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HEAT TRANSFER IN A BANK OF TUBES WITH INTEGRAL WAKE SPLITTERS

Aly Mustafa Ibrahim[1], Mohammad Kamil[2], Mohd Faizal Mohideen Batcha[2], Vijay R. Raghavan[2]
Faculty of Mechanical Engineering
[1]University of Mansoura, Egypt
[2]Universiti Tun Hussein Onn, Malaysia
E-mail: vijay@uthm.edu.my

ABSTRACT

The paper deals with the investigations on heat transfer characteristics of a circular tube as well as tube banks with integral downstream splitter plates in cross flow of air in a rectangular duct. The experiments were carried out in the Reynolds number range 5×10^3 to 10^5 on single plain cylinder and 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 arrangement 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. They were also superior to single tubes with L/D = 1.0.

INTRODUCTION

Crossflow tubular heat exchangers are found in such diverse equipment as economizer of steam boilers, air conditioning coils, indirect fired heaters, waste heat recovery systems and gas cooled reactors to name but a 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.

LITERATURE REVIEW

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 of pumping of gases across a given pressure difference is higher than for liquids. The splitter plate in the wake of the tube reduces the pressure drop by modifying the wake flow.

The study of the effect of splitter plates is relatively more recent, compared to study of plain cylinder and banks of plain tube in crossflow. The first recorded of investigation of flow around a tube with non-integral wake splitter was by Roshko [1]. It included the placement of a splitter plate in the wake of a circular cylinder in a two-dimensional crossflow at a Reynolds number of 1.34×10^4. He reported experiments on the turbulent structure and velocity distribution in the wake and found that they had a large influence on the overall drag coefficient. It was found that a splitter plate of length 5D in contact with the cylinder inhibited vortex formation and caused the pressure drag experienced by the cylinder to be reduced approximately to 63% of the value for the cylinder alone. A splitter plate of length 1.14D in contact with the cylinder did not inhibit vortex formation, but caused an increase in base pressure and a reduction in the Strouhal number as compared with the value for the plain cylinder. When a short splitter plate was moved downstream leaving a gap between it and the cylinder, Roshko [1] observed a complicated and interesting sequence of changes in the base pressure on the cylinder and in Strouhal number.

Seban and Levy [2] studied the effect of a very long splitter plate on the heat transfer from a cylinder in crossflow. The investigation was also conducted with the plate separated from the cylinder by a small gap. It was found that the heat transfer was reduced significantly on the rear portion of the cylinder with the forward portion relatively unaffected. 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 [3] measured the frequency of vortex shedding from a circular cylinder in crossflow at a Reynolds number of 2×10^4 with the 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 approximately D and then to increase as the splitter plate length was increased to 2D.

Apelt and Isaacs [4] found that very short splitter plates attached to a circular cylinder in crossflow of water at a Reynolds number of 1.58×10^4 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.

Geiger and Collucio [5] experimentally investigated the heat transfer and drag behaviour of a heated circular cylinder with an integral heat conducting downstream splitter plate in transverse airflow. The model was a thick-walled aluminium cylinder internally heated by electric resistance heaters. They developed an advantageous situation when the splitter plate was made of a good conductor and was made integral with the tube. Splitter plates of optimum length reduced the pressure drop. The advantage lay in the fact that the integral splitter plate also served as fin and hence the relationship between changes in pressure drop and heat transfer became significantly favourable. The test results indicated that the addition of a splitter plate improved heat transfer characteristics with increased plate length, the most dramatic increase in Nusselt number being at the transition from a plate length of D/3 to 2D/3 at the higher velocities. Drag reduction was, in general, also evident with increased plate length.

The foregoing studies are concerned with a single tube. Relating to a tube bank, only two studies on pressure drop and heat transfer are reported. Sparrow and Kang [6] performed mass transfer experiments on tube banks in crossflow in which individual tubes were equipped with longitudinal fins at the front, at the rear or both front and rear. The range of Reynolds number investigated was 1200–8600. The mass transfer analogy using naphthalene sublimation was employed. The enhancement for various bank geometries was compared at fixed pumping power, fixed pressure drop and fixed mass flow rate. Finning was found to be especially advantageous when the comparison is made at fixed pumping power. In the bundle of equilateral triangular pitch of P/D = 2 and splitter plate L/D = 1.0 that they used, the forward fin configuration yielded the maximum increase in mass transfer but also produced a lower pressure drop than the unfinned tube bank.

The second study, by Qiu and Chen [7] involved detailed row by row determination of mean Nusselt number via the mass transfer analogy for a staggered tube bank with longitudinal fins at the front and rear. Studies were carried out in the Reynolds number range 6000–20000. Quasi–local convective heat transfer coefficients for the front fins, base tube as well as rear fins were deduced. Five different fin lengths in the range 0.438 < L/D < 1.06 were used. Flow visualization studies were also performed to help explain and rationalize the heat transfer results. A correlation was given for the effect of fin L/D ratio on mean Nusselt number. The bundle pressure drop increased slightly with the fin height. The quasi-local Nusselt number for the front fin was higher than that of a bare tube mean Nusselt number only for the shortest fin. The quasi-local values for the base tube and rear fin were below that for the bare tube.

EXPERIMENTAL SET–UP

The experimental investigations were carried out in an open circuit low speed wind tunnel, specially built for this work. The experimental setup is shown schematically in **Figure 1** which illustrates a general view of the testing installation. The test section has a cross sectional area of 152.4 mm x 157 mm and is 1000 mm long with aluminum walls, **Figure 2**

The investigations were carried out on plain cylinder and cylinders having splitters in the downstream wake. The cylinder diameter D was 25.4 mm and the cylinder length l, 157mm. The splitter lengths used were, L = 0.5D and 1D and the plate thickness was 1.6 mm. It was placed in a longitudinal slot milled into the base cylinder at $\Phi = 180^o$ as shown in **Figure 3**. 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 ***a*** was 2.0 and longitudinal pitch b = 1.73. **Figure 4** shows schematically the arrangement of the tubes. Half tubes were arranged to minimize the bypass flow near the walls. **Figure 5** illustrates the position of different lengths of the splitter in staggered arrangement.

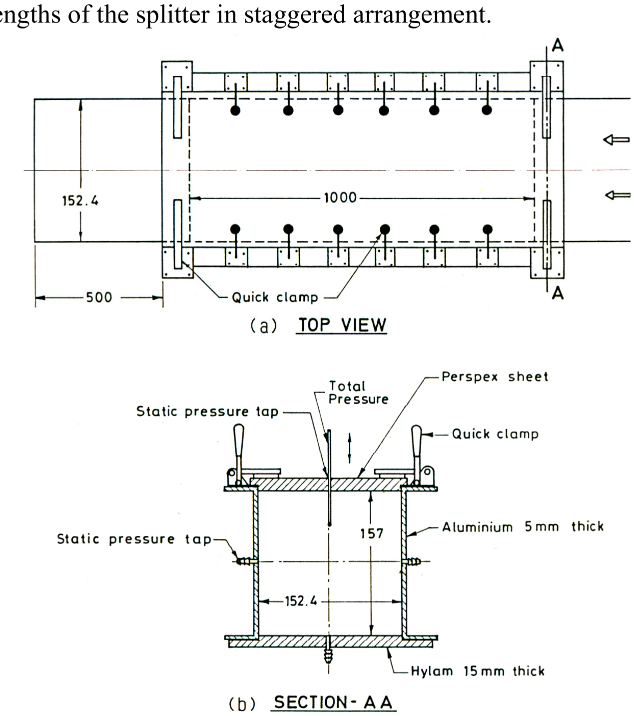


Figure 2: Schematic representation of test section
(a) top view, (b) cross sectional view

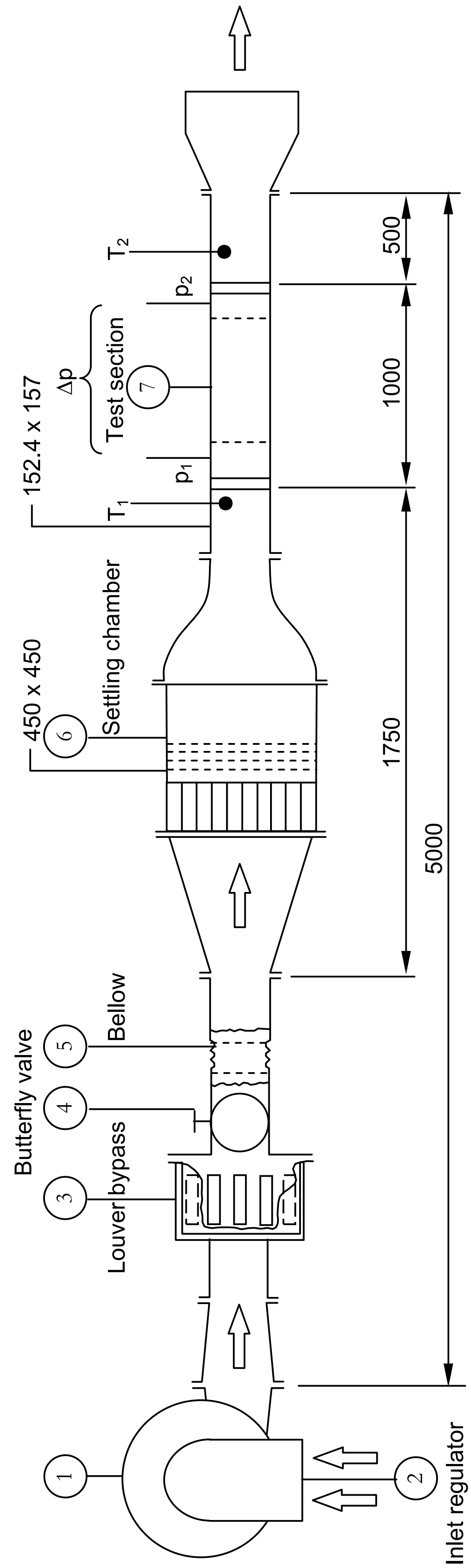
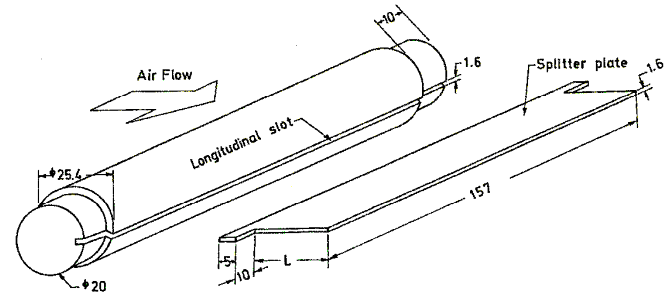


Figure 1: Schematic diagram of the test set-up



RESULTS AND DISCUSSION

Figure 3: Base cylinder with longitudinal slot for splitter plate assembly

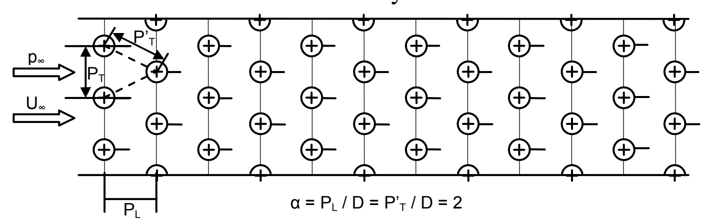


Figure 4: Flow configuration of staggered tube bank

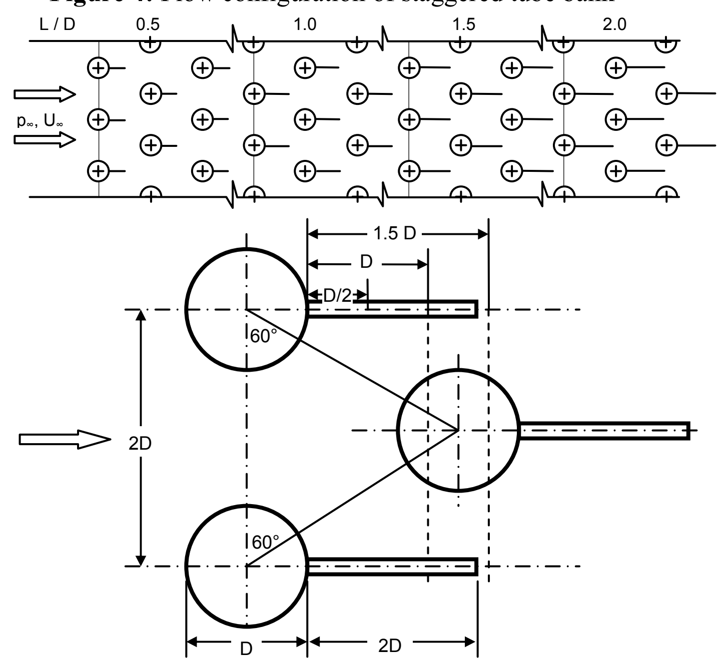


Figure 5: The position of different lengths of the splitter plate in the staggered tube bank

SINGLE CYLINDER

Local Nusselt Number Distribution

Figure 6 and **7** present the local Nusselt number as a function of angle measured from the forward stagnation point, for single tubes at L/D = 0 and L/D = 0.5 respectively. Plotted for three Reynolds numbers each, spanning the Re range of experiments, the two figures show certain points of commonality. The Nusselt number distributions at similar

Reynolds numbers are quite similar. Transition to turbulence sets in for both cases at Reynolds number well below 10^5, obviously due to turbulence level of 0.6-0.7%, measured in the approach flow in the wind tunnel. The minima in Nusselt number curves occur at about 110^0 for the lower Reynolds numbers where no turbulence transition has occurred.

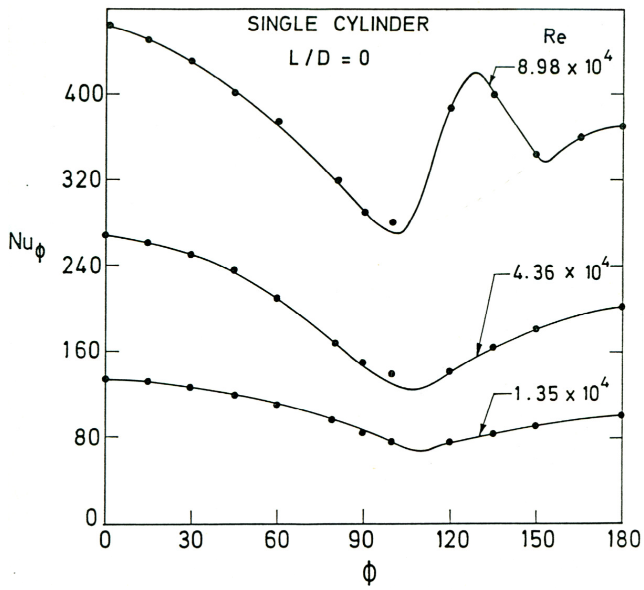


Figure 6: Local Nusselt number around single plane tube at various Reynolds numbers

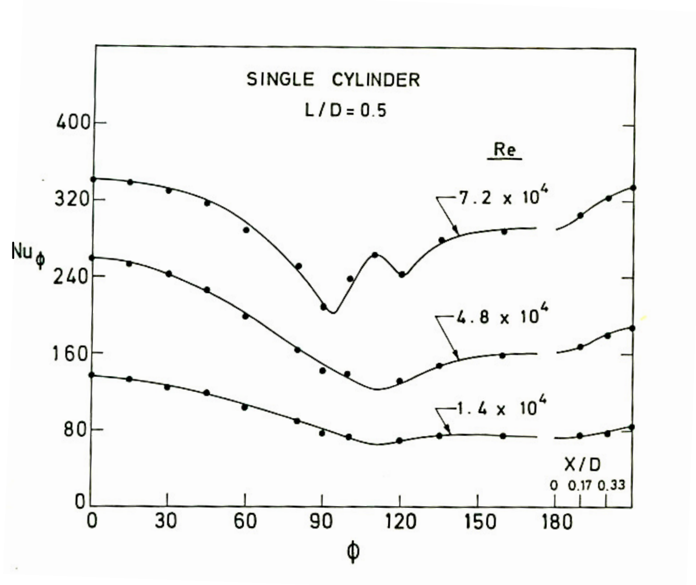


Figure 7: Local Nusselt number around the cylinder and along the splitter plate at various Reynolds numbers

Being different from observations of the separation point from hydrodynamics data [8], this minimum at 110^0 has been recorded for constant heat flux condition also by Boulos and Pei [9], Perkins and Leppert [10], Matsui et al [11] and Dyban and Epick [12]. The location of the minimum Nusselt number seems to be unaffected by the splitter plate. Qualitatively, this is similar to the behavior of the separation point locations obtained in the hydrodynamics studies on single cylinders [8].

The points of deviation between these two figures occur at the rear of the cylinder. It is expected that the forward stagnation point and the front half of various L/D tubes will give essentially the same local Nusselt numbers at the same Reynolds number. However for the L/D = 0.5 tube as compared to the L/D = 0 case, after separation, the local Nusselt number does not rise as steeply on the rear of the tube. In **Figure 7** a break is shown in the curve in Nusselt number at $\Phi = 180^0$ to indicated that this point does not exist on L/D = 0.5 tube, having been covered by the splitter rib. Typically, at a Reynolds number of 4.8 x 10^4, the local Nusselt number is lower by 20% in the region of rear stagnation point.

Nusselt number distributions over the splitter, not reported hitherto, yield interesting results. Being immersed in the wake, the splitter, also termed as longitudinal fin could be expected to give Nusselt numbers as low as that on the rear of the tube. This does not appear to be the case at Reynolds number 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 ($\Phi = 170^o$), 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 cylinder, the fin tip Nusselt number is always lower than the forward stagnation point Nusselt number.

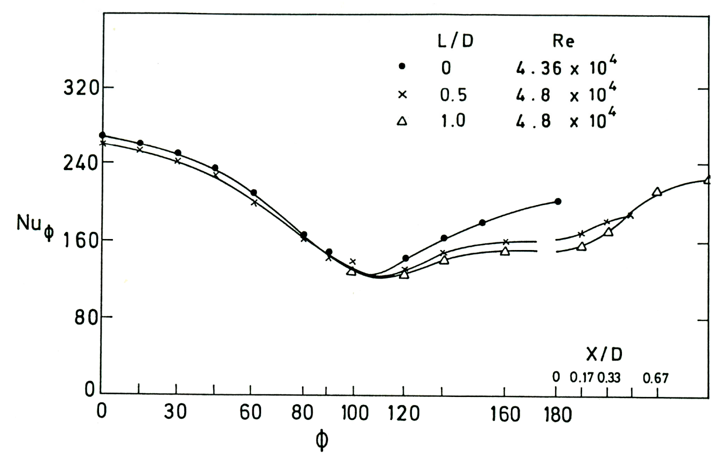


Figure 8: Comparison of local Nusselt number over single cylinders having different splitter lengths

Figure 8 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. The further drop in the Nusselt number 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 L/D = 1.0 case, starts out at a lower value than 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.

Mean Nusselt Number

Mean values of the Nusselt number over sections such as the front half, the rear half and the fin could be obtained by integration of the local Nusselt number over those sections. The sectional mean Nusselt number, also termed as quasi-local values, are plotted for all L/D in **Figures 9** and **10**.

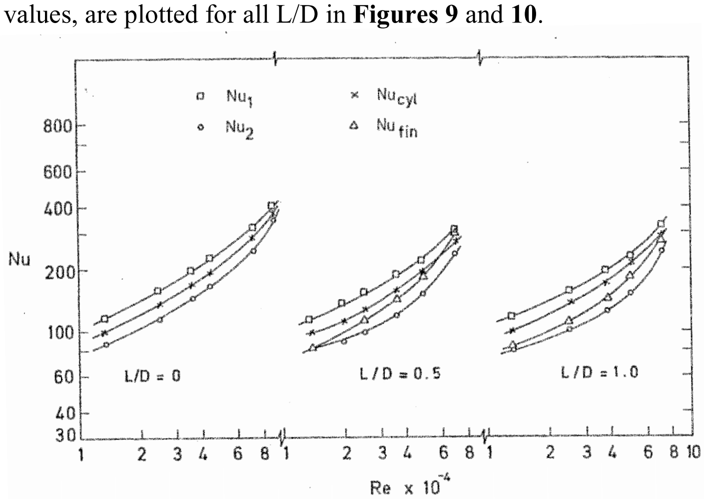


Figure 9: The sectional mean Nusselt number versus Reynolds number for single cylinder

Average Nusselt number as a function of Reynolds number has been given by Morgan [13] for a plain single cylinder and by Geiger and Collucio [5] for single cylinders with splitter plates. In the latter work, the lowest Reynolds number is 5 x 10^4. Morgan [13] recommends a slope of 0.814 for his Re range. The present results commence from Re = 1.3 x 10^4. Hence the change in slope of the Nu_m vs Re curve is to be anticipated. These faired curves can be seen to have an increasing slope with increasing Reynolds number.

Sectional mean heat transfer coefficients were calculated for the front half of the cylinder (0<Φ<90), the rear half of the cylinder (Φ> 90) and the fin portion. These results, given in **Figure 9** provide an idea of the relative thermal performance of each of these sections. For the plain cylinder, the mean Nusselt number for the rear half, designated as Nu_2 is lower than Nu_1, the mean Nusselt number for the forward half, by about 25% over most of the range. Once turbulent transition sets in, Nu_2 tends to become equal to Nu_1.

The front Nusselt number, Nu_1 for L/D=0.5 and 1.0 are practically the same as for the plain single cylinder, L/D = 0. The rear Nusselt numbers Nu_2 for the finned tube, L/D = 0.5 and 1.0 are lower than the Nu_2 values for the plain tube, L/D = 0. They are also lower than respective Nu_1 values by about 35%. As stated earlier, this decrease in Nu_2 is attributable to the suppressed activity in the wake due to the action of the splitter plate and is responsible for the lower values of mean Nusselt number on tubes when splitters are added.

The mean Nusselt numbers of both fins, Nu_{fin} for L/D = 0.5 and 1.0 have nearly the same values. However Nu_{fin} values are higher than Nu_2 values at the same Reynolds number and are only slightly lower than the mean Nusselt number for the cylinder, Nu_{cyl} which includes the front and rear halves (**Figure 9**). Nu_{fin} rises more steeply with Reynolds number than Nu_2 or Nu_{cyl}.

A comment is in order in respect of the determination of Nu_{fin}. In the present experiments, the fin surface experienced constant heat flux. No fin efficiencies were considered, i.e: the fin was taken to have an efficiency of unity. This condition will be approached for fins of good conducting materials having a low heat transfer coefficient over their surface. For fin efficiencies of less than unity, this can be accounted for suitably by weighting the fin Nusselt number. It is worth noting that Sparrow and Kang [6] as well as Qiu and Chen[7] used naphthalene sublimation over the fin too. A fin efficiency of unity is inherent in their calculations since the naphthalene concentration on the fin surface is constant unlike its thermal analogue, where the driving temperature difference decays along the fin.

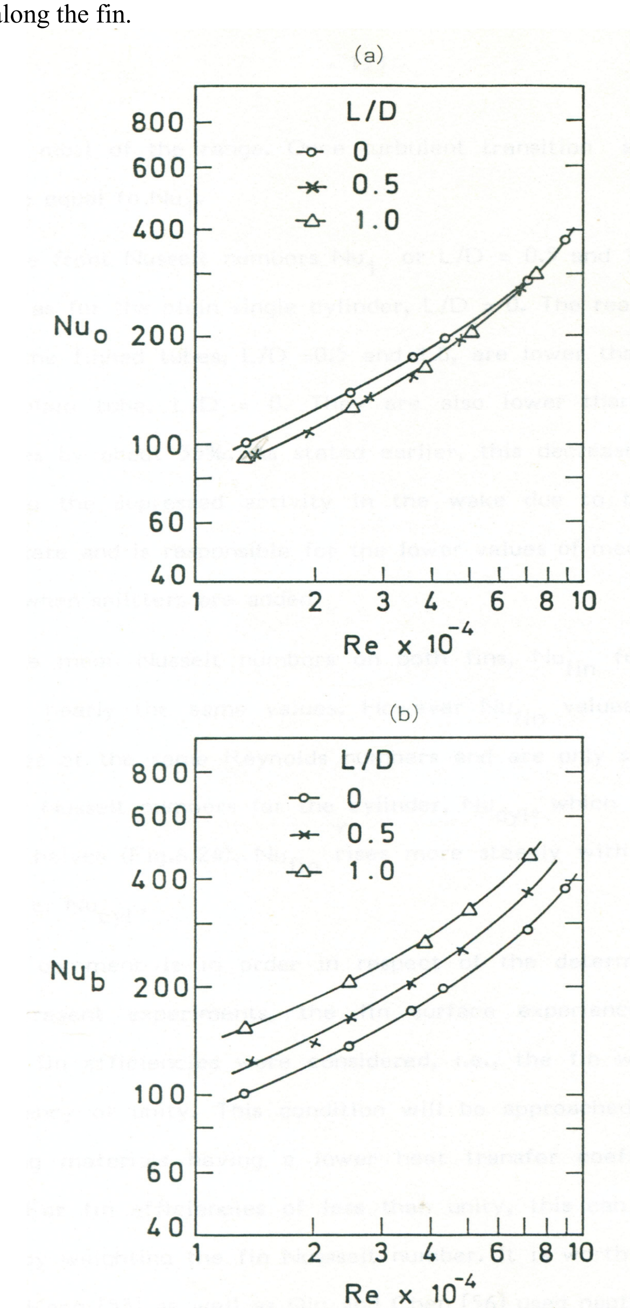


Figure 10: (a) Average Nusselt numbers based on cylinder and plate surface area
(b) Average Nusselt numbers based on cylinder surface area

Average Nusselt numbers for the whole tube, including the fin portion where relevant, are plotted in **Figure 10**. Nu_o is the mean Nusselt number reckoned on bare tube area. Comparison of Nu_o shows that the curve for L/D = 0 lies highest and those for L/D = 0.5 and 1.0 lie slightly lower. In fact, the small difference between L/D =0.5 and 1.0 is not discernible on the plot. The L/D = 1.0 tube picks up on the fin whatever it has yielded to the L/D = 0.5 tube on the rear of the tube, so that overall coefficients are nearly the same. The superior performance of the L/D = 1.0 tube can be gauged from the fact that its drag [8] is the least among all the tubes without any major sacrifice in the heat transfer performance. The Nu_b plots, based on bare tube area, are an index of the increase in the quantity of heat transferred. Here too, L/D = 1.0 has the highest performance.

TUBE BANKS

This part of the results is concerned with local Nusselt numbers at several rows and different L/D values of a tube bank. The hydrodynamic advantages of a tube bank with splitters, L/D = 1.0 have been set out earlier [8]. This section studies the local Nu variations and will lead to the conclusion that the fin performs even better in a tube bank than on a single cylinder, due to the main flow being directed on them by the downstream tube row.

First Row in Tube Banks

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 11** shows the heat transfer in the first row for all the tubes with and without splitter. For clarity of presentation, the results for two Reynolds numbers are shown. The trends are similar for the other Reynolds numbers for all the tubes and the following comments apply to the entire range of Reynolds numbers investigated.

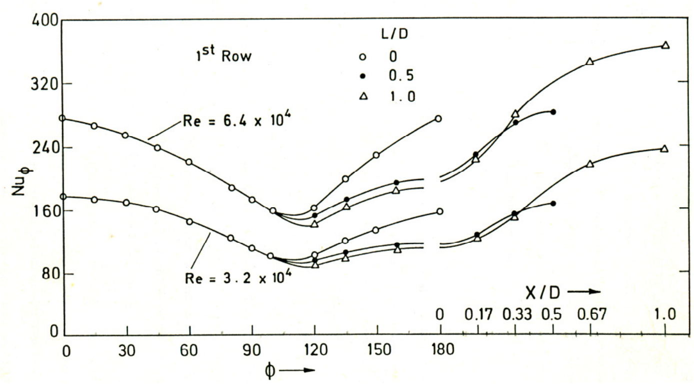


Figure 11: Local Nusselt number at the first row for different L/D

The local Nusselt number distribution is practically identical up to an angle of 90^0 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. This result is agreement with the conclusions from the hydrodynamic studies. Measurements could not be made in the vicinity of the 180^0 location as the splitter was joined to the tube there. This region is featured by a break in the curve for L/D = 0.5 and 1.0.

However, the curves drawn smoothly through the experimental points seem to indicate a continuity as in the case of the studies of hydrodynamics and heat transfer for a single cylinder. 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. At the lowest Reynolds number investigated for tube bundle heat transfer, Re = 1.35 x 10^4, the fin tip Nusselt number was lower than the forward stagnation point value even for L/D = 1.0. On the other hand, the fin tip value even for L/D= 0.5 was higher than for forward stagnation point for Re = 7.2 x10^4, which was characterized by transition to turbulence in the boundary layer. This increase of the 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 the heat transfer in the first row, over a Reynolds number range higher than 2 x10^4.

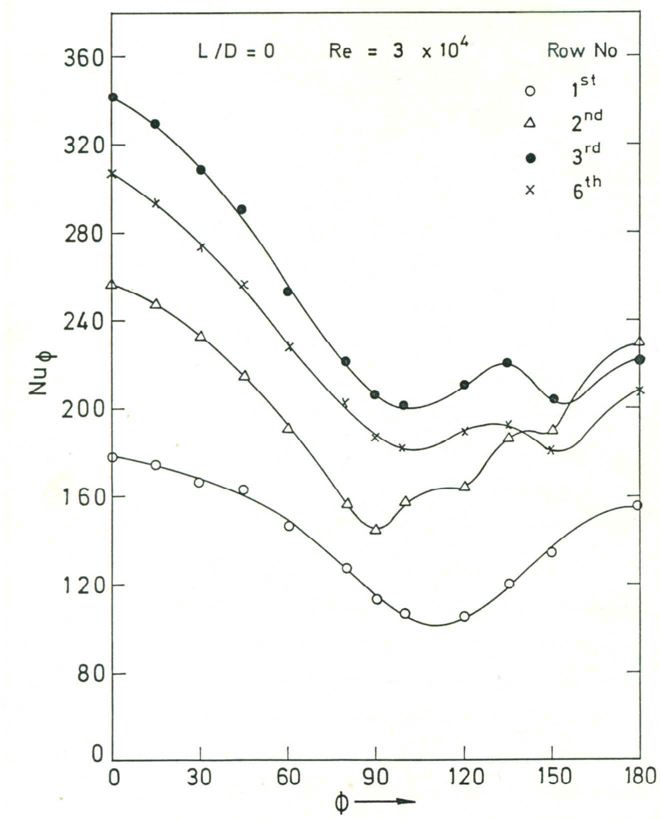


Figure 12: Local Nusselt number for the plain tube

Second Row

Figure 12 and **13** present the local Nusselt numbers at two Reynolds numbers for several rows of a plain tube bank and a bank with L/D=1.0 respectively. For L/D = 0.5, the results were intermediate between those for L/D = 0 and 1.0, as been expected. For all tube banks, the heat transfer coefficient near the rear stagnation point is the highest at the second row. This is a consequence of the highest turbulence and lowest base pressure in the wake of the second row. In this row too, the Nusselt number in the forward portion shows little difference for different L/D for given Reynolds number. On the rear of the second row tubes, attenuation of turbulence by the splitter causes a lower heat transfer coefficient than in the plain tube.

For both values of L/D, the heat transfer coefficient for the fin tip is higher than of the corresponding forward stagnation point (**Figure 14**). 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.

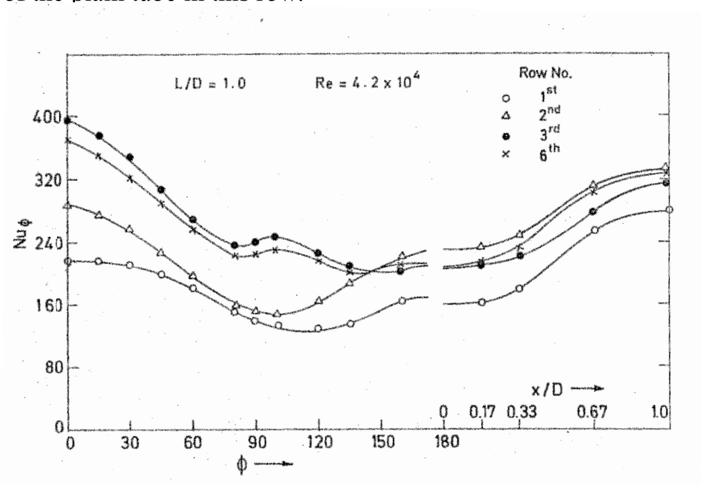


Figure 13: Local Nusselt number at various rows of a tube bank, L/D = 1.0

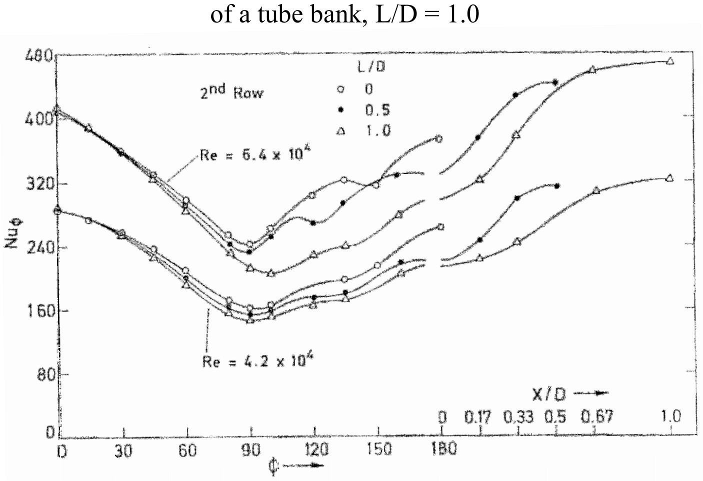


Figure 14: Local Nusselt number in the second row for different L/D

Third and Inner Rows

For all the tube banks, the local Nusselt number distribution shows little difference for the third and inner rows. The twin minima which are characteristic of the third row of plain tube bank are also clearly observed in the tube banks with splitters, but at a higher Reynolds number than the plain tube bank. As reported by Aiba et al [14], the highest Nusselt number occurs in the third row of a plain tube bank, at least for their pitch ratio. 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 by Aiba et al [14] seem to give this effect of the highest coefficient in the third row. Other cases of plain tube bundles have been reported in which the coefficients increase to an asymptotic value in the inner rows [15].

In the case of tube banks with splitters the Nusselt numbers at the forward stagnation are lower than that of a plain cylinder (**Figure 15**). This is evidently due to reduction in turbulence induced by the splitter in the second row. On the rear half of the tube in the third row, heat transfer for a plain tube is highest and it is least for L/D = 1.0. This again is a manifestation of the turbulence reduction by the splitter in this row. The fin tip heat transfer coefficient is lower than that for the forward stagnation point. It is also lower than the fin tip heat transfer coefficient of the second row.

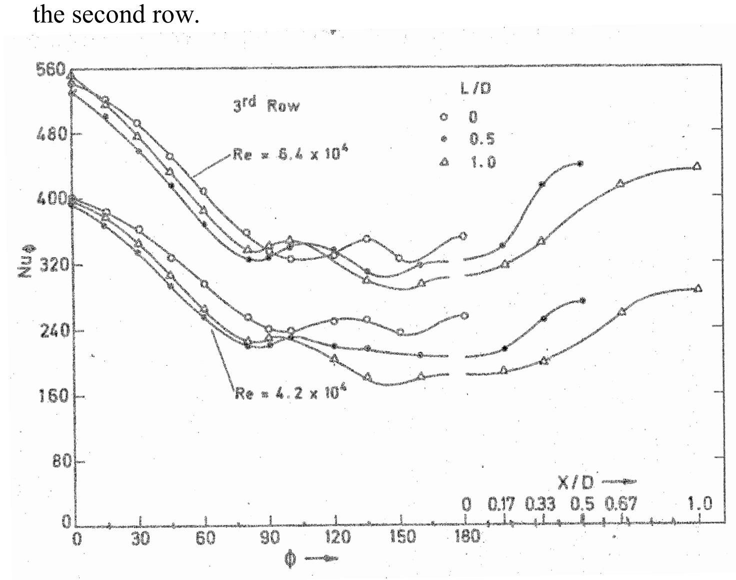


Figure 15: Local Nusselt number in the third row for different L/D

Addition of longitudinal fins caused a reduction in overall coefficients in the mass transfer studies of Sparrow and Kang [6]. The quasi-local values put forward by Qui and Chen [7] also showed that the transfer rate from the fin was less than from the base tube (**Figure 16**). The maximum Reynolds numbers in the two studies were 8 600 and 20 000 respectively. Notwithstanding the difference in geometries, in our case, the same trends were obtained at a Reynolds number 0f 13 500. However at higher Reynolds number considerable increases were observed.

The mean Nusselt number for the base cylinder 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 result, heat transferred by the L/D = 0.5 fin is very different from the tube with no splitter. For L/D = 1.0, an even more favorable situation emerges, in that the tube in the bank yields a higher heat transfer than a plain tube in the bank, despite the greater reduction of pumping power.

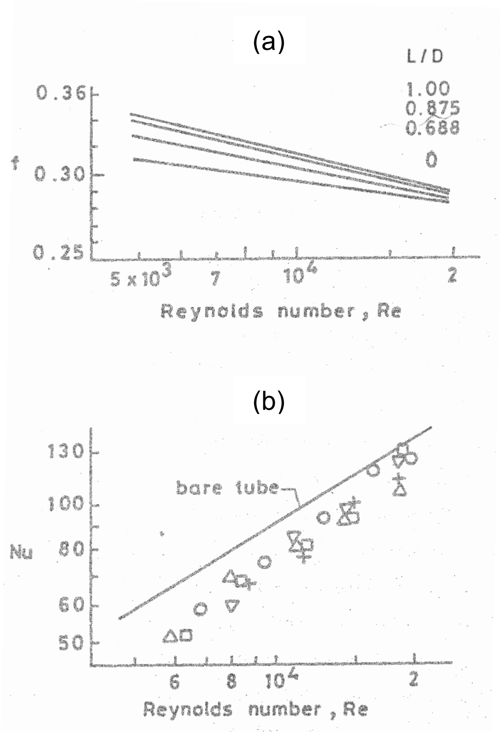


Figure 16: (a) Variation of friction factor with L/D and Reynolds number
(b) Nusselt number of splitter

ERROR ESTIMATION

In the present work, velocity, static pressure, pressure difference, wall temperature, fluid temperature, heating current and voltage are the measured quantities. Taking cognizance of the accuracy of the instruments used and uncertainty in the value of physical properties, the maximum errors calculated as in [16] are: Reynolds number – 2 %, Euler number – 2% , pressure coefficient- 0.9 % and Nusselt number – 2.5%.

CONCLUSION

With integral wake splitters, performance improvement is seen to occur on two fronts: reduced pumping power and increased heat transfer on account of the fin effect. The mechanical energy thus saved is more valuable than an equivalent amount of thermal energy.

The heat exchanger configuration has practical advantages to recommend it. Since the circular tube configuration is retained, the tubes as well as the plates in a heat exchanger are easily fabricated. The overall dimensions of the heat exchanger in any tube pitch layout need not be altered to accommodate the splitter plates.

A further way in which a heat exchanger constructed from such tubes will perform well is with respect to flow induced vibration. By reducing the energy of eddies and also providing additional stiffness to the tubes, integral splitter plates minimize the problem of tube vibration.

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