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LOOKING FOR CORRECT DIMENSIONLESS PARAMETERS FOR TUBE-BANK FLOW ANALYSIS

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In this paper, the sensibility of diagrams of experimental results of pressure and velocity fluctuations of the turbulent cross-flow through tube banks with aspect ratios (P/D) ranging from 1·05 to 1·60, with triangular and square arrangements, by changing reference length and velocity, is investigated, with the purpose of obtaining the optimum form for the presentation of dimensionless power spectral densities. Dimensionless results of mean-square value of pressure fluctuations and of autospectral densities of pressure and velocity fluctuations are presented. Based on these results, the choice of the adequate reference velocity and length scales for tube bank flow is discussed.

1. INTRODUCTION

BANKS OF TUBES OR RODS are found in the nuclear and process industries, being the most common geometry used in heat exchangers. Tube banks are the usual simplification for fluid flow and heat transfer in the study of shell-and-tube heat exchangers, where the coolant is forced to flow transversely to the tubes by the action of baffle plates.

In the last 50 years fluid flow processes in heat exchangers have been widely studied. Attempts to increase heat exchange ratios in heat transfer equipment do not consider, as a priority of project criteria, structural effects caused by the turbulent fluid flow across the tube bank, unless failures occur (Païdoussis 1982).

By reducing the aspect ratio (pitch-to-diameter ratio) of a tube bank, to improve the heat transfer process, dynamic loads will appear, which are not associated to vortex shedding, as in large aspect ratios tube banks. The influence of the other hydrodynamic processes, such as fluidelastic instabilities, turbulent buffeting and acoustic resonance, may also be present in tube banks of all aspect ratios. Therefore, new information about the hydrodynamic phenomena in tube banks is needed for the design and development of new equipment. The generalization of the large amount of experimental data, through similarity studies, and the constant improvement of existing experimental techniques in the last few decades, allowed a better understanding of thermal and hydrodynamic phenomena in tube banks, leading also to the need of knowing the features of the resulting flow and its effects on the walls of the tubes in the bank.

The choice of physical quantities for the definition of dimensionless parameters, such as the mean and friction velocities, combined with physical properties of the fluids, can be considered classic for a certain number of problems, for example, the turbulent flow through pipes (Zierep 1982). The first systematic studies of fluid flow distribution and pressure drop

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in tube banks of shell and tube heat exchangers and steam generators, known to the authors, are the doctoral thesis of Wiemer (1937) and the work of Grimison (1937).

Žukauskas (1972) represents the pressure drop as a function of the flow velocity, the bank arrangement, including longitudinal and transverse gap-distance between tubes and the number of tube rows through which the flow passes. In dimensionless form, the classical relation between Euler and Reynolds numbers was used, these quantities being defined with the tube diameter and a reference velocity, this being, usually, the mean velocity in the narrow gaps between the tubes. The number of rows and transverse and longitudinal pitch-to-diameter ratios complemented the analysis.

On the other hand, it seems to be reasonable to choose as reference, instead of this gap velocity, the mean velocity in the bank or the incidence velocity. In the literature, it can be found that, in several problems, the boundary layer is not self-similar, in the sense that mean quantities cannot be described as functions of a unique reference length or velocity, making it difficult, or even impossible, to attempt the presentation of experimental results from turbulent flows in dimensionless form. This can become more critical when results, like power spectra or correlations, from several experiments, must be compared to obtain general features of a phenomenon, not being obvious *a priori* which parameters must be used in the presentation of the results.

Another important issue in the analysis of the turbulent flow through tube banks is the number of tube rows. Achenbach (1969) observed in a staggered tube bank that the first two rows produce such a level of turbulence due to boundary-layer separation, that the subsequent rows do not experience this process and the flow remains practically unaltered for the subsequent rows. On the other hand, results by Fitzpatrick *et al.* (1988), obtained in a 20-row square arrangement tube bank, showed significant variations according to the position in the bank. However, in the results of Endres *et al.* (1995), the pressure distribution on the fourth rows of $P/D = 1{\cdot}26$ tube banks, with triangular and square arrangements, are similar to those observed in the third rows, in accordance with Achenbach's observations cited above.

The cross-flow passing a tube in a bank is strongly influenced by the presence of the neighbouring tubes. In the narrow gap between two tubes in a row, the strong pressure gradient will influence not only the flow in that region, but also the flow distribution downstream of this point, in the narrow gap between two tubes in the next row, and so on. Žukauskas (1972) compares the flow through tube banks with staggered arrays to the flow in a curved channel with periodically converging and diverging cross-sections. For in-line arrays, the comparison is made with a straight channel, and the velocity distribution is strongly influenced by the velocity in the narrow gaps.

The concern about heat transfer equipment integrity is due to the close relationship between fluid flow around a solid surface or a structural element and the vibrations induced by the flow in the structure. Unlike large aspect ratio tube banks, where dynamic loads are mainly associated with the vortex shedding process, the turbulent flow in tube banks with small aspect ratios is characterized as broad-band turbulence, without a defined shedding frequency (Blevins 1990). Therefore, dynamic loads will be associated with the fluctuating pressure field, of concern for equipment integrity and more important than static loads; this pressure field is produced by the total pressure field (Endres *et al.* 1995).

Pressure fluctuations result from velocity fluctuations at several points in the flow field. The resulting pressure field is described by Poisson's equation, obtained from the divergence of the Navier–Stokes equation (Willmarth 1975). By introducing the Reynolds statement, representing velocity components and pressure by their time average value and the fluctuating part, an equation for the pressure fluctuation distribution will be obtained:

$$\nabla^2 p' = -\,2\rho\,\frac{\partial \bar{u}_i}{\partial x_j}\,\frac{\partial u'_j}{\partial x_i} - \rho\,\frac{\partial^2 (u'_i u'_j)}{\partial x_i \partial x_j} + \rho\,\frac{\partial^2 \overline{u'_i u'_j}}{\partial x_i \partial x_j}. \tag{1}$$

Pressure fluctuations are, thus, produced by the interaction of velocity gradients with velocity fluctuations and Reynolds stresses (Rotta 1972). According to Townsend (1976), the amplitude of the pressure fluctuations may be influenced by velocity fluctuations at a distance comparable to the wavelength of these fluctuations. The search of form and magnitude of pressure and velocity fluctuations and the interdependence between these quantities is necessary for the comprehension of the complex phenomena occurring in the turbulent flow through tube banks. Their correct representation in dimensionless form is a natural additional challenging problem, due to the many different velocities and physical dimensions (lengths) involved. The result is, therefore, the difficulty in obtaining similarity laws for these quantities.

Basically, in fluid flow analysis, similarity is obtained by reproducing the same Reynolds number, between two different flows through similar geometries. The mathematical formulation for geometrical similarity between two fields is (Zierep 1982)

$$(x_2,\, y_2,\, z_2) = \alpha(x_1,\, y_1,\, z_1), \tag{2}$$

where α is the scale factor between fields 1 and 2.

For two physically (dynamically) similar flow fields, the proportionality will appear in the pressure and velocity fields, as well as in the physical properties, like viscosity and density, when the fluid is not the same in both cases.

In this paper, the sensitivity of diagrams of experimental results of pressure and velocity fluctuations in tube banks is investigated by changing reference length and velocity. The banks have triangular and square arrangements and several aspect ratios. The purpose of this experimental investigation is to obtain an optimum form for the presentation of dimensionless r.m.s. values and power spectral densities of pressure and velocity fluctuations. By this means, a discussion on more suitable forms of definition of Reynolds and Euler numbers, and the corresponding choice of reference length and velocity, for tube bank flow analysis is presented.

2. EXPERIMENTAL TECHNIQUE

The test-section is a 1370 mm long rectangular channel, with 146 mm height and a maximal (adjustable) width of 193 mm. Air is the working fluid, driven by a centrifugal blower, passing through settling chamber and a set of honeycombs and screens, before reaching the tube bank at an incidence angle of 90° and about 2% turbulence intensity. The angle of incidence of the air on the tubes was 90°. The flow rate, and thus the Reynolds number, was controlled with the help of a gate valve. Before the tube bank, a Pitot tube was placed, at a fixed position to measure the reference velocity for the experiments. For both the triangular and square arrangements, the tube banks were five rows deep for the measurements of fluctuating quantities. A schematic of the test-section is shown in Figure 1. The geometries investigated and presented in this paper had aspect ratios (P/D) of 1·60, 1·26, 1·16 and 1·05, with tubes of 32·1 mm diameter. The geometries are listed in Table 1. The values of the corresponding Reynolds numbers and their definition will be presented later.

For the measurement of velocity and velocity fluctuations between the tubes, a DANTEC constant-temperature hot wire anemometer was used. Wall pressure fluctuations were

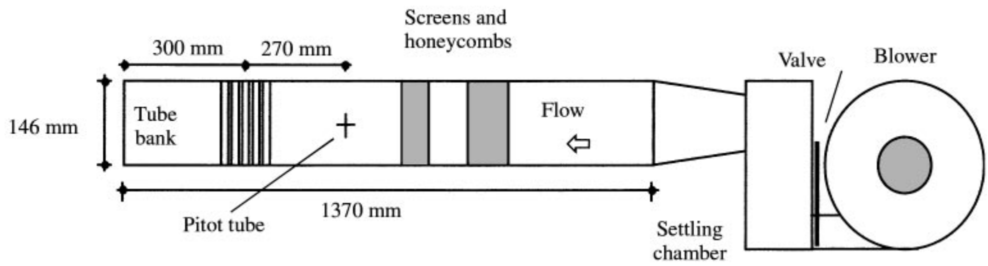

Figure 1. Schematic view of the test-section.

TABLE 1
Geometric parameters for the test-sections investigated

P/D	1·60	1·26	1·16	1·05
N_{square}	20	25	25	25
$N_{\mathrm{triangular}}$	18	23	23	23
M	4	5	5	5
B (mm)	193·0	193·0	182·7	172·9
D (mm)	32·1	32·1	32·1	32·1
P (mm)	51·4	40·3	37·2	33·7
S (mm)	19·3	8·2	5·1	1·6
W (mm)	3·3	0·0	0·9	3·0

N = Number of tubes in the test-section.
M = Number of tubes in the first row.

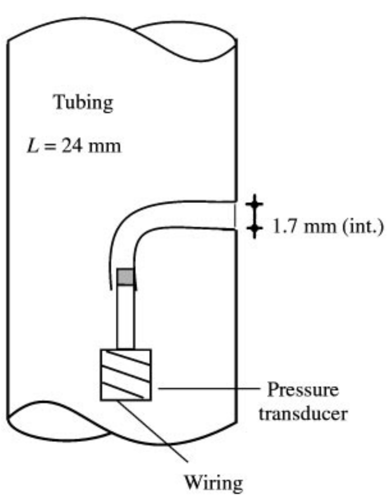

Figure 2. Instrumented tube (schematic).

measured by ENDEVCO piezoresistive pressure transducers, mounted inside the tubes and connected to a pressure tap by plastic tubes, as shown in Figure 2.

The use of tubing was necessary due to the dimensions of the test-section used, although pressure transducers are preferably mounted flush to the walls. Prior measurements in pipe

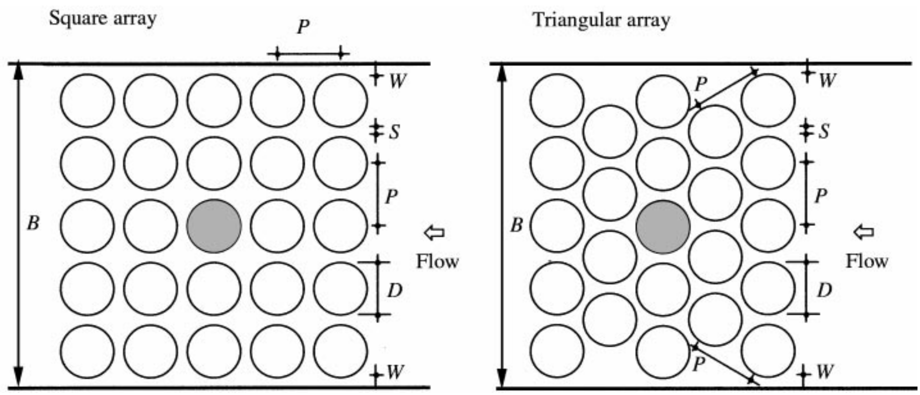

Figure 3. Schematic representation of the tube banks with the instrumented tube shaded.

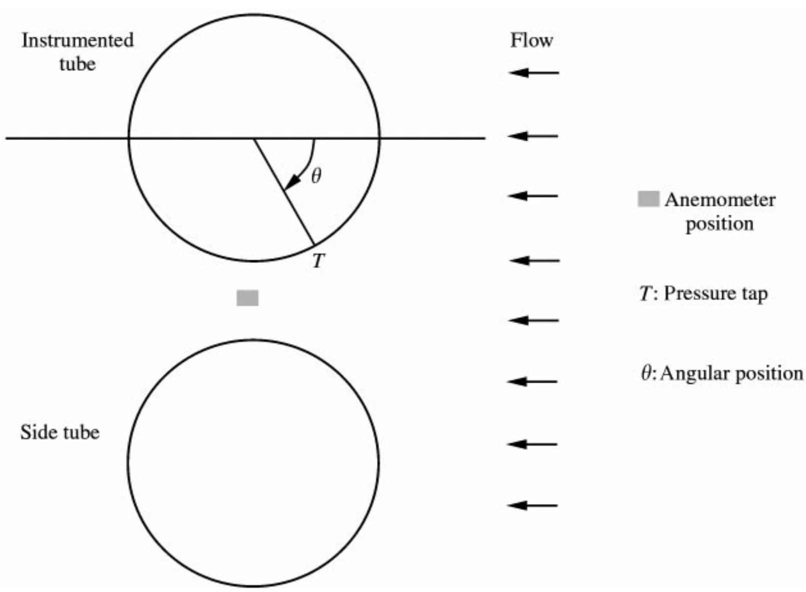

Figure 4. Schematic representation of probes positioning inside the tube bank.

flow showed that this mounting technique was adequate for the measurements to be performed (Endres & Möller 1994).

The tube instrumented with the pressure transducer was the central one in the third row, as shown in Figure 3. Measurements were performed at each 10° by turning the tube about its axis. These angles are measured clockwise between the direction of the main flow and the position of the pressure tap. Zero degrees (0°) corresponds to the position where the pressure tap faces the main flow. Velocity and velocity fluctuations were measured in the centre of the narrow gap between the tubes, directly in front of the pressure tap at position 90, as shown in Figure 4.

Data acquisition of pressure and velocity fluctuations was performed simultaneously by a Keithley DAS-58 A/D-converter board controlled by a personal computer, which was also used for the evaluation of the results.

Experimental results in this paper are characterized by the mean-square value of wall pressure fluctuations and by autospectral (power spectral) density functions of velocity and pressure fluctuations. The sampling frequency was 16·1 kHz, while the signals of the instruments were high-pass filtered at 1 Hz and low-pass filtered at 8·05 kHz.

Uncertainties in the results have a contribution of 1·4% from the measurement equipment (including hot wire, pressure transducer and A/D converter). In the measurements of pressure fluctuations, tubings are responsible for 5% of the uncertainties, leading to a total value for the spectra of pressure fluctuations, up to 1000 Hz, of 6·4%.

3. DIMENSIONLESS PARAMETERS

Characteristic dimensionless parameters found in the fluid flow analysis through tube banks are, among others, the Reynolds and Euler numbers, used to describe flow and pressure drop characteristics. For the definition of these quantities, reference velocities and lengths are required and the choice, in tube banks, is quite varied, leading to several possibilities of definitions of dimensionless parameters.

In this work, the need for dimensionless parameters is not restricted to Reynolds and Euler numbers is indeed, experimental results like the fluctuating pressure field must be interpreted and the results must be presented so that they can feed the engineering work in the form of project criteria.

Length parameters are directly obtained from Figure 3, with the pitch P and tube diameter D. Obviously, the aspect or geometric ratio of a tube bank will be given by the pitch-to-diameter ratio P/D. An additional characteristic length can be obtained from the difference of pitch and diameter, namely the gap spacing S. The use of S can introduce variations of the length parameter in this analysis without the influence of tube diameter D.

The choice of a characteristic velocity can become an additional problem in tube-bank flow analysis. In this research work, four different velocities were chosen:

(i) U_{ref}: Reference velocity, defined by the velocity field impinging on the tube bank; it is measured by a Pitot tube fixed upstream of the bank;

(ii) U_{per}: Percolation velocity, defined by the mean flow velocity crossing the tube bank,

$$U_{\text{per}} = U_{\text{ref}} \frac{A_p}{A_p - N\pi D^2/4}, \tag{3}$$

where A_p is the total area of the longitudinal section of the bank (cross-section of the tubes), N is the number of tubes of the test-section and D is the diameter of the tubes;

(iii) U_{gap}: Gap velocity, defined by the calculated velocity in the narrow gaps between the tubes in a row, from the reference velocity U_{ref}, based on the effective free area of flow, i.e.,

$$U_{\text{gap}} = U_{\text{ref}} \frac{B}{B - MD}, \tag{4}$$

where B is the channel width, M is the number of tubes in the first row, and D is the tube diameter;

(iv) U_{mea}: Measured gap velocity, defined by the measured velocity in the centre of the narrow gap between the tubes in a row; it is obtained from hot wire measurements except for $P/D = 1{\cdot}05$, since in that case there, the gap is too narrow to place the hot wire probe and a Pitot tube is used instead.

Physical properties in this research study are taken at flow temperature. In problems with heat transfer, the choice of the reference temperature will be an additional problem that will not be discussed in this work.

4. RESULTS AND DISCUSSION

Before starting the experiments, the flow distribution and the turbulence intensity in the test-section were measured, showing a uniform velocity profile with 2% of turbulence intensity. Measurements of vibrations of the test-section (including the tubes of the tube bank) were also performed with the help of a METRA accelerometer, to identify the possible influence of the blower or of its electrical motor. Two important resonance frequencies of the test-section were detected, these being about 2 and 8 kHz.

Prior measurements were also performed in air flow ($\mathrm{Re} = 4{\cdot}9 \times 10^4$) in a 5·4 m long PVC tube with 32·5 mm internal diameter to investigate if the mounting technique of the pressure transducer was adequate (Endres & Möller 1994). The pressure transducers were mounted similarly as shown in Figure 2 at a distance of one diameter before the tube outlet, with three different tubing lengths: 12, 29 and 46 mm. A single hot wire probe was placed at a distance of 2 mm from the pressure transducer.

The use of the 29 mm long plastic tube proved to be adequate for the experiment in tube banks, based on the comparison of the results of spectra from the two longest tubes with those from the shortest one. Discrepancies appeared only after a frequency of 1500 Hz for the tube with 29 mm and after 400 Hz for the longest tube. The amplifying effect was in accordance with predictions by the method proposed by Holmes & Lewis (1987). Cross-correlations between velocity and pressure fluctuations showed no appreciable difference, except for the fact that, as the length of the plastic tube was increased, the curves became rough, without altering their shape.

Figure 5(a, b) shows the behaviour of the four proposed velocities in square and triangular arrays. Due to the different values of pressure drop across the banks, which influenced the absolute value of the velocity, they are scaled with the reference velocity U_{ref}. Lines are for visualization of the results only.

The results present similar behaviour in both geometries. By increasing the P/D ratio, the curves tend to the same value, namely U_{ref}, corresponding to the extreme situation where no tube bank was placed in the channel. This fact shows that, in tube banks with a large P/D ratio, the choice of any of the proposed velocities will lead to similar results. In tube banks

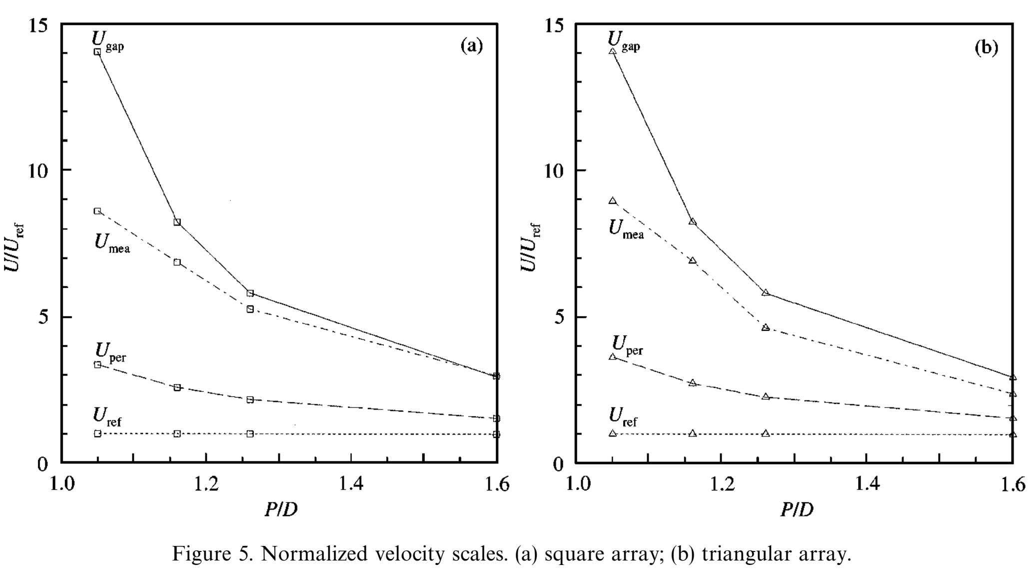

Figure 5. Normalized velocity scales. (a) square array; (b) triangular array.

TABLE 2
Flow parameters for square arrays.

P/D	1·60	1·26	1·16	1·05
U_{ref} (m/s)	7·997	5·679	4·438	2·567
Re(D,U_{ref})	$1{\cdot}70 \times 10^4$	$1{\cdot}17 \times 10^4$	$9{\cdot}29 \times 10^3$	$5{\cdot}34 \times 10^3$
Re(S,U_{ref})	$1{\cdot}02 \times 10^4$	$3{\cdot}05 \times 10^3$	$1{\cdot}49 \times 10^3$	$2{\cdot}67 \times 10^2$
Re(D,U_{per})	$2{\cdot}62 \times 10^4$	$2{\cdot}54 \times 10^4$	$2{\cdot}39 \times 10^4$	$1{\cdot}79 \times 10^4$
Re(S,U_{per})	$1{\cdot}57 \times 10^4$	$6{\cdot}60 \times 10^3$	$3{\cdot}83 \times 10^3$	$8{\cdot}95 \times 10^2$
Re(D,U_{gap})	$5{\cdot}01 \times 10^4$	$6{\cdot}82 \times 10^4$	$7{\cdot}64 \times 10^4$	$7{\cdot}51 \times 10^4$
Re(S,U_{gap})	$3{\cdot}00 \times 10^4$	$1{\cdot}77 \times 10^4$	$1{\cdot}22 \times 10^4$	$3{\cdot}75 \times 10^3$
Re(D,U_{mea})	$5{\cdot}09 \times 10^4$	$6{\cdot}17 \times 10^4$	$6{\cdot}37 \times 10^4$	$4{\cdot}60 \times 10^4$
Re(S,U_{mea})	$3{\cdot}06 \times 10^4$	$1{\cdot}60 \times 10^4$	$1{\cdot}02 \times 10^4$	$2{\cdot}30 \times 10^3$

TABLE 3
Flow parameters for triangular arrays

P/D	1·60	1·26	1·16	1·05
U_{ref} (m/s)	7·968	5·986	4·882	2·872
Re(D,U_{ref})	$1{\cdot}69 \times 10^4$	$1{\cdot}23 \times 10^4$	$1{\cdot}02 \times 10^4$	$5{\cdot}98 \times 10^3$
Re(S,U_{ref})	$1{\cdot}01 \times 10^4$	$3{\cdot}20 \times 10^3$	$1{\cdot}63 \times 10^3$	$2{\cdot}99 \times 10^2$
Re(D,U_{per})	$2{\cdot}63 \times 10^4$	$2{\cdot}78 \times 10^4$	$2{\cdot}77 \times 10^4$	$2{\cdot}16 \times 10^4$
Re(S,U_{per})	$1{\cdot}58 \times 10^4$	$7{\cdot}22 \times 10^3$	$4{\cdot}44 \times 10^3$	$1{\cdot}08 \times 10^3$
Re(D,U_{gap})	$4{\cdot}98 \times 10^4$	$7{\cdot}14 \times 10^4$	$8{\cdot}38 \times 10^4$	$8{\cdot}40 \times 10^4$
Re(S,U_{gap})	$2{\cdot}99 \times 10^4$	$1{\cdot}86 \times 10^4$	$1{\cdot}34 \times 10^4$	$4{\cdot}20 \times 10^3$
Re(D,U_{mea})	$4{\cdot}03 \times 10^4$	$5{\cdot}68 \times 10^4$	$7{\cdot}04 \times 10^4$	$5{\cdot}34 \times 10^4$
Re(S,U_{mea})	$2{\cdot}42 \times 10^4$	$1{\cdot}48 \times 10^4$	$1{\cdot}13 \times 10^4$	$2{\cdot}67 \times 10^3$

with small aspect ratios, important differences in the presentation of the results of fluctuating quantities will arise from the use of any of the proposed velocities. The resulting Reynolds numbers for all experiments, defined with the velocities listed above, using the tube diameter D and gap spacing S are listed in Tables 2 and 3.

The influence of the scale velocity can be observed in the plots of the mean-square value of the wall-pressure fluctuations in square and triangular arrays, shown in Figures 6 and 7, where the defined velocities were used for obtaining dimensionless r.m.s. values of the wall-pressure fluctuation, given by $\overline{p'^2}^{1/2}/\rho U^2$, where U is the chosen scale velocity. No characteristic length is needed. The values of the dimensionless pressure fluctuations change with the scale velocity, so that local maxima of the pressure fluctuations can be clearly observed in square arrays at about 30° for $P/D = 1{\cdot}26$ and at 100° for all geometries.

The same feature appears for $P/D = 1{\cdot}60$ at 60° in triangular arrays. In square arrays, the use of U_{gap} tends to group the curves, with the exception of the smallest P/D. By using U_{mea} the curves are grouped and the maxima can be clearly observed. The use of U_{ref} separates the curves showing the influence of P/D.

In triangular arrays, the use of U_{per} groups the curves, while U_{ref} separates them, showing the influence of increasing P/D ratio. U_{mea} and U_{gap} influence the curves in the same way as U_{ref}.

Figures 8 and 9 show results of spectra of pressure fluctuations for square and triangular arrays. Scale velocities were chosen from the discussion on Figures 6 and 7. All curves for square arrays have the same decay pattern after a Strouhal number, fD/U, St $= 0{\cdot}1$, with dimensionless values of the autospectral densities of the same order, while in the results for

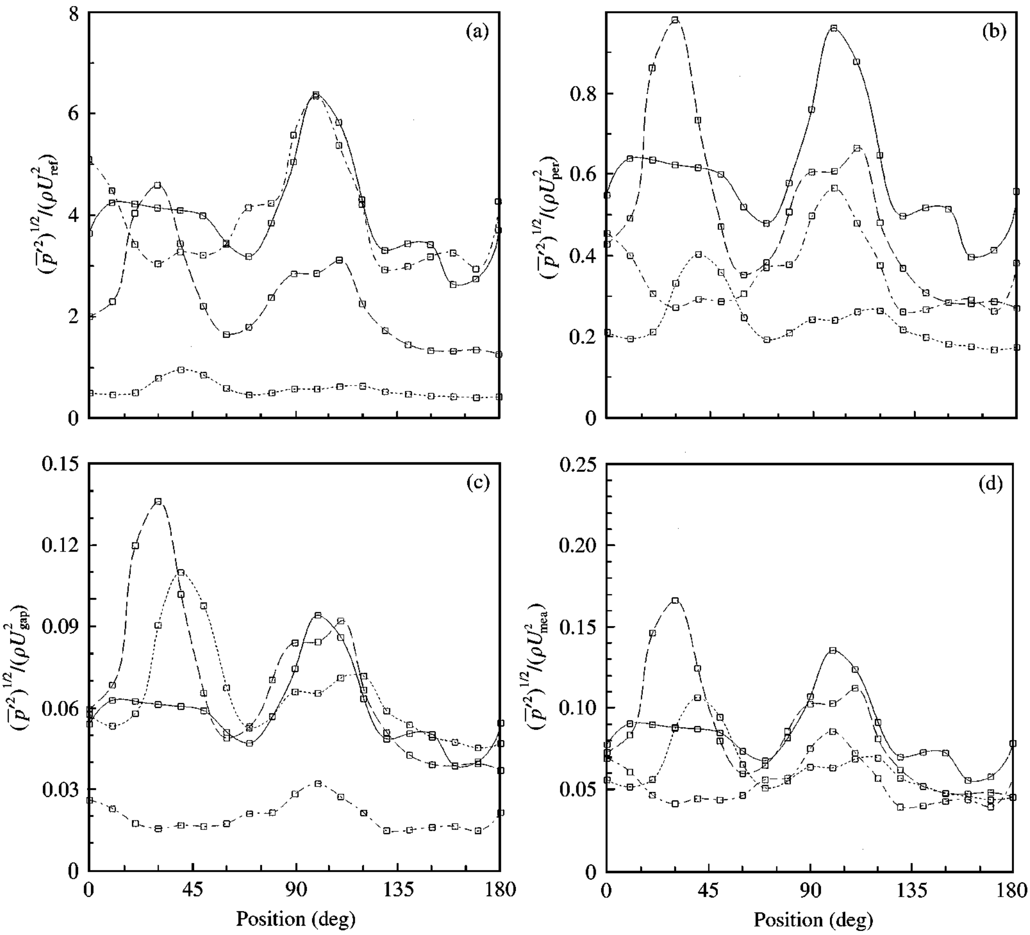

Figure 6. R.m.s. values of the wall pressure fluctuations in square arrays, normalized with the different velocity scales. (a) U_{ref}; (b) U_{per}; (c) U_{gap}; (d) U_{mea}. In this and subsequent figures: – · – · –, $P/D = 1{\cdot}05$; ——, $P/D = 1{\cdot}16$; – – –, $P/D = 1{\cdot}26$; · · · ·, $P/D = 1{\cdot}60$

triangular arrays the spectra have different features for each P/D ratio analysed. All the curves show pronounced peaks at about St = 2·0 and 9·0. These values correspond to the calculated resonance frequencies (Strasberg 1963), inherent of the use of tubings to connect pressure transducer to pressure tap, Figure 2. These peaks can be completely disregarded in this analysis

As expected, in square arrays, Figures 8(a, b), the use of U_{ref} separates the curves showing the influence of P/D. By using U_{mea}, the curves are grouped and a general feature for the spectra can be observed.

In triangular arrays, Figure 9(a, b), the use of U_{ref} separate the curves, showing the influence of increasing P/D ratio. The use of U_{per} groups the curves similarly to U_{mea} for square arrays, emphasizing the different features of the flow in both geometries (Žukauskas 1972).

The influence observed in pressure fluctuation spectra is not so strong in the spectra of velocity fluctuations in both square and triangular arrays, as shown in Figures 10 and 11. This is due to the fact that the chosen reference velocity appears in the denominator of both

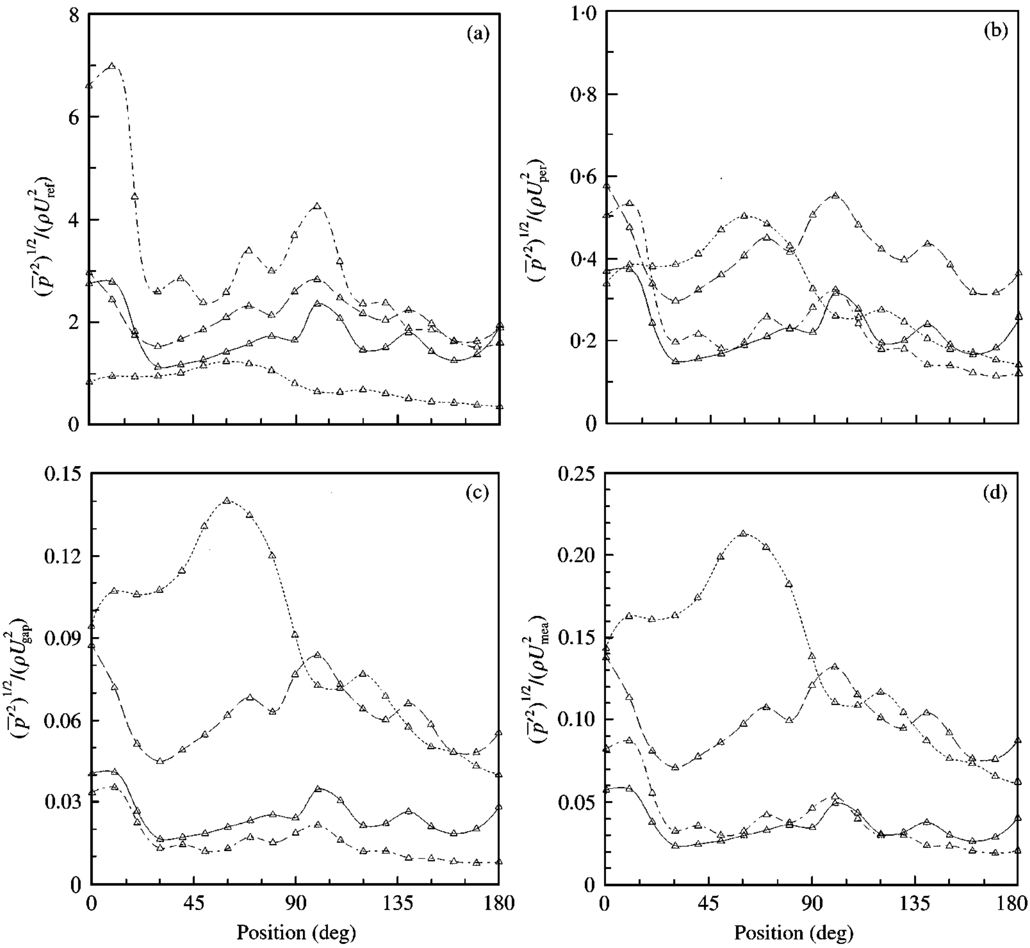

Figure 7. R.m.s. values of the wall pressure fluctuations in triangular arrays, normalized with different velocity scales. (a) U_{ref}, (b) U_{per}, (c) U_{gap}, (d) U_{mea}.

dimensionless spectra and Strouhal numbers, at the first power, in contrast to the spectra of pressure fluctuations where the velocity appears at -3 power. In general, velocity spectra tend to be grouped independently of the chosen reference velocity, in contrast with the results for square array, where the results of the banks with P/D ratios of 1·26 and 1·16 were more close, while the curve for $P/D = 1{\cdot}60$ showed lower values, with a particular behaviour.

For both triangular and square arrays, the influence of reference velocity did not influence the behaviour of the spectra for $P/D = 1{\cdot}60$. This can be explained by the fact that all velocities tend to the same value as the P/D ratio increases, as shown in Figure 5(a, b). No velocity measurements inside the tube bank were performed for $P/D = 1{\cdot}05$ since the gap was too narrow to place the hot wire probe there, as already discussed. The use of the gap spacing in triangular arrays, together with U_{per} grouped the curves better than the tube diameter. Nevertheless, it is worth noticing that the behaviour of the spectra for low P/D ratios is different from the tube bank with $P/D = 1{\cdot}6$, with different slopes. No significant influence was observed in the spectra of velocity fluctuations in square arrangements by the use of the gap spacing as reference length.

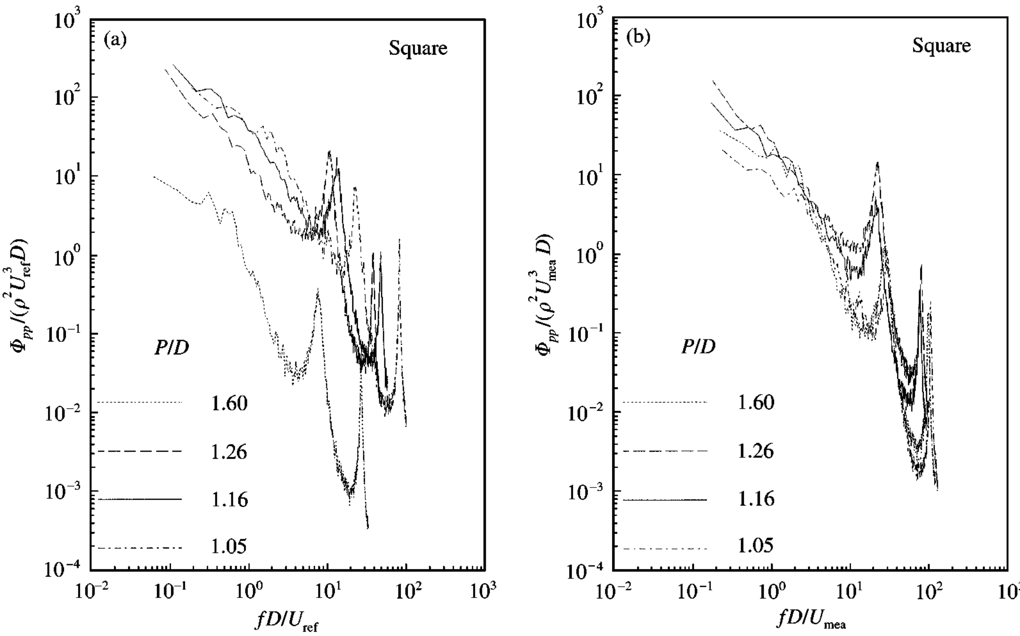

Figure 8. Spectra of wall pressure fluctuations at 90° for square array: (a) normalized with the tube diameter D and U_{ref}; (b) normalized with the tube diameter D and U_{mea}.

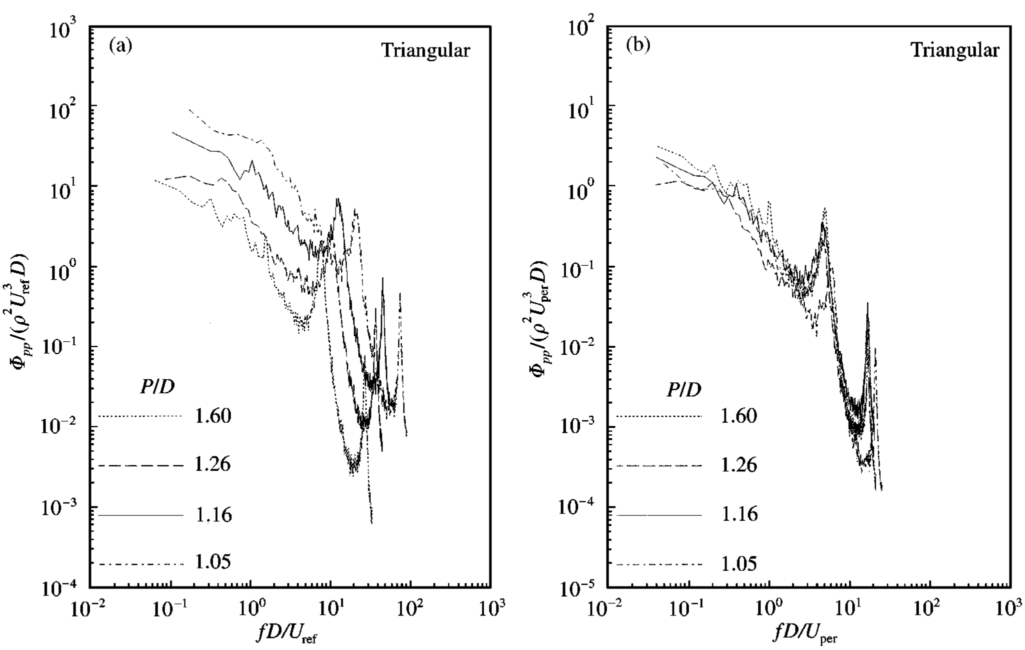

Figure 9. Spectra of wall pressure fluctuations at 90° for triangular array (a) normalized with the tube diameter D and U_{ref}; (b) normalized with the tube diameter D and U_{per}.

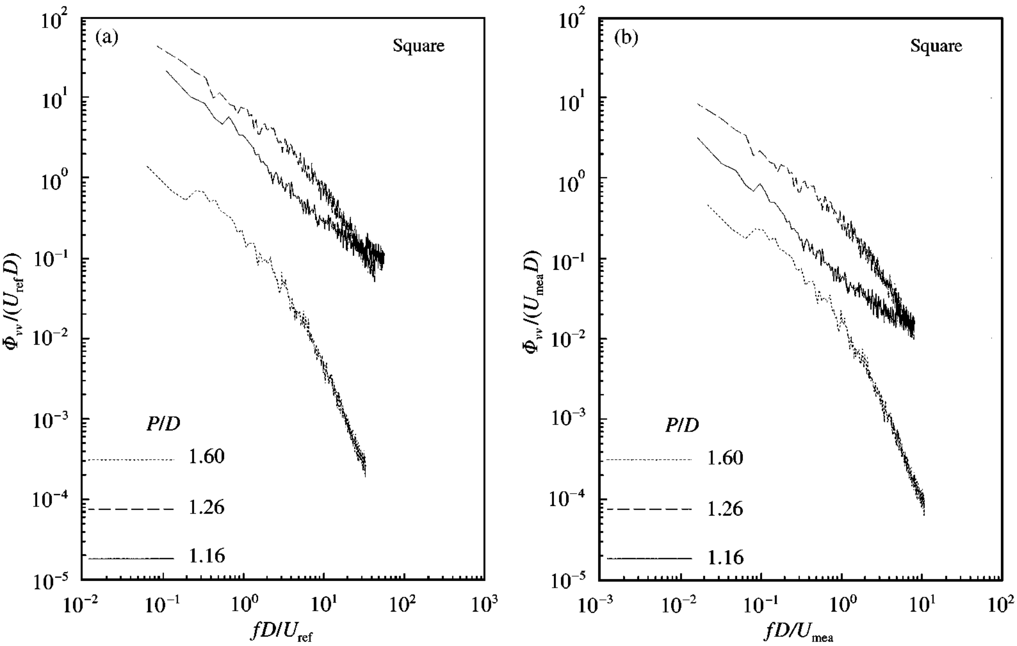

Figure 10. Spectra of velocity fluctuations at 90° for square array (a) normalized with the tube diameter D and U_{ref}; (b) normalized with the tube diameter D and U_{mea}.

5. CONCLUDING REMARKS

This paper presents a discussion about the scales used to present dimensionless experimental results of pressure and velocity fluctuations in the turbulent cross-flow through tube banks with triangular and square arrangements and four different aspect ratios.

In general, the results confirm the description of the flow features, made by Žukauskas (1972) about the different natures of flow in square and in triangular arrays.

Results of spectra of pressure and velocity fluctuations show, in general, that the choice of different velocity or length scales does not affect the decay pattern of the curves. The influence is very similar to that observed in the plots of r.m.s. values of wall pressure fluctuations, grouping the curves or separating them. Therefore, the choice of a certain scale will be related to the interpretation of the phenomenon, in other words, if the curves must appear separated in the plots, in order to describe the influence of, say, P/D ratio, or if it is desired that they appear superposed (or nearly) to determine a universal law.

In this sense, it appears that the definition of velocity and length scales for Reynolds and Euler numbers is not critical in large aspect tube banks, since all velocities tend to the same value, as shown in Figure 5(a,b). Nevertheless, this cannot be observed, neither in the plots of normalized r.m.s. values, nor in the dimensionless spectra of pressure fluctuations, where, as discussed above, by choosing different velocities and length scales, plots will be grouped, leading to a more general definition of dimensionless quantities. From this point of view, triangular and square arrangements will have different reference velocities, U_{per} and U_{mea} (percolation and gap velocity), respectively. This may not be surprising, as described by Žukauskas (1972) in the comparison of the flow through staggered tube banks with the flow in a periodically converging and diverging channel, and of the flow through square arrays with a straight channel.

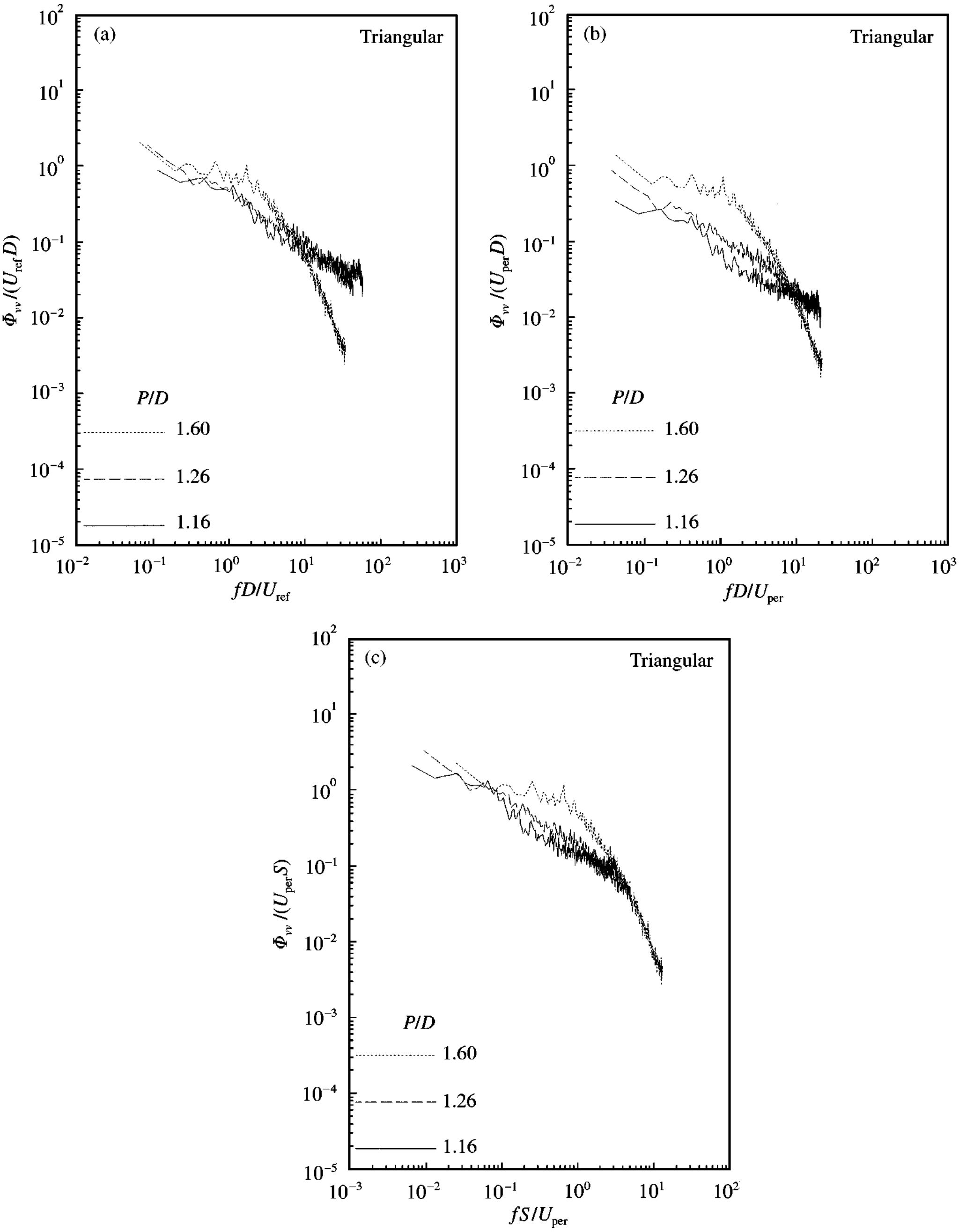

Figure 11. Spectra of velocity fluctuations at 90° for triangular array (a) normalized with the tube diameter D and U_{ref}; (b) normalized with the tube diameter D and U_{per}; (c) normalized with the gap spacing, S and U_{per}.

The use of the gap spacing as the reference length instead of the tube diameter grouped the curves of velocity spectra more closely in triangular arrays. The use of S showed no influence in square arrangements.

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REFERENCES

ACHENBACH, E. 1969 Investigations of the flow through a staggered tube bundle at Reynolds numbers up to $Re = 10^7$. *Wärme- und Stoffübertragung*. **2**, 47–52.

BLEVINS, R. D. 1990 *Flow Induced Vibrations*. New York: van Nostrand-Reinhold.

ENDRES, L. & MOLLER, S. V. 1994 Experimental study of the influence of connections between pressure taps and transducers (in Portuguese). *Proceedings SIBRAT - 8th Brazilian Symposium of Pressure Vessels and Piping*, pp. 416–424, Gramado, RS, Brazil.

ENDRES, L. A. M., SILVA, C. & MÖLLER, S. V. 1995 Experimental study of static and dynamic fluid flow loads in tube banks. In *Transactions SMiRT* 13, *International Conference of Structural Mechanics in Reactor Technology*, Vol. **4**, pp. 451–456, Porto Alegre, BR.

FITZPATRICK, J. A., DONALDSON, I. S., MCKNIGHT, W. 1988 Strouhal numbers for flows in deep tube array models. *Journal of Fluids and Structures*, **2**, 145–160.

GRIMISON, E. D. 1937 Correlation and utilization of new data on flow resistance and heat transfer for cross flow of gases over tube banks. *Transactions of the American Society of Mechanical Engineers*, pp. 583–594. New York: Process Industries Division.

HOLMES, J. D. & LEWIS, R. E. 1987 Optimization of dynamic-pressure-measurement systems, I Single point measurements. *Journal of Wind Engineering and Industrial Aerodynamics* **25**, 249–273.

PAÏDOUSSIS, M. P. 1982 A review of flow-induced vibrations in reactors and reactor components. *Nuclear Engineering and Design* **74**, 31–60.

ROTTA, J. C. 1972 *Turbulente Strömungen*. Stuttgart: B. G. Teubner.

STRASBERG, M. 1963 *Measurement of fluctuating 'static' and total-head pressure in a turbulent wake*. NATO Report - AGARD-R-464, Paris

TOWNSEND, A. A. 1976 *The Structure of Turbulent Shear Flow*. 2nd edition. Cambridge: Cambridge University Press.

WIEMER, P. 1937 Untersuchung über den Zugwiederstand von Wasserrohrkesseln. Dissertation, RWTH - Aachen, Aachen, GER.

WILLMARTH, W. W. 1975 Pressure fluctuations beneath turbulent boundary layers. *Annual Review of Fluid Mechanics* **7**, 13–88.

ZIEREP, J. 1982 *Änlichkeitsgesetze und Modellregeln der Strömungslehre*. Karlsruhe: Braun.

ŽUKAUSKAS, A. A. 1972 Heat transfer from tubes in crossflow. *Advances in Heat Transfer*, **8**, 93–160.