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Heat Transfer and Flow Structures Around Circular Cylinders in Cross-Flow

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Abstract

An experimental study was carried out to investigate heat transfer and flow characteristics from one tube within a staggered tube bundle and within a row of similar tubes. The tube spacing examined S_t and S_l are 1.5×1.5 and 1.5×1.25 where S_l and S_t denote the longitudinal and transverse pitches respectively. The variation of local Nusselt number was predicted with Reynolds number 4.8×10^4. The aim of the second part of the investigation was to examined the influence of the blockage of a single tube in a duct and transverse pitch for a single tube row with Reynolds number range of 7960 to 47770. Blockage ratio was varied from 0.131 to 0.843. Variation of local Nusselt number and local pressure coefficient were shown with different blockages and Reynolds numbers. The main results are described below.

For single tube row experiments, if the blockage ratio is less than 0.5, the general shape of local Nusselt number distribution around the cylinder varies only slightly with blockage. However the local Nusselt number and pressure coefficient distributions are remarkably different for the blockage ratio in the range of 0.668-0.843. Tube bundle experiments showed that changing longitudinal ratio did not affect the mean Nusselt number.

Key Words: Heat Exchangers, Effect of Blockage, Cross-flow, Tube Bundle

Çapraz Akışta Borular Etrafındaki Isı Transferi ve Akış Yapıları

Özet

Şaşırtılmış boru demeti ve tek sıralı borular içerisine yerleştirilmiş olan bir borudan ısı transferi ve akış yapılarının araştırılması için deneysel bir çalışma yapılmıştır. Sırasıyla S_t ve S_l düşey eksenel boşluk ve yatay eksenel boşluk olmak üzere bu mesafelerin incelenmesi S_t ve S_l, 1.5×1.5 ve 1.5×1.25 şeklinde olmuştur. Reynolds sayısının 4.8×10^4 değeri için noktasal Nusselt sayısının değişimi gösterilmiştir. Araştırmanın ikinci bölümünde Reynolds sayısının 7960 dan 47770 değerleri arasında bir kanal içine yerleştirilen tek boru ve tek sıralı boru demetinde düşey eksenel boşluğun değiştirilmesinin etkisi incelenmiştir. Blokaj 0.131 den 0.843 arasında değişmiştir. Farklı blokaj oranlarında ve Reynolds sayısında noktasal Nusselt sayısı ve noktasal basınç katsayısı değişimleri gösterilmiştir. Elde edilen genel sonuçlar şu şekildedir;

Tek sıra boru dizini deneyleri sonucunda eğer blokaj oranı 0.5 den daha az ise boru etrafındaki noktasal Nusselt sayısının genel şekli değişen blokaj ile birbirine çok yakındır. Bununla birlikte blokajın 0.668-0.843 arasındaki durumunda noktasal Nusselt sayısı ve noktasal basınç katsayısı dikkate değer bir şekilde farklılık göstermiştir. Boru demeti deneylerinde, yatay eksenel mesafenin değiştirilmesi ile ortalama Nusselt sayısında kaydadeğer bir farklılık olmadığı gözlenmiştir.

Anahtar Sözcükler: Isı Eşanjörleri, Blokaj Etkisi, Çapraz akış , Boru Demeti

Introduction

Prediction of heat transfer and flow characteristics around cylinders in tube bundles are important in relation to various engineering aspects. A large number of studies have been carried out concerning the features of heat transfer of tube bundles. Despite all the experimental data, it is not yet possible to get a clear idea about the flow and heat transfer processes in the tube bundle because of the very complicated geometry and the large number of parameters involved.

As is well known from the literature, the laminar boundary layer over the front stagnation point of a tube in cross-flow is the thinnest and its thickness increases with displacement downstream. Separation of the laminar boundary layer takes place when low velocity fluid close to the tube wall cannot overcome the adverse pressure gradient over the rear portion of the tube and eventually the flow stops and begins to move in the opposite direction. Fluid movement starts to curl and gives rise to vortices that shed from the tube.

Flow around circular cylinders and boundary layer separation have been investigated by Kraabel et al. (1982), Zukauskas (1972), Schimith and Wenner (1951), Boulos and Pei (1974). Variation in the local Nusselt number around the cylinder is affected by boundary layer development in the front of the tube and by separation and vortex shedding over the side and the wake region. The maximum Nusselt number occurs at the front stagnation point where the boundary layer and resistance to heat transfer is minimum. The minimum Nusselt number is found to correspond to the point before the boundary layer separates. The minimum Nusselt number location is dependent on the Reynolds number of the flow.

The variations in the hydrodynamic conditions in the flow around the tube are described by the distribution of local pressure and local velocity. The boundary layer separation is due to internal friction within the boundary layer and is also related to the pressure and velocity distribution around the cylinder. A certain amount of energy is consumed in overcoming the internal friction in the boundary layer, at the rear of tube the flow velocity decreases and pressure increases ($dP/dx > 0$) the energy in the flow is insufficient to overcome the increasing pressure and the flow separates from the surface. Local pressure and also velocity distribution around the tube up to 50° (from the front stagnation point) does not depend on the Reynolds number. It has been observed (Zukauskas (1972), Fage and Falkner (1931), Achenbach (1968)) that the effect of Reynolds number begins when $\Theta > 50°$.

Variation of heat transfer and flow around a tube in a bank is determined by the flow pattern which depends greatly on the arrangements of the tubes in the bank. In both the in-line and staggered tube arrangement, flow around a tube in the first row is similar to the single tube case but with delayed separation due to the increased blockage. A tube in one of the inner rows is affected by the highly turbulent flow. Considering the staggered tube bank case, the front of the tubes in the second row is influenced by the fluid acceleration and the blockage from the first row. The rear side of an internal tube is affected by high turbulence from the other inner tubes. A tube in the third row of a staggered array is influenced by very high turbulence, therefore for a tube in the third row a higher level heat transfer is observed. This has been observed by Murray and Fitzpatrick (1988), Baughn (1986) and Zukauskas (1972). Downstream from the third row, the heat transfer becomes stable and equal to the value for the third row. Heat transfer from a tube in a bank is influenced by the same parameters that are observed for a single tube. Increasing the Reynolds number produces a higher Nusselt number. Changing both longitudinal and transverse spacing causes great changes in the velocity distributions around the inner tubes in the bank. The effect of the longitudinal spacing on the flow pattern for second and subsequent tubes in a column has been investigated by, for example, Aiba et al. (1980) and Ishigashi (1986). Furthermore, the mean Nusselt number has been investigated for a complete array of 10 or more rows, Grimison (1937), Bergelin et al. (1952) and Zukauskas (1972). The results obtained have shown that the mean Nusselt number for a tube in a bank increases with an increasing of Reynolds number and is enhanced by an increase in the transverse pitch ratio and a decrease in the longitudinal pitch ratio. Pressure drop across the tube bank has been investigated by Grimison (1937) and Zukauskas (1972). Results have shown that increasing Reynolds number and decreasing transverse tube spacing causes an increase in the pressure drop. The longitudinal spacing is found to have less of an effect on the tube array pressure drop. The largest pressure drop is observed across a bank with closely spaced tubes.

The purpose of the present study was to clarify the heat transfer and flow characteristics around a tube in a tube bundle and in a single row. For single tube row experiments, the effect of the blockage ratio is considered in the sub-critical Reynolds number range and for the tube bundle experiment, the effect of longitudinal tube spacing is examined.

2. Experimental Procedure

Single Tube and Single Tube Row Measurement:

The aim of this part of the investigation was to present results the work carried out by Buyruk et al. (1995). They investigated the influence of the blockage of a single tube in a duct and the transverse pitch for a single tube row and also Buyruk et al. carried out a calculation of prediction of the heat transfer rates for the stagnation and laminar boundary layer regions with different blockage ratios. In this report, the effect of the Reynolds number was investigated with different blockage ratios. Experiments were completed in the sub-critical region of flow, that is in a Reynolds number range from $10^3$to $2x10^5$ using a low-speed wind tunnel and a number of 0.0485 m diameter tubes arranged horizontally in a single row across the duct. The spacing of the tubes could be adjusted and slender wedges were used at the end of the row to ensure the correct gap was maintained. The central tube was instrumented and it was rotated about its axis. A schematic view of the single tube and a single tube row can be seen in Figure 1.

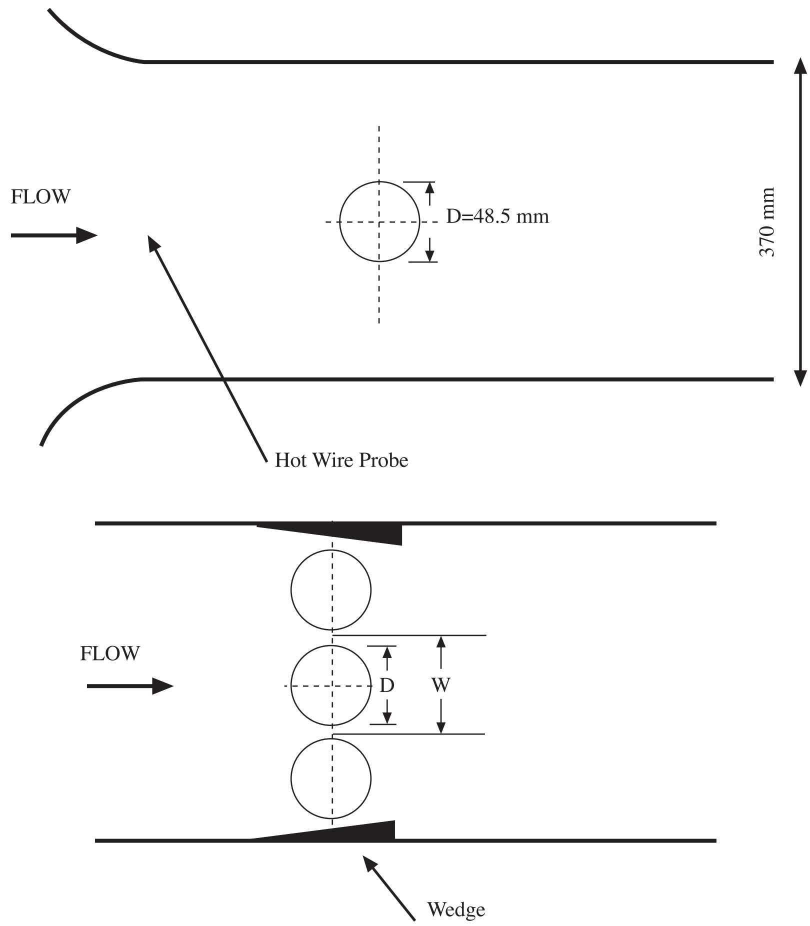

Figure 1. Schematic View of Single Tube and Single Tube Row

A thick-walled copper cylinder heated internally by an electric cartridge heater was used as the heat transfer section. The heater was inserted into the centre of the instrumented tube. The copper section was held in two PVC tubes to form a single tube assembly. The heat flux from the surface of the cylinder was measured using an RdF micro-foil heat flux sensor type 27034 attached to the surface with an epoxy resin adhesive. Only one heat flux sensor was used on the tube surface and the angular position of the tube was varied from 0° to 360° in 10° steps. A protractor was fitted to one end of the tube to allow the angular location to be monitored. Air temperature was recorded by a mercury thermometer placed into the upstream section. A single pressure tapping was used to measure static pressure at the tube surface with an inclined alcohol manometer. Details of the heat flow sensor are given by Buyruk (1996)

Experiments began with a single instrumented tube over a Reynolds number range 7960 to 47770. The heat transfer and pressure characteristics obtained were used as a benchmark against which the remaining data could be compared. In this initial test the blockage ratio, D/W, for the single tube in the duct gave value a of 0.131. Tubes were then added and the gaps adjusted to give different blockage ratios until the tubes were quite close with a blockage ratio of 0.843. A Disa 55M10 type hot-wire anemometer was used for the measurement of the flow velocities. According to the velocity measurement, turbulence level was approximately 1-2%

The results are presented in a dimensionless form. The following definitions are used for static pressure and heat transfer for single tube and single tube row case:

$$C_p = \frac{P - P_\infty}{\frac{\rho}{2} V_\infty^2} \text{ and } Nu = h_\theta \frac{D}{k}$$

$P = P(\theta)$ is the static pressure at the peripherical angle of θ of the cylinder and P_∞ the static pressure of the upstream flow.

Tube Bundle Measurement

In this section, the effect of longitudinal tube spacing on the heat transfer measurements for a staggered tube bundle was considered. The experimental apparatus for a tube bundle was essentially the same as used in the single-tube row experiments. The geometrical configuration is illustrated in Figure 2.

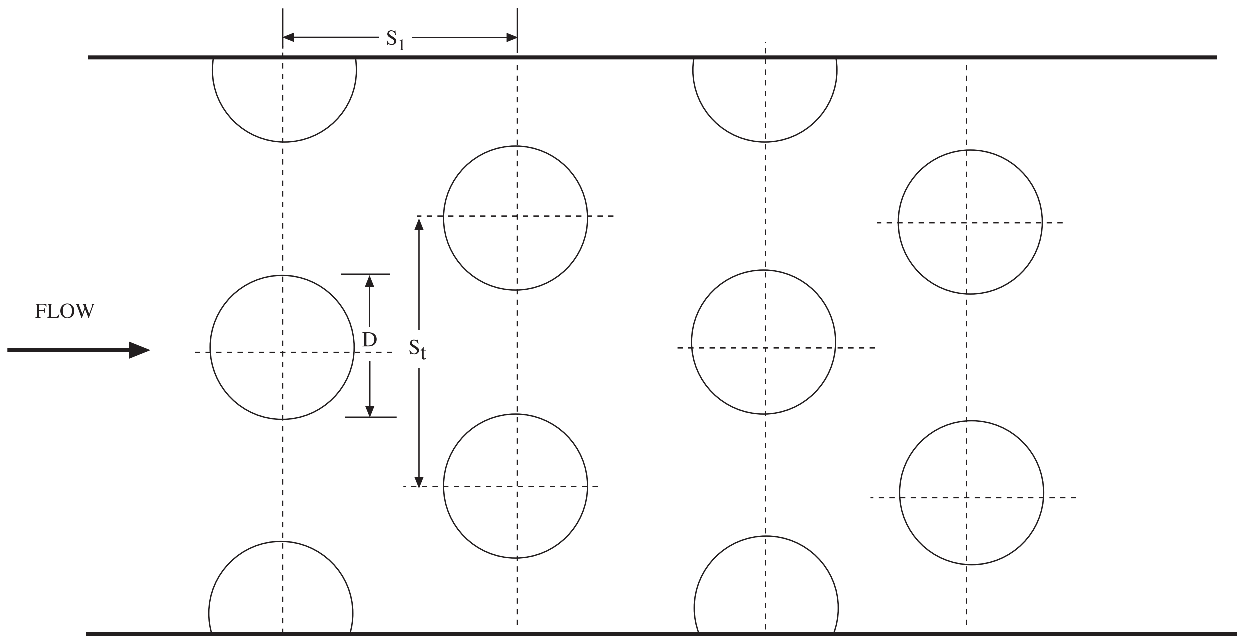

Figure 2. Bank of Cylinders in Cross-flow vs. Staggered Array

The geometry of the tube bundle is specified in terms of D, S_t and S_l. Thirty one tubes were located vertically in the same duct as had been used in previous measurements. Eight half tubes were used to ensure the correct gap at the wall. Seven tube rows were assembled and the instrumented tube was used in the various positions to measure local heat transfer The configuration of the bundle was that the transverse pitch was kept constant for all measurements at S_t of 1.5 and the longitudinal pitch, S_l was 1.25 and 1.5 where the dimensionless parameters S_t and S_l are defined as:

$$S_l = \frac{S_l}{D} \quad S_t = \frac{S_t}{D}$$

The reynolds number was kept constant at 4.8x104 was been calculated by:

$$Re_D = \frac{\rho V_{\max} D}{\mu} \quad [V_{\max} = V\frac{S_t}{S_t - D}]$$

For all measurements, constant up-stream velocity conditions were used and the instrumented tube was heated until thermal equilibrium was obtained while the wind tunnel was running. During the experiments up-stream air temperatures were recorded of approximately 18-20° and turbulence levels were measured for different configurations. Symmetry of the readings was monitored over 360° of the tube.

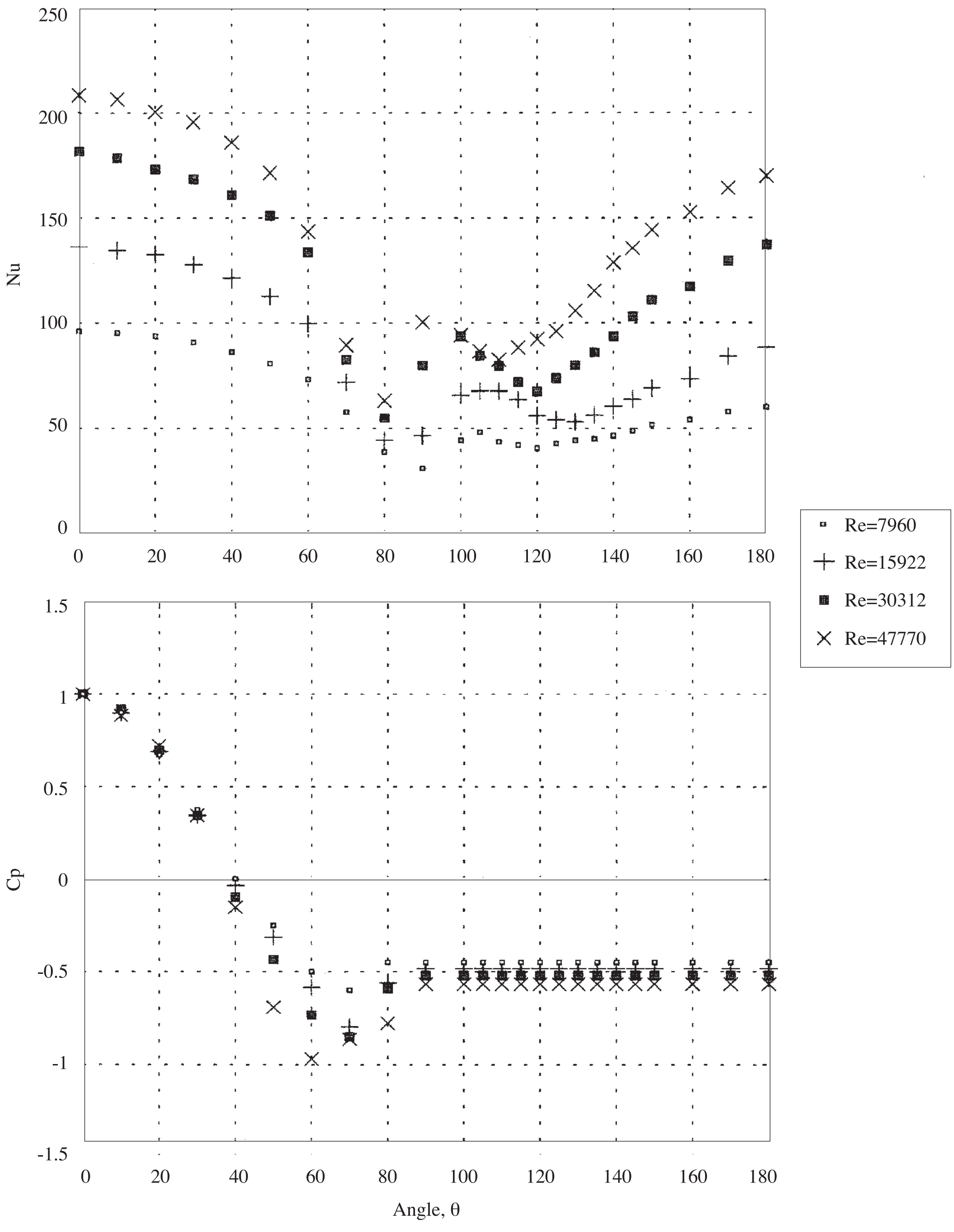

Figure 3. Distribution of Local Nusselt Number and Pressure Coefficient for Single Tube

3. Results and Discussions

3.1 Single Tube and Single Row of Cylinders

Figure 3 shows the local Nusselt number and local pressure coefficient for a cylinder in Reynolds number range of 7960 to 47770. For the single cylinder, the blockage ratio was 0.131 (corresponding to D/W).

The local static pressure drops with increasing distance from the stagnation point. As the flow accelerates around the cylinder the static pressure deviates more from the distribution given by potential theory. At about 70° the pressure becomes a minimum value of between -0.5 and -1 (for varying Reynolds number range) this is considerably higher than the potential flow value of -3.

For a smooth surfaced cylinder, four flow ranges can be obtained (Achenbach (1975)), the critical, the subcritical, the supercritical and the transcritical. Here, typical subcritical results were presented where the laminar boundary layer separates at an angular position of nearly 80°. As is known the local heat transfer is largest at the front stagnation point and decreases with distance along the surface as boundary layer thickness increases. The heat transfer reaches a minimum on the sides of the cylinder near the separation point. After the separation point the local heat transfer increases because considerable turbulence exists over the rear side of the tube where the eddies of the wake sweep the surface. However, the heat transfer over the rear is not higher than over the front, because the eddies recirculate part of the heated fluid. Increasing Reynolds number causes an increase in heat transfer due to the thinning of the laminar boundary layer. In the present study, the separation point was identified from the local Nusselt number graphs where it reaches the first minimum value. In Figure 3 with the lowest Reynolds number case the laminar boundary layer separates at nearly 90° and for the highest Reynolds number case the separation point is moved to 80°. This is also consistent with the previous work of Chen (1972) and Zukauskas (1972).

As pointed out by other authors, such as Murray and Fitzpatrick (1988), Boulos and Pei (1974), and Baughn et al. (1986), the local Nusselt number in the wake region shows a second minimum. This second minimum point was explained by Kraabel et al. (1982) to be a consequence of the laminar nature of the reattaching shear layer at the rear of the tube. Baughn et al. (1986) explained further that the initially laminar shear layer that exists for low Reynolds numbers gives less mixing than the turbulent shear layer experienced at higher Reynolds numbers.

In this study, the instrumented tube was placed at the centre of the duct in a single transverse row of tubes. Five different blockage ratios were used and results have been presented as local Nusselt number and local static pressure coefficient against angle for different Reynolds numbers. As expected, increasing blockage markedly increased the velocity around the cylinders. It is seen from the results that the separation point is dependent on the blockage, as higher blockage causes separation to be delayed (for example compare Figure 4 and Figure 8). Figures 4-8 show the influence of blockage on the local heat transfer and local pressure coefficient. The blockage ratio is from 0.131 to 0.843 and the Reynolds number is from 7960 to 47770 (defined on upstream velocity and tube diameter). In the high blockage case high velocities could not be reached due to the resistance to the flow. The 0.131 blockage represents the single tube case that was seen in Figure 3. As blockage increases so does the heat transfer due to higher flow velocities in the vicinity of the cylinder.

When the blockage ratio increases from 0.131 to 0.5 the average Nusselt number changes by about 10 %, and the separation point is seen to move. However the distribution of local heat transfer and local static pressure values are similar in character to those of the single tube. This can be seen in Figures 4 and 5, blockage ratios of 0.395 and 0.5 respectively; compared with Figure 3. When the blockage is 0.395, on the front side of the cylinder the laminar boundary layer develops, and the separation point moves back with increasing Reynolds number. When the blockage ratio increases from 0.395 to 0.5, local static pressure distribution changes considerably. Both the minimum local Nusselt number and minimum local static pressure are moved to the downstream side of the tube about by 10°.

Figure 6 shows results at a blockage ratio of 0.668 and the effect of the changing flow can be seen on both the heat transfer in the wake region and in the pressure distribution where the value of local static pressure coefficient falls as the flow accelerates through the narrowest gap. Over the front face of the cylinder the heat transfer coefficient is relatively constant due to the accelerating flow and the resulting thinner boundary layer.

Figures 7 and 8 show the results for blockage ratios of 0.75 and 0.843 respectively. When the gap gets smaller, the distribution of local Nusselt number and local pressure coefficient are quite different to those seen earlier. The flow through the row of cylinders with such a small transverse gap is very similar to that through a venturi type of nozzle. Increasing the blockage ratio from 0.131 to 0.843 shows the minimum pressure point to have moved from 70° to 90°, and the separation point moves downstream to about 105°. Especially when the blockage ratio is 0.843, the front portion of the tube's heat transfer distribution increases, and front and rear side values are no longer the maxima.

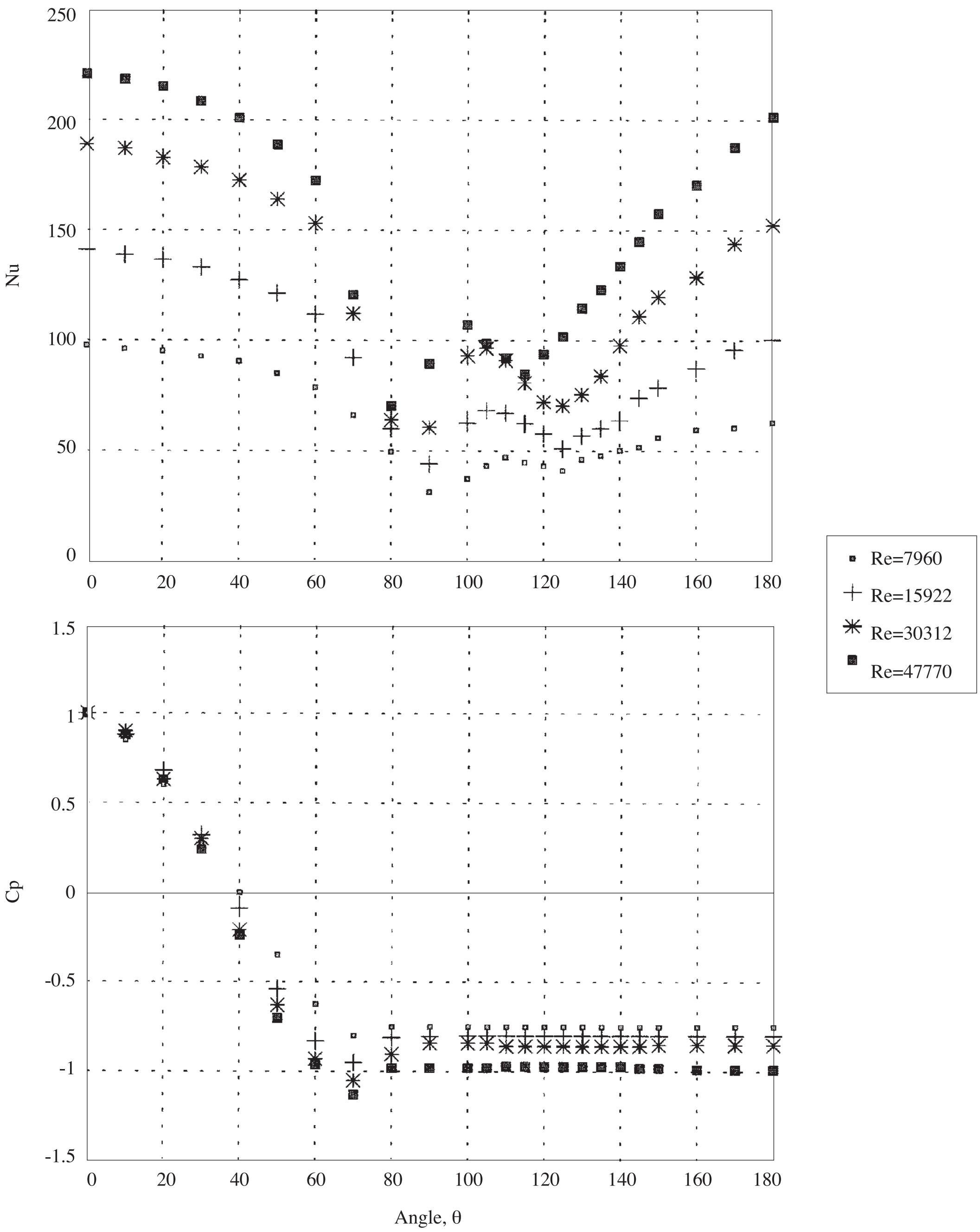

Figure 4. Distribution of Local the Nusselt Number and Pressure Coefficient for B=0.395

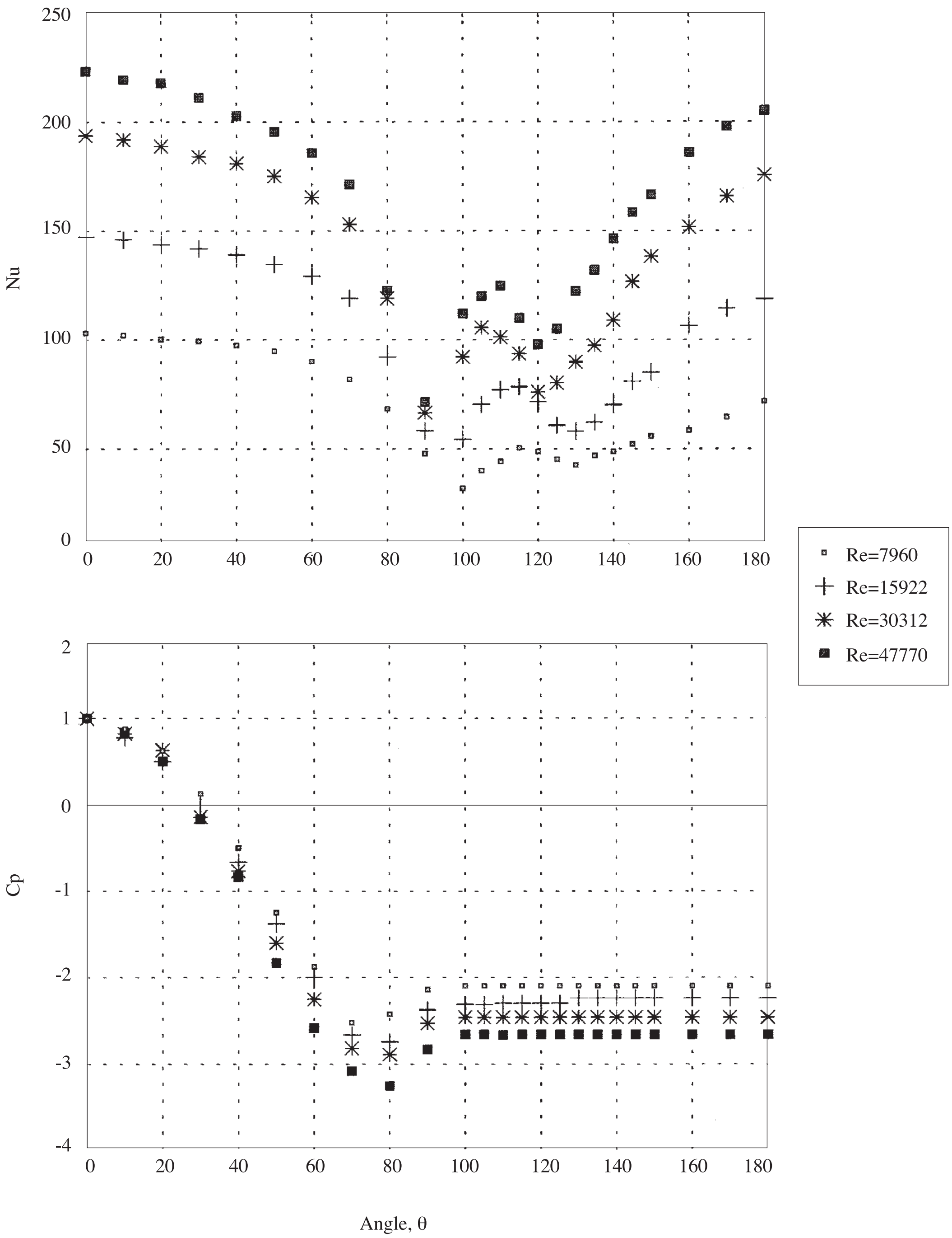

Figure 5. Distribution of Local Nusselt Number and Pressure Coefficient for B=0.5

3.2 Tube Bundle

The heat transfer in the flow over the tube bank depends largely on the flow pattern and the intensity of turbulence that in turn are functions of the velocity of the fluid and size and arrangement of the tubes. As observed by many investigators flow structure around the first transverse rows is similar to flow around a single tube. The turbulent wakes extend to tubes located in the second transverse row. As a result of the high turbulence in the wakes, the boundary layer around the tubes in the second and subsequent rows becomes thinner. Therefore it is expected that heat transfer coefficients of tubes in the first row are smaller than those of the tubes in subsequent rows. In this section, the results are reported for 1.5x1.5 and 1.5x1.25 ($S_t x S_l$) staggered tube bundle configurations for Reynolds number 4.8x10^4 corresponding to maximum flow velocity at the minimum sectional area.

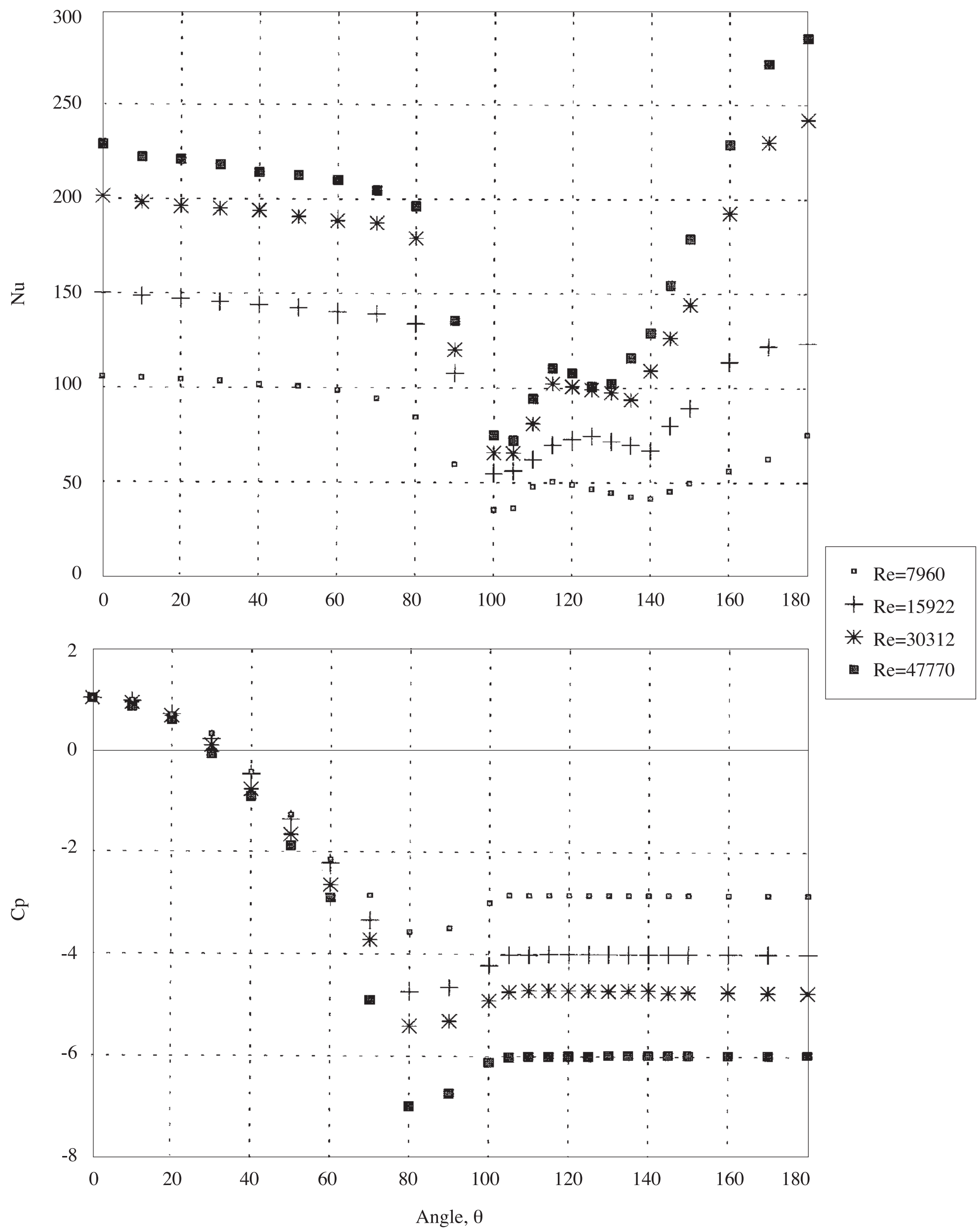

Figure 6. Distribution of the Local Nusselt Number and Pressure Coefficient for B=0.668

First Row The distribution of local Nusselt number around the tube in the first row for both configurations is shown in Figure 9. The results for the tube in the first row are very similar to the single row case when blockage ratio is the equivalent 0.668 and Reynolds number of 15922. Two minimum Nusselt numbers occur at 100° and 140°, as observed in single tube and single tube row cases. It can be seen in other experimental work (Kraabel et al. (1982), Baughn et al.(1986)) that this may be thought to result from the laminar free shear layer. After this second minimum point, the local Nusselt number on the rear side of the tube is higher than the front half of the tube due to the influence of the subsequent inner tubes. In both configurations the results are rather similar but there is a small effect due to the influence of different longitudinal spacing.

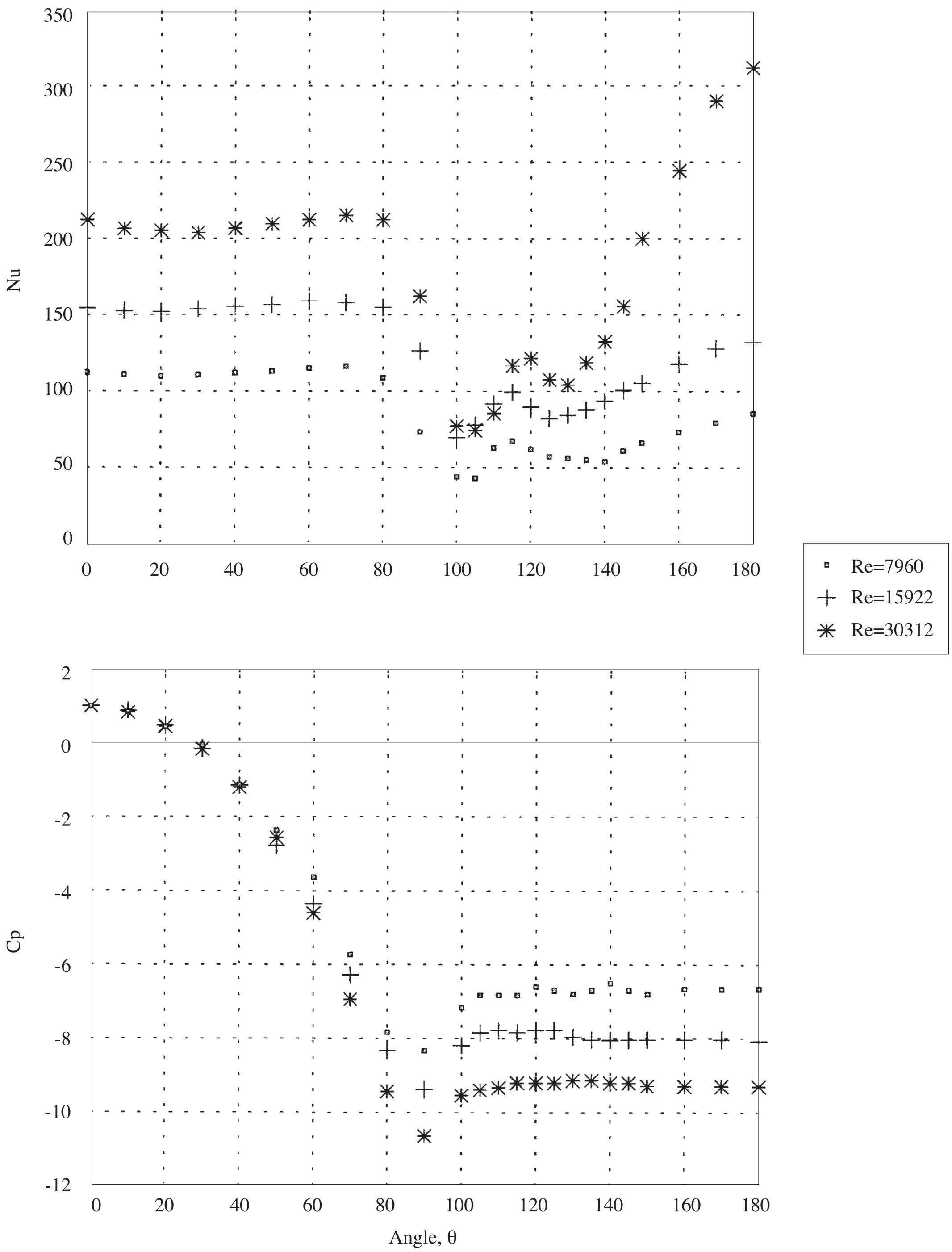

Figure 7. Distribution of Local Nusselt Number and Pressure Coefficient for B=0.75

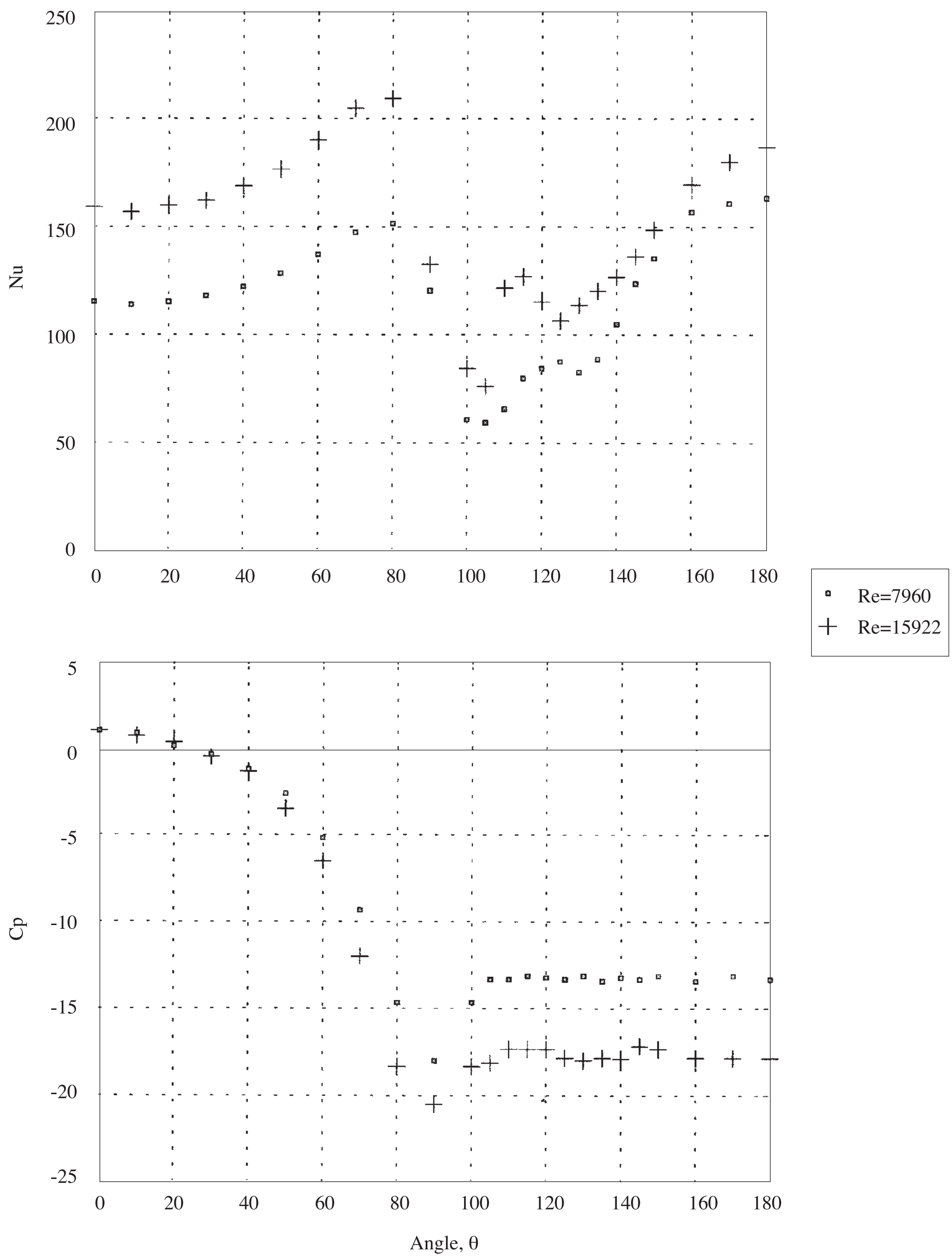

Figure 8. Distribution of Local Nusselt Number and Pressure Coefficient for B=0.843

Second Row As seen in Figure 9 from the local Nusselt number distribution around the second row in a bank, two local minimum points occur. Baughn et al. (1986) commented that this could be due to the turbulence being more effective on the inner rows although where these points are due to transition from the laminar to turbulent boundary layer and then separation or laminar separation followed by turbulent reattachment and separation cannot be determined. In both configurations, local Nusselt number at the front stagnation point is 45% higher than the first row front stagnation point due to flow acceleration caused by the blockage. The decrease in the Nusselt number as the laminar boundary layer develops from the front stagnation points is more noticeable than in the first row. The minimum Nusselt number occurs in both cases at around 80° from the stagnation point.

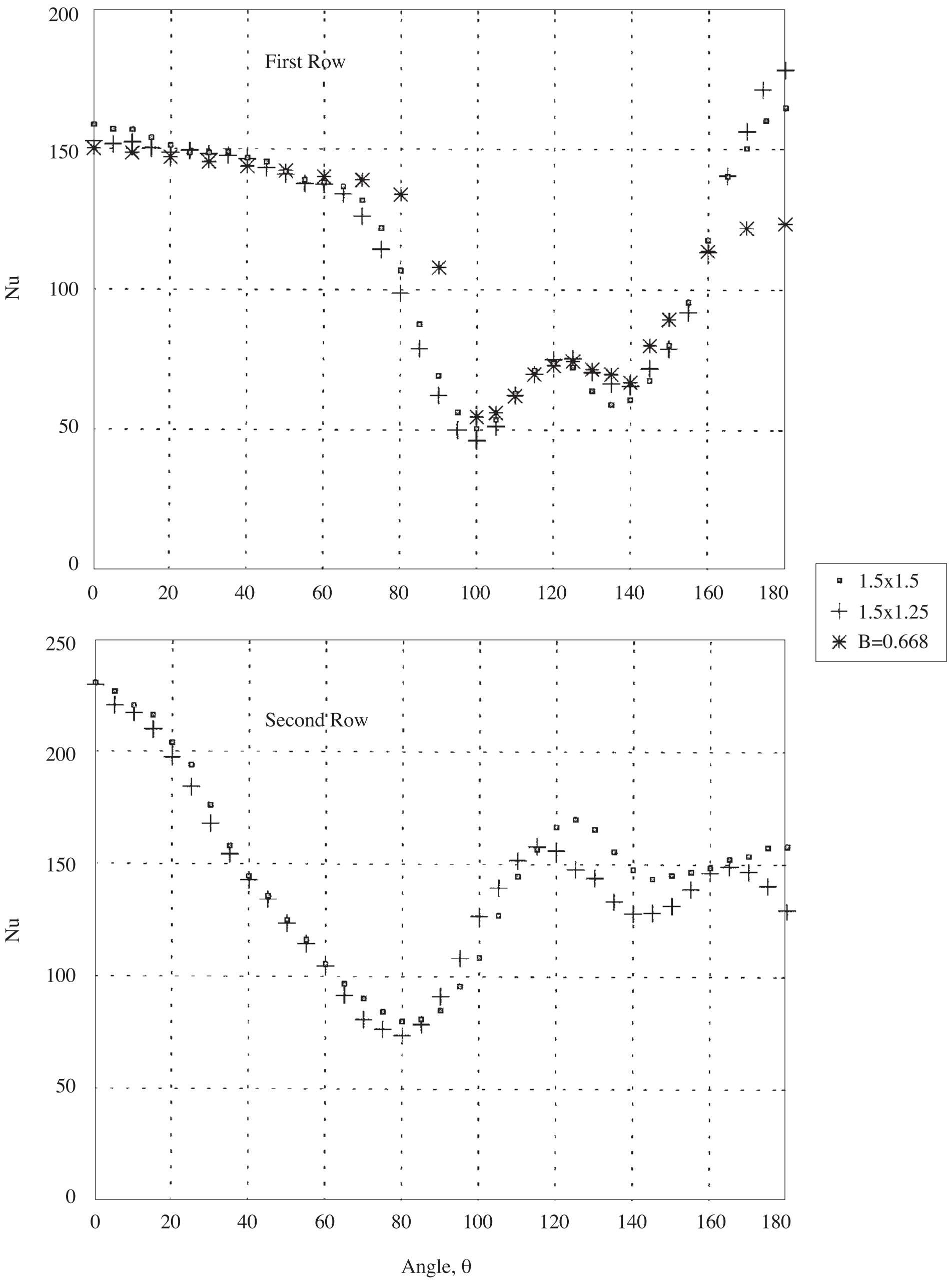

Figure 9. Distribution of Local Nusselt Number for First and Second Row Figure 10 Distribution of Local Nusselt Number for Third and Fourth Row

Third Row The distribution of local Nusselt number with angle over the tube in the third row are shown in Figure 10. The shape of the distributions are similar to the first and second row but at the front of the tube Nusselt number is higher than in the second row due to the high turbulence produced by the previous two rows. The decreasing nature of Nusselt number as the laminar boundary layer develops is more pronounced. As mentioned for the second row, two minimum Nusselt number are seen at about 105° and 140°. After the second minimum point, it increases to rear stagnation point. The first minimum point occurs at 105° and moves to the downstream side of the tube compared with the second row. This is considered to be a result of the high turbulence from the second row.

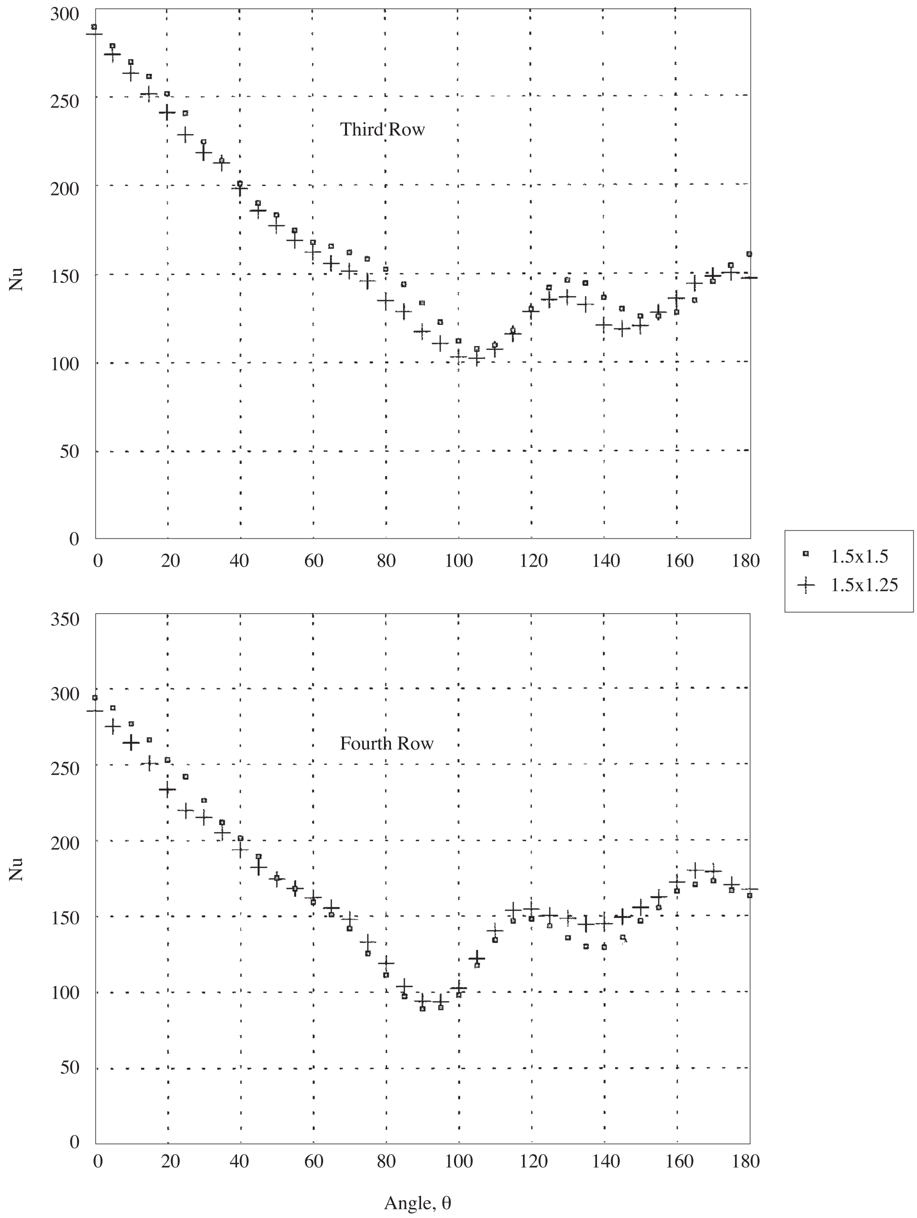

Figure 10. Distribution of Local Nusselt Number for Third and Fourth Row

Fourth Row The variations in the local Nusselt number with angular position in the fourth row are shown in Figure 10. Again the laminar boundary layer develops quickly. The nusselt number is very similar to in the third row but the first minimum Nusselt number moves to the upstream side at nearly 95° in both configurations. In the wake region slightly higher Nusselt values are seen compared with the third row.

Fifth Row The local Nusselt number in the fifth row is shown in Figure 11. The distribution of the Nusselt number over the front side of tube is slightly lower than in the fourth row but is essentially the same and is considered to be constant thereafter. The minimum Nusselt number occurs at the same point as in the fourth row and the wake region is the same as observed in the fourth row.

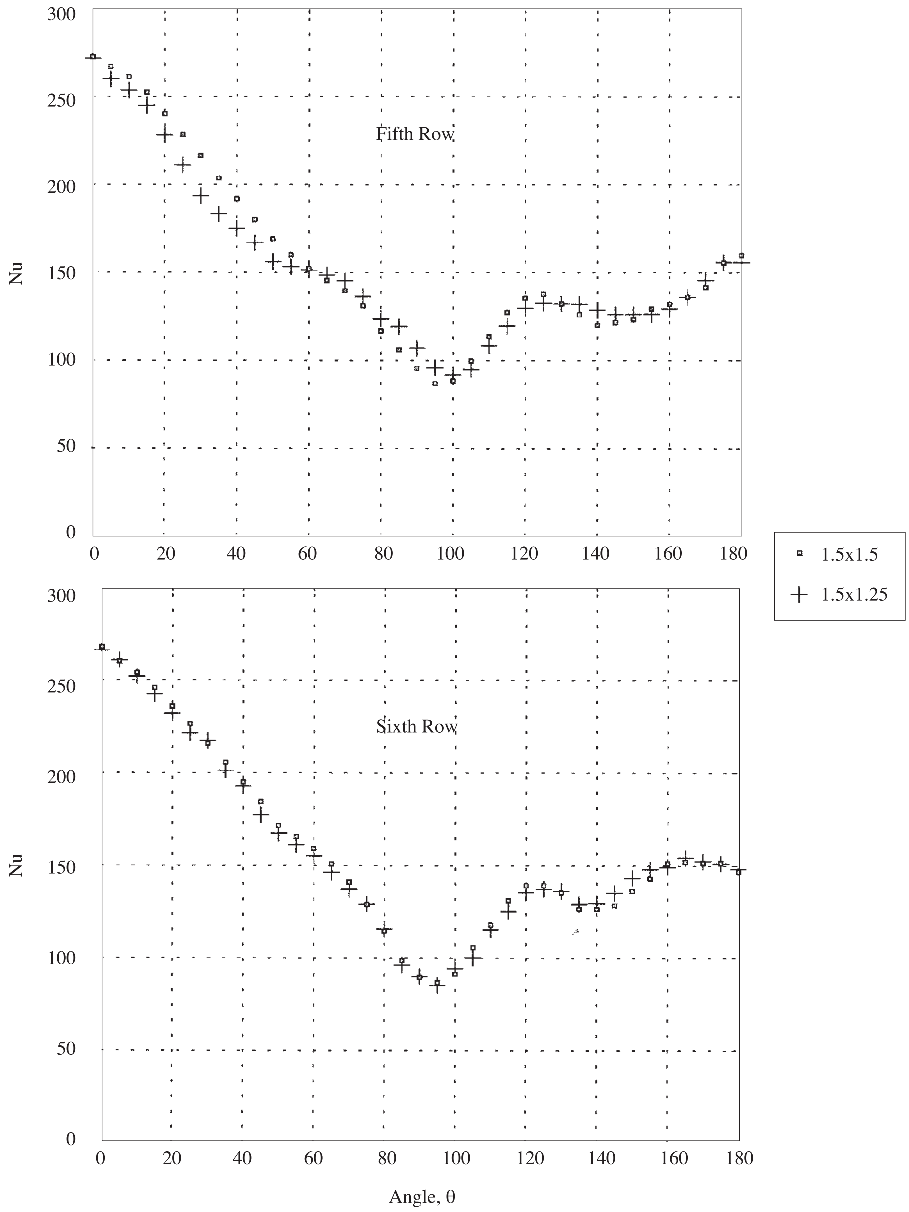

Figure 11. Distribution of Local Nusselt Number for Fifth and Sixth Row

Sixth and Seventh Row The local Nusselt number in the sixth and seventh rows are shown in Figures 11 and 12. The local Nusselt numbers vary with a similar trend as the fifth row. The laminar boundary layer develops and a separation point occurs at nearly 95° for both rows. The local Nusselt numbers at the rear side of the tube are identical compared with the previous row. For the sixth row and seventh row, front stagnation point Nusselt number values are slightly lower than the fifth row.

Figure 13 shows the comparison of the average Nusselt number for both geometries and different rows of tubes. It is clear that the mean Nusselt number increases until the fourth row, after this, a decrease is seen for both geometries. Also altering the longitudinal pitch ratio from 1.25 to 1.5 did not give any noticeable effect on the mean Nusselt number. The tube bundle results showed that the highest Nusselt number was observed in the fourth row in the bank. Zukauskas (1972), Murray and Fitzpatrick (1988) have observed that the highest heat transfer takes place in the third row of a tube bank.

Poskas and Survilo (1983) stated that on the effect of turbulence becomes steady from the fourth or fifth row onwards. Aiba et al. (1982) have found that the average heat transfer rate was the lowest for the first cylinder and the highest for the third one. Knudsen and Katz (1958) measured of average heat transfer coefficients on a single tube for air flowing across a bank of staggered tubes. According to their results the average Nusselt number increases up to the third row, decreases slightly and then remains essentially constant beyond in the fifth row.

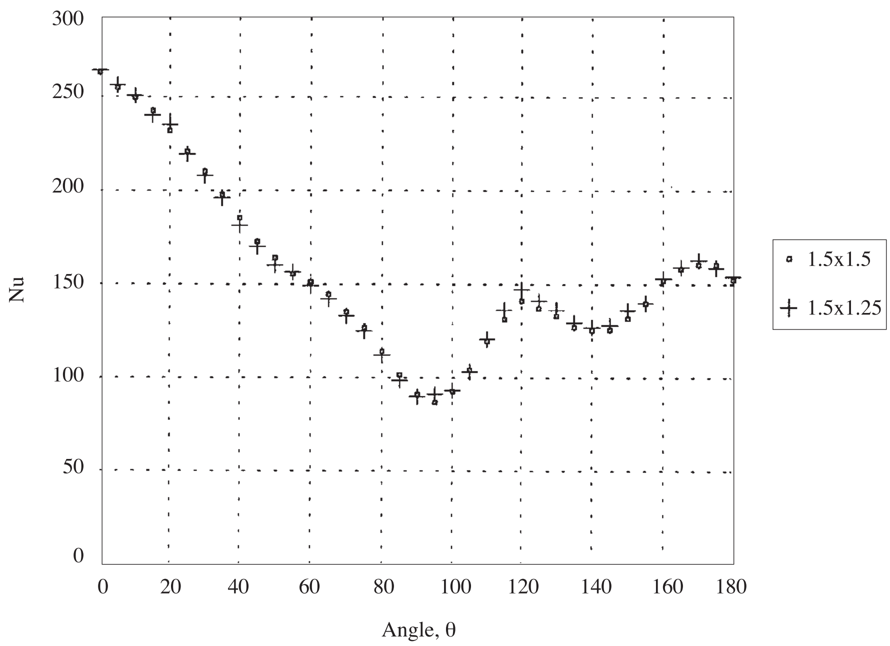

Figure 12. Distribution of Local Nusselt Number for Seventh Row

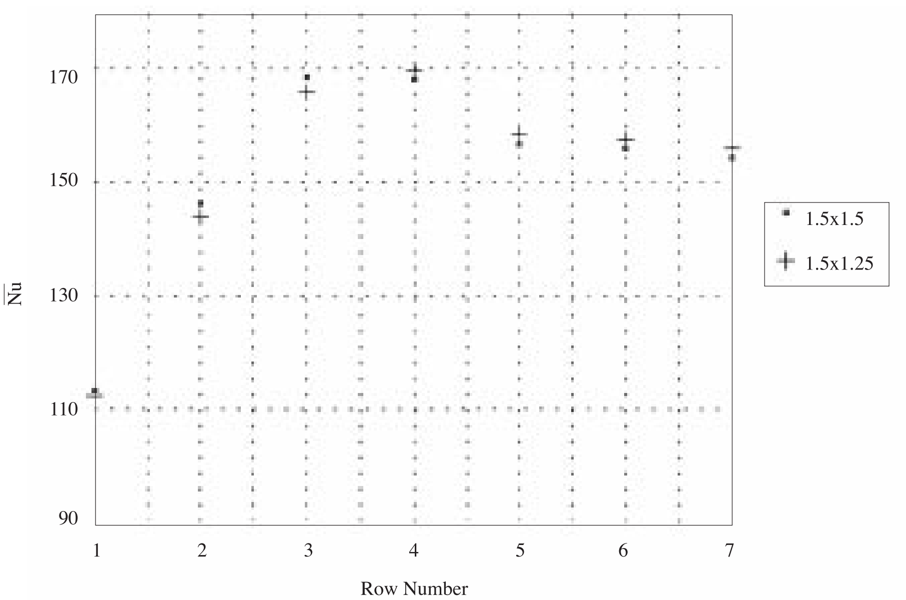

Figure 13. Comparison of Average Nusselt Number for 1.5x1.5 and 1.5x1.25 Staggered Tube Bundle Geometries

4. Conclusions

For the single row experiments, if the blockage ratio is less than 0.5, the general shape of local Nusselt number distribution around the cylinder varies only slightly with blockage. When the blockage ratio is greater than 0.395, increasing blockage causes the minimum pressure and minimum Nusselt number to move to the downstream side of the cylinder. For the blockage ratio in the range of 0.668-0.843, there is a distinct change in the flow compared with the lower blockage cases. Local Nusselt number and static pressure are remarkably different. Due to the great acceleration of the flow minimum local static pressure values reached very low values.

Changes in both longitudinal and transverse pitch were observed to have a noticeable influence on the velocity distribution around the tubes in a bank. For example for the staggered tube bundle geometry, the mean Nusselt number of the inner tubes becomes higher by increasing the transverse pitch and decreasing longitudinal pitch ratios. In the present study, only the longitudinal ratio was changed from 1.5 to 1.25 and the results showed that did not affect the mean Nusselt number. The investigation needs to be extended for different geometries.

5. Nomenclature

C_p : Local Pressure Coefficient
D : Tube Diameter
h_θ : Local Heat Transfer coefficient around the tube
k : Thermal Conductivity
P : Static Pressure
P_∞ : Total Pressure
S_l : Longitudinal Pitch
S_t : Transverse Pitch
s_l : Longitudinal distance of tube bank
s_t : Transverse distance of tube bank
T : Temperature
V : Mean Velocity
$V_{\max}$: Maximum Velocity
W : Height of the Duct

Greek Letters

μ : Dynamic Viscosity
ρ : Density
θ : Peripherical Angle

Non-dimensional Parameters

Re_D : Reynolds Number
Nu : Nusselt Number

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