Thermal Performance Augmentation of Tube Heat Exchanger using Novel Oval Profile Insert

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Abstract

Improvement in heat transfer and fluid flow characteristics of heat exchangers has always been the prime interest of researchers. In the present work, CFD simulations are performed with "FLUENT" module of ANSYS to improve the heat exchanger performance through the passive method. Analysis of novel Oval Profile insert in round tube heat exchanger has been done over Reynolds number range within 3000 to 24000. Non- perforated, as well as non-inclined OP inserts, are used with a constant pitch in the heat exchanger having a diameter of 68.1mm, length of 1.5m and constant heat flux of 1000 W/m2. Furthermore, OP insert cases are investigated when geometrical parameters such as PI = 4%, 6%, 8% within the profile is considered. The introduction of PI results in eddies generation and thus improves heat transfer. Thermal performance (TPF) of 2.214 times over the smooth tube is found for PI = 8% at Re = 3000.Present work TPF comparison with some previous studies is also done to analyze the performance of oval profile insert.

Keywords. Perforation index, heat transfer enhancement, vortex, swirl, mixing of fluid, thermal performance, Nusselt number.

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Nomenclature

A	Area of pipe cross-section, mm^2	L	Length of test section, mm
		u	Velocity of fluid, m/s
CFD	Computational fluid dynamics	PI	Perforation Index
D	Diameter, mm	L_a	Axial length of OP insert, mm
DWT	Delta Wing Tape	L_b	Transverse length of OP insert, mm
d_s	Circular diameter of semicircles of Oval profile, mm	Nu	Nusselt Number
		p	Pressure, Pa
h	heat transfer coefficient,W/m^2-K	P	Pitch, mm
f	Friction factor	Re	Reynolds Number
f_s	friction factor for smooth tube	α	Angle of inclination with respect to transverse axis
k	Turbulent kinetic energy, m^2/s^2	ρ	Density of fluid, kg/m^3
Nu_s	Nusselt number for smooth tube	s	Smooth tube
T	Temperature, K	in	Inlet

Pr	Prandtl Number	out	Outlet
T_{mw}	mean wall temperature	μ	Viscosity, N-s/m^2
Q	Heat transfer rate, W	η	TEF
λ	Thermal conductivity of fluid, W/m-K	mf	mean film
TT	Twisted Tape	m w	wall mean
OP	Oval Profile	f	Fluid
TPF	Thermal Performance Factor	ε	Rate of energy Dissipation, m^2/s^3

1. INTRODUCTION

A heat exchanger is a device that is used to transfer heat from one fluid to another. The heat transfer enhancement methods generally consist of (a) Active method (b) Passive method and (c) combined method [1]. In the Active method, disturbance in fluid flow is provided externally which requires some energy. The passive method is a convenient way to increase the convective heat transfer rate by inserting geometries and obstacles in the path of fluid flow which increases the turbulence and promotes fluid mixing. [2] Numerically studied Plate fin heat exchanger with vortex generator with rectangular wings as vortex generator channels and found that wings width, height, pitch, and attack angle have a noticeable effect on heat transfer and fluid flow characteristics. In a different study [3], inclined projected winglet pair vortex generators are numerically analyzed with protrusions and results are compared with delta winglet type insert, found that the former resulted in 7.1 percent heat transfer augmentation as compared to the latter. In a turbulent flow regime, the influence of twisted tape inserts was numerically investigated, resulting in a 20 to 62 percent increase in heat transfer and a 185 to 245 percent increase in friction factor over plain tube values [4].In a study, numerical analysis of rectangular cut twisted tape inserts with various cut ratios revealed that rectangular cut twisted tape resulted in better fluid mixing, swirl formation near the wall region, and improved heat transfer and thermal performance [5]. Experimental analysis of solid hollow circular disc (with wings of rectangular shape inserted inside the circular tube) is done and perceived thermal performance factors of 1.95 and 3.47 times improved heat transfer over smooth tube values[6]. [7] experimentally studied wire coil inserts with gradually varying width, diverging arrangement, converging arrangement, diverging converging arrangement inside a traverse corrugated tube and found that converging wire coil insert has the least thermal performance while the maximum thermal performance of 1.13 was achieved for wire coil with gradually varying width and drop in augmented entropy generation number was noticed.

Backflow generation and recirculation of fluid stream for proper mixing of fluid are vital for designing of heat exchanger which is taken into account by most of the researchers but some of them resulted in higher friction factor due to blockage provided in the path of fluid flow. Present work proposes a novel insert profile with a diverging section at the primary face where the fluid strikes first and then bring the fluid towards the core region for proper mixing and this process is repeated throughout the heat exchanger length. The proposed novel Oval Profile insert offers less blockage in the path of fluid flow and produces recirculation and fluid mixing with reattachment of fluid streams for enhancing heat transfer and thermal performance.

2. MATERIAL AND METHODS

2.1 Physical model Description

The dimension of the model in the coordinate system is shown in fig.1. [13-17]The test section has a length L=1500 mm, while the inner diameter of the tube is taken as D_{in} = 68 mm and the outer diameter is taken to be D_{out} =71 mm. The insert is of oval profile (OP) with axial length L_a=50 mm and transverse length L_b=20 mm. The Length in the transverse direction is determined

by taking a fraction of the length of the minimum area of a rectangle which can be circumscribed in the circle of diameter equal to the tube internal diameter. In the present investigation the Pitch length, P=140 mm is considered in all the cases. The circular diameter of both semicircles is d_s = 5 mm each which is also the thickness of the oval-shaped insert. The fluid will strike the transverse region of the OP insert with the semicircular transverse region in front-facing direction. Primarily, non-perforated OP inserts are analyzed and later axial perforations of diameter, d_p =4mm are introduced in the OP insert which has values of Perforation index, PI = 4%, 6%, 8%. In this work, the Reynolds number is varied from 3000 to 24000 with corresponding velocities of 0.644 ms^{-1} to 5.154 ms^{-1}. Using ANSYS Fluent setup, the boundary condition for the wall fluid domain is defined by providing a constant wall heat flux condition with no-slip condition for the tube wall, the temperature of the fluid at the inlet section of the fluid domain is taken as T_{in} = 300 K. A constant heat flux, Q=1000 $W.m^{-2}$ is provided on the tube wall.

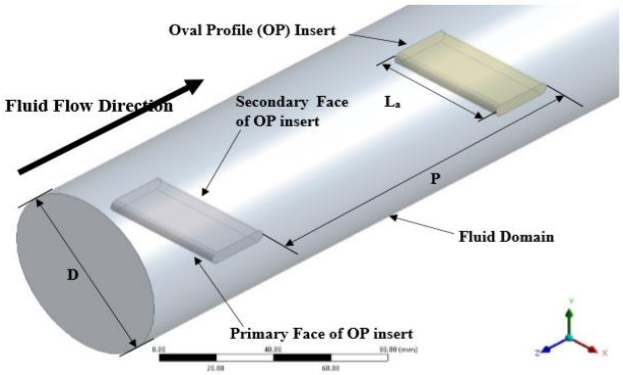

Fig.1 Physical model of OP insert with dimensions in the rectangular cartesian coordinate system

2.2 Governing Equations and Important terminology

Continuity and Momentum equation for the flow and energy equation which governs the temperature distribution of the flow are given as:

(1) Continuity Equation: $\frac{\partial}{\partial x_i}(\rho u_i) = 0$

(2) Momentum Equations: $\frac{\partial}{\partial x_j}(\rho\, u_i u_j) = \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right]$

(3) Energy Equation: $\frac{\partial p c_p u_j T}{\partial x_j} = \frac{\partial}{\partial x_j}\left[k\left(\frac{\partial T}{\partial x_j}\right)\right]$

(4) Reynolds number expression is given by: $Re = \frac{\rho u D}{\mu}$

(5) Nusselt Number expression is given by: $Nu = \frac{hD}{\gamma}$

(6) Friction Factor: $f = \frac{\Delta p}{\left(\frac{L}{D}\right)\left(\frac{\rho u^2}{2}\right)}$

(7) Thermal Performance Factor (TPF): TPF = $\eta = \frac{\frac{Nu}{Nu_S}}{\left(\frac{f}{f_S}\right)^{\left(\frac{1}{3}\right)}}$

(8)PI (Perforation Index): $PI = \frac{\text{Area of holes at a surface or contour}}{\text{Total area of the surface or contour}}$

2.3 Validation of the Numerical Model

To validate the proposed model, CFD results and standard mathematical equation results are compared for the Plain tube. A very small deviation of about ±5% is found for Nu and f values. Furthermore, CFD results are compared with experimental plain tube results of [6], [9], [10], [11], [1] and again smaller deviation is found for Nu as well as f values.

3. RESULTS AND DISCUSSION

Non-perforated cases of OP insert however bring improvement in the rate of heat transfer as well as improved friction factor since the surface is smooth over the oval profile of the insert. The reason behind the higher heat transfer rate is the smooth surface of OP insert. But when Perforations are introduced then the surface smoothness or evenness at either face of OP inserts is reduced and a slight friction factor is increased. However, the introduction of Perforations also enhances the heat transfer since a fraction of fluid striking at the primary face will pass through the holes of the perforations of the inserts and another fraction is then diverged towards the wall of the tube absorbing heat. The incoming fluid from holes is then mixed with the higher heat content fluid and directed towards the wall of tube. The pressure at the back of the secondary face is lower and hence swirl is generated as shown in fig. 3. Heat transfer rate is increased due to proper mixing of fluid and swirl generation. The highest and lowest heat transfer enhancement of

3.296 times and 1.471 times of smooth tube results while the highest and lowest thermal performance of 2.214 times (at Re = 3000) and 1.196 times (at Re = 24000) over smooth tube results are obtained for PI = 8% and 0% respectively. Increment in PI results in an increase in heat transfer rate as well as thermal performance as shown in fig.4. Referring to the contour of outlet temperature as depicted in fig.2, the heat transfer enhancement inside the tube due to perforations can be understood. As the number of perforations increases, the number of cylindrical holes increases, the more fluid is passed through cylindrical holes inside the OP insert and directed towards the wall of tube in combination with fluid guidance due to surface of OP insert, swirl generated and vortex formation behind secondary face of OP insert. These phenomena are responsible for the enhancement of heat transfer due to proper fluid mixing of fluid. The temperature at the outlet face of tube increases with an increase in PI. The pressure inside the tube changes as PI increases as shown in fig.3. Pressure is gradually decreased as PI increases. The pressure behind the OP insert is lower, which is the reason behind the fluid coming at the core region but again due to the shape of OP insert this fluid is directed towards the wall of tube.

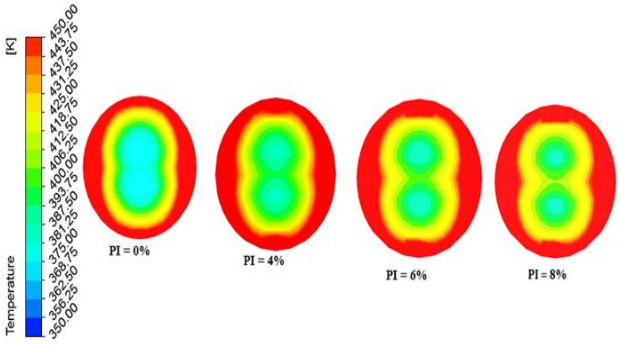

Fig.2Temperature contours for different cases of OP insert at Re =3000

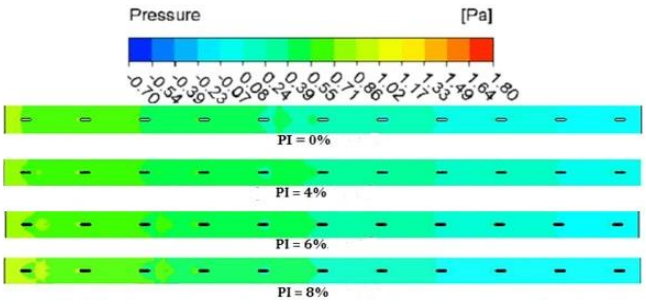

Fig.3 Pressure contours at Z-Y plane for different cases of OP insert at Re =3000

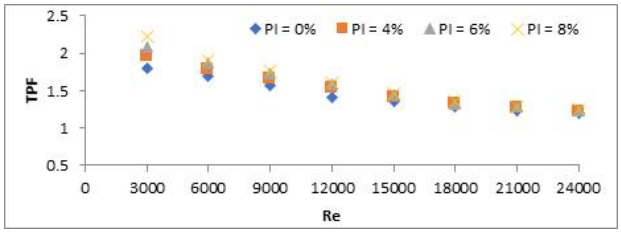

Fig.4 TPF variation with Reynolds number (Re) for different cases

TPF is found higher for higher PI as shown in fig.5. The higher the PI higher is the TPF obtained for the constant value of inclination angle. This is due to the fact of higher heat transfer rate and moderate friction factor as PI increases for particular inclination angle of OP insert.

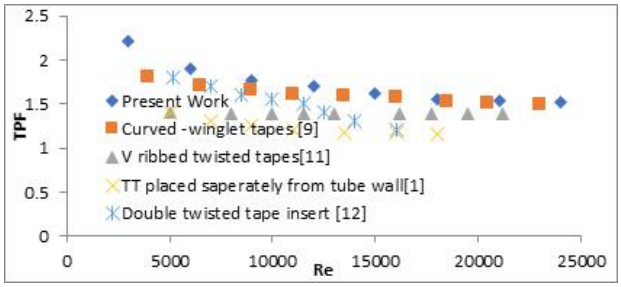

Fig.5 Present work TPF comparison with previous findings

4. PRESENT RESEARCH WORK COMPARISON WITH EARLIER FINDINGS

Fig.5 presents the comparison of the present article thermal performance factor using OP insert with some previous findings by [9],[11], [1],[12] and the present work is found to be superior in terms of overall thermal performance than other inserts used.

5. CONCLUSIONS

The heat transfer and fluid flow characteristics of the round tube heat exchanger using an oval profile insert is investigated. The important findings are as follows:

- Oval profile insert augments the rate of heat transfer by 2.633 times (at Re = 3000) and 1.471 times (at Re= 24000) compared to a smooth tube.
- The friction factor for perforated cases is slightly higher than those of non-perforated cases counterparts due to uneven surface created at the primary as well as secondary faces of OP insert.
- Heat transfer rate with the introduction of Perforations results in a higher heat

transfer rate because a fraction of fluid striking the primary face will go through the length of OP insert and more mixing of fluid occurs.

- TPF is increased with an increase in the perforation index for perforated OP inserts due to higher Nusselt number, fluid mixing, vortex formation, swirl generation.
- The highest and lowest heat transfer enhancement of 3.296 times and 1.471 times of smooth tube results while the highest and lowest thermal performance of 2.214 times (at Re = 3000) and 1.196 times (at Re = 24000) over smooth tube results are obtained for PI = 8% and 0% respectively.

The present research work found the best results for the case PI = 8% in terms of TPF and compared to previous findings TPF results are superior. So, the current proposed novel oval profile insert could be used for achieving a high heat transfer rate for a heat exchanger.

6. ACKNOWLEGDEMENTS

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