RESPONSE OF A CYLINDRICAL CANTILEVER TO AXIAL AIR WATER FLOW

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ABSTRACT

A review of existing work in the field of flow-induced vibration indicates that the significant driving forces in a two-phase flow originate from turbulence and bubble impacts, and the dominant damping mechanism is associated with the drag forces. Spectral and correlation analysis are important aspects of the experiments performed because the physical processes are random in nature.

Flow-induced vibrations of a cylindrical cantilever in an axially flowing air/water mixture were measured, as well as wall pressure fluctuations near the free end of the vibrating cantilever. The results obtained for void fractions up to approximately 25% and constant mass flow (equivalent to 63.6 litres/min. of water) are presented for the two flow conditions of flow on to the free end of the cantilever ("free-end-flow") and directly on to the fixed end (and thence over the free end) ("fixed-end-flow").

The theoretical prediction of an increase in the amplitude of vibration for increasing voidage was observed, with deviations at low voidages due to turbulence effects and at higher voidages due to flow separation. The theory predicted that the damping and the resonant frequency of the cantilever would increase with increasing voidage, and these general trends were observed.

1. INTRODUCTION

The fact that vibrations may be induced in an elastic solid body by fluid flowing over the body has important consequences in nuclear reactor design. Excessive vibration can lead to premature failure through such mechanisms as rattling, fretting and fatigue. In some applications flow-induced vibrations have led to substantial damage (see Wambsganss [1]). For the satisfactory design of reactors it is very desirable to have available methods for determining the magnitude of vibrations to be expected in a given situation, and this in turn demands a knowledge of the mechanism causing the flow-induced vibrations.

In those types of reactors where boiling occurs, two factors in addition to the effects of single-phase flow have to be considered. The first is the effect of vapour bubble production at the heat transfer surfaces; the second is the impactive effect of the flowing mixture of vapour voids (or bubbles) and liquid phase.

Theoretical models and experimental investigations reported to date have not accounted for this all-inclusive case. Few indeed have proceeded beyond a consideration of a single-phase water system, although there are a limited number of references to two-phase situations.

The investigation reported in this paper examined the vibrations induced in a plain cylinder supported as a simple cantilever by a flowing air/water mixture for axial flow conditions. The cantilever support was chosen to minimise the effect of supporting structures, etc. on the flow.

Measurements were made of vibration amplitude and dynamic wall pressure. The pressure fluctuations were observed at the free end of the cantilever because the impactive effects of void/liquid flow may be expected to have their greatest effect at the least constrained section of the structure.

2. THEORETICAL MODELS

2.1 Review of Previous Theories

The construction of an adequate theoretical model to explain flow-induced vibration presents many difficulties. A particular problem is the form of the coupling between the fluid and the solid. The analysis given here is restricted to parallel or axial flow as this is most typical of the reactor situation and is relevant to the experimental results which are to be discussed. However, there is perhaps a slightly better understanding of cross-flow conditions where the solid reacts to periodic vortex shedding (a decidedly nonlinear process), Marris [2]. One of the difficulties is to ascertain whether the vibrations are simply the direct passive response of a cantilever to pressure fluctuations in the fluid, or whether there exist some forms of self-sustained oscillations excited by the pressure fluctuations.

Historically, the first investigations considered that the equation of motion for flexural vibration of a rod had an extra dynamic term, to explain the hydrodynamic interaction, of the form $\rho_w DV^2$, where ρ_w is the density of water, D the hydraulic diameter, and V the velocity of the water (see Burgreen, Byrnes, Benforado [3]). In this approach the equation of motion was not solved, but rather used to create a series of dimensionless parameters characterising the fluid flow and the vibration of the body. Combinations of these parameters were used, together with an end-fixity factor (expressing the constraints arising from the method of fixing the rod) to provide a formula to fit the measured data. This formula was then referred to as a correlation.

Quinn [4] considered that centrifugal forces arising from the motion of the fluid around bowed rods was a dominant factor and developed expressions for the characteristics of the flow-induced vibration after some elegant mathematics. He reported some increase in vibration level when using a 7% voidage air/water mixture exciting a rod bundle. We may call Quinn the first exponent of the self-sustained oscillation theory.

Paidoussis [5] also developed a theory for self-sustained oscillations taking into account the forces due to bowing as considered by Quinn and also the velocity dependent drag forces which can act on a body across which fluid is flowing. The results were then used to derive a correlation (see Paidoussis [6]) superior to that used by Burgreen et al. [3]. The effect of two-phase flow in this correlation was allowed for by using the effective two-phase density in the expression.

Reavis [7] developed a correlation by considering that the vibration in a rod was due to the forces exerted on the rod due to the turbulent nature of fluid flow, and he used the data from gas studies in his calculations. Whereas Reavis [7] used a simple model to describe the motion of the rod, Chen and Wambsganss [8] attempted a more advanced analysis by adopting the model of Paidoussis [5] and incorporating the additional forces due to turbulence. Basile, Fauré and Ohlmer [9] made measurements on flow-induced vibrations for changes in such parameters as flow rate and hydraulic diameter.

To date, no correlation has been vindicated as successful in all cases even after correcting for the fact that the various investigators often used different measures to specify the motion (for example: average, true r.m.s., peak).

One worker (Gorman [10]) carried out a more basic investigation and utilized differential wall pressure data, obtained from the actual experimental rig, in a turbulence correlation formula. He obtained reasonable agreement between theory and experiment and thus felt that his results verified the hypothesis that a rod responds to fluctuating pressures in the turbulent layer near the rod for one-phase flow.

2.2 Theory for Two-Phase Flow Induced Vibrations

Having regard to the present relatively undeveloped state of the theoretical models, especially in relation to two-phase flow, an essentially morphological approach was employed in the preliminary work reported here. The approach was to monitor parameters of interest (flow, voidage, wall pressure and amplitude of vibration) and then, using noise analysis techniques, to try to discern by inspection what possible relationships might exist between the various parameters. Nevertheless, before reporting on the results of the investigations, it seems useful to attempt some analytical conjectures as to what effects should be expected for vibrations induced by two-phase flow.

2.2.1 Excitation processes

We will assume that the dominant mechanism for rod excitation in isothermal two-phase flow arises from the impulses that the rod receives on contact with a bubble due to a sudden localised transition from liquid to gas. We will further assume that the effect of changing the voidage is only to change the number of bubbles and not their size (at least for up to 25% voidage). In these circumstances we are dealing with a single degree of freedom system (a rod vibrating in its fundamental flexural mode) excited by a random binary sequence. Following the approach of Rice [11] and assuming the impulses from the gas-liquid interfaces are random in time and space, we expect the r.m.s. vibration amplitude to increase as the voidage, α, because the energy transferred to the rod is proportional to the rate of bubble impact (since we have assumed all bubbles are the same size). The vibration frequency of the rod will then be at its own mechanical resonant frequency and will not be influenced by the impact rate. There is also an extra effect, particularly at low values of voidage, due to turbulence.

2.2.2 Resonant frequency of the rod

The resonant frequency of the rod depends both on its inherent characteristics and on the mass loading by the fluid. Therefore we may expect an increase in this frequency as the fluid density decreases. Since we are dealing with a displacement resonance, the change in resonant frequency will also depend on the change in damping, which is also a function of voidage.

2.2.3 Damping mechanisms

The energy loss mechanisms for a rod vibrating in water are internal friction in the solid material, external fluid friction due to the viscosity of the fluid, and drag forces on the rod due to the bulk flow of fluid. The drag forces are a function of the fluid velocity and are the major contribution the flowing fluid makes to the damping of the rod. The drag forces depend on the density of the fluid, and hence it appears that the average damping would decrease as the void fraction increases. It can be demonstrated fairly readily that the fluid damping probably increases with increasing voidage up to 50%, and thereafter decreases to zero when no liquid, only gas, is present.

To show this effect we follow Paidoussis [5]. The drag forces are divided into normal (transverse) and tangential (longitudinal) components. The normal component is of prime importance in determining the damping of flexural vibrations, and is of the form

$$F_N = (1/2)C_N \rho \left[U^2 \partial y/\partial x + U \partial y/\partial t\right] , \qquad (1)$$

where F_N is the normal drag force per unit projected area of the rod, C_N is a drag coefficient, U is the fluid velocity, ρ the average fluid density of the two-phase mixture, y the flexural (transverse) displacement and x is measured along the rod. We identify $\partial y/\partial x$ as the local instantaneous inclination of an element of the rod to the average direction of flow and $\partial y/\partial t$ as the instantaneous transverse velocity of an element of the rod. Now for voidage fractions up to 25% the fluid velocity U is approximately constant, and the average value of the fluid density is given by

$$\rho = \rho_w (1 - \alpha) ,$$

where ρ_w is the density of the water (the effect of the gas density is assumed negligible). Since the displacement has been assumed to be proportional to voidage, the r.m.s. value of $\partial y/\partial t$ and hence also $\partial y/\partial x$ will also be proportional to voidage. A term has to be included to represent the turbulence contribution and this is assumed constant for a given mass flow. We therefore assume that the r.m.s. values of both $\partial y/\partial t$ and $\partial y/\partial x$ are proportional to $(K_o + C\alpha)$ where K_o and C are constants; K_o relating to the turbulence contribution and C to the two-phase contribution. Now we can write down the ratio of the r.m.s. drag forces for voidage α and single-phase flow using eq. (1) (letting # imply r.m.s.):

$$\frac{\left(F_N^{\#}\right)_{\alpha=\alpha}}{\left(F_N^{\#}\right)_{\alpha=o}} = \frac{(1 - \alpha)\ (K_o + C\alpha)}{K_o} . \qquad (2)$$

Now, for increasing voidage, the K_o representing the turbulence contribution should become negligible in the numerator of eq. (2), which then becomes

$$\frac{\left(F_N^{\#}\right)_{\alpha=\alpha}}{\left(F_N^{\#}\right)_{\alpha=o}} \doteqdot \left(\frac{C}{K_o}\right)\alpha \quad (1-\alpha) \quad . \tag{2'}$$

The exact form of the damping as a function of voidage for small void fractions depends on the relative sizes of K_o and C. Since we have no reliable estimate of these, no quantitative predictions can be made. However, for larger void fractions (which are, of course, beyond the validity of the assumptions for this derivation) eq. (2') predicts that the damping will increase with voidage up to a voidage fraction of 50% and then decrease. The experiments reported here were limited to values of voidage up to 25% and over this range a general increase in the damping was to be expected. It will be seen that this increase did occur.

3. ANALYTICAL TECHNIQUES USED ON RESULTS

Since the signals received in experiments on flow-induced vibrations are predominantly random, two complementary techniques of signal analysis were employed; correlation and spectral (frequency) analysis. The autocorrelation coefficient, $R_{xx}(\tau)$, is defined in the usual way for a signal, $x(t)$, as

$$R_{xx}(\tau) = \frac{1}{T}\int_o^T x(t)\, x(t+\tau)dt \quad , \tag{3}$$

where τ is the time lag. Strictly, eq. (3) only provides an estimate of $R_{xx}(\tau)$ since we really require the limit as T tends to infinity. However, we assume that T is large enough for any error to be negligible. Several useful parameters are immediately available from the autocorrelation. The autocorrelation for zero time lag ($\tau = 0$) for signals of zero mean is the variance ($\sigma^2 = R_{xx}(o)$), and the square root (σ) is the true r.m.s. value. The measurement of $R_{xx}(o)$ for very large values of T gives a much better estimate of the r.m.s. value of a signal than conventional r.m.s. meters. For a single-degree-of-freedom system excited by a white noise source for the range of frequencies of significance to the system, the autocorrelation will be in the form of a cosine wave with a decaying exponential envelope. The damping coefficient of this cosine wave (usually computed via calculations of the log decrement) is the same as that of the system itself. The periodicity in the autocorrelation has the same frequency as the resonant frequency of the system.

The cross-correlation coefficient, $R_{xy}(\tau)$, for two signals, $x(t)$ and $y(t)$, is defined in the usual way as

$$R_{xy}(\tau) = \frac{1}{T}\int_o^T x(t)\, y(t+\tau)dt \quad . \tag{4}$$

If there is some relationship between two variables, in particular the wall pressure fluctuations and the induced vibration, then the cross-correlation should reveal it.

Spectral analysis is usually stated in terms of the power spectral density, $S_{xx}(\omega)$, which with $R_{xx}(\tau)$ forms a Fourier transform pair. The energy lying in the frequency range ω to $\omega + \delta\omega$ is given by $S_{xx}(\omega).\delta\omega$. A power spectral density analyser gives an output in $(\text{volts})^2/\text{Hz}$. The device employed in this study did not have a squaring amplifier so the output was in volts/Hz. An examination of the power spectral density curve for a particular signal yields the following:

(i) the value of any discrete frequencies present and their magnitude,

(ii) the range of frequencies present in the signal,

(iii) the damping of the system by measurements of the bandwidth of a resonance (frequency separation of -3 dB points),

(iv) the effective amplitude for the pressure which is in the frequency range to which the rod could respond.

A further parameter that can be employed is the probability density function, $p(x)$, where the probability that a value of x in the range x to $x + \delta x$ may occur, is $p(x).\delta x$. The form of $p(x)$ for most random signals is a Gaussian curve and often only deviates from this form where there is some nonlinearity in the system.

4. EXPERIMENTAL RIG

4.1 Basic Experimental Arrangement

Figure 1 is a diagram of the test section. The fluid circuit was constructed of Perspex pipe with bolted flange connections. The fluid flow could be arranged to be upwards (flow onto the free end of the cantilever) or downwards (flow over support on to the free end). The fluid (water or air/water mixture) entering at the bottom of the rig passed through a flow straightener and then after a 0.61m (2 ft) riser entered the 0.91m (3 ft) test section. The fluid continued through another short vertical section of pipe before returning to the rest of the circuit.

The circulation pump was an Ajax type CA1 single impeller centrifugal pump operating at a constant speed. The water flow rate was controlled by both a main valve and a bypass valve, which returned water to the header tank. The conductivity of the water was adjusted by circulating the water, as required, by means of a small positive displacement pump from the header through a de-ionising column. This enabled a constant low conductivity to be maintained and prevented any mineral deposition in the system. Air was introduced into the water system by means of a porous sinter in the form of a cylindrical bush and the volume flow rate of air was measured by air rotameters. The water flow was measured by a turbine flowmeter. The circuit contained a voidage calibration section which enabled a measured length of transparent vertical pipe to be isolated from the system by a pair of pneumatically operated spade valves. The air and water phases separated out in the isolated section and thus an accurate measure of voidage could be made from measurements of the height of water in the section.

4.2 Test Section

The test section consisted of a 0.91m (3 ft) length of Perspex tube of 2.54 cm (1 inch) I.D. which contained a 0.953 cm (3/8 inch) diameter aluminium rod of 50.8 cm (20 inch) length concentric with the tube. The rod was supported rigidly at one end by a triple-arched support shaped aerodynamically to minimise its effect on the flow. The other end of the rod was free.

The upper half of the sealing flange contained a number of radially orientated copper leads bonded by an epoxy resin to form insulated lead-throughs. The output leads from the strain gauges on the fixed end of the cantilever were soldered to these lead-throughs.

Special attention was paid to the mountings of the test section to achieve vibration isolation. The sealing flanges at each end of the test section were clamped to a wooden mounting board by means of massive split metal blocks mounted on slabs of rubber. The mounting board was in turn supported by a wooden stand which was wedged between the floor and the ceiling by means of wooden plates and slabs of rubber.

4.3 Instrumentation

The vibration of the cantilever was measured by means of two pairs of Philips semi-conductor strain gauges (type PR 9860). Each pair was mounted on opposite sides of the rods and the axes joining the two pairs were set at right angles to each other. The strain gauge assembly was coated with a waterproofing cement. The strain gauge pairs were incorporated in a Wheatstone bridge and the excitation voltage was 5 volts d.c. The output signals were amplified by Preston differential input amplifiers (type 8300 XWB).

The wall pressure at the free end of the cantilever was measured by two differential pressure transducers. The four pressure tappings were spaced uniformly around the wall of the test section and each transducer used the pair of tappings at opposite sides. Any harmful effects arising from the use of short tubes from the pressure tappings to the transducers would occur at frequencies well above those of significance. It was found that by slightly inclining these tubes no air bubbles were trapped when using an air/water mixture. The pressure transducers (S.E. Laboratories type 42/L/V/BB) were of the differential diaphragm type employing strain gauges to sense diaphragm displacement, and the outputs were amplified by Brookdeal amplifiers (type 450). The signals from the strain gauges and the pressure transducers were recorded on an Ampex FR1300 tape recorder and subsequently analysed using a Hewlett-Packard type 3721A correlator and a Spectral Dynamics type SD301B real time spectrum analyser.

5. RESULTS AND DISCUSSION

5.1 Test Conditions

The mass flow was kept approximately constant. For the single-phase (water) flow the flow rate was 63.6 litres/minute (14 gpm) which gave a flow velocity of 2.06 m/s (6.76 ft/sec). The Reynolds number for single-phase flow (based on the annular gap) was 69,100.

5.2 Basic Vibration Characteristics of Rod

The basic vibration characteristics of the cantilever were measured both in still air and stagnant water using external impact excitation and analysis via the autocorrelation. The results obtained are summarised below.

	Resonant Frequency	Logarithmic Decrement
Air	19.7 Hz	1.63
Stagnant water	16.4 Hz	2.09

5.3 Typical Results for Vibration and Wall Pressure (Frequency Spectra, Autocorrelation and Probability Density Function)

Figure 2 is a typical amplitude spectral analysis for the differential wall pressure and the dynamic strain resulting from the vibrations of the rod. The hump in the pressure spectrum was present for all voidage fractions being apparently associated with the annular fluid channel geometry in the experimental rig. A typical autocorrelation of the flow-induced strain in the cantilever is shown in Figure 3 and a classical exponentially decaying cosine wave is apparent. Figure 4 gives the probability distribution function, p(X),for the strain, X, in the cantilever, and shows a typical Gaussian characteristic.

5.4 Results for Total Vibration for Two Flow Conditions

Since comparison tests showed that the separate pairs of transducers had similar outputs, the results that follow are presented for one set of strain gauges and one differential pressure transducer. It was also evident from comparison tests that the plane of vibration of the rod was random, but unfortunately no technique was available to determine the nature of this randomness (for example, whether the plane of vibration precesses).

Figure 5 gives the total r.m.s. strain in the cantilever, which is a direct measure of the amplitude of vibration, for the two flow conditions; henceforth flow on to the free end is referred to as 'free-end-flow' and flow over the support on to the fixed end is called 'fixed-end-flow'. The free-end-flow situation shows a general increase in vibration with increasing voidage as the theory predicted but with two important departures from the trend; first at the lower voidages and then at the highest.

A possible explanation of these effects is as follows. There was severe turbulence at the free end of the rod when the fluid passed over it and this gave rise to the vibration at zero voidage. Hence, because of this turbulence, it seems reasonable to expect that the addition to the flow of a small volume of bubbles would initially serve only to absorb energy from the turbulence and so decrease the vibration. Eventually this effect would be overtaken when sufficient bubbles were present for the two-phase forces to become significant.

The decrease of the magnitude of the induced vibration at the higher voidage may be associated with the flow separation which started at about 10% voidage. The form of this separation is depicted in Figure 6.

The fixed-end-flow situation shows a more predictable characteristic with the amplitude of vibration (r.m.s. strain) increasing approximately linearly with void fraction, essentially as predicted by the theory. However, the magnitude of the vibration does not go to zero for zero void fraction, owing to the turbulence contributions.

5.5 Results for Total Differential Wall Pressure for Two Flow Conditions

The differential wall pressure fluctuations are assumed to be similar to those at the rod, and hence give a measure of the forces causing the flow-induced vibration. Figure 7 gives the total r.m.s. differential wall pressure for the two flow conditions; at about 10%

voidage a change occurred in the expected rising characteristic of the wall pressure as a function of voidage. It evidently occurred after separation, and was very dramatic for free-end-flow when the separated air layer extended over a substantial part of the cantilever. This effect increased as the voidage increased.

5.6 Results for Vibration and Wall Pressure at Resonance of the Rod

It is expected that the rod will respond primarily at its resonant frequency to both turbulent and bubble impact forces. Some insight into the basic mechanisms should be obtained from an examination of the amplitude response of the rod at its resonant frequency in relation to the magnitude of the average wall pressure fluctuations for a band of frequencies centred on the rod resonant frequency.

Figure 8 gives the amplitude of vibration (r.m.s. strain) at resonance (found by measuring the height of the resonance peak) and also the r.m.s. wall pressure around resonance for free-end-flow. (The r.m.s. wall pressure was found from the pressure spectrum by taking an average over the frequencies corresponding to the -3 dB points of the vibration resonance peak.) The strain shows the predicted increase in vibration amplitude with voidage except for low void fractions where the gas bubbles simply reduce the turbulence contribution. The wall pressure also shows the rising characteristic until flow separation occurs, after which the wall pressure is no longer a good measure of the forces acting on the cantilever.

Figure 9 gives the relative amplitude of vibration (r.m.s. strain) at resonance and r.m.s. wall pressure around resonance as a function of voidage for the fixed-end-flow. Both curves are now similar with rising characteristics, except for a slight dip in the strain curve at low void fraction, again probably due to the absorption of turbulence energy by the gas bubbles. The phenomenon of separation did not drastically upset the wall pressure measurements in this case as the separated layer did not extend over any part of the rod.

5.7 Resonant Frequency and Damping for Two Flow Conditions

Figure 10 gives both the resonant frequency and logarithmic decrement calculated from both $R_{xx}(\tau)$ and $S_{xx}(\omega)$ as a function of voidage for free-end-flow. The apparent 1.4 Hz offset between the two estimates of the resonant frequency is not yet understood, but both estimates show an almost identical upward trend as predicted. The errors in the frequency measurement are of the order of 0.4 Hz for the determination from $R_{xx}(\tau)$ and 0.3 Hz for the determination from $S_{xx}(\omega)$. The logarithmic decrement calculated from the two measurements shows a generally upward trend as predicted.

Figure 11 gives both the resonant frequency and logarithmic decrement for fixed-end-flow. The results are much more erratic than for free-end-flow although the overall trend is for a rise in both the resonant frequency and decrement as a function of void fraction. Note the uncertainty in the logarithmic decrement at the highest void fraction as calculated from $R_{xx}(\tau)$ where the decay was apparently not exponential. The initial (zero voidage) resonant frequency and decrement are higher than for free-end-flow, which is anomalous as an increase in one quantity should lead to a decrease in the other. However, the results must reflect the fact that the equations of motion are basically nonlinear (see Paidoussis [5]), and also that the vibrations of the cantilever and the characteristics of the fluid flow are more strongly coupled for free-end-flow.

5.8 Results of Cross-Correlation Experiments

Some cross-correlation analyses were carried out between the rod vibration and wall pressure signals but no significant correlation was apparent. This does not imply that no correlation exists, but could mean that some pre-processing of the signals is required (see Ledwidge [12]).

5.9 A General Observation

An interesting tentative observation appears to be that for free-end-flow the amplitude of rod vibration (Figure 5) does not follow the predicted trends as a function of voidage whereas the resonant frequency and logarithmic decrement (Figure 10) do. For fixed-end-flow the amplitude of rod vibration (Figure 5) follows the predicted trend whereas the resonant frequency and logarithmic decrement (Figure 11) do not.

6. CONCLUSIONS

For the case of a cantilever-mounted rod placed in a flowing two-phase mixture it has been shown by theory that the following effects may be expected as the voidage fraction of the fluid mixture increases over the range zero to 25%:

(i) The vibration amplitude should increase.

(ii) The resonant frequency of the rod should increase.

(iii) The damping should increase, at least initially.

All these trends were observed experimentally for two flow conditions; free-end-flow in which the fluid flows on to the free end of the cantilever, and fixed-end-flow in which the fluid flows over the support (fixed end) and on to the free end. Some deviations observed at low voidages are believed to be due to turbulence effects, and for free-end-flow some observed deviations are due to the flow separation that occurs along the body. This separation for free-end-flow also makes the differential wall pressure a poor indicator of the driving forces for this case.

The theoretical expressions derived in this paper rely on the assumption that the energy transferred to the rod is dominated by momentum fluctuations in the fluid. The available experimental observations are not inconsistent with this theory. It seems therefore that air/water mixtures may be satisfactory for use as an analogue of steam/water mixtures in situations where temperature effects are unimportant.

7. ACKNOWLEDGEMENTS

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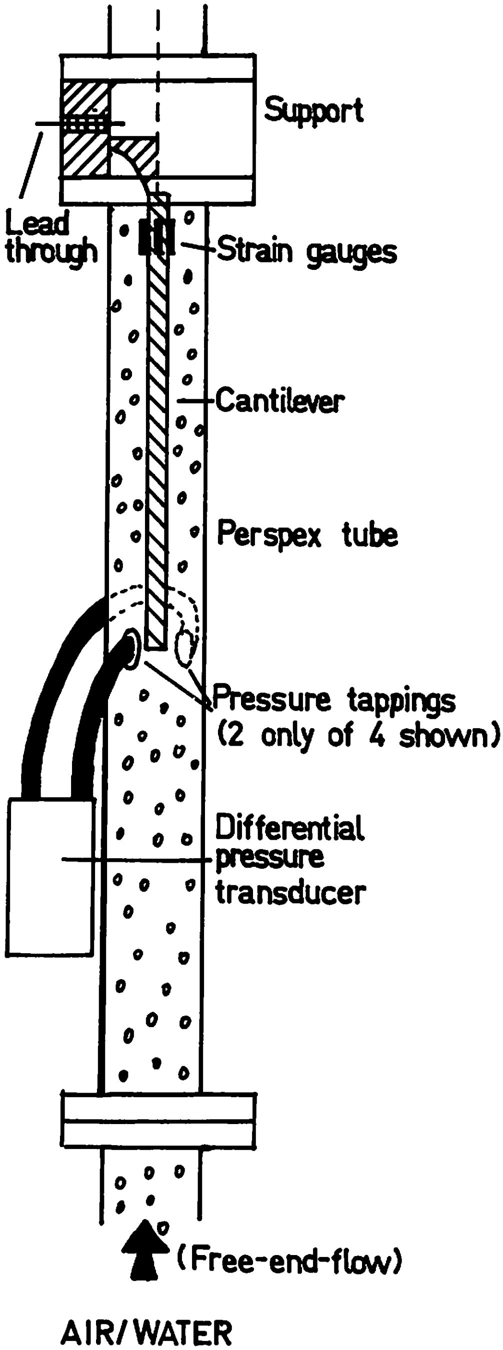


FIGURE 1 Diagram showing experimental arrangement.

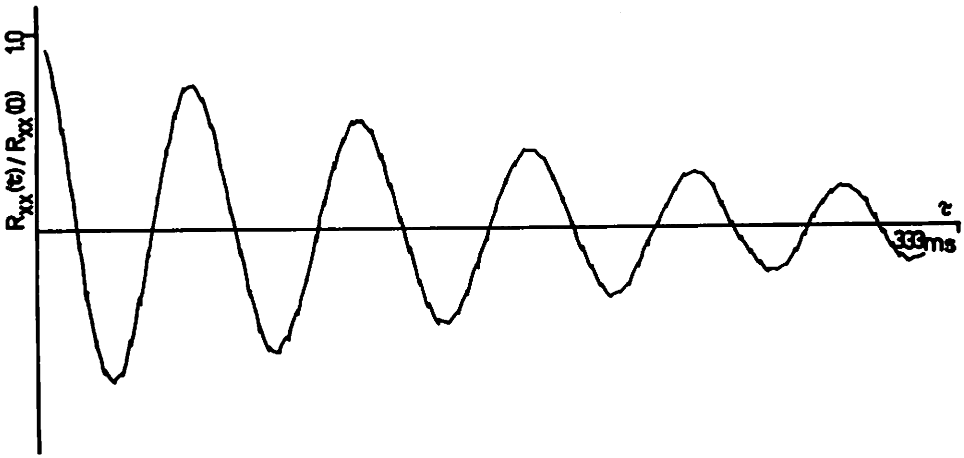


FIGURE 2 Frequency spectra for wall pressure and vibration for 23% voidage and free-end-flow.

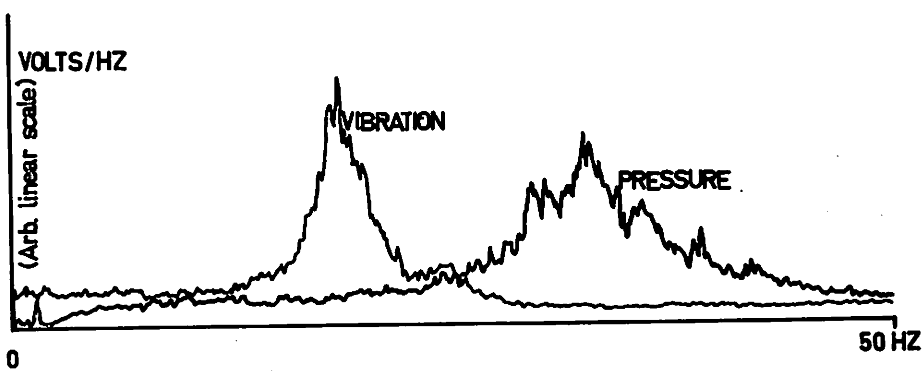


FIGURE 3 Typical autocorrelation for vibration of cantilever.

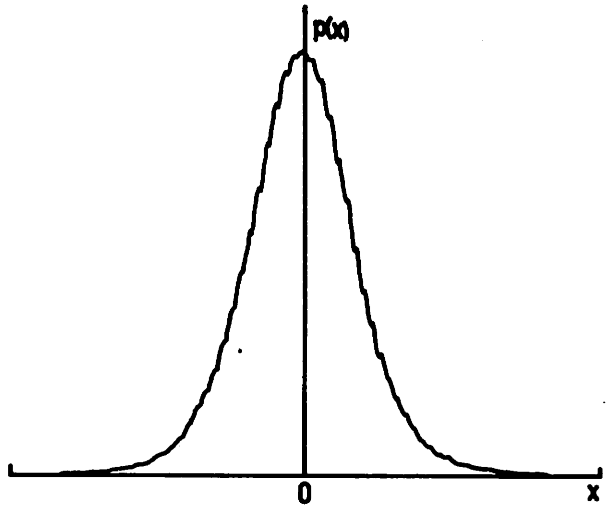
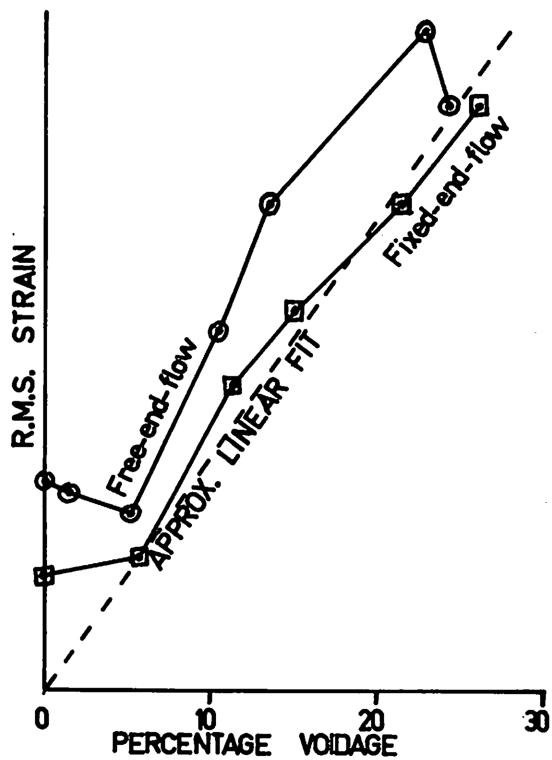


FIGURE 4 Probability distribution function for strain gauge output.



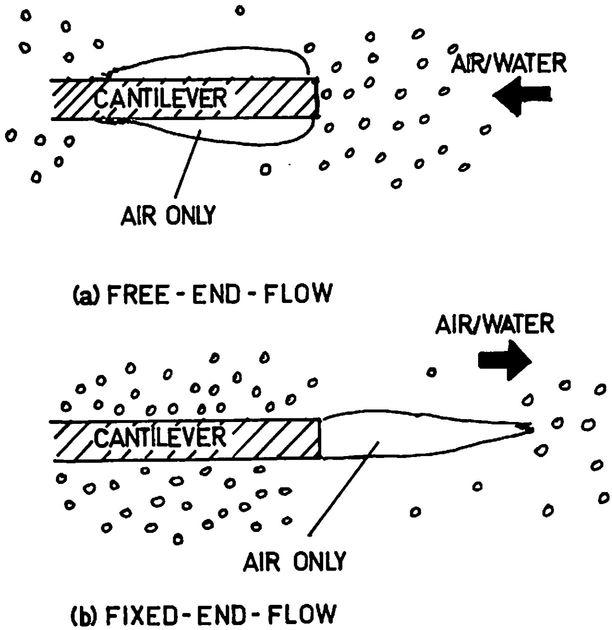


FIGURE 6 Illustration of form of flow separation at free end of cantilever.

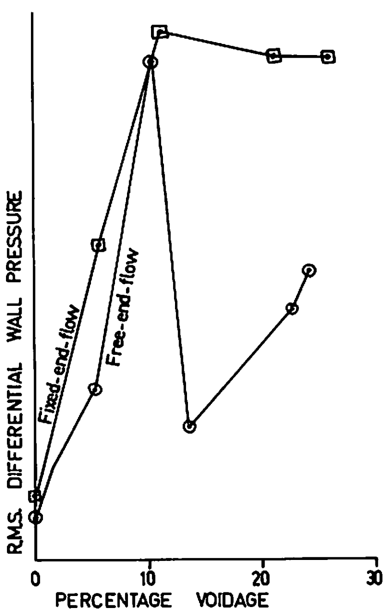


FIGURE 7 Differential wall pressure (r.m.s.) as a function of voidage.

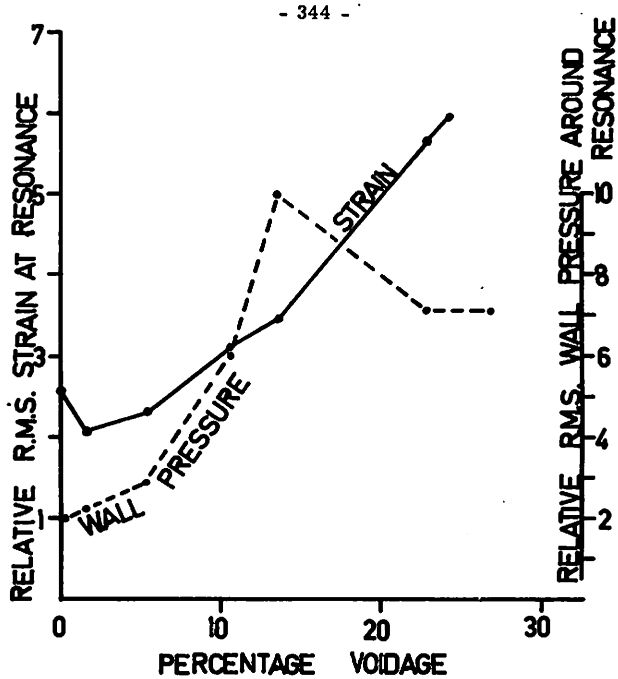


FIGURE 8 Free-end-flow. Vibration amplitude (r.m.s. strain) at resonance and wall pressure fluctuation around rod resonance frequency as functions of voidage.

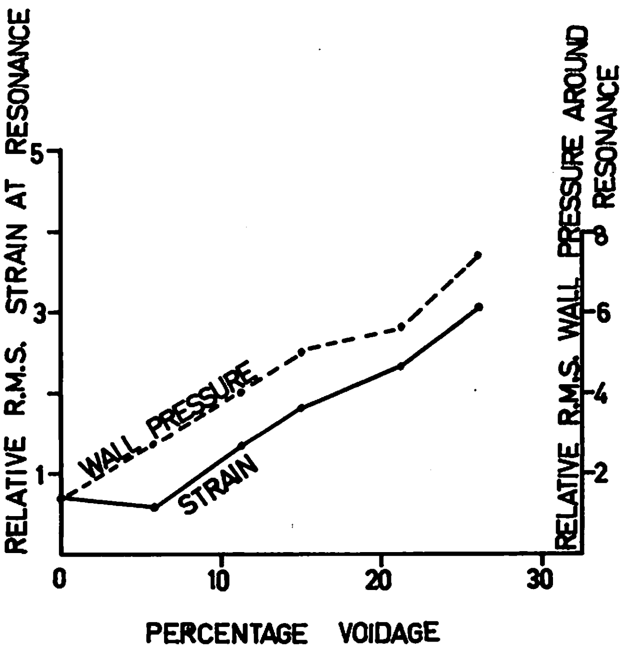


FIGURE 9 Fixed-end-flow. Vibration amplitude (r.m.s. strain) at resonance and wall pressu fluctuation around rod resonance frequency as functions of voidage.

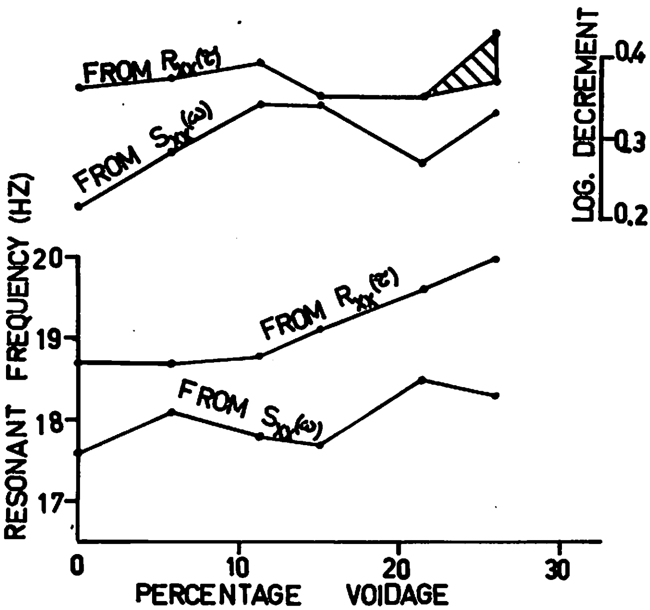


FIGURE 10 Free-end-flow. Rod resonant frequency and log decrement as functions of voidage.

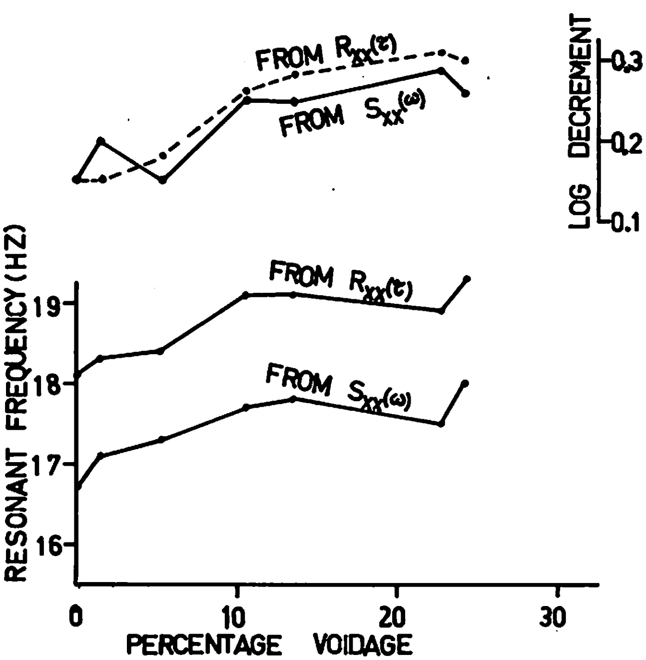


FIGURE 11 Fixed-end-flow. Rod resonant frequency and log decrement as functions of voidage.

DISCUSSION

Q T. H. LEE, U. S. A.

In your spectral analysis, did you consider the cross correlation function ? Is it true that only auto-correlation function is considered in this analysis ? What are your comments about the effects from the correlation between components of flow displacements in this particular problem ?

A T. J. LEDWIDGE, Australia

The prime object was to characterise the response of the rod and for this purpose auto-correlation function is sufficient.

Q S. CURIONI, Italy

In your investigation you have found some characteristic numbers (for example: Reynolds, Froude, etc. or a combination of these). To study this problem in other scale or with different flow velocities, do you maintain the same deflection of cantilever ?

A T. J. LEDWIDGE, Australia

The initial objective was to investigate the physics of the driving mechanism. The quantitative characterisation as a function of non-dimensional groups is clearly the next step.

Q H. SHIBATA, Japan

Is there any instability of the relation of rms of strain waves to p. v. , or a step change phenomenon even in bubble flow region ?

A T. J. LEDWIDGE, Australia

I suspect that discontinuties will only exist where there is a regime change from say bubble to slug, annular, etc.. In the present experiments the regime was at all times bubbly.

Q L. CEDOLIN, Italy

The results obtained with air-water mixtures could be not consistent with the ones of steam-water mixtures on the basis presented in the paper. We had, in fact, experimental evidence that in steam-water mixtures (isothermal) pressure fluctuations were entirely different from air-water mixtures.

A T. J. LEDWIDGE, Australia

The form of the relationship should still hold although the magnitude will clearly depend on bubble size distribution.

Q R. A. VALENTIN, U. S. A.

Do you believe that your experiments - at least the phenomenological aspects - would be applicable to sodium systems ? In particular, if one passes superheated sodium over a cantilever and if it produces large voids at the rod end, what do you feel would be the order of magnitude effect of this process on the pressure fluctuations seen by the rod.

A T. J. LEDWIDGE, Australia

The proposed mechanism is only valid for an isothermal fixed bubble distribution situation. I believe that flashing flow, bubble growth have such different amplitude statistics that the effects are not predictable with the present model.