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A REVIEW OF CROSS-FLOW INDUCED VIBRATIONS IN HEAT EXCHANGER TUBE ARRAYS†

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Flow induced vibrations are widely recognized as a major concern in the design of modern tube-and-shell heat exchangers. Tube failures caused by excessive vibrations are relatively commonplace and often very expensive to repair. While considerable progress has been made in the development of predictive tools, many uncertainties still remain. This paper reviews our state of understanding of the cross flow excitation mechanisms and presents design guidelines. Also discussed are the research needs in this field.

1. INTRODUCTION

FLOW INDUCED VIBRATIONS are widely recognized as a major concern in the design of modern tube-and-shell heat exchangers. These problems are especially acute in nuclear steam generators where it has become rather commonplace to shut down nuclear power stations in order to effect very expensive repairs to leaking tubes. While some tube failures occur due to fatigue or thinning and splitting at mid-span as a result of tube-to-tube clashing, most failures are due to fretting wear at the tube supports. Problems typically arise in U-bend regions, where the tube natural frequencies tend to be low, or in areas producing localized high velocities such as entrance and exit nozzles, baffle plates or open tube lanes. These tube failures are invariably associated with high velocity cross-flows.

As a result of all these problems, a great deal of research has been conducted and, especially in the last decade, a substantial literature has developed. A number of papers have summarized the current state of knowledge and presented design guidelines, as in References [1–3], for example. Singh and Soler [4] have written a book on the mechanical design of heat exchangers which includes a chapter on flow induced vibrations. Several recent symposia have been held [5–8]. Probably the most extensive overview has been given by Païdoussis [9]. This research effort has culminated in computer codes for predicting tube response and fretting wear such as reported by Frick *et al.* [10] and Axisa *et al.* [11]. However, the power of such codes often exceeds the quality of the input data. Significant uncertainties remain in nearly every step of the

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prediction process. Indeed, we still do not have a clear understanding of the flow excitation mechanisms.

This paper reviews our state of knowledge of the flow induced vibration phenomena in heat exchangers. Because of the quality and completeness of the review by Païdoussis [9], the emphasis here will be on those contributions and developments since the time of that paper, 1983. Design guidelines will be presented and the areas requiring further research will be discussed.

While some confusion still seems to exist, the flow excitation mechanisms can be broadly classified as (a) turbulent buffeting, (b) Strouhal periodicity, (c) fluidelastic instability and (d) acoustic resonance. These classifications will be used here and treated under separate headings below.

The definitions used in this paper for the various standard tube array patterns are shown in Figure 1.

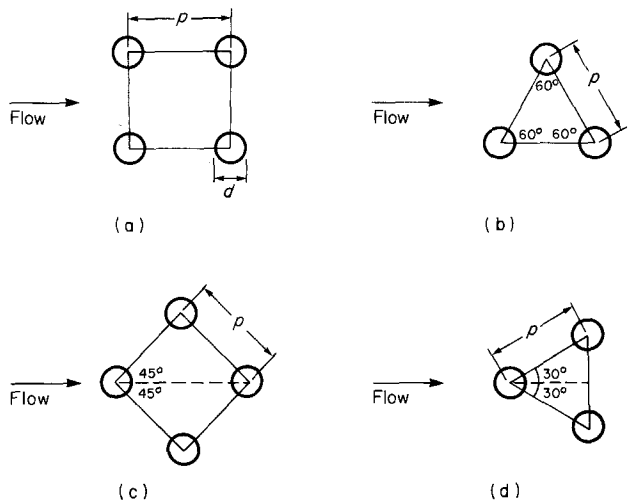

Figure 1. Standard tube array patterns. (a) Normal square (90°). (b) Parallel triangle (60°). (c) Rotated square (45°). (d) Normal triangle (30°).

2. TURBULENT BUFFETING

2.1. State of Knowledge and Design Guidelines

The tubes in tube-and-shell heat exchangers act as turbulence generators and thus, at all practical Reynolds numbers, will be subjected to broad band turbulent buffeting. The nature of this buffeting and its relation to resonant tube vibration and acoustic resonance has been the subject of considerable controversy, the history of which is well covered by Païdoussis [9]. Basically, Owen [12] presented some very cogent arguments to attribute the observed tube vibration and acoustic resonance problems to a relatively broad band peak in the turbulence spectrum. At the same time, he speculated that the coherent wake structures associated with discrete vortex shedding could not occur inside closely spaced tube arrays. Owen used his concept to predict the discrete frequency resonance phenomena called vortex shedding by Chen [13] and, since he used the same basic data to develop his semi-empirical predictive formula, the Owen and Chen predictions agree reasonably well. Païdoussis [9] correctly pointed out that these two approaches are simply different physical explanations for the same unexplained phenomenon. However, the result was a period of confusion in which

relatively broad band turbulence and discrete excitation tended to be treated as one excitation mechanism.

It is now known that turbulence and discrete excitation exist as separate mechanisms as shown, for example, in the tube response spectrum of Figure 2 [14]. This particular array was exposed to a water flow and hence exhibits a cluster of fluid-coupled natural frequencies corresponding to the various relative tube modes. It should be noted that the relatively broad band turbulence excitation has a flattish peak which broadens and shifts to higher frequencies with increasing Reynolds numbers. On the other hand, the so-called "vorticity" peak is a very narrow band process and is seen as essentially a discrete frequency in both hot wire spectra of the interstitial flow and off-resonance tube response spectra. It is important to note that these "turbulence" and "vorticity" peaks in the tube response spectrum are well removed from the tube natural frequencies (Figure 2) and hence are a direct reflection of the flow excitation spectrum. Owen [12] argued that only the broad band turbulence phenomenon could occur inside tube bundles but used this physical explanation to predict the discrete phenomenon called "vorticity" in Figure 2. Only broad band turbulence will be treated in this section.

Since high turbulence levels are omnipresent in heat exchanger tube bundles, some turbulence response will always exist. If this is capable of producing tube failures, it will undoubtedly be due to fretting wear at the tube supports and will only occur after many years of service. Given the fact that equipment such as nuclear steam generators often have amortization periods of 20 to 40 years, turbulence excitation becomes an important design consideration.

Turbulence response design equations have been developed by Pettigrew and Gorman [15] and Blevins *et al.* [16]. As these are discussed and compared by Païdoussis [9], the details will not be repeated here. Suffice it to say that they are both essentially the same, being based on elementary random vibration theory, and predict the mean square resonant response of a lightly damped structure. In order to obtain the necessary force coefficients from response spectra, both approaches require a number of assumptions:

(a) there is no significant off-resonant tube response such as would be produced by a peak in the turbulence spectrum remote from the tube natural frequency;
(b) the turbulence is isotropic and perfectly correlated along the tube length;

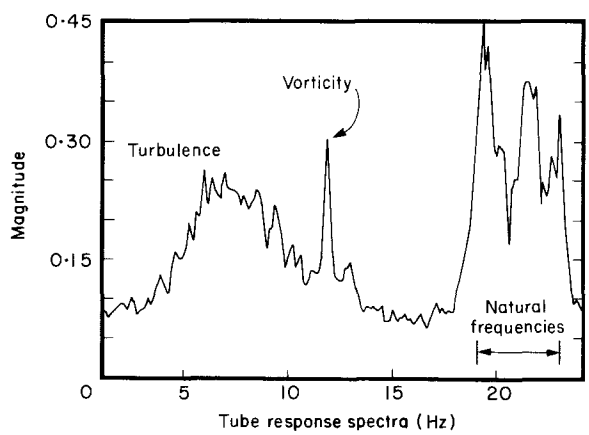

Figure 2. Response spectrum of a tube in a 1·5 pitch ratio rotated square array with water cross flow (Reference [14]).

(c) the power spectral density of the excitation force per unit length is proportional to the dynamic head.

These assumptions permit a force coefficient per unit length to be determined from measured response spectra and plots of these are provided in References [15] and [16] for tube arrays in water and in air respectively. Païdoussis' comparison suggests that these two prediction formulae give mid-span buffeting amplitudes which are of the same order of magnitude [9]. However, our calculations for interior tubes show that the Pettigrew and Gorman curves consistently give predicted amplitudes which are more than 10 times those predicted by Blevins *et al.* This difference is entirely due to the difference in the empirical force coefficients developed by these two groups, as their methodologies are the same. It should be noted that the force coefficient curves of Pettigrew and Gorman indicate that the amplitude response of upstream tubes will be greater than that for interior tubes, while the opposite trend is shown in the curves of Blevins *et al.* This difference is undoubtedly the result of the former data having been obtained from water flow experiments in which the upstream tubes experienced Strouhal-periodicity-excited resonance. On the other hand, the latter data were obtained from wind tunnel tests in which no Strouhal resonance occurred and the upstream turbulence levels were relatively low.

As predictive tools improve for the large amplitude vibration mechanisms which cause relatively short term tube failures, it will become increasingly important to more precisely predict turbulence response. To this end, a number of recent papers have been published. Sandifer and Bailey [17] reported the results of tests in water of a parallel triangular array of tubes with a pitch ratio of 1·5. They found that the theoretical predictions of Pettigrew and Gorman [15] were substantially conservative. Savkar [18] conducted a fundamental study of various tube arrays in an air flow up to Reynolds number over 2×10^5. He found that staggered arrays exhibited a drop in lift and drag coefficients characteristic of the subcritical to supercritical transition occurring for isolated cylinders. He did not observe such a transition for in-line arrays. Savkar also found that, while the lift and drag coefficients were of the same order, the lift coefficients were larger and both coefficients increased with decreasing pitch ratio.

Price *et al.* [19] studied the response of a single flexible tube in a rotated square array with a pitch ratio of 2·12. They found that the turbulence response amplitude increased approximately linearly with flow velocity, rather than velocity squared as is often assumed.

Taylor *et al.* [20] reported some preliminary results of a comprehensive study of buffeting response of tube arrays in liquid and two phase (air–water) cross-flows. While results have only been reported for tube rows, they are quite revealing. As found by Savkar [18], the lift and drag coefficients increase with decreasing pitch ratio. Taylor *et al.* [20] use the same bounding spectra approach used by Blevins *et al.* [16]. The data collapse well for water flows but not for two phase flows where the force coefficients are strongly dependent on void fraction.

The vast majority of the research quoted above has dealt with single span tubes. Au-Yang [21] has extended the approach of Blevins *et al.* [16] and presented analytical expressions for the turbulence response of single span and multi-span tubes which include the effects of correlation length and variable tube mass from span to span.

2.2. Turbulent Buffeting Research Requirements

The design approach proposed by Pettigrew and Gorman [15] and Blevins *et al.* [16] would appear to be appropriate and has the advantage of being quite simple. The use

of bounding spectra to predict upper bounds on tube vibration response appears to work quite well for single phase fluids and it is preferable to plot these in dimensionless form as in Reference [16]. However, the data are sparse and a number of questions must be answered in order to improve the predictions, as follows.

(a) The correlation length of turbulence excitation is likely to be of the order of a few tube diameters while tube span-to-diameter ratios in typical heat exchangers are greater than 30. It follows that the fluid force coefficients which are determined from tube response spectra without regard to correlation length will be of questionable validity. Force coefficients obtained from relatively short laboratory models will likely be significantly higher than those obtained from long tubes. Research is required to determine correlation lengths for turbulence in tube arrays and how these are affected by Reynolds number and tube array geometry.

(b) It is clear that the fluid force coefficients in the lift and drag directions are not the same, especially at larger pitch ratios. It has also been shown that these coefficients increase with decreasing pitch ratio. Additionally, there is increasing evidence that the fluid force coefficients are not linearly proportional to dynamic head, but a more complicated function of flow velocity. Further research is required to quantify these effects.

(c) The turbulence excitation of tube arrays in two phase flows is known to be strongly dependent on void fraction. It has also been shown that the bounding spectra approach used for single phase flows does not collapse the data for different mass fluxed. Research must be conducted to develop a better understanding of turbulence excitation in two phase flows. Care should be taken to determine the effects of both void fraction and pitch ratio.

(d) At present, typical design criteria are based on maximum permissible mid-span tube response. It is not clear that a unique relationship exists between mid-span tube response and fretting wear at the tube supports. Research is required to establish reliable design criteria for tube response due to turbulence excitation.

3. STROUHAL PERIODICITY

3.1. State of Knowledge

As discussed in Section 2.1, a periodic excitation mechanism exists in heat exchanger tube arrays which is distinct from turbulent buffeting. As its precise nature is not known, this mechanism has been variously referred to as vortex shedding, periodic wake shedding, Strouhal excitation and, erroneously, turbulent buffeting. However, tube response data in water and hot wire anemometer data in air show that it is a narrow band (essentially periodic) phenomenon that occurs at a constant Strouhal number.

The effect in water flows across certain tube array geometries is to produce a resonance peak in the response curve very similar to that observed for isolated circular cylinders. An example of such a resonant response is shown in Figure 3 [14]. The vorticity or "Strouhal excitation" points shown in this figure are obtained from response spectra such as shown in Figure 2. Note that the resonant peak in the tube response curve coincides with the intersection of the Strouhal line with the tube natural frequency band and that the Strouhal excitation reappears after a short "lock-in" region. Unfortunately, such a clear resonant peak does not occur for most tube array geometries and mass ratios. Rather, there may be no resonant response at all (typical of gas flows), or a coincidence of resonance with fluidelastic instability (typical of water

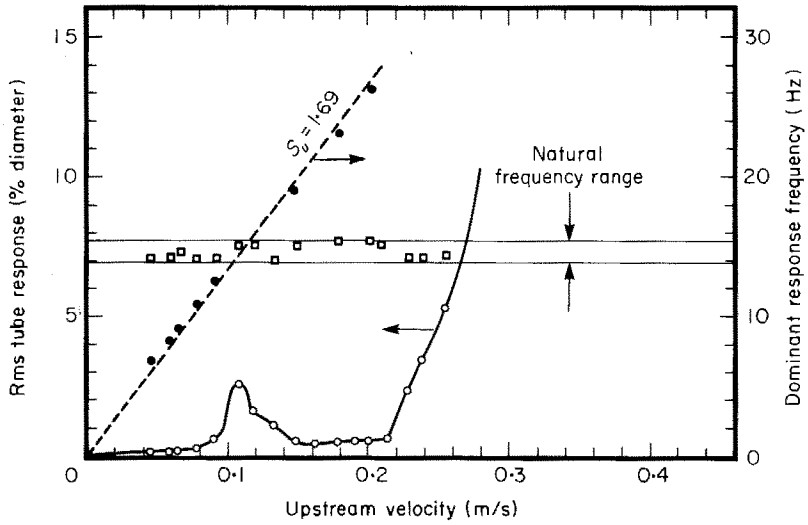

Figure 3. Tube amplitude and frequency response, normal triangular array in water, pitch ratio 1·5 (Reference [14]).

flows). The result of the latter is often a series of lumps in the response curves which make it extremely difficult to distinguish between Strouhal resonance and instability and, of course, to obtain reliable Strouhal numbers. Much of the early Strouhal data was obtained from heat exchangers with acoustic resonance. As discussed by Païdoussis [9], this has partly been responsible for creating such confusion in the Strouhal maps of Fitz-Hugh [22] and Chen [23]. Clearly, Strouhal numbers which have been obtained from either tube resonance or acoustic resonance data must be considered as suspect.

A number of recent papers have been published which provide corroboration of results discussed in Païdoussis' review [9], as well as new information and insights, as follows.

(a) Weaver and Yeung [14] have developed an expression which relates the maximum tube displacement/diameter ratio to the mass damping parameter similar to that used for isolated single cylinders. While the required data for lift coefficients are sparse, the predictions were a reasonable upper bound for the resonant amplitudes observed in their experiments. The important conclusion drawn was that negligible response amplitudes are expected for mass damping parameters greater than about unity. This agrees with the general observation that no significant Strouhal periodicity excited resonance occurs for tube arrays in gas flows.

(b) Axisa *et al.* [24] found that no Strouhal excitation occurred for their square array bundles in air-water and steam-water flow experiments for any but the lowest void fraction experiments. Pettigrew *et al.* [25] reported similar findings for their square and normal triangular arrays in air-water flows. These results corroborate the previous findings reported by Païdoussis [9].

(c) Savkar [18] found that upstream turbulence grids essentially eliminated the Strouhal peaks in his air flow experiments. On the other hand, Gorman [26] found that a number of different turbulence grids did not remove the tube resonance in the first couple of tube rows of arrays subject to water flows. Indeed, only upstream rods with their axes at right angles to the heat exchanger tube banks appeared effective.

(d) Ziada *et al.* [27] conducted experiments on yawed tube banks and found that the independence principle as applied to isolated cylinders is valid for tube arrays. Thus, the Strouhal number for tube arrays with yaw angles up to at least 30° to the flow direction may be computed using the component of the flow velocity which is normal to the tube axis.

(e) Kouba [28] conducted experiments on three different sets of finned tube arrays in a wind tunnel over the Reynolds number range of 10^4–10^5. He found that his Strouhal numbers agreed with those published in the literature for smooth tubes as long as the tube diameter used was the root diameter of the fins.

(f) Weaver *et al.* [29] reported the results of air flow experiments on eight different tube bundles, two different pitch ratios for each of the four standard patterns. Their hot-wire anemometer data showed multiple Strouhal numbers for a square array ($p/d = 1{\cdot}5$), rotated square array ($p/d = 1{\cdot}7$) and a normal triangular array ($p/d = 1{\cdot}5$). They also found that the Strouhal number(s) for a given array may be dependent on Reynolds number. These measurements were all taken behind the third row of four row arrays. Fitzpatrick *et al.* [30] reported hot wire measurements from various in-line arrays in which multiple Strouhal numbers were measured. These were found to be dependent on both position in the tube bank and Reynolds number. Price *et al.* [19] also found multiple Strouhal numbers in their experiments on a rotated square array of tubes with a pitch ratio of 2·12. Agreement between their results in air and water was excellent as one would expect, as long as the Reynolds numbers are comparable. An important finding in those experiments is that only one Strouhal number was found after the fourth row and that one corresponds to that reported in the literature.

(g) Weaver and Abd Rabbo [31] used flow visualization to study flow development in a square array of tubes with a pitch ratio of 1·5. This array in water flow exhibits a large apparent resonant peak in the response curves. The tube motion is highly organized with all tubes in a row oscillating in phase with one another in the streamwise direction and 180° out-of-phase with tubes in adjacent rows. The flow visualization clearly shows that this motion is associated with symmetric vortex shedding.

(h) Abd Rabbo and Weaver [32] published flow visualization photographs of laminar vortex shedding in water flows over a rotated square array of tubes with a pitch ratio of 1·41. Attempts to determine the nature of the Strouhal excitation in the turbulence regime (higher Reynolds numbers) failed to reveal any coherent flow phenomena. However, the current work of Scott and Weaver [33] for several staggered tube arrays has shown that the Strouhal number of the discrete excitation in the turbulent regime is the same as that observed for laminar vortex shedding. Thus, we have at least circumstantial evidence that the much debated Strouhal number excitation phenomenon is turbulent vortex shedding, at least in some array geometries.

3.2. Strouhal Periodicity Design Guidelines

The research to date suggests that Strouhal periodicity, or the vorticity excitation mechanism, is only a potential design problem in the early tube rows of tube arrays in liquid flows (or more precisely, tube arrays with mass-damping parameters less than unity) or as a noise source for inducing acoustic resonance in gas flows. The latter phenomenon will be dealt with below in Section 5.

In order to design against resonance due to flow periodicity, one can attempt to destroy the periodicity with suitable turbulence grids [26]. However, care must be taken in using this approach, as the data are very limited and some grids have been shown to

be ineffective [26]. Alternatively, one can avoid tube resonance by mismatching the frequency of flow periodicity and tube natural frequency. This approach requires knowledge of the appropriate Strouhal number and, to this end, the curves adapted from Weaver *et al.* [29] are given in Figure 4. The Strouhal number used in these plots, S_u, is that based on upstream flow velocity, V_u. The solid lines in these curves were adapted by Weaver *et al.* [29] on the basis of the suggestion by Owen [12] that the frequency should be equal to the interstitial gas velocity divided by twice the distance between successive tube rows. Note that this is not the empirical equation given in Owen's paper and the phenomenon is *not* turbulent buffeting. For comparison purposes, the empirical curves of Žukauskas and Katinas [34] are also plotted. The simple approach of Weaver *et al.* [29] seems to be as good as any, noting of course the scatter in the data and the remarks made earlier regarding multiple Strouhal numbers and Reynolds number dependence.

It may be that mismatching frequencies leads to excessive limitations on flow velocities for certain arrays. In such cases it will be necessary to determine the resonant amplitudes and their potential for causing tube damage. Païdoussis [9] discusses the details, including the general scarcity of the necessary empirical data.

3.3. Strouhal Periodicity Research Requirements

While some real progress has been made in terms of understanding and predicting the Strouhal excitation phenomenon, numerous questions remain unanswered, as follows.

(a) There is now real evidence to suggest that this periodic excitation mechanism is due to vortex shedding, at least for some arrays. An alternative candidate mechanism might be shear layer instability with associated feedback amplification. This latter mechanism has not been observed and the lack of distinct dimensions for such a fluid-resonant phenomenon makes this explanation questionable. (Normally well defined points of flow separation and reattachment are required and these are unlikely in tube arrays.) In any event, further research is required to improve our understanding of the mechanism(s) responsible for Strouhal excitation.

(b) While there exists a substantial amount of Strouhal number data for the various array geometries, there is considerable scatter and evidence of multiple Strouhal numbers for certain arrays. Additional research is required to refine Strouhal number predictions. Part of this must be to determine which Strouhal numbers are dominant in cases of multiple Strouhal numbers and which arrays are susceptible to significant resonant amplitudes. (Some array geometries which exhibit well defined Strouhal excitation are found to have no significant resonant tube response.)

(c) The design approach presented by Pettigrew and Gorman [15] appears quite reasonable, but data are required for force coefficients. These are likely to be strongly dependent on tube pattern and pitch. Reliable design criteria for limiting amplitudes of vibration also need to be determined.

4. FLUIDELASTIC INSTABILITY

4.1. State of Knowledge

Fluidelastic instability is the excitation mechanism with the greatest potential for short term damage to heat exchangers. Païdoussis [35] has reviewed numerous examples of tube failures attributed to this mechanism and a recent paper by Yu [36] graphically illustrates the destructive nature of fluidelastic instability. A U-tube,

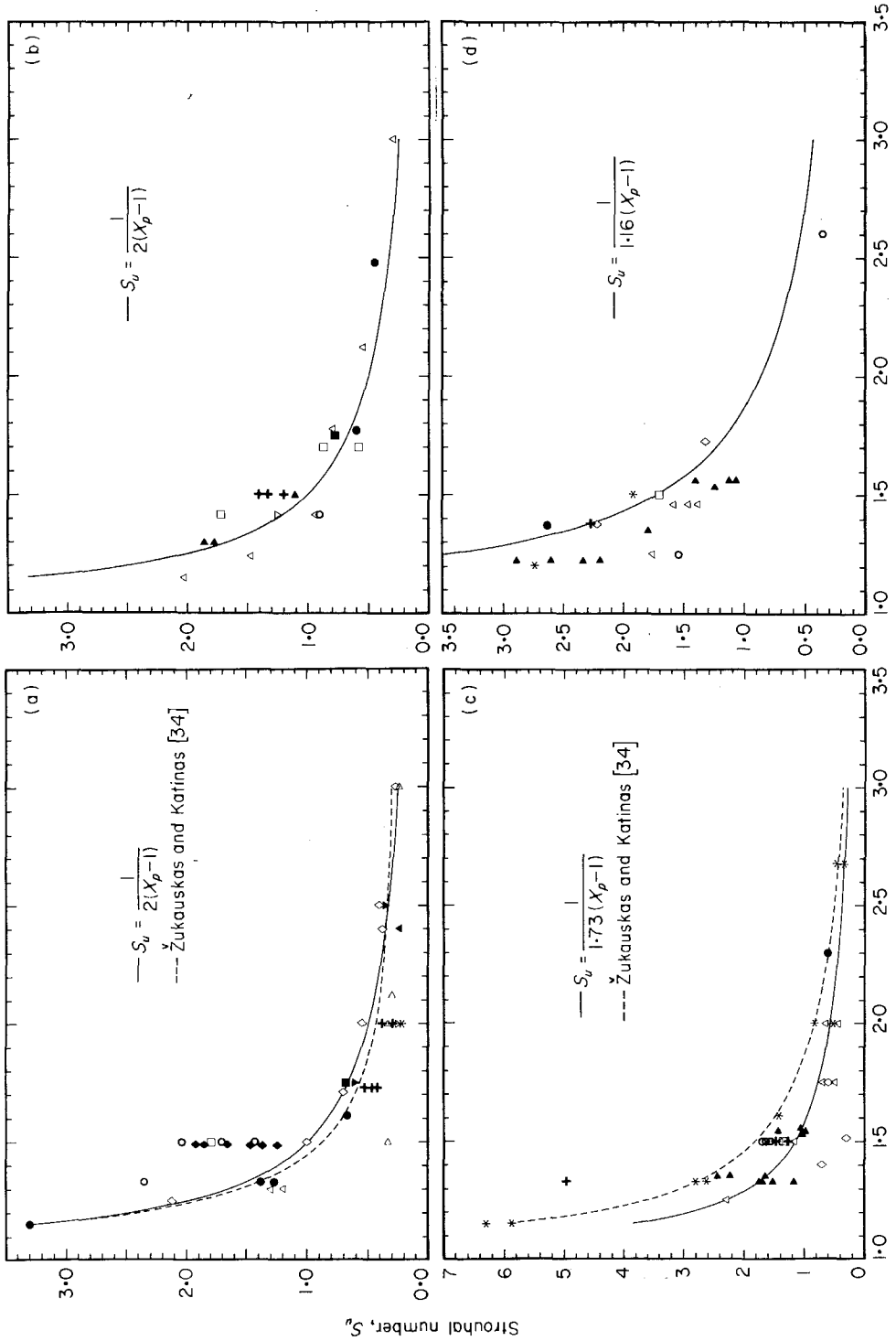

Figure 4. Strouhal number relationships for various array geometries (adapted from Reference [29]. (a) Square arrays. (b) Rotated square arrays. (c) Normal triangular arrays. (d) Parallel triangular arrays.

water-to-water heat exchanger with an equilateral triangular array of 1·5 pitch ratio tubes was found to develop leaks after 60 hours of service. The offending tubes were plugged and the heat exchanger put back into service (or what could be called the ultimate destructive test). After another 290 hours the heat exchanger was disassembled to find that 60 pairs of tubes had failed, seven of these being completely severed through fatigue. Dozens of tubes were worn through from fretting at the support plates or from tube-to-tube clashing in the U-bends. An additional 98 tube pairs were found to have significant wear and the tube support plates literally fell into pieces upon disassembly.

With this potential for short term failure, it is not surprising that this mechanism has received the most attention. Païdoussis [9] has also made it clear that the lack of a uniform definition of stability as well as a lack of consistency in defining the various physical parameters, not to mention the difficulty in measuring some of these, has led to significant scatter in the reported data.

Recent research has largely concentrated on theoretical and experimental work to develop a better understanding of the excitation mechanism and practical work to improve our predictive capabilities. The theoretical work has ranged from models which require an enormous amount of empirical input, to those which require little or none. The model developed by Chen [37] represents the former and, while its publication precedes Païdoussis' paper and was discussed therein, a few words are worthwhile here. Chen's model treats an array of tubes as a full unsteady second order system with matrices of coefficients which account for the motion of a tube in flowing fluid as well as the influence of the motion of surrounding tubes. Unfortunately, many of these coefficients must be determined experimentally and are strongly dependent on tube pattern and pitch. Thus while the stability threshold predictions are good, the results are not generalizable and new measurements need to be made for every tube pattern and pitch of interest. Nevertheless, this work by Chen and the associated results of Tanaka and Takahara [38] have provided some useful insights into the fluidelastic excitation mechanism. In particular, they have elucidated some of the effects on a tube's stability of the motion of neighbouring tubes and provided information relating to the phase relationship between a tube's motion and the net fluid force acting on it. Chen concludes that the dominant influence in heavy fluids is a type of negative damping (tube velocity dependent) while the dominant effect in very light fluids is fluid stiffness related (relative tube displacement dependent).

At the opposite end of the spectrum is the first principles model developed by Lever and Weaver [39]. Based on experimental observations, these authors assumed that each tube in an array becomes independently unstable, the effect of neighbouring tubes being primarily to determine the flow field unique to each array pattern. The fluid mechanics are modelled using the unsteady Bernoulli equation with a phase lag, based on a hydraulic transient analog, to account for the effect of fluid inertia. In spite of the many simplifying assumptions, the predictions for a parallel triangular array with a pitch ratio of 1·375 were shown to agree quite well with experiments.This theoretical model was subsequently modified slightly and extended to examine transverse and streamwise dynamic stability as well as static instability for all of the standard array geometries [40, 41]. It was found that transverse dynamic instability was always critical and that the predictions for several parallel triangular and square arrays was good. Lubin *et al.* [42] plotted this theoretical model for square arrays and found that the predictions for pitch ratios of 1·2 and 2·0 bracketed nearly all of the available experimental data for square arrays. However, the agreement of this theory with experimental results for rotated square and normal triangular arrays is not as good, probably because of the more tortuous flow paths through these geometries.

Motivated by a similar desire to obtain a purely theoretical model for instability, Paidoussis *et al.* [43] derived a theory based on potential flow and incorporating a phase lag between cylinder displacement and displacement dependent fluid forces. While the results were very interesting and the analysis pointed out the errors in previous attempts at such a theory, the agreement with experiments was not particularly good. Thus, these authors modified their theory using the superposition of a viscous cross-flow about a stationary cylinder and a vibration induced potential flow [44]. Again, the agreement with experiments was not good unless empirically based fluid dynamic stiffness terms were incorporated into the analysis. As this defeats the purpose of this approach, the authors concluded that this theory was more useful for the insights gained than for practical purposes. Particularly noted was the importance of fluid dynamic stiffness terms at high values of the mass-damping parameter.

Price *et al.* [45–48, 19] have published a series of papers which report the results of an intermediate approach, i.e., a simplified theoretical model with measured quasi-state fluid force coefficients. They have variously treated a single flexible cylinder in a rigid array [19, 48], a double row of flexible cylinders in a rigid array [45, 46] and full flexible array [47]. In each case, the fluid force coefficients are determined experimentally using a linearized quasi-static approach. The frequency dependent terms which lead to instability arise from an assumed time delay and flow retardation effects. Computational efficiency is achieved by using what the authors refer to as a "constrained mode" analysis. Thus, the analysis considers only certain relative tube modes and the system is reduced to four degrees of freedom. This approach appears to be a reasonable compromise with minimum computational effort and moderate experimental input. In principle, this approach ought to work well and, indeed, a qualitative comparison with experimental data shows the correct trends. For certain arrays, quantitative comparison is good as well, although it is a curious fact that the simpler theory for a single flexible cylinder in a rigid array fits the data better than the more elaborate theories [47].

This theoretical and experimental work of Price *et al.* [19, 45–48] has provided a number of insights into the nature of fluidelastic instability. They found that the measured force coefficients for a double row of flexible tubes were only applicable to a full array of tubes if the measurements were taken with a double row surrounded by rigid tubes [46]. They also reported that fluid stiffness terms became dominant for larger values of the mass-damping parameter. This effect is produced by relative tube motion and it is concluded that the use of a single flexible cylinder in a rigid array of tubes to model fluidelastic instability is not generally applicable and, in particular, is not valid for configurations in which the mass-damping parameter exceeds 300 ($m\delta/\rho d^2 > 300$). Interestingly, they found that a single flexible cylinder in a rotated square array of cylinders with a pitch ratio of 2·12 was stable at all flow velocities [46]. This is in apparent contradiction of the observations of Abd Rabbo and Weaver [32] for a similar array, but with a pitch ratio of 1·41. Clearly, there is a need for further research in this area.

Quite a different approach has been taken by Goyder and Teh [49] although it is based on a single flexible cylinder in a rigid array. With a cleverly designed apparatus, a tube is oscillated at its natural frequency and the net fluid dynamic force measured as a function of flow velocity. The result is used in a design equation to predict the fluidelastic threshold. For a normal triangular array with a pitch ratio of 1·35, instability is predicted in the transverse direction. Additionally, the fluid forces were found to be independent of Reynolds number and to increase linearly with tube displacement. This approach has substantial potential for revealing information regarding the nature of the fluid phenomena leading to instability.

Finally, some interesting practical research has been conducted to evaluate the effects of various heat exchanger design details. Johnson and Schneider [50] and Huff *et al.* [51] have studied the influence of open lanes. Chen *et al.* [52] have examined vibrations in helical tube steam generators and Žukauskas and Katinas [53] studied radial tube bundles. The effects of tubes in loose supports have also received considerable attention [54–58]. It is now possible to compute the nonlinear dynamics and resulting force of tube-baffle interaction and it is generally recommended that tube support clearances be kept as small as possible.

4.2. Fluidelastic Instability Design Guidelines

Despite all of the theoretical work on the subject, the most reliable approach to design against fluidelastic instability is still empirical. Chen [3] collected all of the available experimental data and provided different plots for the standard array geometries. From these plots he determined lower bounds on the data and provided simple analytical expressions for the stability thresholds. Blevins [59] argued that the "inherent variability in the data exceeds any dependence on tube pattern". He proceeded to plot all of the data points on a single graph and computed different power law fits over different ranges of the mass damping parameter. Blevins then recommended the 90% confidence curves for design purposes. In his response, Chen argued that this was a retrograde step [59].

There is no denying that there is significant scatter in the available data. However, there is also substantial evidence to show that the stability threshold is a function of tube array pattern and pitch. Thus, some of the "inherent" scatter in Blevins' curves [57] is not random and the use of these curves may lead to excessive conservatism.

The present authors agree with Chen's approach and gratefully acknowledge Dr Chen's providing them with his data. To this, we have added some recent results including those from References [60, 61] and replotted the stability curves shown in Figure 5. Note that the damping used is that for tubes in still air and that the reduced velocity is based on pitch velocity, $V_p = [p/(p-d)]V_u$, for all arrays. Chen [3] had used different definitions of reduced velocity for the various array geometries in order to account directly for the effects of pitch ratio. It has generally been found that the critical reduced velocity for a given array geometry increases with increased pitch ratio and, hence, it is highly desirable to include pitch ratio in an efficient design guideline. However, for the reasons outlined by Chen [3], there is significant scatter in the data and the present authors feel that there is insufficient consistently obtained data to reliably account for pitch ratio effects. The reduced velocity data supplied by Chen were therefore all converted to pitch velocity for the purpose of the plots presented here. The damping value of the tubes in-vacuo is an appropriate measure of their energy dissipation capacity at the stability threshold, but little error will be introduced by using the damping as measured in still air. The effective tube mass per unit length, m and natural frequency, f, should be those at the stability threshold and, again, little error will be associated with using the values in still fluid. Tube arrays in liquid exhibit a group of fluid coupled natural frequencies depending on relative tube mode. In such cases, the mode with the lowest natural frequency is usually excited first by the flow and it is this frequency which should be used.

The equations for the lower bound curves in Figure 5 are given in Table 1. The inflection point in the curves for all geometries has been taken at the same value of the mass-damping parameter for simplicity. In using these curves, it should be noted that the approach flow direction is very important [62]. If this is not ideal relative to the

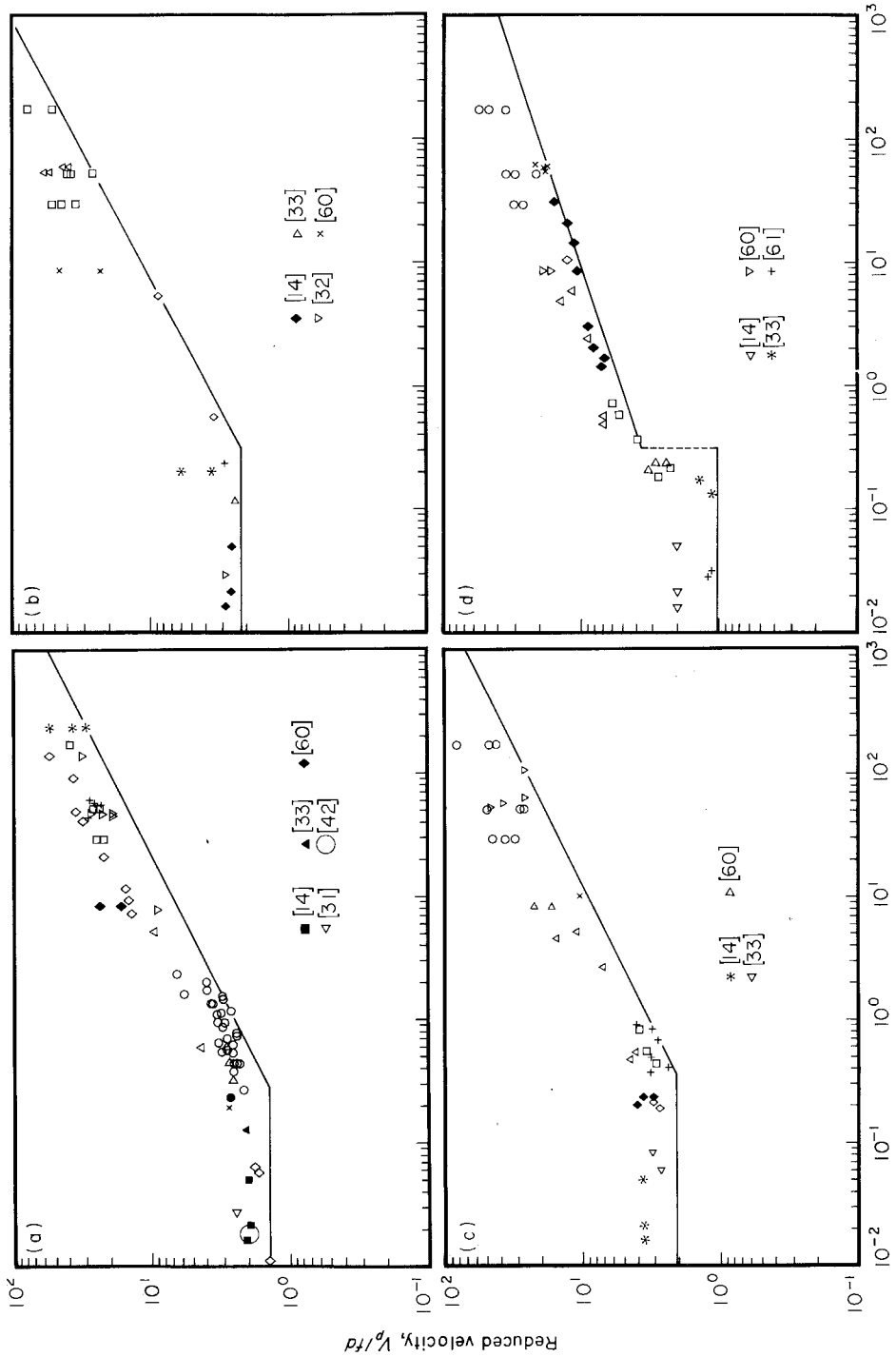

Figure 5. Critical flow velocities for fluidelastic instability for various array geometries (references as shown against symbols; remaining data from Chen [3]). (a) Square arrays. (b) Rotated square arrays. (c) Normal triangular arrays. (d) Parallel triangular arrays.

TABLE 1
Summary of stability thresholds from Figure 5

Array geometry	$m\delta_a/\rho d^2 < 0{\cdot}3$	$m\delta_a/\rho d^2 > 0{\cdot}3$
Square	$V_p/fd = 1{\cdot}4$	$V_p/fd = 2{\cdot}5\,(m\delta_a/\rho d^2)^{0{\cdot}48}$
Rotated square	$V_p/fd = 2{\cdot}2$	$V_p/fd = 4{\cdot}0\,(m\delta_a/\rho d^2)^{0{\cdot}48}$
Normal triangle	$V_p/fd = 2{\cdot}0$	$V_p/fd = 3{\cdot}2\,(m\delta_a/\rho d^2)^{0{\cdot}40}$
Parallel triangle	$V_p/fd = 1{\cdot}0$	$V_p/fd = 4{\cdot}8\,(m\delta_a/\rho d^2)^{0{\cdot}30}$

standard pattern as shown in Figure 1, then the lower of the two possible bounds should be used.

Comparison of these results with those of Chen [3] shows some important differences. One obvious difference is that the present curves are horizontal for values of the mass-damping parameter less than 0·3. This is supported by the systematic study of mass ratio effects by Weaver and Yeung [14] and several of the theoretical predictions [41, 46, 47]. Chen [3] and Tanaka and Takahara [38] found very small slopes for certain arrays at low mass-damping parameters. This insensitivity of the critical reduced velocity to the mass-damping parameter at low values of the latter parameter also makes sense intuitively. For values of reduced velocity less than unity and amplitudes of vibration typical of the onset of fluid-elastic instability, the component of velocity induced by tube oscillation is no longer negligible in comparison with the mean approach flow velocity. The tubes generally become unstable in the direction transverse to the flow. The transverse relative velocity component induced by tube motion, opposes the tube motion and is therefore stabilizing in the absence of a relatively large streamwise flow velocity component.

Other differences between these results and those of Chen are the slope of the curves for higher mass-damping parameters and the existence or not of a vertical jump. Some of these differences may be due to the reduced velocity definitions used by Chen. However, some are also due to interpretation and these serve to emphasize the subjective nature of determining such lower bonds in the presence of scattered and, in some regions, sparse data.

4.3. FLUIDELASTIC INSTABILITY RESEARCH NEEDS

Substantial progress has been made in our ability to predict fluidelastic instability and, with the use of suitable factors of safety, it should be possible to avoid this phenomenon in most conventionally designed equipment. However, a number of uncertainties still exist.

(a) Further fundamental research is required to develop a better understanding of the phenomenon. This will require systematic experimentation and improvements in theoretical models. Aside from academic interests, this will permit prediction of the effects of non-uniform flows produced by open lanes, etc., the effects of frequency detuning of tubes, and will be especially important in the evaluation of novel designs.

(b) Additional carefully determined data are required for improving empirical guidelines. This will permit reduction of the conservativeness typical of many current design approaches. Particularly sparse are data at very low and higher values of the mass-damping parameter. While it has been argued that fluid stiffness is dominant at values of the mass-damping parameter greater than about 300, there are no published data for tube arrays in this range. Considering that many heat exchangers have hot

gases flowing across the tubes, this is a serious deficiency. Also required are additional data to improve our predictions of the effects of pitch ratio.

5. ACOUSTIC RESONANCE

5.1. State of Knowledge

Acoustic resonances may occur in heat exchangers with high velocity gas cross flows and are invariably associated with acoustic standing waves at right angles to both the tubes and the mean flow direction. While the problem may manifest itself in the form of excessive noise levels, often exceeding 160 dB, it may also lead to substantial structural damage to tubes or ductwork. Païdoussis' review [9] outlines many of the uncertainties and controversies which existed at that time. Some interesting progress has been made in the intervening years.

The first concern in designing against acoustic resonance is predicting the excitation frequency for a given tube array and determining whether or not this coincides with an acoustic natural frequency of the tube shell. This excitation mechanism has been the subject of considerable controversy as discussed in Sections 2 and 3 above and, for example, by Païdoussis [9]. The various candidates include turbulence, vortex shedding and more recently, turbulent jet instability [63]. The appellations given have been even more diverse and, indeed, it would appear that the diversity of opinion regarding the mechanism has sometimes obscured the real purpose of the research. Quite correctly, Païdoussis [9] pointed out that various authors were apparently giving different physical explanations for the same phenomenon. In a recent paper, Fitzpatrick [64] has shown that Chen's "modified Strouhal number" [63] based on transverse gap for in-line arrays reduces to a simple relationship based on longitudinal pitch, plotted in reference [29] and reproduced in this paper. Thus, again we have different physical explanations for the same phenomenon.

Studies on the effect of sound on vortex shedding have been conducted by Kacker *et al.* [65] who reported that sound levels of 162 dB were sufficient to correlate the vortex shedding along the length of a cylinder in cross flow, resulting in a factor of 6·5 increase in the force exerted on the cylinder. More recently, Blevins [66, 67] has reported that the effect of sound on vortex shedding from cylinders is similar to that of tube vibration in that there is a "lock-on" effect in which the sound frequency controls the vortex shedding frequency and increases the strength and correlation of the vorticity. Furthermore, he confirmed that the effect of the sound is greatest when the cylinder is located at a pressure node, suggesting that vortex shedding is a dipole type source. Blevins' finding that free stream turbulence suppresses the influence of impressed sound when the turbulence velocity fluctuations exceed the sound particle velocity follows logically. Similar studies have been reported by Archibald [68] and Welsh *et al.* [69] for flow-acoustic interaction with flat plates in ducts.

Blevins [70–72] has extended these fundamental studies to tube arrays. He experimentally confirmed expressions for the effect of tubes on the acoustic velocity in tube banks as derived by Parker [73] and showed that a finite element analysis using this corrected acoustic velocity accurately predicts the acoustic modes and frequencies [70]. In their studies of nine different tube banks, Blevins and Bressler [71] found that for the widely spaced arrays (pitch ratios greater than 2), the acoustic resonance spanned a velocity range 19% below to 29% above the velocity required for coincidence of the Strouhal frequency with the acoustic natural frequency. However, more closely spaces arrays were reported to be "quieter". Blevins and Bressler [72]

found that the Strouhal maps of Fitz-Hugh [22] were better predictors than the available alternatives. However, it must be said that most of their tube banks were widely spaced with pitch ratios between 2 and 4 and Strouhal numbers between 0·19 and 0·27. On the other hand, most of the uncertainty in Strouhal numbers is for tube arrays with pitch ratios less than 2. Blevins and Bressler conclude that, for widely spaced arrays, acoustic resonance is most likely vortex induced. The flow visualization studies by Weaver *et al.* [31, 33] support this conviction.

Fitzpatrick [64] has reported on acoustic resonances for closely spaced in-line arrays and noted that they may be associated with coincidence of a Strouhal frequency with the acoustic natural frequency, with the Strouhal frequency at half the acoustic frequency, or with no integer relationship between these two frequencies. His recommendation for the prediction of resonance for in-line arrays is to use the simple relationship between the Strouhal number and the longitudinal pitch ratio, as detailed in Section 3.2, and to calculate a safe limit velocity using the correlation first suggested by Putnam [74].

The potential for suppression of an acoustic resonance in a given heat exchanger has been addressed by both Grotz and Arnold [75] and Chen and Young [76]. Fitzpatrick [77] suggested an alternative which allows for geometric scaling. Some preliminary tests reported by Fitzpatrick and Donaldson [78] have indicated the likely resonant and non-resonant zones for this criterion. Blevins and Bressler [72] found that both the Grotz and Arnold [75] and Chen and Young [76] criteria predicted resonance in three of their arrays which did not resonate. They presented maps of potential resonance as a function of tube spacing for staggered and in-line arrays.

5.2. Acoustic Resonance Design Guidelines

The design approach is then to compute the transverse acoustic natural frequencies of the heat exchanger shell including the effects of tubes and, using the appropriate Strouhal number for the array from graphs such as Figure 4, determine whether or not the operating flow velocities will produce a coincidence of the Strouhal frequency with the acoustic natural frequency. Care should be taken at this step to examine actual data for similar arrays, as some uncertainty exists in all of the available predictors and exact coincidence of frequencies is not required. For in-line arrays, Fitzpatrick [64] has proposed an equation for calculating a limit velocity. If resonance is predicted, then it is appropriate to consider a damping criterion to see if resonance is likely. Fitzpatrick [64] proposed a criterion for in-line arrays which allows for the calculation of critical duct sizes. Blevins and Bressler [72] presented maps for predicting resonance potential for both in-line and staggered arrays.

When it becomes apparent that a heat exchanger unit will be susceptible to acoustic resonance, some form of remedial action is required. A number of techniques have been suggested, including the installation of baffles [72, 79, 80] and the removal of tubes [79]. The most common method recommended is to use baffles parallel to the flow direction and perpendicular to the standing wave direction. These baffles are located so as to increase the transverse acoustic natural frequencies well above any excitation frequency and are very effective. Eisinger [79] discussed the installation of baffles and outlined the drawbacks of limited service life due to vibration and thermal effects. He suggested a novel approach of attaching fore and aft fins to the tubes in a column parallel to the flow. The result is essentially a perforated baffle which increases the acoustic damping potential of the system. On the other hand, Byrne [80] suggested increasing the acoustic damping using a porous blanket of ceramic or glass fibre.

Blevins and Bressler [72] found that solid baffles were more effective than perforated baffles.

A different approach was proposed by Zdravkovich and Nuttall [81] in which selected tubes are removed from the array such that the acoustic feedback mechanism is destroyed. However, Blevins and Bressler [72] found that removal of 3% to 6% of the tubes in several of their arrays had a negligible effect on sound levels. Removal of 16% of the tubes in one of their arrays did reduce the sound level appreciably.

Blevins and Bressler [72] also discuss the use of a tuned Helmholtz resonator in order to increase the acoustic damping. This approach may be effective in reducing the sound of a specific acoustic mode if the resonator volume is sufficiently large.

5.3. Acoustic Resonance Research Needs

It is clear that uncertainty still remains in both the prediction of Stouhal numbers, especially for smaller pitched arrays, and the damping criteria. It would appear that some minimum energy level (dynamic head) in the turbulence spectrum at the excitation frequency is required to induce resonance and that this is associated with the available acoustic damping. Further fundamental research is required to improve our understanding of the Strouhal phenomenon and thereby the critical Strouhal number predictors. Additionally, research is required to better establish those conditions under which acoustic resonance will occur, given coincidence of the Strouhal frequency with acoustic natural frequencies. It is possible that more than one distinct phenomenon can excite acoustic resonance and that the dominant excitation mechanism may depend on array geometry.

6. DAMPING AND FRETTING WEAR

The review in this paper has emphasized excitation mechanisms and their prediction. In the cases of fluidelastic stability and acoustic resonance, this is generally sufficient, as these excitation mechanisms must be avoided. In the case of vortex shedding resonance in liquid flows, avoiding resonance may lead to excessive conservatism while, in the case of turbulence, avoidance is impossible. For these latter cases, accurate prediction of vibration amplitude and determination of permissible amplitude limits are necessary. While a detailed account of these is beyond the scope of this paper, some brief remarks about recent work may be useful.

In order to predict vibration amplitudes, an estimate of damping is necessary. Unfortunately, damping is not well understood and is generally very difficult to measure, especially for loosely supported heat exchanger tubes. The uncertainty associated with the damping values used could explain some of the scatter observed in the data of Figure 5. Pettigrew *et al.* have analysed a vast amount of damping data for multispan heat exchanger tubes in gases [82] and liquids [83]. They found that the principal damping mechanism in gases is friction between tubes and supports and that this is strongly related to support thickness. In liquid flows, viscous damping and squeeze-film damping at the supports are added to support friction as the most important dissipation mechanisms. Semi-empirical expressions are given for estimating the damping for design purposes.

In heat exchangers which have been designed to avoid fluidelastic instability and acoustic resonance, tube damage will most likely be due to fretting wear at the tube supports. The recent papers by Ko [84], Cha *et al.* [85] and Hofmann *et al.* [86] provide an up-to-date picture of the pertinent research on fretting wear. While significant

progress has been made, further research is required to produce wear data for various materials at conditions typical of heat exchangers in service and to relate the vibration characteristics of heat exchanger tubes in service to material removal rates.

References

1. M. J. Pettigrew 1981 Flow-induced vibration phenomena in nuclear power station components. *Power Industry Research* **1,** 97–133.
2. M. P. Paidoussis 1981 Fluidelastic vibration of cylinder arrays in axial and cross flow: state of the art. *Journal of Sound and Vibration* **76,** 329–360.
3. S. S. Chen 1984 Guidelines for the instability flow velocity of tube arrays in cross flow. *Journal of Sound and Vibration* **93,** 439–455.
4. K. P. Singh and A. I. Soler 1984 *Mechanical Design of Heat Exchangers and Pressure Vessel Components.* New Jersey: Arcutus Publishers.
5. M. P. Paidoussis et al (eds) 1984 ASME *Symposium on Flow Induced Vibrations.* ASME *Winter Annual Meeting,* New Orleans.
6. E. H. Dowell and M. K. Au-Yang (eds) 1985 *ASME Symposium on Fluid-Structure Interaction and Aerodynamic Damping,* Cincinnati, Ohio.
7. S. M. Cho, A. J. Friedland and D. A. Steininger (eds) 1986 ASME Symposium on Thermal Hydraulics and Effects of Nuclear Steam Generators and Heat Exchangers. *ASME Winter Annual Meeting,* HTD **51** Miami Beach, Florida.
8. S. S. Chen, J. C. Simonis and Y. S. Shin (eds) 1986 *ASME Symposium on Flow Induced Vibrations,* PVP 104.
9. M. P. Paidoussis 1983 A review of flow-induced vibrations in reactors and reactor components. *Nuclear Engineering and Design* **74,** 31–60.
10. T. M. Frick, T. E. Sobeck and J. R. Reavis 1984 Overview on the development and implementation of methodologies to compute vibration and wear of steam generator tubes. In Reference 5, Vol. 3, pp. 149–161.
11. F. Axisa, J. Antunes and B. Villard 1986 Overview of numerical methods for predicting flow-induced vibration and wear of heat exchanger tubes. In Reference 8, pp. 147–159.
12. P. R. Owen 1985 Buffeting excitation of boiler tube vibration *Journal of Mechanical Engineering Science,* **7,** 431–439.
13. Y. N. Chen 1968 Flow induced vibration and noise in tube bank heat exchangers due to von Karman streets. *ASME Journal of Engineering for Industry* **90,** 134–146.
14. D. S. Weaver and H. C. Yeung 1984 The effect of tube mass on the flow induced response of various tube arrays in water. *Journal of Sound Vibration* **93,** 409–425.
15. M. J. Pettigrew and D. J. Gorman 1978 Vibration of heat exchanger tube bundles in liquid and two-phase cross-flow *Proceedings of the BNES International Conference on Vibration in Nuclear Plant,* Keswick, U.K., paper 2.3.
16. R. D. Blevins, R. J. Gibert and B. Villard 1981 Experiments on vibration of heat exchanger tube arrays in cross flow. *Transactions of 6th International Conference on Structural Mechanics in Reactor Technology (SMIRT),* paper B6/9.
17. J. B. Sandifer and R. T. Bailey 1984 Turbulent buffeting of tube arrays in liquid cross flow. In Reference 5, Vol. 3, pp. 211–266.
18. S. D. Savkar 1984 Buffeting of cylindrical arrays in cross flow. In Reference 5, Vol. 3, pp. 195–210.
19. S. J. Price, M. P. Paidoussis, R. Macdonald and B. Mark 1987 The flow-induced vibration of a single flexible cylinder in a rotated square array of rigid cylinders with pitch-to-diameter ratio of 2.12. *Journal of Fluids and Structures* **1,** 359–378.
20. C. E. Taylor, M. J. Pettigrew, F. Axisa and B. Villard 1986 Experimental determinations of single and two-phase cross flow-induced forces on tube rows. In Reference 8, 31–40.
21. M. K. Au-Yang 1986 Turbulent buffeting of a multispan tube bundle. *ASME Journal of Vibration, Stress and Reliability in Design* **108,** 150–168.
22. J. S. Fitz-Hugh 1973 Flow-induced vibrations in heat exchangers. Proceedings *UKAEA/NPL International Symposium on Vibration Problems in Industry,* Keswick, U.K., paper 427.
23. Y. N. Chen 1977 The sensitive tube spacing region of the tube bank heat exchangers for fluid-elastic coupling in cross-flow. *ASME Symposium on Fluid-Structure Interaction*