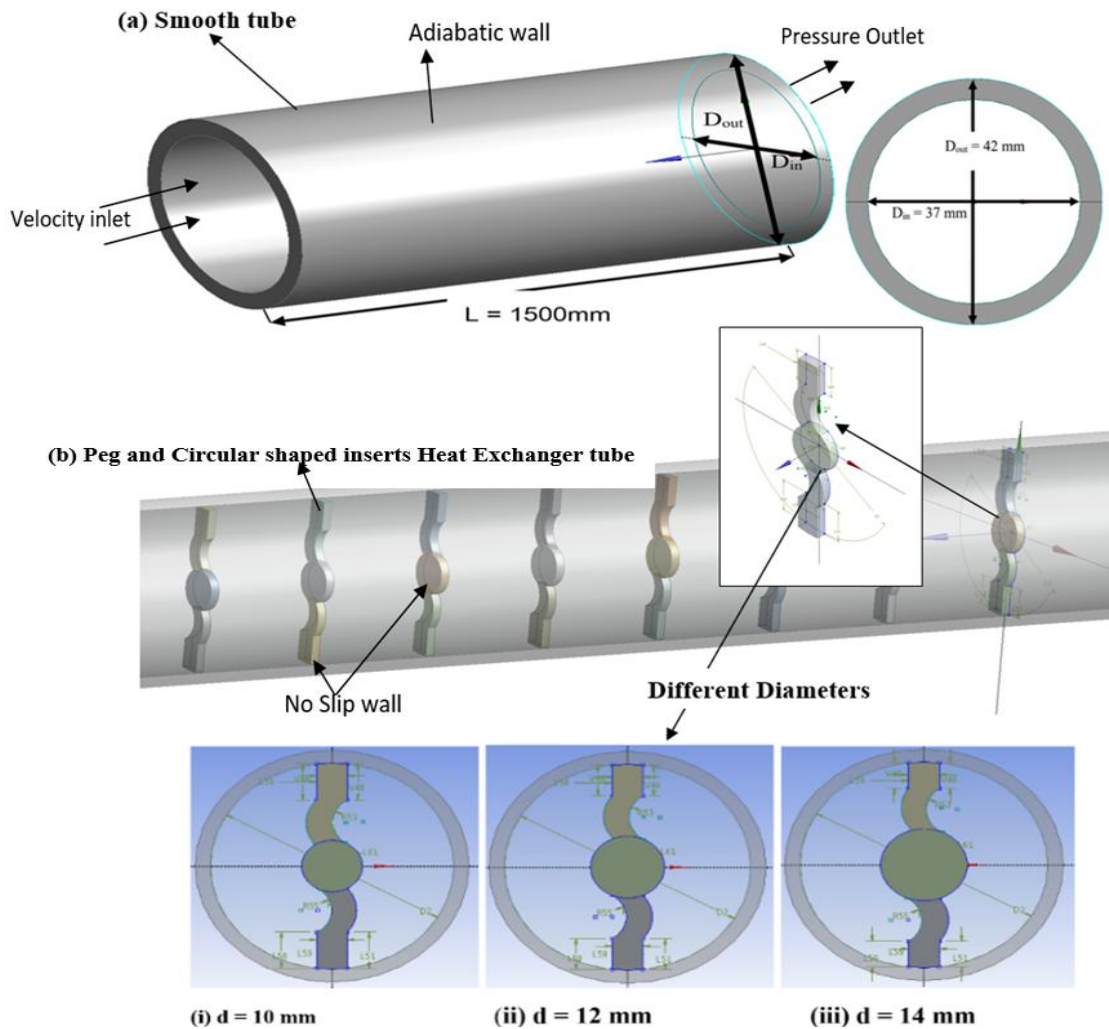


Fig. 1. (a and b) Geometry of Smooth tube with round Peg and ring shaped heat exchanger tube.

2.1 Mathematical Data

2.1.1 Governing Equation

In order to create the governing equation for a flow system that includes turbulent, laminar, and laminar to turbulent flow, the section discusses the conservation equation for laminar and turbulent flow scenarios. This section examines the governing equations for flow systems, emphasising turbulent, laminar, and transitional states. The K- ϵ model is employed to tackle residual stress concerns adjacent to the insert boundary. A computational examination of tubes with inserts demonstrates fluctuations in turbulent viscosity and energy efficiency. The K- ϵ model addresses residual stress near roughness boundaries, while turbulence models solve fluid-related problems, improving system characteristics like pressure drop, temperature, kinetic performance, and thermal performance. Ansys workbench solves continuity equations, while momentum equations explain acceleration changes. RNG k- ϵ model based on its suitability for the characteristics of our problem and its widely recognized accuracy in predicting turbulent flow phenomena as suggested in [1, 2, 15, 30].

2.1.2 Equation of continuity used in HET

The equation often articulates the relationships of velocity in various directions within a fluid dynamics model. The continuity equation is resolved by Ansys Workbench based on the control volume of the system, as seen in Equation 1.

$$\frac{\partial \bar{u}}{\partial x} + \frac{\partial \bar{v}}{\partial y} + \frac{\partial \bar{w}}{\partial z} = 0 \quad (1)$$

2.1.3 Equation of momentum of HET

These equations explained average change in acceleration with w.r to total length of tube and average acceleration is submission of fluid particles in turbulent shape character.

X- Momentum equations:

$$\left(\bar{u} \frac{\partial \bar{u}}{\partial x} + \bar{v} \frac{\partial \bar{u}}{\partial y} + \bar{w} \frac{\partial \bar{u}}{\partial z} \right) = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \left(\frac{\partial^2 \bar{u}}{\partial x^2} + \frac{\partial^2 \bar{u}}{\partial y^2} + \frac{\partial^2 \bar{u}}{\partial z^2} \right) \quad (2)$$

Y- Momentum equation:

$$\left(\bar{u} \frac{\partial \bar{v}}{\partial x} + \bar{v} \frac{\partial \bar{v}}{\partial y} + \bar{w} \frac{\partial \bar{v}}{\partial z} \right) = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \left(\frac{\partial^2 \bar{v}}{\partial x^2} + \frac{\partial^2 \bar{v}}{\partial y^2} + \frac{\partial^2 \bar{v}}{\partial z^2} \right) \quad (3)$$

Z – Momentum equations:

$$\left(\bar{u} \frac{\partial \bar{w}}{\partial x} + \bar{v} \frac{\partial \bar{w}}{\partial y} + \bar{w} \frac{\partial \bar{w}}{\partial z} \right) = -\frac{1}{\rho} \frac{\partial p}{\partial z} + \nu \left(\frac{\partial^2 \bar{w}}{\partial x^2} + \frac{\partial^2 \bar{w}}{\partial y^2} + \frac{\partial^2 \bar{w}}{\partial z^2} \right) \quad (4)$$

2.1.4 Equation of energy for HET

This equation observed all performance of energy in design system which is calculated by Ansys software under all applied parameters of Peg and ring shaped inserts and finally observed best energy performance with energy equation.

$$\bar{u} \frac{\partial t}{\partial x} + \bar{v} \frac{\partial t}{\partial y} + \bar{w} \frac{\partial t}{\partial z} = \alpha \left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2} \right) \quad (5)$$

2.2 Properties of material

2.2.1 Physical properties of inserted inserts

In this chapter peg and ring shaped inserts is inserted in smooth tube were used of copper solid material due to better thermo-physical characteristics, high thermal conductivity, machinability and low cost. Following for copper are used in numerical calculation as show in table: 2, 3 and 4.

Table 2: Thermo-physical properties of round Peg shaped inserts

Properties	Value
Density, ρ	8978
Specific heat	381
Thermal conductivity k	387.6
Electrical conductivity	5.8e+07
Magnetic Permeability	1.257e-06

Table 3: Thermo-physical properties of smooth tube.

Properties	Value
Density, ρ	8978
Specific heat	381
Thermal conductivity, k	387.6
Electrical conductivity	5.8e+07
Magnetic Permeability	1.257e-06

Table 4: Thermal physical properties of fluid.

Properties	Value
Density (ρ)	1.225
Specific heat	1006.4
Viscosity (μ)	1.7894e-05
Thermal conductivity	0.707

2.3 Boundary Conditions

The boundary condition is constrained for the circular tube with peg-shaped inserts. This numerical simulation involves a tube equipped with coupled peg-shaped inserts under a flowing fluid, where a no-slip requirement is disregarded with a 5% allowance for all solid walls. The velocity is characterised by Reynolds numbers of $Re = 4000, 7000, 10000, 13000, 16000, 19000,$ and 22000 . The boundary condition comprises the following terms: The inlet is characterised by velocity, while the outlet is defined by pressure. The absorber features an inner source tube and an upper half tube with heat flux, along with an upper side tube exhibiting heat flux and a cutting cone, all adhering to a no-slip condition designated during meshing. The inlet temperature is set at 300 K.

2.4 Mesh Generation

Meshing involves balancing automation, validity, accuracy, and mathematical foundation in a discrete tube representation, using 3-D uniform meshing techniques and regular tetrahedron meshing to smooth nodes and restore boundaries. Grid test is a condition of best mesh as well as results when both are independence to each other. In presented design model mesh is applied at 0.003 m face elements where 905656 number of elements found at these elements, the result is start maximally independence of other variations in meshing. In order to achieve smooth transition, virtual topology, and an element size of 0.003 m, the design model for a circular tube of a combining peg shaped with circular ball inserts used.

Table 5: Grid independence test for tube with roughness.

Number of elements	Nusselt Numbers
351326	58.65665
453969	80.35562
905656	25.56650
1026256	26.02656
1258946	39.56510

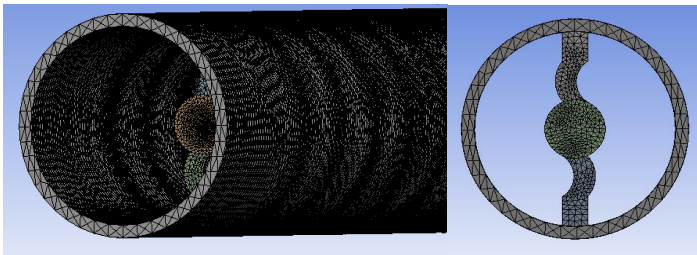


Fig. 2. Meshing of heat exchanger tube with Peg and ring shaped inserts.

3. Results and discussion

Inserts create turbulence in fluid, increasing the Nusselt Number and causing temperature rise due to increased convective heat transfer. Inserts lowers friction coefficient, lowering temperature. Using pitch space shortens equipment lifespan but reduces effectiveness. The tube as heat exchanger consists inserts as vortex generator as inserts which was made of copper to give maximum efficiency for peg type tube and various analyses were performed by varying relative pitch space to hydraulic ratio $p/D_h = 1.89, 2.43, 2.97$ and 3.51 pipe diameter size, variation in temperature to create the warm condition for air due to this process the finite temperature difference was used to create the good condition for better heat transfer enhancement analysis in the tube.

3.1 Evaluating the Dittus-Boelter Equation and the Smooth Tube in Parallel

Figure 3 displays a comparison of the Dittus-Boelter equation correlation and the Nusselt number and smoothness of the Dittus-Boelter equations are crucial factors to consider ($Nu = 0.023 \times Re^{0.8} \times Pr^{0.4}$) for heat exchangers. Figure 5 illustrates the updated single tube heat-transfer enhancement study based on the Dittus-Boelter correlation.

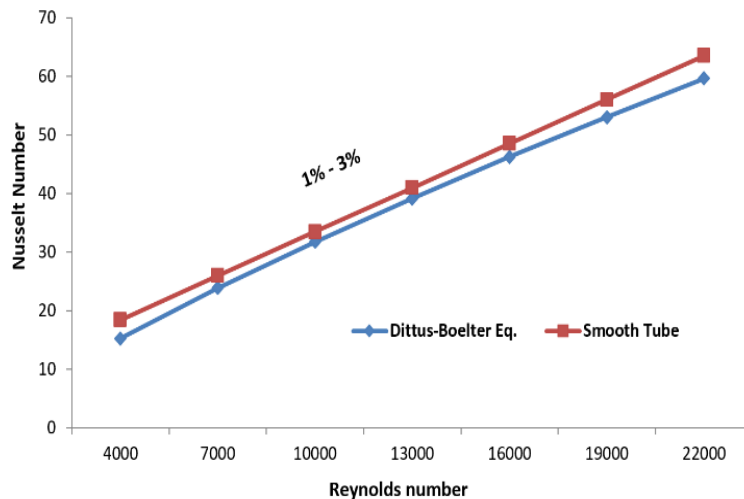


Fig. 3. Variation of Nusselt number with increasing flow velocity.

3.2 Velocity Effects of Fluid around Peg Shaped inserts in HET

Useful values for the velocity pattern surrounding the heat exchanger tube inserts with respect to $Re = 4000$ – 22000 are shown in Figure 4. The graph indicates that a maximum velocity of 11.32 m/s is attained at $Re = 22000$ when pitch separation between inserts is 90 mm. The combined inserts work in two sections - first is Peg shaped that collects the fluid and increases the velocity when it comes in arc contact and second part is round shaped which is used to give continuity of fluid in circulation zone and also reduces factors resistance in turbulence fluid. In fig. 4, the heat transfer rate is generated lower than figure 5 because of less recirculation of fluid flow inside the heat exchanger tube at $d = 10$ mm.

Fig. 5 explains phenomena of velocity after striking on inserts and inlet velocity of fluid is 8.6 m/s that increased and found maximum velocity of HET 13.7 m/s because of length of peg leg ($l_p = 6$ mm) and arc ($R_{p1} = 5$ mm and $R_{p2} = 6.5$ mm) of peg is suitable for creating a reattachment zone of fluid and this reattachment zone contain maximum energy of transfer fluid of design model at Reynolds number ($Re = 22000$). The velocity increased in compare of round peg diameter $d = 10$ mm because increased in diameter starts to creates a turbulent path for fluid and this turbulent fluid is recalculating with maximum energy when it comes in contact of peg shaped inserts. The integrated inserts functions in two sections: the first is peg-shaped, designed to collect fluid and enhance velocity upon arc contact; the second is ring-shaped, which maintains fluid continuity in the recirculation zone and diminishes resistance factors in turbulent fluid, as well as reducing the heat transfer rate and residual stress in the heat exchanger tube. Fig. 6. Observed decrement in velocity at $d = 14$ mm and $l_p = 5$ mm as compared to $d = 10$, $l_p = 7$ mm and $d = 12$ mm, $l_p = 14$ mm. Figure 7 illustrates temperature variations in a tube due to friction or stress created by fluid particles when moving on an inserts surface. The peg-shaped inserts design maintains a constant maximum temperature, with the maximum temperature altering at $T_{max} = 346.6$ K at various points. This variation increases with increasing Nusselt number. The turbulent kinetic energy is a generation of energy with the help of movement of turbulent fluid and formation of eddies per unit mass of fluid after contacting Peg shape inserts. The eddies of fluid are swirling of convective fluid in turbulent regime due to this kinetic energy is defined in form of turbulent. In figure 8 observed maximum TKE = 16.11456 j/kg at Reynolds number 22000 , $P = 90$ mm, and $d = 12$ mm.

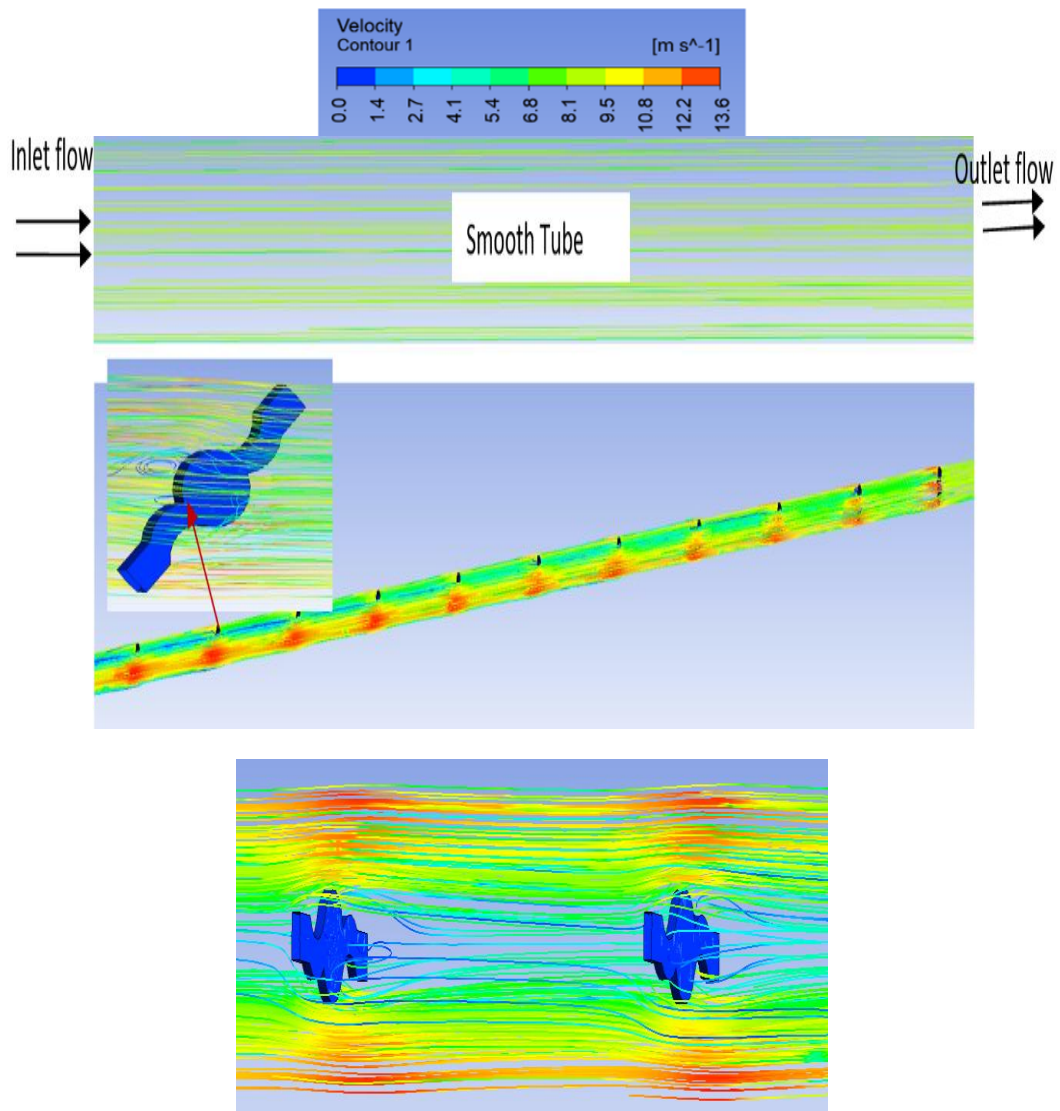


Fig. 4. Variations in Velocity of fluid after striking on combined inserts at ring's diameter $d = 10$ mm and length of Peg leg shaped $l_p = 7$ mm.

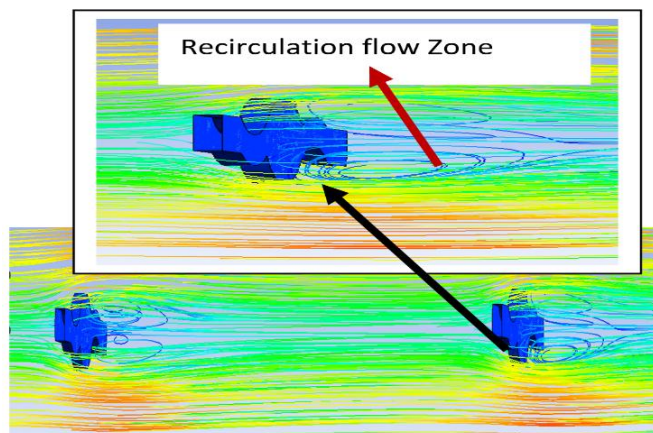


Fig. 5. Variations of velocity of fluid in HET at $d = 12$ mm and $l_p = 6$ mm.

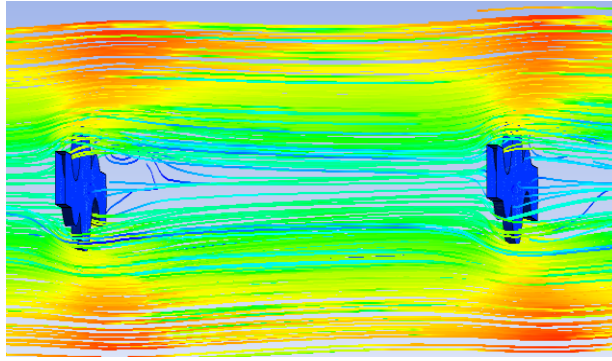


Fig. 6. Velocity contour at $d = 14$ mm, $l_p = 5$ mm.

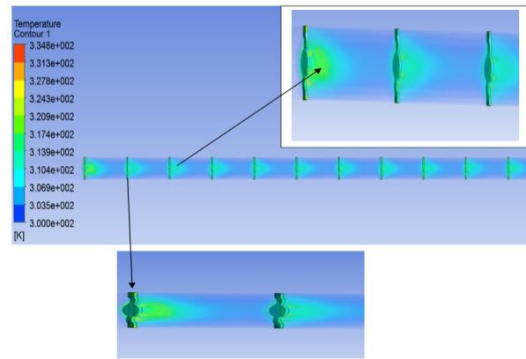


Fig. 7. Variations of contour of temperature at $P = 90$ mm, $d = 12$ mm, and $l_p = 6$ mm.

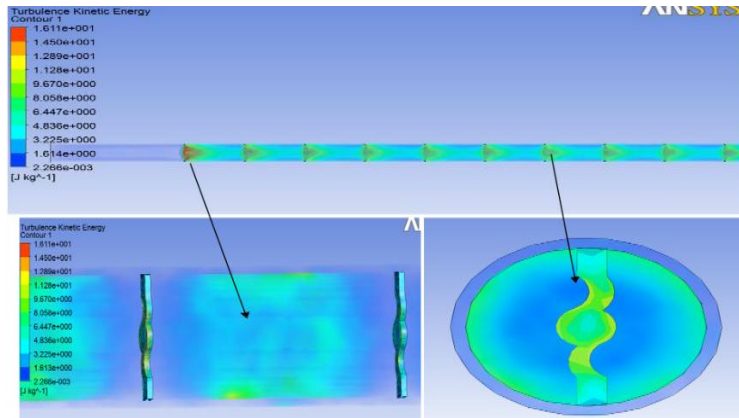


Fig. 8. Variations of turbulent kinetic energy of HET with Peg shaped inserts.

3.3. Effects of Pitch Shape and inserts on Heat Transfer Rate Characteristics

The figure 9 explains the variation between heat transfer rate as Nusselt number (Nu) of inserts with respect to the variation in Reynolds numbers $Re = 4000, 7000, 10000, 13000, 16000, 19000$ and 22000 variations in different Pitch inserts ratio $P/D_h = 1.89, 2.43, 2.97$ and 3.51 and constant roundness peg diameter $d = 10$ mm. The figure 10 provides an explanation of the variation in heat transfer rate as measured by the Nusselt number (Nu) of roughness in relation to the variation in Reynolds numbers $Re = 4000, 7000, 10000, 13000, 16000, 19000$ and 22000 . Additionally, the figure illustrates the variation in different Pitch roughness ratios $P/D_h = 1.89, 2.43, 2.97$ and 3.51 , as well as the constant roundness peg diameter $d = 10$ mm. Furthermore, the figure demonstrates that the maximum Nusselt number $Nu = 182.9$ of roughness was obtained at $p/D_h = 2.43$ and 2.9 times is greater than the result of a smooth tube at $Re = 22000$.

The Figs 10-11 explain phenomena of velocity after striking on inserts and inlet velocity of fluid increased and found maximum velocity of HET because of length of peg leg ($l_p = 6$ mm) and arc ($R_{p1} = 5$ mm and $R_{p2} = 6.5$ mm) of peg is suitable for creating a reattachment zone of fluid and this reattachment zone contain maximum energy of transfer fluid of design model at Reynolds number ($Re = 22000$). The figures 10-11 explain the variation between heat transfer rate as Nusselt number (Nu) of inserts. The Nusselt number diminishes with an increase in pitch space due to the absence of streamline and rotational fluid flow between the two rough surfaces. Consequently, heat transfer escalates with the Reynolds number, not with the pitch and pitch-to-hydraulic diameter ratio (P/D_h). In Fig. 10. The maximum heat transfer occurs when the diameter of the round peg roughness is 12 mm, yielding maximum Nusselt numbers of 151.01, 210.1, 184.6, 124.2 and 63.6 at P/D_h ratios of 1.89, 2.43, 2.93, 3.51 and for a smooth tube, respectively, with a Reynolds number of $Re = 22000$. The highest Nusselt number for inserts is $Nu = 176.2523396$ at $P/D_h = 2.43$, ($P = 90$ mm), respectively. In comparison to plain tube, these results showed increments of 2.7 times. Figures 12 compare the friction factors of a peg-shaped inserts tube with Reynolds number $Re = 4000-22000$, focusing on the pitch to hydraulic diameter ratio $P/D_h = 1.89 - 3.51$. The minimal friction factor of the inserts tube is $Fr = 0.023$ at Reynolds number $Re = 22000$, $d = 12$ mm, and $l_p = 6$ mm. The results are compared to the Plain Tube $Fr = 0.0078$, increasing the smooth tube by 2.9, 3.5, 3.3 and 2.6 times.

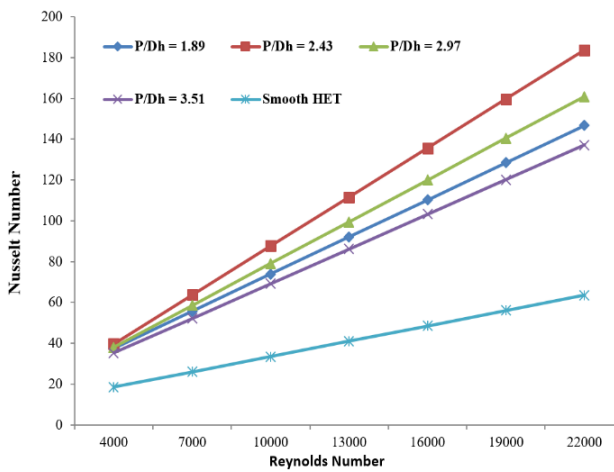


Fig. 9. Variations in heat transfer rate at $d = 10$ mm, $l_p = 7$ mm.

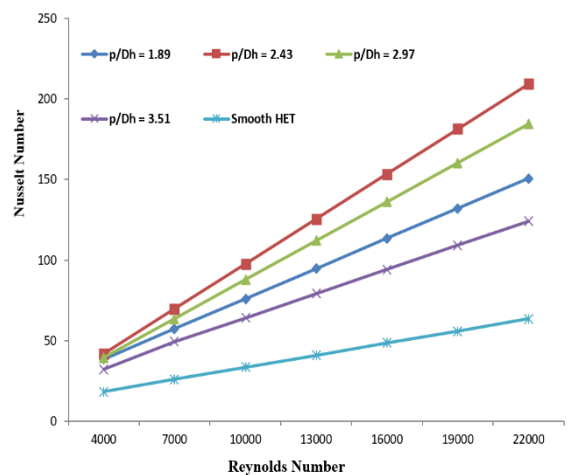


Fig. 10. Variations of Nusselt number and Reynolds number at $d = 12$ mm, $l_p = 6$ mm.

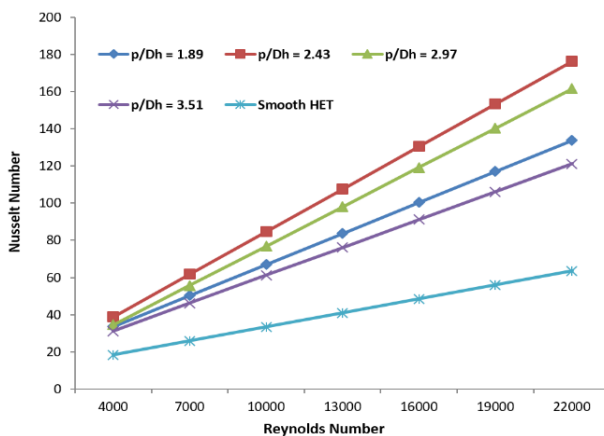


Fig. 11. Variations of Nusselt number with Reynolds number at $d = 14$ mm, $l_p = 5$ mm.

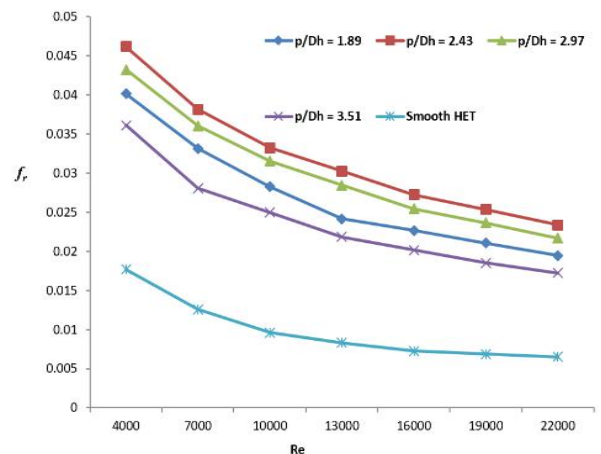


Fig. 12. Variations of Friction factors w.r.to Reynolds number at $d = 12$ mm, $l_p = 6$ mm.

4. Thermal Hydraulic Performance

In fig. 13. The study explains the THP of a heat exchanger tube with circular Peg-shaped inserts, highlighting that factors like pitch gap variations, ball diameters, and tube insert types can enhance thermal hydraulic performance, providing a basis for evaluating performance under different conditions. The results that are analysed under various conditions constitute the basis of the thermal hydraulic performance. The total functioning of the tube under all conditions, including fluid flow, friction factors, pressure drop, temperature differential, and turbulent eddies production, is known as thermal hydraulic performance. The higher thermal performance is obtained $\eta = 2.2$ when inserts is used without hole in ball shaped tube and at $Re = 22000$, $P/D_h = 2.4$, $d = 12\text{mm}$, $l_p = 6\text{ mm}$, $D_h = 37\text{ mm}$ and Pitch space between the inserts is $P = 90\text{ mm}$. The minimum hydraulic performance is obtained as 1.44.

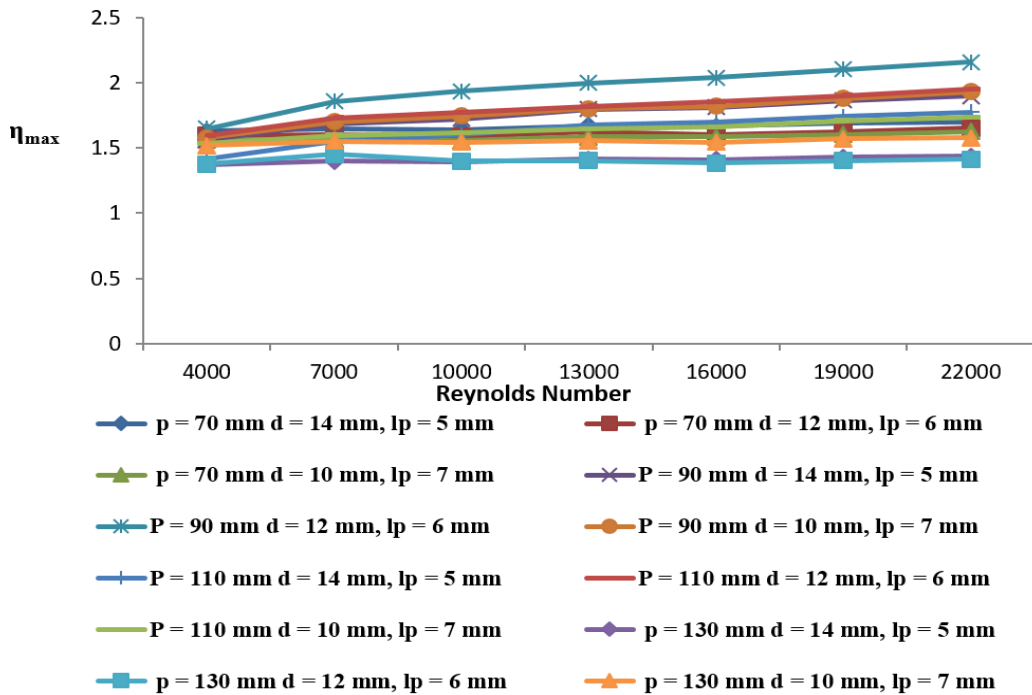


Fig. 13. Maximum thermal hydraulic ($\eta_{\max}$) of HET with inserted Peg shaped inserts.
 $d = 10\text{ mm}$ and length of Peg leg shaped $l_p = 7\text{ mm}$.

5. Conclusion

The design model's results depend on its parameters, and recent research has focused on understanding the focused on inserts $P = 70, 90, 110$ and 130 mm and Reynolds number $Re = 4000, 7000, 10000, 13000, 16000, 19000$ and 22000 . It also looked at the effects of pitch space and Reynolds number on heat transfer and duct passage. These variations are pretended to enhance maximum thermal energy from fluid and its environment with minimum loss of energy and all maximum conditions for maximum performance of HET is conclude below.

The main conclusions can be explained as follows:

- CFD analysis of design model is valuable at k- ϵ model and this model is active in RNG position for complete design of model.
- From all results, it is concluded that heat transfer rate increased with increase in Reynolds number as opposed to friction factors.
- The Nusselt number is increased for all variations in the pitch to hydraulic diameter ratio (P/D_h) and Nusselt numbers is increased 1.91 - 3.29 times of smooth tube with decreasing in friction factor from 2.34 to 3.59 of smooth tube at $Re = 22000$.
- When the Reynolds number and pitch spacing between the inserts ($P = 70$ and 90 mm) vary, the Nusselt number rises.
- The highest Nusselt number is 209.3 for ($d = 12\text{ mm}$ of roundness of peg and $l_p = 6\text{ mm}$ of peg leg) HET.

- The lowest friction factor $f=1.91$ was obtained at $P = 130$ mm, $d = 14$ mm $l_p = 7$ mm.
- Highest hydraulic thermal performance was obtained $\eta = 2.2$ at $Re = 22000$, $P = 90$ mm, $P/D_h = 2.43$, $D_h = 0.037$ m, $d = 12$ mm, $l_p = 6$ mm and round peg shaped inserts.

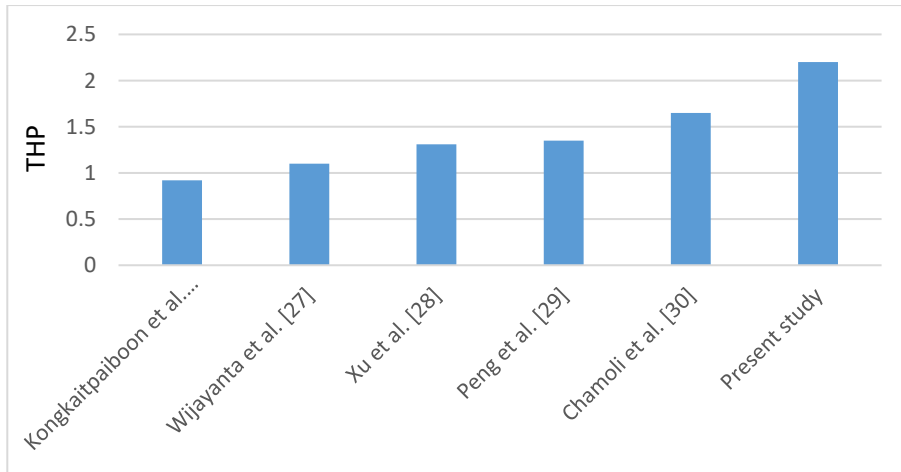


Fig. 14. Thermal Hydraulic performance (THP) results compared with published literature.

Future scope

There are following valuable points:

- Raise the rate of heat transmission by altering the pitch space.
- To enhance thermal performance, modify the inserts parameters of the Peg-shaped structure.
- Use ball-shaped inserts variations to lower friction factors.
- The perforated Peg shaped inserts can have improved the rate of energy transfer without loss of energy of working model.

Nomenclature

L	Total length of smooth tube (mm)
D_{in}	Inner diameter of tube (mm)
D_{out}	Outer diameter of tube (mm)
l	Total length of combined roughness (mm)
l_p	Length of leg of peg shaped roughness (mm)
R_{p1}, R_{p2}	Arc of peg shaped roughness (mm)
d	Diameter of ball shaped roughness (mm)
Nu	Nusselt number
Re	Reynolds number
f_r	Friction factor
t	Thickness of combined roughness (mm)
I	Heat flux (W/m^2)
P	Pitch shape (mm)
D_h	Hydraulic diameter (m)
P/D_h	Pitch to hydraulic diameter ratio
v	Velocity of fluid (m/s)
η	Thermal hydraulic performance
α	Angle of attack ($^\circ$)
ρ	Density [$kg\ m^{-3}$]
λ	Thermal conductivity [$W\ m^{-1}\ K^{-1}$]
HET	Heat exchanger tube
THP	Thermal hydraulic performance

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