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Heat Transfer Studies in Tube Banks with Integral Wake Splitters

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Abstract

This paper reports the findings from heat transfer studies with the presence of extended surfaces from tube banks which are termed as integral wake splitter plates. Employing this type of fins, investigations on heat transfer characteristics on a single circular tube as well as tube banks were carried out in cross flow of air in a rectangular duct. Experiments were carried out in the Reynolds number range 5 x 10^3 to 10^5 on a single cylinder of various splitter length-to-tube diameter ratios, L/D = 0.5, 1.0, 1.5 and 2.0. Further, tube banks consisting of 12 rows and 3 tubes per row in equilateral triangle arrangements with transverse pitch to diameter ratio, a = 2, were also investigated, the banks being made up of plain tubes or tubes with splitters. Heat transfer characteristics were studied for tubes with L/D = 0, 0.5 and 1.0 under constant heat flux conditions. Tube banks with L/D = 1.0 yielded the highest heat transfer rates. Findings from this work may be adopted to be utilized in various industrial applications such as economizer of a steam boiler, air-conditioning coils or waste heat recovery systems.

Keywords: Heat transfer, integral wake splitter, tube bank

1. INTRODUCTION

Crossflow tubular heat exchangers are found in such diverse equipment as economizer of steam boiler, air conditioning coil, indirect fired heater, waste heat recovery systems and gas cooled reactors to name but few. In many applications, gases flow over the tubes. On the gas side, only small pressure drops are permissible if the operating cost due to blower work is to be limited. On the other hand, most gases, when coupled with a low
pressure drop, give a low heat transfer coefficient. High performance crossflow heat exchangers aim to remedy this situation by increasing the heat transfer coefficient without a proportionate increase in drag. Of the various approaches to achieve such an effect, one can list transverse extended surfaces, vortex generators and oval or flattened tubes.

Another method of obtaining a similar effect is to use an integral wake splitter plate which modifies the boundary layer over the tubes. These splitters have also been termed longitudinal fins. Such wake splitters under appropriate conditions can produce the highly desirable result of lower drag yet higher heat transfer, considering the fin effect. Since air flow is a strong function of heat transfer, hydrodynamics of flow in a tube bundle is studied prior to heat transfer investigations.

2. LITERATURE REVIEW

2.1. Tube and Tube Banks

The information accumulated from earlier work serves as a basis to bring into perspective the later work on tubes and tube bundles. In a bank of tubes, each tube has a neighbour in the longitudinal or lateral direction. The influence of the neighbours is considerable for close-pitched tubes. In the direction of flow, a tube is immersed in the wake of another tube. Dayoub [1] examined the two-dimensional, non-homogeneous flow over a circular cylinder immersed in the wake of another identical cylinder. He found that the aerodynamic parameters are determined by the oncoming wake flow which is characterized by variations of velocity, static pressure and turbulence quantities in both the lateral and longitudinal directions. The mean lift on the rear cylinder, when the two cylinders are not aligned with the freestream flow direction, is directed towards the centerline of the oncoming wake, which is opposite to the direction predicted by inviscid flow theory. The main factors contributing to the generation of this lift force are identified as (i) the static pressure gradient in the approaching flow and (ii) the gradient of turbulence intensities between the two sides of the rear cylinder which affects the boundary layer development on both sides, resulting in asymmetrical separation points and pressure distribution.

Aiba et al [2] conducted their studies of heat transfer around four cylinders closely spaced in a crossflow of air. The cylinders were settled in tandem with equal distance between centers. Their in-line pitch ratio was in the range 1.15 to 3.4. Their results showed that the mean Nusselt number for the four cylinders vary discontinuously at about Re = 2.1 x 10^4 with in-line pitch ratio of 1.3. Mean Nusselt number for all the cylinders is lower than that for the single cylinder in the region of Re < 2.1 x 10^4.It increases suddenly thereafter and the values for the second, third and fourth cylinders then become larger than that for the single cylinder by about 10-35%. The heat transfer and flow characteristics change drastically at Re 2.1 x 104 for the pitch ratio of 1.3. Such a Reynolds number is called the critical Reynolds number which decreases with increasing pitch ratio. Beyond the critical Reynolds number, the heat transfer rate increases suddenly on the surface facing the flow between each two cylinders.

Aiba et al [3] carried out an experimental study to investigate the heat transfer and flow around tubes in staggered tube banks in crossflow of air. The cylinder spacings examined were 1.6 x 1.6 and 1.2 x 1.2 with seven rows in each case. Their tests in the Reynolds number range of 8600 to 36000 showed that in the case of, the 1.6 x 1.6 pitch ratio, the data were utilized as input to analyses aimed at establishing performance relationship between in-line and staggered arrays. In the experiments, mass transfer measurements via the naphthalene sublimation technique were employed to determine the row-by-row distribution of the heat transfer coefficient. Fully developed conditions prevailed for the fourth row and beyond. In general, the fully developed heat transfer coefficients for the in-line array are lower than those for the staggered array, but the pressure drop is also lower.

2.2 Performance Improvement

A number of reports have been published in recent years describing attempts to modify the boundary layer on the circular cylinder in transverse air flow. Fage and Warsap [4] which showed that freestream turbulence, surface roughness and surface roughness elements all had a systematic effect on the relationship of drag coefficient to Reynolds number. Prior to this study, investigations of the boundary layer on a cylinder where the influence of Reynolds number and turbulence was demonstrated, particularly near the separation point. A most extensive and precise measurement of the local variation of the heat transfer coefficient around a cylinder was made by Schmidt and Werner [5] in which they reported the effect of surface projections on heat transfer for circular cylinders.

Gorla [6] formulated an expression for the eddy diffusivity induced by the freestream turbulence intensity and integral length scale. Lenticular tubes in heat exchangers have been investigated by Ruth [7].

When a circular cylinder is yawed, the flow over it is similar to that over elliptic cylinders. Recognizing this, Ota et al [8] made a study of heat transfer and flow around elliptic cylinders at various angles of attack. The static pressure and heat transfer coefficient around the periphery were measured. While the pressure drag coefficient of the elliptic tube is lower, even the lowest Nusselt number on it was higher than the mean Nusselt number on a circular cylinder of equal circumferential length.

2.3 Tubes with Wake Splitters

The conventional augmentation techniques aim to increase heat transfer without a proportionate increase in pressure drop. Another approach to augmentation of heat exchanger performance is to reduce the pressure drop without a proportionate reduction or even no reduction in heat transfer. This approach is of particular value for gases such as air, since the cost. This was attributed to the fact that the splitter plate altered the downstream flow to a separated and reattached boundary layer enclosing a region of reverse flow between it and the surfaces of the plate and the cylinder. Gerrard [9] measured the frequency of vortex shedding from a circular cylinder in crossflow at a Reynolds number of 2 x 10^4 with splitter plates of different lengths up to a maximum of 2D attached to the cylinder. As the splitter plate length was increased, the Strouhal number was found to decrease to a minimum for a plate length of approximately D and then to increase as the splitter plate length was increased to 2D.

Apelt and Isaacs [10] found that very short splitter plates attached to a circular cylinder in crossflow of water at a Reynolds number of 1.58 x 10^4 caused large reductions in drag. With a splitter plate of D/8 the pressure drag was reduced to 83% of the value measured for the plain cylinder while for the splitter plate length of D the pressure drag was reduced to 68%. But only relatively small further changes in the drag resulted when the splitter plate length was increased in steps up to 8D.

2.4 Present Work

The previous work on tube banks, with longitudinal fins [11, 12] reported gross results with little explanation of the observations. Even more remarkable is the fact that in neither case was it recognized that the rearward fin was cting as a wake splitter. The present work studies the effect of an integral wake splitter in heat removal. Its influence over a range of practical Reynolds numbers and for several splitter plate lengths are investigated for a more complete understanding.

3. EXPERIMENTAL INVESTIGATIONS

3.1Experimental Set-up

The experimental investigations were carried out in an open circuit low speed wind tunnel which is schematically shown in Figure 1. Ambient air was taken from a medium pressure centrifugal blower and the airflow was controlled by a throttle valve at the inlet and a butterfly valve at the delivery side.

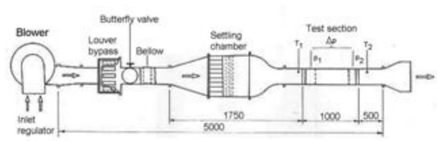

Fig. 1 Schematic diagram of the experimental set-up

The test section has a cross sectional area of 152.4 mm x 157 mm and is 1000 mm long with aluminum walls. Static pressures and temperatures are measured at the inlet and exit of the test section respectively. The investigations were carried out on plain cylinders and cylinders having splitters in the downstream wake. The cylinder diameter was 25.4 mm and 157mm in length while the splitter lengths used were, L = 0.5D and 1D and the plate thickness was 1.6 mm. It was placed in longitudinal slot milled into the base cylinder at $\Phi = 180^o$ as shown in Figure 2.

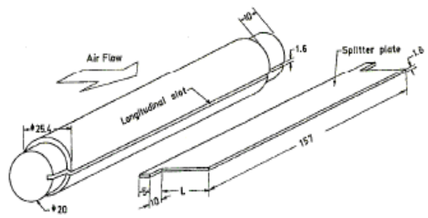

Fig. 2 Base cylinder with longitudinal slot for splitter plate assembly

The splitter plate has an interference fit with the cylinder slot for rigidity. The experiments were carried out with the plain tube bank and tube banks with splitters of L/D = 0.5 and 1.0 consisting of 12 rows and 3 tubes per row in equilateral triangle arrangement. The dimensionless transverse pitch was 2.0 and longitudinal pitch b = 1.73. Figure 3 shows schematically the arrangement of the tubes where half tubes were arranged to minimize the bypass flow near the walls. The figure also illustrates the position of different lengths of the splitter in the staggered arrangement.

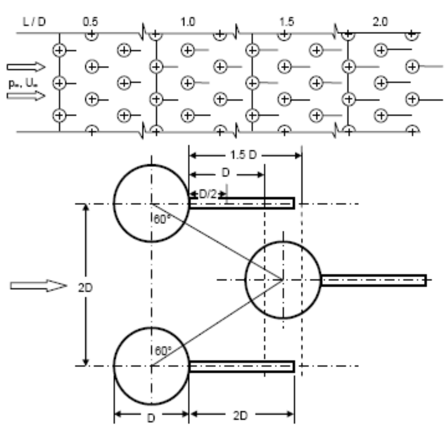

Fig. 3 The position of different lengths of the splitter plate in the staggered tube bank

3.2 Heat Transfer Measurement

Heat transfer measurements were done by using thin constantan foils, of 0.05 mm thick, were applied to the surfaces of the test cylinder made of bakelite and heated direct current to give a constant wall heat flux condition. For measuring the local heat transfer coefficient at different rows in a tube bank, the constant heat flux technique was chosen for its greater convenience as well as its ability to provide point-wise local data. Local heat transfer coefficient values are measured from the forward stagnation point on the tubes. Iron-constantan thermocouples 0.2 mm in diameter were embedded on the cylinder and the splitter surfaces in order to measure the wall temperature. In the 80° to 120° region where the greatest likelihood of gradients would occur, the thermocouples were positioned at intervals of 10° around the test cylinder while the other thermocouples were placed at 15° intervals. All the test cylinders were heated under the condition of the same heat flux, 2.06 Kw/m2. Local heat transfer coefficient, $\alpha_{\varnothing}$ and Nusselt number are defined respectively as follows:

$$\alpha_{\phi} = q''/(T_w - T_{\infty}) \tag{1}$$

$$Nu_{\phi} = \alpha_{\phi} D/\lambda \tag{2}$$

where T_w is surface temperature, T_{∞} and λ is temperature and thermal conductivity of air respectively.

4. RESULTS AND DISCUSSION

Heat transfer characteristics of tubes and tube banks with splitters, L/D = 0.5 and 1.0 are investigated. The results which concerned with local Nusselt numbers for both cases of single tube and tube banks with splitter plates are discussed. The Nusselt numbers presented in this section are calculated from the heat transfer coefficient which is measured as a function of angle, Ø, measured from the forward stagnation point of the tubes.

4.1 Single Tube with Splitter Plate

Unlike flow over a plain tube or tube banks, the presence of splitter plates has shown interesting findings in terms of heat transfer characteristics. Being immersed in the wake, the splitter could be expected to give Nusselt numbers as low as that on the rear of the tube. However, this does not appear to be the case at Reynolds numbers higher than 1.35 x 10^4. While there is a continuity of Nusselt number values at the corner formed by the cylinder and the splitter (Ø ≈170°), the coefficients rise towards the fin tip. In general, the Nusselt numbers on the splitter are higher than on the rear of the tube. However, for the single tube with splitter, the fin tip Nusselt number is always lower than the forward stagnation point Nusselt number. Figure 4 compares the local Nusselt number distribution for L/D = 0, 0.5 and 1.0 at a Reynolds number of 4.8 x 10^4.

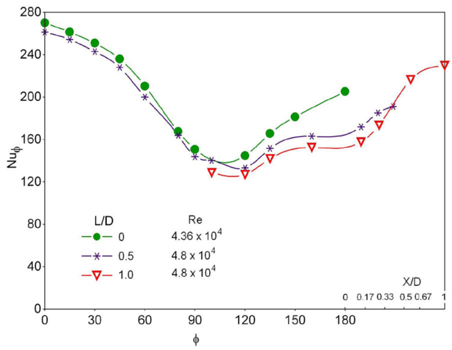

Fig. 4 Comparison of local Nusselt number over a tube with different splitter lengths

The further drop in Nu on the rear surface of the tube for L/D = 1.0 is proof of the greater wake attenuation by the longer splitter. The fin base in the L/D = 1.0 case, starts out at a lower value than for L/D = 0.5, but its tip reaches almost the same value as L/D = 0.5 at a lower Reynolds number and surpasses it at the higher Reynolds number shown in the figure.

4.2 Tube Banks with Splitter Plates

The local Nusselt numbers are investigated in terms rows of tube in the tube banks. The findings lead to the conclusion that fins perform even better in a tube bank than on a single cylinder, due to the main flow being directed on to them by the downstream tube row.

1) First Row in Tube Bank

The heat transfer from a tube in the first row is similar to that of a single tube at low Reynolds number, the heat transfer in the front portion being higher than in the rear. With increasing Re, the heat transfer in the rear portion of the cylinder increases in relation to the front. Figure 5 shows the heat transfer in the first row for all the tubes with and without splitter.

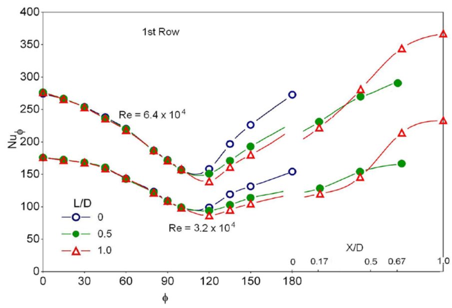

Fig. 5 Local Nusselt number at the first row for different L/D

For clarity of presentation, the results for two Reynolds numbers are shown. The local Nusselt number distribution is practically identical up to an angle of 90° for all the tubes while the heat transfer in the rear portion of the plain tube is higher than that of L/D = 0.5 and 1.0. Furthermore the tube of L/D = 1.0 indicates the lowest heat transfer in the rear portion, due to the maximum reduction of turbulence in the wake by the splitter. Measurements could not be made in the vicinity of the 180° location, as the splitter was joined to the tube there. This region is featured by a break in the curves for L/D = 0.5 and 1.0. However, the curves drawn smoothly through the experimental points seem to indicate continuity.

The fin tip Nusselt number is higher than the forward stagnation point value for L/D = 1.0 and somewhat lower in the case of L/D = 0.5 for the Reynolds numbers shown. On the other hand, the fin tip value for L/D = 0.5 was higher than for forward stagnation for Re = 7.2 x 10^4, which was characterized by transition to turbulence in the boundary layer. This increase of fin Nusselt number is due to increase of turbulence in the wake with Reynolds number. If fins of effectiveness close to unity are considered, the L/D= 1.0 situation will be more advantageous for heat transfer in the first row, over a Reynolds number range higher than 2 x 10^4.

2) Second Row in Tube Bank

Figures 6 and 7 present the local Nusselt numbers at two Reynolds numbers for several rows of a tube bank with L/D = 1.0 and several L/D values for the second row respectively.

For L/D = 0.5, the results were intermediate between those for L/D = 0 and 1.0, as is to be expected. For all the tube banks, the heat transfer coefficient near the rear stagnation point is the highest at the second row. In this row too, the Nusselt number in the forward portion shows little difference for different L/D for a given Reynolds number.

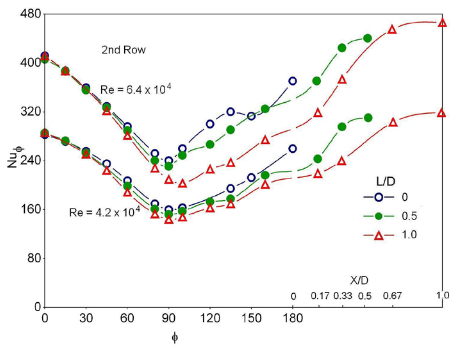

Fig. 7 Local Nusselt number in the second row for different L/D

For both values of L/D, the heat transfer coefficient for the fin tip is higher than that of the corresponding forward stagnation point (Figure 7). Hence the overall heat transfer coefficient for the tube with splitters, L/D = 0.5 and 1.0, is higher than that of the plain tube in this row.

3) Third and Inner Rows in Tube Bank

For all the tube banks, the local Nusselt number distribution shows little difference for the third and inner rows. The twin mimima which are characteristic of the third row of the plain tube bank are also clearly observed in tube banks with splitters, but at a higher Reynolds number than the plain tube bank. For the sixth row, the Nusselt number distribution for L/D = 0, 0.5 and 1.0 are similar to, but somewhat lower than those obtained for the third row. The longitudinal pitch ratio 1.73 used in this work seems to give the effect of highest coefficients in the third row. In the case of tube banks with splitters, Figure 8 shows that the Nusselt numbers at forward stagnationare higher than that of a single tube (Figure 4). This is evidently due to flow air directed to the tubes in the 3 row onwards.

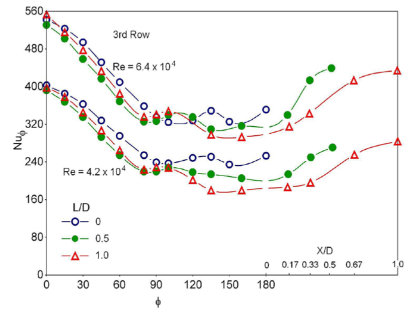

Fig. 8 Local Nusselt number at various rows of a tube bank for L/D = 1.0

The mean Nusselt number for the tubes in all the rows of the tube banks with splitters, L/D = 0.5 and 1.0, were lower than that of the plain tube. However, heat transfer from the fin part compensates for the reduction over the base tube. As a result, heat transferred by the L/D = 0.5 fin is not very different from the tube with no splitter. For L/D = 1.0, an even more favourable situation emerges, in that the tube in the bank yields a higher heat transfer than a plain tube in a bank, despite the greater reduction of pumping power.

5. CONCLUSIONS

From the present work, it can be concluded that the local Nusselt numbers on the rear of a single tube are depressed when a splitter is attached to the tube. The reduction being greatest at rear stagnation but the Nusselt numbers rise steadily from base of the splitter plate towards the tip. In the first row of tube banks, heat transfer behaviour is similar to that of the single tube. The fin tip Nusselt numbers, at the middle to high range of the investigated Reynolds numbers, are higher than the forward stagnation point values. For all tube banks, Nusselt number at the rear stagnation point of any row is highest at the second row due to the greatest intensity of turbulence at this location.

This effect, which is responsible for the third row Nusselt numbers being the highest, also produces the highest heat transfer coefficients at the tip of the second row fins. For the third row the fin tip heat transfer coefficients are always lower than the forward stagnation point, provided turbulent transition has not occured. In the inner rows, heat transferred by the tubes with splitters can be as much as or more than the plain tubes, depending on the Reynolds number of flow. The beneficial effect of the splitter plate on a single cylinder has been attributed to the attenuation of vortex shedding in the wake. That similar benefits would be derived from splitter plates attached to the tubes of a bank is not expected since the wake flow in a tube bank after the first row is highly disorganized.

REFERENCES

[1] A.H. Dayoub, "Aerodynamic Forces in a Cylinder Structure in the Wake of another Cylinder Structure," in *Applied Scientific Research*, vol. 39, 1982, pp. 3–20.

[2] S. Aiba, T. Ota, and H. Tsuchida, "Heat Transfer of Tubes Closely Spaces in In-line Bank", Int. *Journal of Heat & Mass Transfer*, vol.23, 1982, pp. 311–319.

[3] S. Aiba, T. Ota, and H. Tsuchida, "Heat Transfer around Tubes in Staggered Tube Banks", *Bulletin of JSME*, vol.25, 1982, pp. 927–933.

[4] A.Page and J.H.Warsap, "Effect of Roughness and Stream Turbulence on the Drag of Circular Cylinders", 1930, British Aero. Res. Council, R&M, pp1283.

[5] F. Schmidt and K. Werner, "Heat Transfer Over a Heated Cylinder in Crossflow", *NACA Technical Memo 1050*, 1943.

[6] R. S.Gorla, "Stagnation Point Heat Transfer Augmentation Due to Freestream Turbulence, *Applied Scientific Research*, vol.39, 1982, pp 143-153.

[7] E.K. Ruth, "Experiments on a Crossflow Heat Exchanger with Tubes of Lenticular Shape, Journal of Heat Transfer, Trans. ASME, vol.105, 1983, pp 571-575.

[8] T. Ota, H. Nishiyama, and Y. Taoka "Heat Transfer and Flow around an Elliptic Cylinder", *Int. Journal of Heat & Mass Transfer*, vol.27, 1982, pp 1771-1779.

[9] J.H. Gerrard, "The Mechanics of the Formation of Vortices Behind Bluff Bodies", *Journal of Fluid Mech*., vol.25, 1966, pp 401-413Y.

[10] C.J.Apelt and L.T. Isaacs, "Effect of Splitter Plates Placed in the Wake of Bluff Cylinders",1970, CAARC Symp. on Separated Flows and Wakes, Univ. of Melbourne.

[11] E.M.Sparrow, and S.S. Kang "Longitudinally Finned Crossflow Tube Banks and Their Heat Transfer and Pressure Drop Characteristics", *Int. Journal of Heat & Mass Transfer*, vol.28, 1985, pp 339-350.

[12] J. Qiu, and Z. Chen, "Heat Transfer and Pressure Drop of A Staggered Longitudinally Finned Tube Bank", *Int. Heat Transfer Conf.*, San Fransisco.