



2-Dimensional Numerical Study of Cylindrical Tubes in Crossflows Arranged in Staggered Formation

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Abstract: 2-dimensional numerical study of cylindrical tubes in crossflows arranged in a staggered formation that aims to investigate the characteristic of thermal and flow of crossflow tubular heat exchangers in staggered arrangements. Prediction of heat transfer and flow around cylinders in tube banks is significant to various engineering aspects. 4 rows of cylindrical tubes with equilateral triangular arrangements were set up with a pitch distance-to-diameter ratio, P/d , between 1.2 to 2.2, respectively. The Reynolds numbers used are between 100 to 50000. In the present study, the coefficient of performance (COP) of the setups was obtained. The Reynolds number of 10000 has the highest value of the coefficient of performance at a pitch distance-to-diameter of 1.2. The highest value of the average Nusselt number goes to the pitch distance-to-diameter ratio of 2.2 at the Reynolds Number of 50000. The value of the average Nusselt number for the pitch distance-to-diameter ratio of 2 is slightly the same as the pitch distance-to-diameter ratio of 2.2. The pitch distance-to-diameter ratio of 2.2 has the highest value than the other pitch-distance-to-diameter ratio of 1.2, 1.5, and 2.

Keywords: Cylindrical Tubes, Heat Exchanger, Crossflows, Staggered Formation, Tube Banks

1. Introduction

Prediction of heat transfer and flow around tube bank cylinders is crucial for various engineering aspects. The recent energy crisis has promoted the development of high-performance heat exchangers for the effective use of energy. For that purpose, it is, necessary to clarify the detailed heat transfer characteristics of cylinders in crossflow heat exchangers in staggered formations. Earlier works focus primarily on pressure loss and heat transfer through the bank [1-3] There isn't much knowledge on how the fluid flow behaviours relate to the heat transfer process.

Additionally, there is a lack of information on the narrow-spaced tube banks, which may be crucial for developing compact heat exchangers. From this perspective, the current authors thoroughly researched the heat transfer and flow around a set of three or four cylinders in a straight line or spaced apart configuration in a crossflow of air [4] The cylinder distance was extensively altered to gather essential information about heat transport in tube banks. The heat transfer characteristics at a tight spacing in the case of the staggered arrangement resemble those of a cylinder in a two-dimensional impinging jet. On the other hand, they approach those of a single cylinder in a uniform flow for a broad separation [5-7].

However, in a staggered tube bank made of several cylinders, particularly in the tight spacing, the effects of surrounding cylinders on the heat transfer properties may be substantial.

2. Methodology and Geometry Modelling

The domain is modelled using the Design Modeler. It consists of 4 main parts: inlet, outlet, cylindrical tubes, and walls. The geometry that has been modelled and used throughout this project is shown in figure 1. The geometry shown in the figure is drawn and modelled by the following parameters.

- Diameter of the tube, $d = 0.025$ mm
- Pitch distance-to-diameter ratio, $P = 1.2d, 1.5d, 2.0d, 2.2d, 2.2d$



Fig. 1 – Dimension sketching of 2-dimensional model

2.1 Meshing and boundary conditions

The viscous model and Solver Set-up was set as the following figure 2 and table 1, while the boundary conditions are shown in table 2.

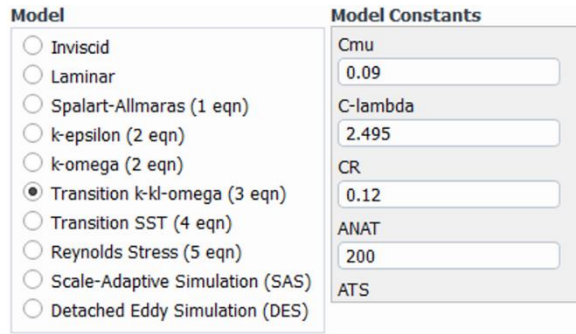


Fig. 2 – Viscous model setup

Table 1 – Solver setting

Pressure	Second Order
Momentum	Second Order Upwind
Turbulent Kinetic Energy	1st Order Upwind
Laminar Kinetic Energy	1st Order Upwind
Specific Dissipation Rate	1st Order Upwind
Energy	2nd Order Upwind
Pressure	2nd Order
Momentum	2nd Order Upwind
Turbulent Kinetic Energy	1st Order Upwind
Laminar Kinetic Energy	1st Order Upwind
Specific Dissipation Rate	1st Order Upwind
Energy	2nd Order Upwind

Table 2 – Boundary conditions setting.

Pitch distance-to-diameter ratio	1.2 $d = 0.03$ mm
	1.5 $d = 0.0375$ mm
	2.0 $d = 0.05$ mm
	2.2 $d = 0.055$ mm
Reynolds Number	100
	1000
	10000
	30000
	50000
Velocity Inlet	0.00973 m/s
	0.09738 m/s
	2.9215 m/s
	4.869115646 m/s
Inlet Temperature	300 K
Wall Tube temperature	400 K
Length	0.025 m
Density of air	1.225 kg/m ³
Viscosity	1.7894×10 ⁻⁵ kg/ms

The fluid used for the simulation is air with the density, $\rho = 1.225$ kg/m³ and viscosity, $\mu = 1.7894 \times 10^{-5}$ kg/ms. By comparing the value of the local pressure distribution with experimental data, standard- k -epsilon and four turbulence models were utilized for validation. The results for numerical (k-kl-omega) are slightly different from those from Shinya Aiba, Hajime Tsuchida, and Terukazu Ota [8] as shown in figure 3 below.

The outcomes for $Re = 3.6 \times 10^4$ were compared to the Local Pressure Coefficient of different turbulence model.

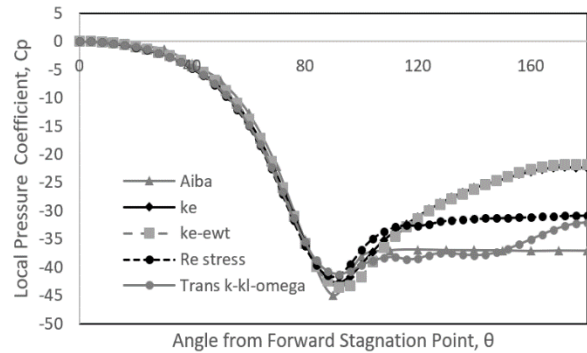


Fig. 3 – Local Pressure Coefficient (C_p) against Angle from Forward Stagnation Point, θ

2.2 Pressure coefficient calculation and NRSME

The relationship between the kinetic energy per volume of the flow and the pressure decrease brought on by obstacles in the fluid flow path is expressed by the pressure coefficient, which is a representation of the pressure traveling in a fluid [9,10].

$$C_p = \frac{p - p_\infty}{\frac{1}{2} \rho u_\infty^2} \quad (1)$$

The values for each validation model and the NRSME percentage error are listed in the table. The transition-k-kl-omega turbulence model has an NRSME percentage error value of 0.0486, according to the table. To test the effectiveness of a crossflow tubular heat exchanger in a staggered design, transition-k-kl-omega was chosen [11,12].

$$NRSME = \frac{RSME}{y_{\max} - y_{\min}} \quad (2)$$

$$RSME = \sqrt{\frac{\sum_{t=1}^T (\bar{y}_t - y_t)^2}{T}} \quad (3)$$

Table 3 – Value of NRSME error

Turbulence model	NRSME error
Standard-k-epsilon	0.1772
k-epsilon-Enhanced Wall Treatment	0.1781
Reynolds Stress	0.0879
Transition k-kl-omega	0.0463

2.3 Grid independency test (GIT)

Figure 4 shows the graph of pressure coefficient of fluid domain around cylindrical tubes with different grid of meshing while figures 5 and 6 are the mesh of the model with the grid size of 1.0 and 1.05 respectively. Table 4 below is the summary of the number of elements with a different mesh size.

Table 4 – Number of elements of different mesh grid

Mesh grid	Grid 1.0	Grid 1.01	Grid 1.02	Grid 1.05	Grid 1.1	Grid 1.2
No of elements	62114	20369	15768	12188	10942	9830

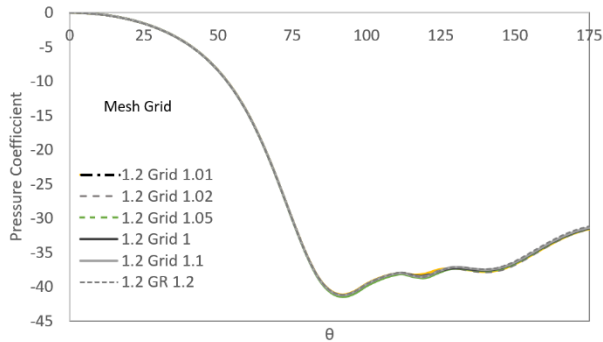


Fig. 4 – Local Pressure Coefficient (C_p) against Angle from Forward Stagnation Point, θ

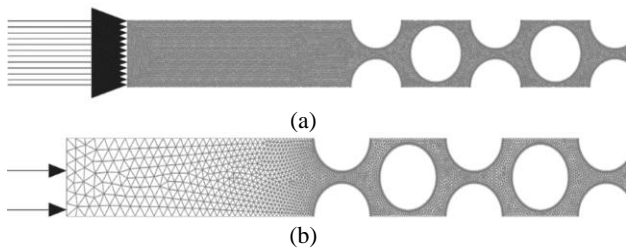


Fig. 5 – Example of meshing size for the model (a) Grid 1.0 (b) Grid 1.05

3. Results and Discussion

Figure 6 shows a graph of average Nusselt Number distributions of all tubes is obtained. The highest value of the average Nusselt number is 186.2467 for the pitch distance-to-diameter ratio of 2.2 at Reynolds number of 50000. The value of average Nusselt number for the pitch distance-to-diameter ratio of 2 is slightly the same as the pitch distance-to-diameter ratio of 2.2. The lowest value of the average Nusselt number is 1.059323 for the pitch distance-to-diameter ratio of 1.2 at Reynolds number of 100.

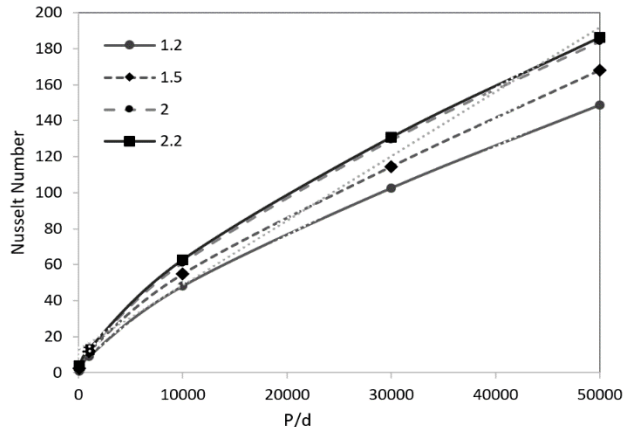


Fig. 6 – Graph of average Nusselt number of all tubes

Figure 7 shows a graph of the average Nusselt number against the number of rows is obtained. The highest value of the average Nusselt number is 167.3129 for the third row at Reynolds number of 50000. The graph pattern of average Nusselt number at Reynolds number of 30000 is slightly the same as the average Nusselt number at Reynolds number of 50000. The Reynolds number of 100 has the lowest average Nusselt number.

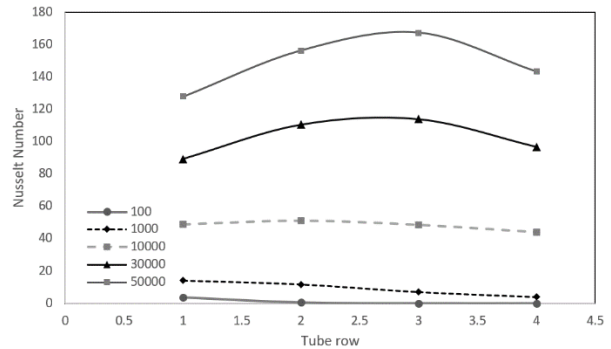


Fig. 7 – Average Nusselt Number distributions of each tube

Figure 8 shows the graph of Surface Heat Flux against Reynolds number. The highest value of surface heat flux is 20163.27 with pitch distance-to-diameter ratio of 2.2 and Reynolds Number of 50000. The lowest value of surface heat flux is 415.2203 with pitch distance-to-diameter ratio of 2.2 with Reynolds Number of 100. The value of surface heat flux with pitch distance-to-diameter ratio of 2 is slightly the same as pitch distance-to-diameter ratio of 2.2. Therefore, Reynolds Number affects the value of surface heat flux with different pitch distance-to-diameter ratio.

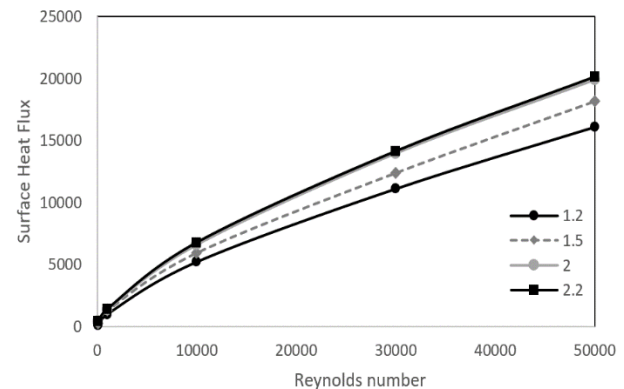


Fig. 8 – Surface heat flux distributions

From the figure 9, a graph of Surface Heat Flux distributions of pitch distance-to-diameter ratio of 1.2 was obtained. The highest value of Reynolds Number at 50000 has the highest value of surface heat flux at the third row of the cylindrical tube. The value of surface heat flux of the first row of cylindrical tube with Reynolds Number at 50000 is lower compared to the other row of the cylindrical tubes. Therefore, the third row of cylindrical tube has the higher value of surface heat flux with pitch distance-to-diameter ratio of 1.2.

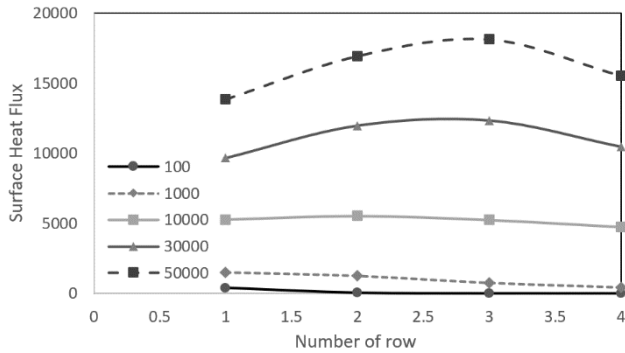


Fig. 9 – Surface Heat Flux distributions of pitch distance-to-diameter ratio of 1.2

Figure 10 shows the graph of coefficient of performance distributions. From the graph, it shows that Reynolds Number of 10000 has the highest value of coefficient of performance at pitch distance-to-diameter of 1.2. The coefficient of performance with Reynolds Number of 10000 with pitch distance-to-diameter of 2.2 has the lowest value compared to the value of coefficient of performance with Reynolds Number of 10000 with pitch distance-to-diameter of 1.2. Therefore, the higher the pitch distance-to-diameter ratio, the lower the value of coefficient of performance of cylindrical tubes in the staggered formation.

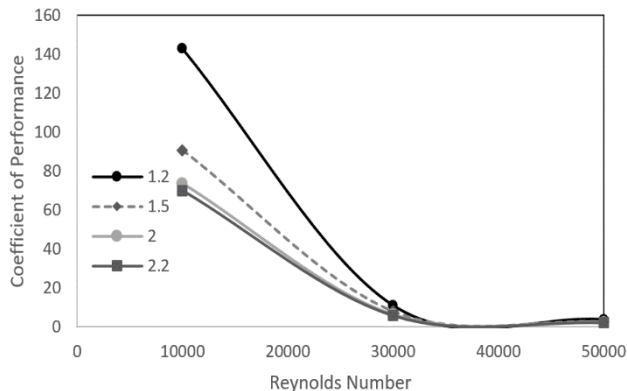


Fig. 10 – Coefficient of Performance distributions

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4. Conclusion

The numerical study was carried out to investigate the characteristic of thermal and flow field of crossflow tubular heat exchanger in staggered arrangement and to obtain the overall performance of crossflow tubular heat exchanger in the staggered formation. The numerical results show the effect of Re, pitch distance-to-diameter ratio and the rows of cylindrical tubes with equilateral triangular arrangement. These parameters were observed for Reynolds number in the ranged of 100 to 50000 with different value of pitch distance-to-diameter such as 1.2, 1.5, 2.0 and 2.2. The number of cylindrical tube rows were considered are 4 rows. Therefore, 2-dimensional numerical study of crossflows in staggered formation has been investigated with different pitch distance-to-diameter and different Reynolds number among 4 rows of cylindrical tubes of crossflows in staggered arrangement. In the present study, the coefficient of performance (COP) behavior throughout the crossflow in cylindrical tubes in staggered formation was obtained. In a comparison between different parameters such as Reynolds number, pitch distance-to-diameter ratio and number of cylindrical tube rows, the highest effect on the value of surface heat flux is found when the Reynolds Number increases.

The value of velocity inlet is depending on the value of Reynolds Number. The higher the Reynolds Number, the higher the value of velocity inlet. The crossflow of turbulent heat exchanger in staggered formation have been successfully generated and gone through the analysis using ANSYS Fluent. The method has been validated successfully. Thus, the method is acceptable. The pressure and velocity distributions are dependent on each other. The value of coefficient of performance is depending on the value of surface heat flux and the pumping power per area.

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