Numerical simulation of cross-flow in a bank of tubes with three rows in the subcritical region of Reynolds

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Abstract. The present work focused on 2-dimensional unsteady numerical simulation in predicting hydrodynamics and thermal characteristics of air flow across circular tube banks with integral wake splitters. The tube banks studied consist of three rows of tubes in staggered arrangement. The lengths of the splitters are 0, 0.5, 1.0, 1.5 and 2.0 times the tube diameter. The range of Reynolds number investigated is in the range of 1000 to 10000, which is in the sub-critical region of Reynolds number. The flow condition within this range is incompressible since the maximum Mach number is less than 0.3. The numerical approach was validated against the experimental works of Zukauskas (1985) and Anderson (1997). Local pressure coefficient for flow around a single tube with integral wake splitter is also presented for comparison. It was found that the present of the wake splitters was able to improve the overall heat transfer of the system.

1. Introduction

Cross-flow heat exchanger is a common type of heat exchanger with wide range of applications. Cross-flow means that one fluid flow perpendicular to the second fluid flow direction. The most common tube used is circular in shape. The fluid flow across this tube is very complex and was subjected to many research works [1-3]. The hydrodynamics of the flow give effect to thermal characteristics of the system. Rigorous studies on this have been done by many in order to understand the underlying phenomena. Many have attempted to improve its overall performance [3-5]. Flow across a circular cylinder is a classical problem, and has been studied experimentally, visually and numerically covering various aspects of flow behavior. The development of the vortex street, many studies on the cylinder flow problem have been undertaken. Such flow in a heat exchanger has been studied for the purpose of getting high heat transfer efficiency i.e. high heat transfer rate at low required pumping power. For flow around a single tube, at very low Reynolds number in the order of one, the laminar boundary layer that starts from the forward stagnation point may not separate from the tube surface up until the rear stagnation point. As the Reynolds number increases, the boundary layer starts to detach and form symmetrical recirculation regions or vortices. These vortices which initially small begin to increase in size and shed alternately at Reynolds number in the order of two. This particular region is known as the wake region. The vortices periodically shed in a deterministic
manner and the point of separation maintained at an angle of 80° from the forward stagnation point at large range of Reynolds number until critical value of $2 \times 10^5$.

Many researchers have done a lot of studies to improve the performance of cross-flow heat exchanger. Some added additional feature to heat exchanger in order to improve the heat transfer performance. Zhang et al. [6] added vortex generators to promote higher turbulence flow in order to enhance heat transfer rate. Tiwari et al. [7] integrated longitudinal plate to circular tube in order to split the wake region for reducing form drag. Jacimovic et al. [8] investigated the effect of plate fin to the pressure drop. Kongkaitpaiboon et al. [9] used circular-ring turbulator to improve the efficiency of heat exchanger. Some researchers modified the shape of the traditional circular tube to enhance the overall heat exchanger performance. Yajima and Sano [10], and Valera et al. [11] studied the effect of circular tube perforation towards pressure drag and heat transfer rate. Khan et al. [12] and Matos et al. [13] investigated elliptical tube array for cross-flow cooling. Shirani and Nasibi [14] studied the hydrodynamics of flow across in-line tube banks at low Reynolds number ranging from 25 to 2500 numerically. They suggested that there is an optimum value for longitudinal pitch at which drag is at minimum. Khoddamrezzaee et al. [15] investigated numerically the heat transfer characteristics of nanofluid flowing across tubes in a rectangular arrangement. It was predicted that the separation points were postponed and the heat transfer coefficients were higher than that of base fluid without nano particle. High performance cross-flow heat exchanger means that the heat transfer capability is high. However heat transfer rate almost always proportional to high pressure drop. All previous works in attempt to achieve heat exchanger design with high efficiency concentrated in improving heat transfer rate without significant increase in pressure loss, thus the focus of the present work.

2. Approach and Methods

2.1. The Cases Studied

The tube was arranged in staggered formation with the pitch-to-diameter ratio of 2. The splitter plate length-to-tube diameter ratio was varied from 0 to 2 in the increment of 0.5. The splitter plate thickness was 1/25 the tube diameter.

2.2. The Computational Domain and Boundary Conditions

The computational domain is shown in figure 2. Uniform free-stream velocity of air enters the domain at a temperature of 300 K. Meanwhile the temperature of the tubes and splitter plates were set to be 400 K. Periodic boundary conditions were set at both side of the domain in order to reduce the computational cost. Each periodic pairs experiences the same flow and thermal conditions. The outlet boundary was distanced 20 tube diameters from the center of the 3rd row tube to minimize the effects of the outlet boundary condition on the flow in the vicinity of the tubes.

An example of a discretized domain is shown in figure 3. Distance to the nearest tube wall, $Y_{wall}$, is 0.004 times the tube diameter. Structured grid was used to resolve the near-wall region. Unstructured grid with triangular cells was implemented elsewhere to ease the meshing process. This hybrid meshing technique can reduce the processing cost by reducing the total number of cells. Finer cells were created near the wall and the wake region to give better resolution on the large property gradient region.

2.3. The Governing Equations

The present study simulations were performed for unsteady, 2-dimensional flow past unconfined bank of tubes with and without wake splitters using laminar model. The followings are the conservation equation of mass, momentum and energy involved in tensor form.

$$\frac{\partial u_i}{\partial x_i} = 0$$  \hspace{1cm} (1)
\[ \rho \left( \frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} \right) = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} \]  

(2)

where

\[ \tau_{ij} = \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu \frac{\partial u_i}{\partial x_i} \delta_{ij} \right] \]

is the stress tensor

\[ \rho c_p \left( \frac{\partial T}{\partial t} + u_j \frac{\partial T}{\partial x_j} \right) = \frac{\partial}{\partial x_i} \left( k \frac{\partial T}{\partial x_i} \right) \]

(3)

Figure 1. The investigated configuration.

Figure 2. The computational domain.

Figure 3. The discretized computational domain.
3. Numerical Results and Discussions

3.1. The Validations

The present numerical approach was tested by validating it against the experimental works of Anderson and Szewczyk [16], and Zukauskas and Ziugzda [17]. Good agreements were obtained between the numerical and the experimental data. They were evaluated quantitatively by calculating the root mean square of error, RMSE. The RMSE are 0.23 and 0.31 respectively.

\[
RMSE = \sqrt{\frac{\sum_{i=1}^{n} (x_{1,i} - x_{2,i})^2}{n}}
\]  

(a) \(C_p\) distribution around the cylinder for \(Re = 4.2 \times 10^4\) and \(BR = 1/4\)

(b) \(C_p\) distribution around the cylinder for \(Re = 4.6 \times 10^4\) and \(BR = 1/17\)

Figure 4. Validation against a) Anderson et. al. 1997 and b) Zukauskas et. al. 1985.

3.2. The Grid Sensitivity Study

The sensitivity of the numerical results can be observed from figure 5. Apparently, the results are not sensitive to change of grid or cell density. According to table 1, the maximum RMSE of the predicted local pressure coefficient is below 0.1.
Figure 5. $C_p$ distribution around the tubes at 1st, 2nd and 3rd row for $Re = 1000$ and $L/D = 0.0$ for various no of total elements
Table 1. RMSE of $C_p$ distribution around the tubes at 1st, 2nd and 3rd row for Re = 1000 and L/D = 0.0 compared to its mean result for various total no. of cells

| Total no. of cells | RMSE (compared to mean value)  |
|-------------------|--------------------------------|
|                   | 1                              | 2                              | 3                              |
| 25794             | 0.007574                       | 0.01112                        | 0.096945                       |
| 16964             | 0.001859                       | 0.005152                       | 0.009259                       |
| 14170             | 0.002595                       | 0.006167                       | 0.04767                        |
| 12950             | 0.005521                       | 0.007286                       | 0.089357                       |

3.3. Local Pressure Coefficient

For flow around single tube, shown in figure 6, the base pressure (pressure near the rear stagnation point) of tubes with splitter plate are higher than that of plain tube. This indicates that the splitter plates have effectively reduced the pressure loss, hence reducing drag. However, for flow across bank of tubes, illustrated in figure 7, the splitter plates, in most cases increase the pressure loss. Only at the 3rd row, splitter plate of L/D = 1.5 and L/D = 2.0 at Re = 1000, and splitter plate of L/D = 0.5, L/D = 1.0, L/D = 1.5 and L/D = 2.0 at Re = 10000, give positive effects.

![Figure 6](image)

Figure 6. $C_p$ distribution around a single tube with splitter plate of various L/D for Re = $4.6 \times 10^4$ and BR = 1.

3.4. Total Pressure Drop

Energy loss is evaluated by calculating the difference in total pressure at the inlet and the outlet. From figure 8, it can be seen that the total pressure loss increases as the Reynolds number increases. The same trend can be seen as the splitter plate length increases, with the exceptional of L/D = 0.5 at Re = 8000.

3.5. Total Heat Transfer

From figure 9, it is observed that the attachment of wake splitter does not improve the heat transfer process at the surface of the tubes. Since mixing intensity is constrained by the splitter plate, the heat transfer rate is thereby attenuated. However, according to figure 10, the overall heat transfer improves since the splitter plates also act as fins, which increases the surface contact area for better overall heat transfer process.
Figure 7. $C_p$ distribution around the tubes at 1st and 3rd row for Re = 1000 and Re = 10000.
Figure 8. Total pressure drop, $\Delta p$ for various L/D and Reynolds number.

Figure 9. Total Surface Heat Transfer from The Tubes (excluding wake splitters).

Figure 10. Total Surface Heat Transfer from The Tubes (including wake splitters).
4. Conclusions
The present study has arrived at the following conclusions:

- Good agreements between the present numerical work and the existing experimental data of Zukauskas and Ziugzda [17], and Anderson and Szewczyk [16] for flow around a single tube have been obtained.
- For flow around a single tube, wake splitters of all studied length were predicted to be able to reduce pressure loss significantly.
- For cross-flow in a bank of tubes, the addition of splitter plates increases the pressure loss, especially for tubes at the 1st and 2nd row.
- The numerical simulations predicted that the overall heat transfer process can be intensified by attaching wake splitter plates to the tubes.

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