Transactions of the Korean Nuclear Society Autumn Meeting
Gyeongju, Korea, November 2-3, 2006

Effect of longitudinal pitch on the convection heat transfer from the tube banks in crossflow

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1. Introduction

When the tube banks in the heat exchanger are compactly designed, it is known that the average heat transfer coefficient is reduced compared with that of widely-designed tube banks. Thus, the heat transfer rate calculated by the usual heat transfer correlation will be over-estimated more than the actual one and the heat exchanger with such a design will have insufficient heat transfer capacity. Therefore, it is necessary to evaluate the effect of longitudinal and transverse pitches on the heat transfer, quantitatively.

Zukauskas [1] correlated various experimental data for aligned and staggered arrangements of tube banks as a function of Reynolds number and Prandtl number. In addition, Grimison [2] suggested the heat transfer correlation for tube banks whose coefficients are determined by geometrical characteristics. However, Zukauskas correlation does not consider the effect of longitudinal and transverse pitches in the case of the aligned arrangement and Grimison correlation can only be used for specific geometrical arrangement such as 1.25X1.25, 1.50X1.50, and so on. Therefore, additional correlation for a heat transfer coefficient which covers a wide range of a pitch is required to predict the heat transfer rate appropriately.

In this study, as a first step, the effect of a longitudinal pitch on the heat transfer is investigated for aligned tube banks by using CFD (Computational Fluid Dynamics) code.

2. Modeling

Typical aligned tube banks are illustrated in Fig. 1. The analysis was conducted with FLUENT [3] and the grid was generated by GAMBIT which is a pre-processor of FLUENT. As easily recognized, the fluid through the tube banks flows with a symmetric manner. Therefore, the calculation domain could be simplified with a symmetry boundary condition as shown in Fig. 1. In order to investigate the effect of longitudinal pitch, the transverse pitch was fixed at 14.0 mm. The diameter of tube is 10.0 mm and the ratios of the longitudinal pitch with respect to the tube diameter were selected in the range from 1.1 to 3.0 with an increment of 0.1. The wall temperature was assumed as 200 ℃ and the pressure and temperature of bulk fluid were 14.7 MPa and 310 ℃, respectively. The properties of water were obtained under the bulk conditions and the variation of water properties was neglected. The inlet and outlet were modeled with periodic boundary conditions to simulate the repeated geometry so that the calculation domain could be more simplified due to the exclusion of the repeated structure. Four turbulent models, which are standard k-ε model, RNG k-ε model, Realizable k-ε model, and SST (Shear Stress Transport) k-ω model, were used to study the dependency on the turbulent model and the SIMPLE algorithm and second-order upwind scheme were adopted as a pressure-velocity coupling method and discretization scheme, respectively. In order to model the near-wall flow field, the enhanced wall treatment included in FLUENT and the sufficiently fine mesh whose y+ is less than 1.0 were used.

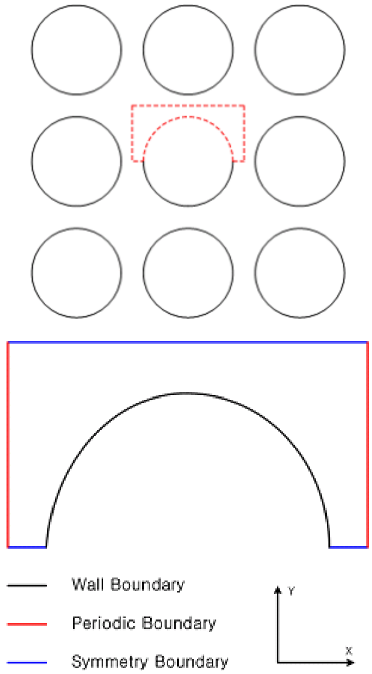

Fig. 1 Geometry and Boundary Conditions

Table 1. Boundary Conditions

Pressure (MPa)	14.7
Bulk Temperature (℃)	310.0
Density (kg/m3)	702.94
Specific Heat (J/kg-K)	5798.6
Thermal Conductivity (W/m-K)	0.53857
Viscosity (Pa-sec)	8.4186E-4
Wall Temperature (℃)	200
Inlet Mass Flow Rate (kg/sec)	1.4
Longitudinal Pitch Ratio	1.1 – 3.0

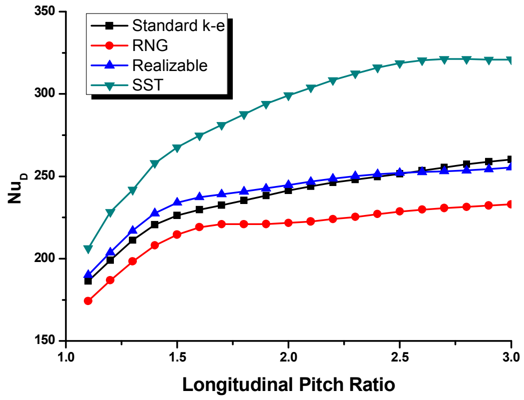

Fig. 2 Nusselt number

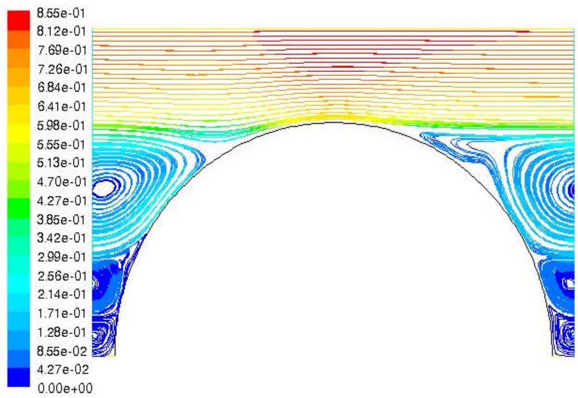

Fig. 4 Streamline for longitudinal pitch ratio of 1.1

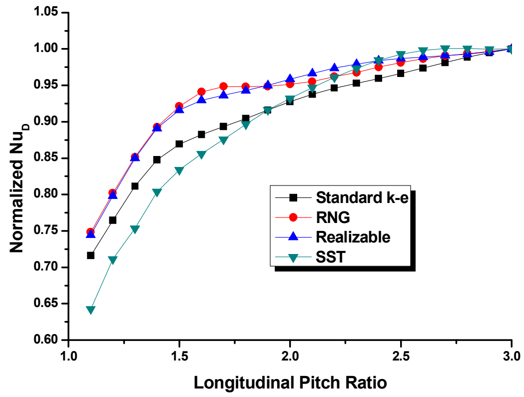

Fig. 3 Normalized Nusselt number

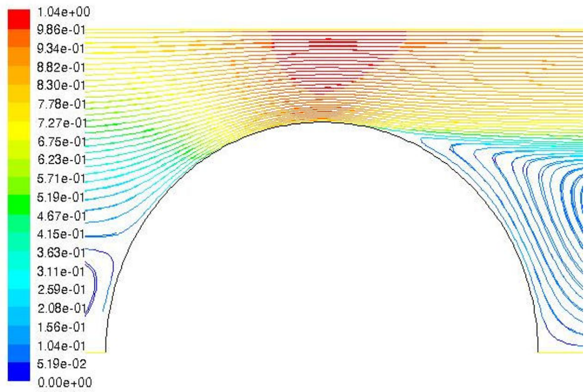

Fig. 5 Streamline for longitudinal pitch ratio of 3.0

3. Calculation Results

In order to interpret the calculation results, Nusselt number defined with the tube diameter was used. As shown in Figs. 1 and 2, Nusselt number decreases as the longitudinal pitch ratio decreases. This result comes from the flow field near the tube. As shown in Figs. 4 and 5, the flow at the azimuthal direction becomes stronger as the longitudinal pitch ratio increases. Therefore, the efficiency of the heat transfer at the surface of the tube which contributes to the heat transfer decreases when the longitudinal pitch ratio is small, so that the average heat transfer coefficient has a smaller value compared with the case of large longitudinal pitch ratio. This shows the same effect with a decrease of the heat transfer area and the heat exchanger designed without a consideration of such an effect will have an actually smaller heat transfer capacity than expected. The maximum Nusselt number difference from the case of the largest longitudinal pitch ratio is 35.7 % for all the cases.

From the results, it is found that there is an effect of a turbulent model to predict the heat transfer coefficient as shown in Figs. 1 and 2. A series of k-ε models predict a similar trend and Nusselt number. However, SST model predicts a considerably higher value of the heat transfer coefficient. This is due to the characteristics of the turbulent model. Generally, it is known that SST model can simulate a flow separation and swirl more accurately than k-ε models. In addition, the results from SST model agree with the results from Zukauskas correlation in the preliminary analysis for further study. Therefore, in this case, the application of SST model is recommended.

4. Conclusion

In order to investigate the effect of longitudinal pitch on the convective heat transfer from the tube banks in crossflow, the analytical approach was made by CFD code. From the result, it was found that the heat transfer coefficient is considerably decreased when the longitudinal pitch ratio decreases. Therefore, the effect of the longitudinal pitch should be considered when the compact heat exchange is designed.

REFERENCES

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