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Convective Heat Transfer from Tube Banks of 8 Rows with In-Lined Arrangements in Crossflow

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Abstract: Due to the superior heat transfer efficiency and cheaper cost compared to other types of heat exchangers, the inline crossflow heat exchanger is often employed in the process sector. Modelling of the flow arrangement is usually used as the foundation for the unit's design. The tube pitch distance ratio, Reynolds number, and fluid entry velocity into the heat exchanger are all modified in this study. The resistance to airflow around a heat exchanger is directly correlated with the distance between tubes. Greater heat transmission is encouraged by smaller spacing since it improves the surface contact area per volume. However, more pumping force is needed to force air through the confined area. The optimum ratio of heat transmission over the necessary pumping power is provided by optimal tube spacing. A flow through a heat model with an in-line structure that has two rows and eight tubes per row was investigated experimentally in the Reynolds number range of 5000 to 30000. In a variety of tube pitch distance ratios, the distribution of local static pressure was experimentally established around the tubes. Additionally, the overall pressure loss was calculated. FLUENT, a programme for computational fluid dynamics (CFD), is being used in this work to evaluate the pressure drop and velocity vectors that occur as fluid flows over a vertical tube.

Keywords: Heat exchanger, Reynold numbers, Airflow, Pumping power, CFD

1. Introduction

Several crucial industrial processes are idealised by the heat transfer and fluid movement in tube bundles. Crossflow heat exchangers, whose design still depends on empirical correlations of pressure drop and heat transfer, openly utilise tube banks. Many chemical and thermal engineering processes are very interested in heat exchangers using tube bundles in crossflow [1-4]. The design of heat exchangers must consider heat transfer in flow over a bank of tubes. There are several industrial uses for heat exchangers, including the production of steam in boilers and the cooling of air in air conditioner coils. The dimensionless transverse, longitudinal, and diagonal pitches of tube banks, which are often organised in a line or staggered fashion, define them. One fluid typically flows over the tubes while another, at a different temperature, goes through them [5-7].

Heat exchangers have many different applications including gas turbines, vehicle radiators, and glass ceramic. To conserve money and energy, it is thus always preferred to boost the efficiency of heat exchange equipment. Due to their enormous technological relevance, academics have long been interested in finding design changes that will increase heat exchangers' performance. The heat transmission rises in both configurations (inline or staggered) mostly with decreasing longitudinal pitch ratio and to a lesser extent with rising transverse pitch ratio. Higher heat transfer is indicated by compact banks (in-line or staggered) than those that are widely spread. Heat transfer is greater in a staggered bank than in an in-line bank for the same pitch ratio. This is because of the main flow's course being more difficult in a staggered bank and more downstream tubes' surface area remaining in this path [8-10].

The Nusselt number is one of the fundamental parameters used to assess the heat transmission in a tube bank [11-13]. The ratio of convective to conductive heat transfer across a barrier is what is used to define it. The Nusselt number is a dimensionless heat transfer coefficient, to put it another way. Utilizing dimensionless numbers allows for the comparison of the outcomes of the applied amounts on their own. The Nusselt number can be obtained in two well-known ways. The first one employs

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the Reynolds and Prandtl numbers as the basis for its empirical correlations, while the second one uses the heat transfer coefficient, characteristic length, and thermal conductivity [14-17].

This study was conducted to achieve several objectives that has been set which to explore the characteristic of flow field and thermal characteristics of crossflow tubular heat exchangers in in-lined arrangements and to achieve a correlation between thermal performance and the flow geometries of crossflow tubular heat exchangers in in-lined arrangements. This study was handled in simulation through ANSYS with some scopes need to be emphasized to build the desired model of heat exchangers. The models were built with an arrangement of 8 rows of cylindrical tubes with in-lined. The longitudinal and transverse ratio of pitch distance-to-diameter is between 1 to 2. The last one is the most important to set the velocity in the ANSYS which to set the Reynold numbers, Re between 5000 to 30 000.

2. Materials and Methods

Heat exchanger in the inline arrangement is modelled according to the dimensions of practically available heat exchanger using 8 rows of cylindrical tubes. To plot the in-lined tube banks crossflow, it is important to consider several parameters, which are the diameter of tube banks, and Pitch-distance-to-diameter ratio. The value has been set as in Table 1 below.

Table 1 – The result of Nusselt number

Parameters	Value
Diameter of tube, *d*	0.025 m
Pitch-distance-to-diameter ratio, *S/d*	1.2, 1.5, 1.8 and 2.0
Distance of inlet and first row of tube	0.125 m
Distance of outlet and last row of tube	0.5 m

After discretization of the computational domain, the boundary conditions were applied. The boundary conditions are given as per requirement and the solution is initialized and calculations are iterated. After the calculation is converged the contours are to be plotted. Configuring settings for turbulent, laminar, and viscous incompressible flow is possible using the Viscous Model dialogue box. This arrangement is used to do the computation for the staggered 2-dimensional model. Select 'Viscous Model' first, followed by 'transition k-kl-omega' from the 'Models' menu. The material properties must be specified during setup. For this condition of simulation, the air domain was used.

The velocity inlet value then computed for the boundary condition. The velocity calculated using an equation. $Re = \rho U_T D/\mu$ and $\boldsymbol{U}_\infty = U_T(C_y - D)/C_y$ with various Reynolds numbers. The temperature then needs to be set to 400 *K* in each tube exist in the designed *2D* tube banks model. A tube bank of inline tubes with a 0.025m diameter, 7 rows, and 4 columns from ratio 1.6 while 7 rows and 5 columns from ratio 1.2 was used for the investigation. A wind tunnel that was 0.225 m wide and 0.12 mm high was also employed in this experiment.

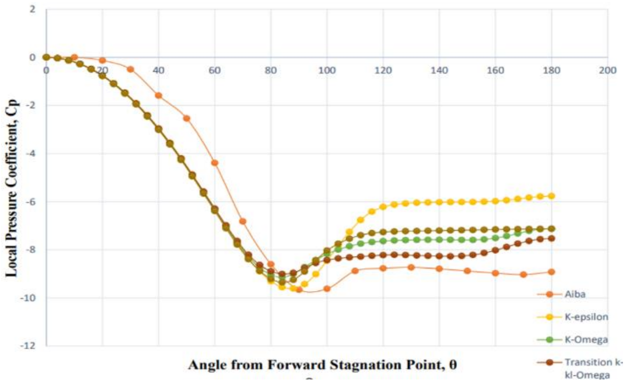

Fig. 1 – Local pressure coefficient, C_p vs angle from forward stagnation point, θ

Since *k-kl-omega* has the best value among the other models in terms of the Normalised Root Mean Square of Error, it was chosen as the model with the best outcome. Validate the result with previous published experimental data. The result from S. Aiba et al. [18] compared to the simulation result that has been conducted. Various value of Reynold number has been used to handle the simulation, which is 5000, 10000, 15000, 20000, 30000. The Reynold number formula can be used to determine the value of mean velocity, U_t which then can be used to calculate the inlet velocity, U_∞.

$$\mathrm{Re} = \frac{\rho U_T D}{\mu} \tag{1}$$

Value from previous U_t can be used to calculate inlet velocity, $\boldsymbol{U}_\infty$ by using the following equation.

$$U_\infty = \frac{U_T(s-D)}{s} \tag{2}$$

Other equations used to obtain the correlation for the convective heat transfer from tube banks of 8 rows with in-lined arrangements in crossflow such as the pumping power per area, efficiency and overall performance were shown as in the following equations.

$$P_{pump} = P_{inlet} - (P_{outlet} \times Vel_{inlet}) \tag{3}$$

$$\eta = \frac{\text{desired output}}{\text{required input}} \tag{4}$$

$$\eta_{overall} = \frac{\text{average heat flux}}{P_{pump}} \times 100\% \tag{5}$$

3. Results and Discussion

The summary for data obtains from the simulation were shown in Table 2 – 7. Table 2 shows the inlet velocity for each ratio used in the simulation while the rest of tables shows the results from the simulation result.

Table 2 – Inlet velocity with different Reynold numbers

	s/D			
	1.2	**1.5**	**1.8**	**2.0**
$\boldsymbol{Re_t}$	0.005	0.0125	0.02	0.025
5000	0.486803	0.973605	1.298141	1.460408
10000	0.973605	1.947211	2.596281	2.920816
15000	1.460408	2.920816	3.894422	4.381224
20000	1.947211	3.894422	5.192562	5.841633
30000	2.920816	5.841633	7.788844	8.762449

Table 3 – Total value for pressure inlet

	s/D			
	1.2	**1.5**	**1.8**	**2.0**
Re_t	0.005	0.0125	0.02	0.025
5000	10.87317	10.47351	9.218599	8.938859
10000	73.10971	42.98234	36.93902	35.31018
15000	161.058	94.87416	81.05738	76.72078
20000	279.1187	164.6468	140.2448	132.0031
30000	595.7637	351.1126	297.3472	279.9426

Table 4 – Pumping power/area

	s/D			
	1.2	**1.5**	**1.8**	**2.0**
Re_t	0.005	0.0125	0.02	0.025
5000	5.29E+00	1.02E+01	1.20E+01	1.31E+01
10000	7.12E+01	8.37E+01	9.59E+01	1.03E+02
15000	2.35E+02	2.77E+02	3.16E+02	3.36E+02
20000	5.44E+02	6.41E+02	7.28E+02	7.71E+02
30000	1.74E+03	2.05E+03	2.32E+03	2.45E+03

Data from Table 3 was the data from pressure inlet which can get from the CFD computation. The average data of pressure inlet was taken from the average value. The value then can be used to calculate for value of pumping power/ area likes had been calculated from Table 4.

Table 5 – Average heat flux

	s/D			
	1.2	**1.5**	**1.8**	**2.0**
Re_t	0.005	0.0125	0.02	0.025
5000	1518.171	2891.08	3261.159	3396.865
10000	3689.668	4957.724	5391.988	5529.487
15000	5086.172	6658.609	7145.032	7286.243
20000	6366.807	8192.314	8717.242	8856.133
30000	8672.478	10913.51	11479.12	11643.58

Data of average Nusselt number can be obtain from the ANSYS software by changing the area and diameter values from reference value tabs. 0.025m is used as the value for both parameters represent the diameter of the tube.

Table 6 – Average Nusselt number

	s/D			
	1.2	**1.5**	**1.8**	**2.0**
Re_t	0.005	0.0125	0.02	0.025
5000	1.40E+01	2.67E+01	3.01E+01	3.14E+01
10000	3.41E+01	4.58E+01	4.98E+01	5.11E+01
15000	4.70E+01	6.15E+01	6.60E+01	6.73E+01
20000	5.88E+01	7.57E+01	8.05E+01	8.18E+01
30000	8.01E+01	1.01E+02	1.06E+02	1.08E+02

Data of overall performance can be obtained from the correlation as above. The values of average heat flux and power pump/area was taken to form a correlation to calculate the efficiency.

Table 7 – Data of overall performance

	s/D			
	1.2	**1.5**	**1.8**	**2.0**
Re_t	0.005	0.0125	0.02	0.025
5000	28682.13	26020.88	27251.18	28352.07
10000	5183.573	5361.43	5622.272	5923.513
15000	2162.392	2167.68	2263.438	2402.876
20000	1171.439	1148.486	1197.045	1277.645
30000	498.385	474.6702	495.6462	532.0885

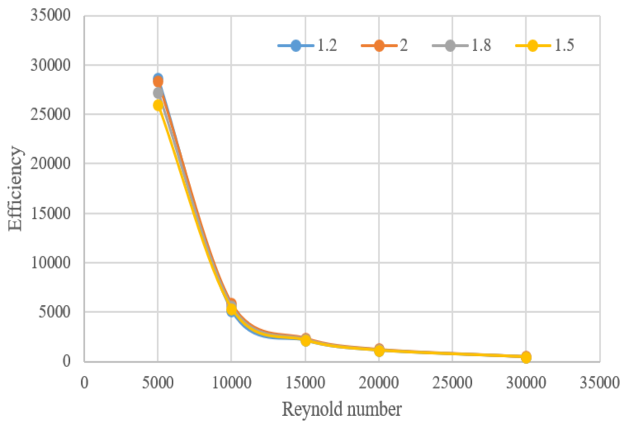

Fig. 2 – Efficiency vs Reynold Number

Figure 2 displayed the graph of the heat exchanger efficiency by using the average heat flux as the output and power pump/area as the output. As the data compared, pitch ratio of 2.0 has the highest efficiency than the others. Although the highest value of efficiency recorded at pitch distance ratio 1.2 at Reynold 5000, but the other Reynold for this pitch ratio is slightly lesser than pitch ratio 2.0.

In this study, the correlation of the data can be produced by using the regression method. The equation can be support by comparing the simulation result and its calculation. The result will be show in Parity plot in figures 4 - 6 to make sure that the equation constructed can be accepted. The R^2 value is a statistical measure that represents the goodness of fit of the trendline to the data points. The table below show the summary of the equation for the correlation of average Nusselt number, average heat flux, pumping power per area, Euler number and the overall performance.

Table 8 – Summary of the equation for correlations

Parameter	Equation	R^2
Ave. Nusselt number	$Nu_{Ave} = 0.069 \times (S/d)^{0.7} \times \mathrm{Re}^{0.7}$	0.97848
Average heat flux	$q_{Ave} = 6.6 \times (S/d)^{0.73} \times \mathrm{Re}^{0.7}$	0.97888
Pumping power	$P_{pump} = (6.64 \times 10^{-11}) \times (S/d)^{0.5} \times \mathrm{Re}^{3}$	0.99741
Euler number	$Eu = (5.53 \times 10^{-7}) \times (S/d)^{2} \times \mathrm{Re}^{-0.3}$	0.94377
Overall performance	$\eta_{overall} = (4 \times 10^{10}) \times \mathrm{Re}^{-2.2}$	0.99238

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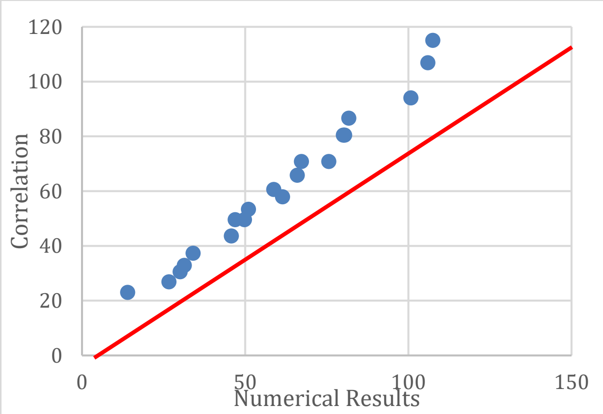

Figure 3: Parity plot of Nusselt number

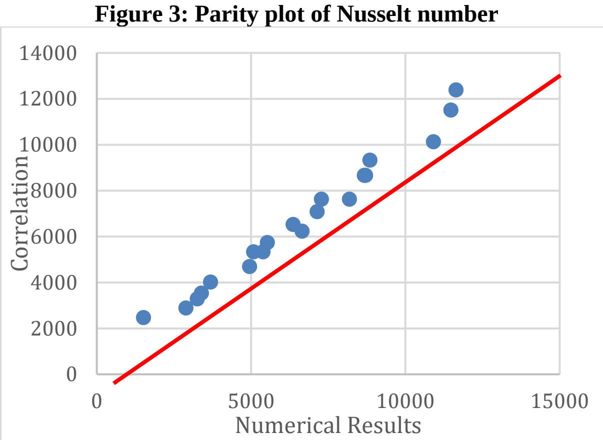

Figure 4: Parity plot of average heat flux

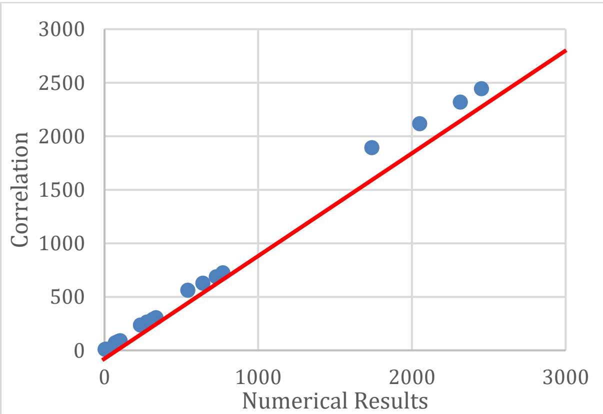

Figure 5: Parity plot of pump power/Area

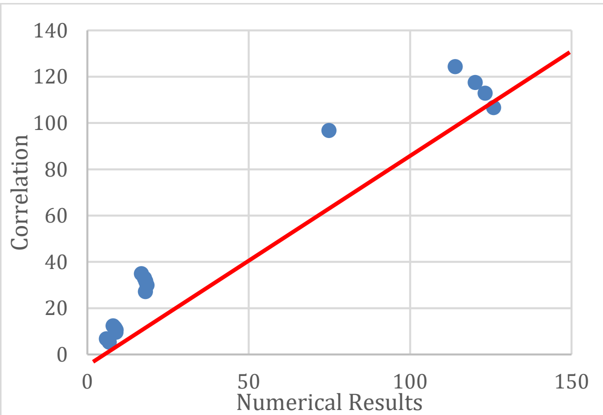

Figure 6: Parity plot of Euler number

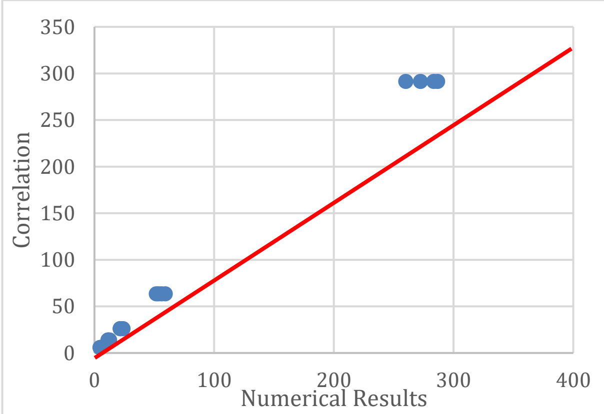

Figure 7: Parity plot of overall performance

4. Conclusion

In conclusion, the study on convective heat transfer from tube banks of 8 rows with in-lined arrangements in crossflow has provided comprehensive insights into the intricacies of heat transfer in this specific configuration. Through a combination of experimental investigations and numerical simulations, the research has yielded valuable data and correlations that contribute significantly to my understanding of heat transfer phenomena in such systems.

The goal of this work has been accomplished because the numerical prediction of pressure drops while fluid flows between tubes has been carried out and findings from experiments have been compared to ensure satisfactory outcomes. Considering the study, the pitch distance ratio of the tube banks and the Reynolds number affect the pressure drop. Based on the results, the model with the highest tube pitch distance ratio is the ideal design for an inline arrangement crossflow tube.

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