The effects of the axis ratio on the flow and heat transfer characteristics around a drop-shaped tube

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Abstract. Flow and heat transfer characteristics around single drop-shaped tubes with different axis ratio ($L/D$) in cross-flow are studied numerically for values of Reynolds number in the range $1.3 \times 10^3$ to $20 \times 10^3$. The results are obtained using the commercial software ANSYS Fluent for a two-dimensional (2D) computational domain. The axis ratio of the studied tubes is varied from 1 to 4, when $L/D = 1$, the tube is circular. The simulation results agree well with the available literature. The distribution of local coefficients of pressure and friction over half of the tube’s surface is plotted and analysed. It found that the drop-shaped tubes delay the separation of the boundary layer from the tube wall. The results confirm that the minimum value of pressure coefficient decreases as $L/D$ decreases, and the maximum value of the friction coefficient gradually increases with the growth of $L/D$. The result of the numerical simulation indicates the superior overall performance of drop-shaped tube with $L/D = 4$ over the rest of the tubes. Correlations of the average Nusselt number and the friction factor in terms of Reynolds number, calculated by the maximum velocity in the minimum free cross-section, and axis ratios for the studied cases are proposed.

1. Introduction

In recent years, non-circular tubes have attracted attentions in the context of saving energies [1–2]. Lavasani [3] experimentally investigated the flow about cam shaped tube bank with inline arrangement. It was noted that friction factor of cam shaped tube bank is approximately 95% lower than that in the circular tube bank. Brauer [4] reported that the pressure drop for elliptical tubes reduced 18% compared to circular ones. Zhukova [5] experimentally and numerically investigated hydrodynamics and heat transfer characteristics of oval and circular tubes. They analyzed total drag, pressure losses, pressure distribution, friction factor, and local heat transfer around the tubes. Deeb and Sidenkov [6–8] numerically studied hydrodynamics and heat transfer characteristics of a drop-shaped tubes bundle of various configurations. Their results indicate that the hydrodynamic characteristics of the drop-shaped tubes is better than the circular ones at angles of attack of 0°, 180°. Their results showed that thermal–hydraulic performance of drop-shaped tube bundle is about 3.5 ~ 4.5 times greater than the circular tube bundle.

The above-mentioned works have pointed out overall flow and heat transfer characteristics. However, the effects of the axis ratio on the flow and heat transfer characteristics of the drop-shaped tubes are still
not fully understood. Thus, the present work sheds light at the properties of local and overall air flow and heat transfer about drop-shaped tubes with various axis ratios for $1.3 \times 10^3 \leq \text{Re} \leq 20 \times 10^3$.

2. Problem definition and boundary conditions

2.1. Geometrical description of the study
In this study, six drop-shaped tubes are considered with axis ratios of 1.5, 2, 2.5, 3, 3.5, 4 (figure 1). When $L/D = 1$, the tube is circular. The tubes are located in a square cross-section channel (305 x 305 mm) and a length of 780 mm. To compare heat transfer and friction factor from each tube, the characteristic length $D_{eq} = P/\pi = 22.5$ mm is defined as the diameter of an equivalent circular tube, where $P$ is the circumference of the tube. The geometric characteristics of the studied tubes are presented in Table 1.

![Figure 1. a) Schematic of the studied tubes, b) Cross-sections of the tube.](image)

| No. tube | 1   | 2   | 3   | 4   | 5   | 6   | 7   |
|----------|-----|-----|-----|-----|-----|-----|-----|
| $L$ (mm) | 22.5| 27.13| 28.822| 29.992| 30.744| 31.397| 31.841|
| $D$ (mm) | 22.5| 17.6| 14.4| 12| 10.4| 8.98| 8|
| $L/D$    | 1| 1.5| 2| 2.5| 3| 3.5| 4|

2.2. Problem description and boundary conditions
The forced convection problem has been solved using ANSYS Fluent [10] in a two-dimensional stationary arrangement. The solver in ANSYS Fluent solves the viscous Navier-Stokes equations for the fluid flow in addition to the equation of state, total energy equation, and the two-equation of $k - \omega$ model [10]. The following assumptions are adopted in this study:

1) The flow is considered to be incompressible as the highest Mach number achieved by the flow is 0.041.
2) The thermo-physical properties of the fluid are calculated at the mean inlet air temperature.
3) The effect of radiation heat transfer was avoided because of the small temperature difference between tube surfaces and the surrounding air inside the channel.
4) The condensation of the incoming air was neglected.
5) Gravity and buoyancy lift are neglected.

The air velocity at the inlet varies from 1 to 15 m/s, which corresponds to Reynolds numbers $\text{Re} = 1.3 \times 10^3 ~ 20 \times 10^3$, at a temperature of 56.5°C and atmospheric pressure. While the temperature at the tube wall is kept at 20.8°C with no-slip condition.
2.3. Mesh generation and numerical results validation

The mesh is generated using ICEM CFD. Figure 2 shows the configuration of the computational domain and mesh for all the studied tubes with boundary conditions and the real dimensions of the 2D channel. The domain is symmetrical rectangle, includes half of the tube. The location of the first node away from a wall were kept at $1 < y^+ < 5$. The mesh quality of 0.926 is maintained for the entire simulation. The solution was considered converged when the scaled residual of all the fluid flow equations attains $10^{-10}$.

The mesh-sensitivity analysis was carried out mainly to check for a mesh independent solution (figure 3). The mesh of 48612 nodes is considered here-onwards to optimize the time and accuracy of the solution.

(a)         (b)

**Figure 2.** Mesh details with boundary conditions: a) $L/D=1$, b) $L/D=2.5$.

In order to validate the numerical model, the results of the present study were validated with [9] in the same range of values of Reynolds number. Figure 4 shows that the numerical predictions conform with the experimental results. Therefore, the numerical model used in this study is valid to carry out the rest of the simulations.

3. Results and discussion

3.1. Characteristics of air flow

The streamlines help to track the fluid particle path over the tube surface within the test channel. The streamlines also provide the details of the separation point and the intensity and location of the vortex formation downstream of the tube. Figure 5 shows the streamlines contours for the studied tubes with different axis ratios at Re=4000. For all cases, a pair of vortices appears behind the tube. By increasing $L/D$, the separation point of the boundary layer moves downstream and the vortex area decreases, which alters the flow field characteristics downstream when using tubes with different axis ratios.
3.2. Heat transfer characteristics

The distribution of a local heat transfer coefficient $\alpha$ over a half surface of each tube at $Re = 9400$ is given in Figure 6. For all tubes, the largest value of the local heat transfer coefficients is achieved at the forward stagnation point then it decreases until the flow reaches the separation point of the boundary layer, after which $\alpha$ increases again as the vortex appears. The lowest point of the curves in figure 6 indicates that the flow is separated from the surface of the tube.

The heat transfer is largely associated with the development of hydrodynamic boundary layer over the tube surface. The variation of the average Nusselt number against the Reynolds number is illustrated in figure 7. It is noticed, at $Re=1300$, that the circular tube ($L/D=1$) has the best $Nu_{av}$ as compared with the drop-shaped tubes. This can be attributed to the low level of turbulence intensity in the case of $L/D > 1$. At $1300 < Re \leq 6700$, the drop-shaped tube with $L/D=1.5$ has the highest value of $Nu_{av}$ compared to other tubes. With an increase in $Re$ ($Re > 6700$), heat transfer rate is noticed to rise for drop-shaped tube with $L/D=2$ (at $Re=20000$, $Nu_{av}$ is about 12.11\% higher than that of a circular tube). The average Nusselt number is determined from the computational results as:

$$Nu_{av} = \frac{\bar{\alpha}D_{eq}}{\lambda}$$  \hspace{1cm} (1)

where $\bar{\alpha} = \frac{1}{F} \int_{0}^{\pi} \alpha dF$ is the heat transfer coefficient averaged over entire tube surface.

Figure 8 presents the variation of $Nu_{av}$ versus $L/D$ for the different $Re$. It is found that for ($L/D>1.5$, $1300 < Re \leq 6700$), average Nusselt number of drop-shaped tubes slightly decreases as the tube axis ratio increases. This effect is noticed for tubes of axis ratio $L/D>2$ but for $Re > 6700$. This is due to the growth of boundary layer on a longer distance of the tube surface with the increase of the axis ratio, resulting in greater thermal resistance on the “flat” parts of the tube’s surface.

Correlation for the average non-dimensional Nusselt number for different axis ratios of tube based on the computational results is predicted by equation (2):

$$Nu_{av} = a Re_{D, max}^{b} Pr^{0.37} \left( c + \frac{L/D}{d} \right)^{e}$$  \hspace{1cm} (2)

All Constants of this equation are listed in Table 2.

| Table 2. Constants for correlation (2) |
|---|---|---|---|---|---|
| $L/D$ | $a$ | $b$ | $c$ | $d$ | $e$ |
| 1 $\leq L/D < 1.5$ | 0.1624 | 0.6172 | 0.4125 | 0.4913 | 0.3448 |
| 1.5 $\leq L/D < 2$ | 0.1793 | 0.6594 | -0.4623 | 1.5275 | 0.1255 |
| 2 $\leq L/D < 2.5$ | 0.1316 | 0.6863 | -0.2311 | 1.2785 | 1.2785 |
| 2.5 $\leq L/D < 3$ | 0.114 | 0.6965 | 0.1444 | 1.118 | 0.0937 |
| 3 $\leq L/D \leq 3.5$ | 0.1047 | 0.706 | 1.8195 | 0.8166 | 0.0521 |
| 3.5 $< L/D \leq 4$ | 0.1047 | 0.706 | 1.8195 | 0.766 | 0.0455 |
3.3. Pressure coefficient distribution and friction factor

The pressure over the tube surface can be expressed through a pressure coefficient as:

\[ C_{Pi} = \frac{P_i - P_{\infty}}{\frac{1}{2} \rho U_{\infty}^2} \]  

(3)

where \( P_i \) is the local static pressure; \( P_{\infty} \) and \( U_{\infty} \) are the pressure and the free-stream velocity, respectively.

Figure 9 shows the distribution of pressure coefficient over half surface of each tube at \( Re = 9400 \). At the forward stagnation point, it is clear that all tubes have the maximum value of the pressure coefficient, which is about 1.1 (point A). Then it decreases to reach a minimum value (point B), the point of transformation from a favorable \( (\frac{dP}{dx} < 0) \) to an adverse \( (\frac{dP}{dx} > 0) \) pressure gradient, and then increases up to the stagnation point at the rear of the tube. Drop-shaped tube with \( L/D = 4 \) has a higher favorable pressure gradient at the front section of the tube while it has the lower adverse pressure gradient in the back of the tube, which prevents an early separation of the fluid from the wall boundary.

The pressure drop across the channel is represented by friction factor, as defined by:

\[ f = \frac{2\Delta P}{(L/D)(\rho U_{\infty}^2)} \]  

(4)

where \( \Delta P \) is the pressure drop across the channel obtained from the results of the computational simulation.

Figure 10 presents the friction factor for the tubes with different \( L/D \) of the tube, at \( Re = 1.3 \times 10^3 - 20 \times 10^3 \). as the Reynolds number of the flow \( Re \) increases, for all cases, the friction factor for the fluid decreases. This is attributed to the fact that the pressure forces are more dominant than the friction forces. In the case of low \( Re \), the effect of viscosity over smarts the pressure influence on the tube wall. For \( 1 \leq L/D \leq 2 \), friction factor decreases rapidly and then settles at \( L/D = 2.5 \) (the change is relatively small). Results show that the friction factor of a drop-shaped tube with \( L/D = 4 \) is about 14 – 18.3 times lower than the circular shaped tube. Correlation for friction factor for the tubes with different axis ratios based on the computational experiment was predicted by equation (5):

\[ f = \left[ a + b \left( \frac{L}{D} \right) + c \ln(Re_D) \right] \left[ 1 + d \left( \frac{L}{D} \right) + e \ln(Re_D) \right]^{-1} \]  

(5)

The obtained correlations (2) and (5) is applicable for \( 1.3 \times 10^3 \leq Re \leq 20 \times 10^3 \) and for Prandtl number \( Pr \approx 0.699 \). All Constants of this equation are listed in Table 3. Reynolds number in equations is calculated by
\[ \text{Re}_{D,\text{max}} = \rho U_{\text{max}} D_{\text{eq}} / \mu, \] where \( \mu \) is the dynamic viscosity, \( U_{\text{max}} \) is the maximum velocity in the minimum free cross-section.

### Table 3. Constants for correlation (5)

| \( L/D \) | \( a \) | \( b \) | \( c \) | \( d \) | \( e \) |
|---------|--------|--------|--------|--------|--------|
| \( 1 \leq L/D < 1.5 \) | 0.7919 | -0.4615 | -0.00756 | -0.9958 | 0.12739 |
| \( 1.5 \leq L/D < 2 \) | -0.045 | 0.00356 | 0.00203 | -0.3147 | -0.0996 |
| \( 2 \leq L/D < 2.5 \) | -0.02 | 0.00021 | 0.00842 | -0.2617 | -0.1003 |
| \( 2.5 \leq L/D \leq 4 \) | -0.0305 | 0.0026 | 0.00065 | -0.137 | -0.1702 |

#### 3.4. Overall thermal performance

There is a need to evaluate the combined effect of heat transfer along with the pressure drop associated with the flow over the studied tubes. The overall thermal performance for drop-shaped tubes based on circular tube is defined by efficiency index \( \eta \) [11]:

\[
\eta = \left[ \frac{\text{Nu}_{\text{av,drop}}}{\text{Nu}_{\text{av,circ}}} \right] \left[ \frac{f_{\text{drop}}}{f_{\text{circ}}} \right]^{-1}
\]

(6)

Figure 11 illustrates the relationship between \( \eta \) and Re for tubes of different axis ratios. The results show that \( \eta \) increases with the increase in axis ratio and/or with the increase of Reynolds number of the fluid flow. It was found that the values of \( \eta \) for drop-shaped tube with \( L/D = 4 \) is about 11.5 ~ 20.2 times greater than those for the circular tube (\( L/D = 1 \)). This can be attributed to its aerodynamic shape and lower friction factor compared to the circular tube.

![Figure 8. Effect of L/D on Nuav](image8.png)

![Figure 9. Pressure coefficient distribution](image9.png)

![Figure 10. Effect of L/D on f](image10.png)

![Figure 11. \( \eta \) versus Re](image11.png)
Conclusions
A numerical investigation is carried out to grasp the fluid flow and heat transfer characteristics of single circular and drop-shaped tubes in crossflow with different axis ratios. The main conclusions are as follows:
1) As the tube shape flattens, it allows the boundary layer to grow, which increases the thermal resistance of the tube. This causes the average Nusselt number to drop marginally. The extent of this effect is governed by the tube axis ratio and Reynold number.
2) $f$ of a drop-shaped tube with $L/D = 4$ is about 14 – 18.3 times lower compared to the circular one.
3) For different tube’s axis ratios, the results indicate that the friction factor decreases with an increase in the Reynolds number or and the tube’s axis ratio.
4) Correlations of $Nu_{av}$ and $f$ in terms of $Re_D$ and $L/D$ for the studied cases are proposed.
5) The overall performance of drop-shaped tube with $L/D = 4$ is about 11.5 – 20.2 times greater than those for the circular tube.

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