Numerical investigation of heat transfer and friction factor characteristics for staggered double drop-shaped tubes bundle in cross-flow

R Deeb and D V Sidenkov

Department of theoretical basis of heat engineering, National Research University “MPEI”, Russia, 111250 Moscow, 14Krasnokazarmennaya Street

E-mail: e.rawad.deeb@yandex.com, sidenkovidv@mpei.ru

Abstract. A numerical study has been conducted to clarify the flow and heat transfer characteristics of a cross-flow heat exchanger with drop-shaped tubes at zero angle of attack. A mathematical and numerical model in software package Ansys for numerical evaluation of heat transfer and flow field of a bundle of drop-shaped tubes, with taking into account the strain caused by different pressures inside and outside the tubes has been developed. The result of the numerical simulation indicates the superior thermal and hydrodynamics performance of the drop-shaped tube bundles over the circular tubes.

1. Introduction

In recent years, the issue of heat recovery from exhaust gas has been considered using cross-flow non-circular tube’s bundles heat exchangers. It is very important to develop efficient heat exchangers which have a high heat transfer rate and low-pressure loss. Heat transfer of circular tubes has been investigated extensively in the literature. Buyruk[1] performed numerical study of heat transfer characteristics on tandem cylinders, inline and staggered tube bank in the cross flow of air at various pitch ratio. Achenbach [2] measured the local static pressure, friction factor and heat transfer of single circular cylinder in crossflow. The results from these studies indicated that the flow over the cylindrical tubes bank reports early separation of the fluid from the wall boundary, which reduces the heat transfer and increases the pressure drop within the tube bank. Hence this forced the researchers to study non-circular tubes to enhance the heat transfer rate.

Lavasani[3] experimentally investigated the flow around cam shaped tube bank with inline arrangement for both longitudinal pitch ratios 1.5 and 2. It was noted that by increasing longitudinal pitch ratios from 1.5 to 2, heat transfer increases about 7-14 %. Furthermore, friction factor of cam shaped tube bank is approximately 95 % lower than circular tube bank. Horvat et al [4] numerically compared the heat transfer conditions for the tube bundle in cross flow for different tube shapes as cylindrical, ellipsoidal, and wing-shaped. The pitch to the diameter ratio in the staggered arrangement was from 1.125 to 2.0. Their results showed that drag coefficient is lower for ellipsoidal and wing-shaped tubes than that for the cylindrical tubes. However, drag coefficient decreases with increasing the Reynolds number.
The effects of angles of attack on the heat transfer characteristics and the drag coefficient for staggered drop-shaped tubes were experimentally and numerically investigated by Sayed Ahmed et al. [5-6]. They found that the bundle with zero angle of attack (θ=0) has increased average Nu values by about 76% compared to elliptical tubes bundle with the same heat transfer surface. In addition, the lowest values of pressure drop were achieved at zero angle for all values of Reynolds numbers.

The purpose of this study is to investigate the heat transfer and friction factor characteristics for staggered double drop-shaped tubes bundle in crossflow using Ansys package with taking into account the effect of deformation caused by pressure drop inside and outside the tubes.

2. Problem definition and boundary conditions

2.1. Geometrical description of the study

Using Ansys, a numerical study of heat transfer and hydrodynamics of a bundle of 44 drop-shaped tubes (figure 1) is carried out. Drop-shaped tubes are located in a square cross-section channel, a side of the square cross-section is 305 mm with the following dimensions: the large radius is 5.8 mm, the small radius is 2.9 mm, the equivalent diameter (Deq) is 22.5 mm (figure 2). The transversal and longitudinal spacing in the range of ST=37 mm and SL=37 ~ 46.25 mm, respectively.

2.2. Problem description and boundary conditions

The forced convection problem has been solved using Ansys Fluent [7] in a two-dimensional stationary formulation assuming a viscous incompressible flow with constant thermophysical properties, with taking into account the possibility of turbulence generation but without heat exchange by radiation. The system of differential conservation equations includes the continuity equation, two momentum equations and the energy equation:

\[
\frac{\partial}{\partial x_i}(\rho U_i) = 0
\]

\[
\frac{\partial}{\partial x_j}(\rho U_i U_j) = -\frac{\partial P}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j}
\]

\[
\frac{\partial}{\partial x_i}[U_i (\rho E + P)] = \frac{\partial}{\partial x_i} (\lambda \frac{\partial T}{\partial x_i})
\]

where i – tensor indicating 1 and 2; U – the air velocity; \(\rho\) – the air density; P – air pressure; \(\tau_{ij}\) – the viscous stress tensor; \(\lambda\) – the fluid effective thermal conductivity and T – temperature of the liquid.
The modeling process is carried out in two stages. Firstly, the stress-strain state modeling has been performed using Ansys Static Structural, the deformations caused by different pressures inside (14 bar) and outside the tubes (1 bar) have been determined. Figure 3 illustrates the cross section of the drop-shaped tube after deformation.

In the second stage, similar to [5], the RNG k-ε model with “Enhanced wall Treatment” function [7–8] is used in the present study. As an external flow, the air flow is used, the initial velocity of the air at the channel’s entrance region varied $u = 1.33 \sim 14 \text{ m/s}$, which corresponds to Reynolds numbers $Re_a = 1.8 \times 10^3 \sim 18.7 \times 10^3$, at a temperature of 56.5 ºC and atmospheric pressure (figure 1).

![Figure 3. Tube status under the load.](image)

### 3. Results and discussion

#### 3.1. Characteristics of air flow

Figure 4 illustrates the streamlines for circular and drop-shaped tubes of the studied bundles. The streamline provides the details of the separation point and it indicates the intensity and location of the vortex formation at the tube downstream. For all cases of longitudinal spacing, it is clear that there are three zones of significant changes in the local hydrodynamic characteristics of the air flowing across the tubes lead to changes in the local values of heat transfer: two at the lateral and one at the rear surfaces of the tubes. It is noted that the area of the vortex zone is greater in the circular tubes.

![Figure 4. Streamlines for circular and drop-shaped tubes at $u=7 \text{ m/s}$ for longitudinal spacing $S_i$ of: (a,c) 37 mm, (b,d) 46.25 mm.](image)

In the case of the longitudinal spacing of 37 mm, the distance between the two adjacent tubes is small, so the vortex effect is weak (figure 4a, c). With the increase in longitudinal spacing to 46.5 mm, the intensity of the vortex within the tube columns increases, which leads to the higher heat transfer rate due to the increase in fluid circulation area (figure 4b, d).

#### 3.2. Heat transfer characteristics over the tubes bundle

In the case of a bundle of drop-shaped tubes, the average Nusselt number increases with the increase in the air velocity. This can be attributed to a high turbulence in wakes of downstream tubes which makes the
boundary layer of tubes rows thinner. In the circular tube bundle, there is almost no effect of increasing the longitudinal spacing on the heat transfer (figure 5). The average Nusselt number of a bundle was determined from the computational experiment results as:

\[
\bar{Nu} = \frac{\bar{\alpha} - D_{eq}}{\lambda}
\]  

where \(\bar{\alpha} = \frac{1}{F} \int \alpha.dF\) is the heat transfer coefficient averaged over whole surface of tubes bundle, \(\frac{W}{m^2 K}\).

Figure 5. Average Nusselt number versus air velocity.

Figure 6 demonstrates the distribution of temperature of circular and drop-shaped tubes bundle for two hydrodynamic regimes and for two cases of the longitudinal spacing. The temperature of the tube surface increases by gaining the heat from the incoming air. As the longitudinal spacing increases, the intensity of the turbulent fluid within the tandem tubes increases, which may enhance the heat transfer. On the other hand, the area of higher temperature air zone increases. The turbulence area behind the circular tubes is larger than that in drop-shaped tubes, which contributes to a further improvement in heat transfer.

\[S_L = 37 \text{ mm}\]

\[u = 1.33 \text{ m/s} \quad u = 14 \text{ m/s}\]

\[S_L = 46.25 \text{ mm}\]

\[u = 1.33 \text{ m/s} \quad u = 14 \text{ m/s}\]

Figure 6. Distribution of the temperature for circular and drop-shaped tubes bundle for \(S_L\) of 37 mm (a-d) and 46.25 mm (e-h) at \(u = 1.33 \sim 14 \text{ m/s}\).
3.3. Friction factor

Figure 8 shows the static pressure distribution for circular and drop-shaped tubes bundles for two cases of the longitudinal spacing and the air velocity. For all studied cases, it can be seen that the pressure has the highest value at the stagnation point on the front of the tube, this is because of the flow velocity at this point tends to zero (figure 7). When the flow passes over the surface of the tube, the pressure decreases to the lowest value on the lateral surface.

Friction factor $f$ is defined as:

$$f = \frac{\Delta P}{0.5 \rho U_{av}^2 N_L}$$  \hspace{1cm} (5)

where $\Delta P$ - pressure drop across the bundle, Pa; $U_{av}$ - average flow velocity in the narrow cross-section of the bundle, m/s; $N_L$ - number of transverse rows.

$S_L = 37 \text{ mm}$

$u = 1.33 \text{ m/s}$

(a) \hspace{1cm} (b) \hspace{1cm} (c) \hspace{1cm} (d)

$u = 14 \text{ m/s}$

$S_L = 46.25 \text{ mm}$

$u = 1.33 \text{ m/s}$

(e) \hspace{1cm} (f) \hspace{1cm} (g) \hspace{1cm} (h)

$u = 14 \text{ m/s}$

Figure 7. Distribution of the velocity for circular and drop-shaped tubes bundle for $S_L$ of 37 mm (a-d) and 46.25 mm (e-h) at $u = 1.33 \sim 14 \text{ m/s}$. 
Figure 8. Distribution of the pressure for circular and drop-shaped tubes bundle for $S_L$ of 37 mm (a-d) and 46.25 mm (e-h) at $u = 1.33 \sim 14$ m/s.

Figure 8 indicates the friction factor for circular and drop-shaped tubes bundles. The friction factor for the fluid decreases, with an increase in the air velocity. This is usually due to the dominant pressure force, which reduces the friction. For all bundles, the increase in longitudinal spacing of the tubes increases the vortex formation between each two adjacent tubes in the same row, and hence, the friction factor increases. It is also clear that the friction factor for the circular tubes is about 5.9 ~ 7.3 times higher than the drop-shaped tubes (Figure 9).

3.4. Thermal-hydraulic performance

The thermal hydraulic performance of drop-shaped tube heat exchanger is proposed by Webb [9] as:

$$\eta = \frac{N_{\text{Uad,drop-shaped}}/N_{\text{Uad,circ}}}{f_{\text{drop-shaped}}/f_{\text{circ}}}$$

(6)

Figure 9. Friction factor versus air velocity. Figure 10. Thermal–hydraulic performance of drop-shaped tube and circular tube bundles.
Thermal-hydraulic performance of drop-shaped tube bundle is compared with circular tube bundle in figure 10. The results show that thermal–hydraulic performance of drop-shaped tube bundle is about 3.5 ~ 4.5 times greater than the circular tube bundle. As a result, drop-shaped tube bundle performs better than circular ones. This can be attributed to its aerodynamic shape and lower friction factor compare to circular tube.

Conclusions
The thermal and fluid flow behaviour in case of a circular and double drop-shaped tube bundles with staggered arrangement have been studied numerically. Mathematical and numerical model has been developed to calculate the heat transfer and friction factor of staggered double drop-shaped tubes bundle using the Ansys package, with taking into account the stress-strain state of the tubes.

The drop-shaped tubes provide the better heat transfer characteristics as compared to that of the circular tubes. A bundle with $S_L = 46.25$ mm has the maximum averaged Nusselt number.

As the air velocity increases, the friction factor decreases. The friction factor is higher in the bundles with a longitudinal spacing of $46.25$ mm relative to a longitudinal spacing of $37$ mm. This is due to more clearance between the tube columns.

Friction factor from drop-shaped tube bundle is about 13.4–28.3% lower than circular tube bank and as a result thermal–hydraulic performance of this tube bank is about 3.5 ~ 4.5 times greater than circular tube bank. The efficiency of drop-shaped tubes with a longitudinal spacing of $46.25$ mm is higher than that of $37$ mm. Consequently, the present analysis suggests that considering both thermal and hydrodynamic parameters overall the drop-shaped tube bundle is superior than the circular tube's one.

References
[1] Buyruk E 2002 *Heat and Mass Transfer* 29(02) 355–66
[2] Achenbach E 1975 *Heat and Mass Transfer* 18 1387–96
[3] Lavasani A M, Bayat H and Maarefdoost T 2016 *Applied thermal engineering* 65(1–2) 85–93
[4] Horvat A, Leskovar M and Mavko B 2006 *Heat and Mass Transfer* 49 1027–38
[5] Sayed A et al 2014 *Heat and Mass Transfer* 50(8) 1091–102
[6] Sayed A et al 2014 *Heat and Mass Transfer* 51(7) 1001–16
[7] ANSYS Inc. 2019 *Ansys Fluent User’s Guide*. In Ansys Aim Student 19.0, 19.2
[8] Yakhot V 1993 *Int. Conf. on Near-Wall Turbulent Flows* (Arizona: Tempe) pp. 1031–46
[9] Webb R L 1981 *Heat and Mass Transfer*. 24 715–26