ANALYSIS OF HEAT TRANSFER IN A HEATED TUBE WITH A DIFFERENT TYPED DISC INSERTION

by

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Heat transfer and fluid flow can be controlled in a tube by inserting different typed passive elements. The main objective of this study is to control heat transfer and fluid flow using cutting edged disc in pipe. Governing equations of laminar, 2-D flow is solved via finite volume technique. The disc is adiabatic and its thickness is 5 mm. It is located into axial axis of the tube. Three cases were applied based on the type of the disc as inclination angle of the top side is 45° and 0°. Calculations were performed for different Reynolds number in the range of 335 < Re < 845. Three cases were tested based on types of discs. It is observed that each position exhibits different heat transfer ratio according to studied Reynolds number. The highest heat transfer is formed when inlet flow impinges to flat side of the cutting edged baffle.

Key words: heat transfer enhancement, disc, numeric, fluid flow

Introduction

Heat transfer enhancement techniques are very important to save more energy and using of optimal energy sources. Heat transfer enhancement techniques are mainly divided into two classes as passive and active techniques. The passive techniques have been usually preferred by many researchers since no additional external power is required as extended surfaces, rough surfaces and swirl flow devices. Coiled wire insert is one of the passive heat transfer enhancement techniques, which is extensively used in various heat transfer applications such as, air conditioning and refrigeration systems, heat recovery processes, food and dairy processes, and chemical process plants. The wire coil is quite simple to manufacture, to insert and remove from the tube which, therefore, justifies its usage in heat transfer enhancement [1-5].

A study on heat transfer enhancement is performed tried to enhance heat transfer in a tube with equilateral triangle cross-sectioned coiled wire inserts [6]. They made several experiments to do this. They indicated that the highest overall enhancement efficiency of
36.5% is achieved for the wire at Reynolds number of 3858. Consequently, the experimental results reveal that the best operating regime of all coiled wire inserts is detected at low Reynolds number, leading to more compact heat exchanger. In another study, delta-winglet twisted tape inserts was used to enhance heat transfer in a tube [7]. They found that the values of Nusselt number and friction factor in the test tube equipped with delta-winglet twisted tape are noticeably higher than those in the plain tube and also tube equipped with typical twisted tape. Investigation of heat transfer enhancement in a tube with an inner tube insertion was made numerically and experimentally by Fu et al. [8, 9], respectively. Both studies indicate that heat enhances by inserting a inner tube. Oztop et al. [10] studied the effects of contraction-expansion-contraction (CEC) pipe insertion in a pipe. They observed that the fluid deviates the outside pipe wall due to insertion of CEC pipe. Varol [11] numerically analysed the ring like body inserted tubes with different positions and he tried to find optimum position for heat and fluid flow. Nguyen et al. [12] prepared a numerical code to solve conjugate heat transfer and fluid flow in rib roughened tube. They indicated that thermal conductivity value is an important parameter on temperature distribution and heat transfer. Fu et al. [13] made a numerical work on heat transfer rate of heated walls in a channel with an oscillating cylinder. An arbitrary Lagrangian-Eulerian kinematic description method is applied. It is shown that the position and the diameter of the cylinder in the cylinder have great effects on the flow and thermal fields [14]. Other related studies can be found in Promvonge et al. [15] and Yu et al. [16].

The main objective of this study is to present heat transfer and fluid flow in a method sharp edged disc inserted pipes. Based on brief above review and authors’ knowledge, the inserted geometry did not studied in earlier works using a numerical technique. Results will be presented in next parts with pressure contours, velocity contours, temperature distribution, heat transfer and temperature and velocity profiles.

**Problem definition**

Physical model of considered problem is depicted in fig. 1. In this figure, a pipe with 2 m length and 40 mm diameter are chosen and a sharp edged disc were inserted its center. The location of center of disc is taken as 0.5 m from inlet. In fig. 1(a), a, b, r, R, and \( \varphi \) stand for short side of the disc, thickness of the disc, higher side of the disc, radius of the pipe, and inclination angle of the disc, respectively. Its thickness is 5 mm and thermal conductivity of disc material is chosen as very low. Physically the disc is fixed using a very thin rope which is not disturbed the flow. Thickness of pipe wall is accepted as very thin. Thus, conduction heat transfer along pipe wall is neglected. Air was chosen as working fluid and its properties are chosen as \( \rho = 1.225 \text{ kg/m}^3 \), \( c_p = 1006.43 \text{ J/kgK} \), and \( \nu = 1.78 \cdot 10^{-5} \text{ m}^2/\text{s} \). Diameter of the pipe is taken as 30 mm.

**Governing equations and solution**

Governing equations as continuity, x- and y-momentum and energy are written in cylindrical co-ordinates by considering incompressible flow, neglecting of radiation of heat transfer and buoyancy forces, steady-state, and laminar flow conditions. Thus, equations are:

\[
\frac{\partial u}{\partial x} + \frac{1}{r} \frac{\partial}{\partial r}(rv) = 0
\]
Figure 1. Physical model, (a) pipe with inner body, (b) Case I, (c) Case II, (d) Case III, (e) boundary conditions

\[
\rho \left( u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial r} \right) = -\frac{\partial P}{\partial x} + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial r^2} + \frac{1}{r} \frac{\partial u}{\partial r} \right)
\]

\[
\rho \left( u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial r} \right) = -\frac{\partial P}{\partial r} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial r^2} + \frac{1}{r} \frac{\partial v}{\partial r} - \frac{v}{r^2} \right)
\]

\[
\rho c_p \left( u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial r} \right) = k \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right)
\]

The boundary conditions assume no-slip conditions for velocity on the surfaces and the passive element. Symmetrical solution is adopted. Velocity and temperature is uniform at the inlet and temperature boundary condition is also constant on the pipe wall. The physical properties of the air have been assumed to remain constant at average bulk temperature. Thus, Prandtl number is taken as 0.72 for whole cases. Velocity is hydrodynamically developed at the pipe exit. Disc is assumed as adiabatic for all cases. Boundary conditions can be defined mathematically as:

- at the inlet \( u = \text{const}., \quad T = \text{const}, \quad \text{and} \quad P = \text{const.} \) \( (5) \)

- at the block \( u = v = 0, \quad \frac{\partial T}{\partial n} = 0 \) \( (6) \)
\[ \frac{\partial u}{\partial x} = \frac{\partial v}{\partial y} = 0, \quad P = 0 \] (7)

\[ u = v = 0, \quad T = \text{const.} \] (8)

Governing equations were solved by finite volume method of Patankar [17]. SIMPLE solution algorithm was applied. Fluent commercial code [18] was used to solve governing equations and simulate the results. The discretization scheme used is hybrid for the convective terms in the momentum and energy equations, and the SIMPLE algorithm for pressure-velocity coupling. The mesh for considered model is generated in the Gambit 2.1.6 preprocessor [19]. Thus, a grid independent solution was performed for 10000 nodes. Convergence criteria is taken as \(10^{-6}\) and \(10^{-3}\) for energy and other parameters, respectively. Convergence is obtained after 200 iterations. Local and average Nusselt numbers are calculated as:

\[
\text{Nu}(x) = \frac{h(x)D}{k}, \quad h(x) = \left[ \frac{q}{T_m - T_b} \right]
\]

\[
T_m = \int_0^r c_p \rho u(r) Tr dr, \quad \overline{\text{Nu}} = \int_0^L \text{Nu}(x) dx
\] (9)

Results and discussion

A numerical work has been performed to investigate the effects of sharp edged disc insertion in a pipe on heat transfer and fluid flow. Three different discs were used to make this analysis. The study was performed for different Reynolds number as \(\text{Re} = 335, 590, \) and \(845\). Pressure contours, velocity contours, isotherms, velocity and temperature profiles will be presented in next part of the study. Validation of computers code was made with literature (fig. 2). Trend is similar between two works but they are little difference near the inlet due to different acceptations.

Figure 2. Comparison of local Nusselt number with literature [20]

Figure 3(a)-(c) illustrates the velocity contours for the same Reynolds number and compares the effects of used types of discs. The flow is maximum near the center of the body and it is almost zero near the impinged region to the disc. Thus, a stagnation point is occurred at the middle but it is difficult to see it on fig. 3(a). The flow goes from top inclined part of the disc and it impinges to the top wall of the pipe. Due to acceleration of the fluid between disc and pipe wall a maximum flow velocity is occurred. Results are shown for sharp edged disc is located as in fig. 3(b), the maximum velocity is occurred only at the edge of the disc. Isotherms are presented in fig. 4 for the same case with velocity contours (fig. 3). It is noted that the fluid inlets to the duct with lower temperature than that of heated boundary. The boundary is heated under constant temperature and the disc is taken as adiabatic. As seen from the figure, the fluid is heated after the disc depends of the types of the inner disc. The disc types affect the temperature at the upstream of the channel. Figure 5 presents the
Figure 3. Velocity contours for three different cases (Re = 335); (a) Case I, (b) Case II, (c) Case III (color image see on our web site)

Figure 4. Isotherms for three different cases (Re = 335); (a) Case I, (b) Case II, (c) Case III (color image see on our web site)
Figure 5. Isotherms for three different cases (Re = 590); (a) Case I, (b) Case II, (c) Case III (color image see on our web site)

Figure 6. Isotherms for three different cases (Re = 845); (a) Case I, (b) Case II, (c) Case III (color image see on our web site)
Isotherms for three different cases at Re = 590. Temperature distribution for different cases are presented for Re = 845 in fig. 6. Both of these figures indicate that thinner temperature boundary layer is obtained with the presence of baffle. The thermal boundary layer becomes thinner with further increasing of Reynolds number. For Re = 845, the flow in front of baffle becomes cold then, a dome typed temperature layer is observed around the baffle. Velocity and temperature profiles for Case I are shown in fig. 7(a-c), for Re = 335, 590, and 845, respectively. The flow is developed at the exit of the duct. The profile is seen around $L = 0.8$ m. Higher velocity values are obtained for Re = 845. Variation of velocity becomes almost constant at $L = 0.4$ m with increasing of Reynolds number. Similarly, fig. 8 is plotted to give

Figure 7. Velocity profiles (left) and temperature profiles (right) along pipe at different stations for Case I; (a) Re = 335, (b) Re = 590, (c) Re = 845
temperature profiles for Case II at different location of pipe length directions and Reynolds number. Temperature values are almost constant at the exit for the lowest value of Reynolds number. Flow is heated along the duct. There is huge difference on temperature between $L = 0.4$ and $L = 0.8$. In a similar manner, fig. 9 gives the profiles for Case III. The trend is almost the same as earlier profiles but values are different from each other depending on the pipe length. Figure 10 presents the variation of local Nusselt numbers along the pipe. Heat transfer increases around the inserted baffle and this increment higher with increasing of Reynolds number. Higher increment can be noticed for Case III. The variation of mean Nusselt number with Reynolds number is presented in tab. 1.
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Table 1. Variation of mean Nusselt number with Reynolds number

<table>
<thead>
<tr>
<th>Re</th>
<th>Case I</th>
<th>Case II</th>
<th>Case III</th>
</tr>
</thead>
<tbody>
<tr>
<td>335</td>
<td>12.71057</td>
<td>12.69113</td>
<td>12.73672</td>
</tr>
<tr>
<td>845</td>
<td>30.94855</td>
<td>31.46152</td>
<td>31.40339</td>
</tr>
</tbody>
</table>

Figure 9. Velocity profiles (on the left) and temperature profiles (on the right) along pipe at different stations for Case III, (a) Re = 335, (b) Re = 590, (c) Re = 845
Conclusions

A numerical study has been performed to examine the heat transfer and fluid flow in a sharp-edged disc inserted channel with different Reynolds numbers. Three types of disc are studied to see the effects of discs on temperature distribution, flow field, and heat transfer. It is found that both flow and heat transfer can exhibit different behavior at the same value of Reynolds number. Case II presents the best results on heat transfer at the highest Reynolds number. For low Reynolds number, Case III gives the better results. A sharp-edged disc can be a good control element for heat transfer and fluid flow.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>short side of the disc</td>
<td></td>
</tr>
<tr>
<td>b</td>
<td>disc thickness</td>
<td></td>
</tr>
<tr>
<td>cp</td>
<td>specific heat capacity of air</td>
<td>[J kg⁻¹ K⁻¹]</td>
</tr>
<tr>
<td>D</td>
<td>diameter of the pipe, [m]</td>
<td></td>
</tr>
<tr>
<td>h</td>
<td>heat transfer coefficient</td>
<td>[W m⁻² K⁻¹]</td>
</tr>
<tr>
<td>h(x)</td>
<td>local heat transfer coefficient</td>
<td>[W m⁻² K⁻¹]</td>
</tr>
<tr>
<td>k</td>
<td>thermal conductivity, [W m⁻¹ K⁻¹]</td>
<td></td>
</tr>
<tr>
<td>L</td>
<td>length of the pipe, [m]</td>
<td></td>
</tr>
<tr>
<td>L₁</td>
<td>location of disc distance from inlet, [m]</td>
<td></td>
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<tr>
<td>Nu(x)</td>
<td>local Nusselt number, [-]</td>
<td></td>
</tr>
<tr>
<td>Nu̇</td>
<td>average Nusselt number, [-]</td>
<td></td>
</tr>
<tr>
<td>n</td>
<td>any co-ordinate, [-]</td>
<td></td>
</tr>
<tr>
<td>P</td>
<td>pressure, [Nm⁻²]</td>
<td></td>
</tr>
<tr>
<td>r</td>
<td>higher side of the disc, radial co-ordinate</td>
<td></td>
</tr>
<tr>
<td>R</td>
<td>radius of the pipe, [mm]</td>
<td></td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number (uD/v)</td>
<td></td>
</tr>
<tr>
<td>T</td>
<td>temperature, [K]</td>
<td></td>
</tr>
<tr>
<td>u, v</td>
<td>velocities, [m s⁻¹]</td>
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Greek symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
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<tbody>
<tr>
<td>μ</td>
<td>dynamic viscosity, [kg m⁻¹ s⁻¹]</td>
<td></td>
</tr>
<tr>
<td>ν</td>
<td>kinematic viscosity, [m² s⁻¹]</td>
<td></td>
</tr>
<tr>
<td>ρ</td>
<td>fluid density, [kg m⁻³]</td>
<td></td>
</tr>
<tr>
<td>φ</td>
<td>inclination angle of the disc, [°]</td>
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References