VIBRATIONS INDUCED BY THE TWO-PHASE (GAS + LIQUID) COOLANT FLOW IN THE POWER CHANNELS OF A PRESSURE TUBE TYPE NUCLEAR REACTOR (°)

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ABSTRACT

For the fuel element development of the CIRENE nuclear reactor, a detailed investigation has been undertaken on the power channel vibrational characteristics. The power channel consists of a vertical pressure tube (10.61 cm I.D.) containing eight 19 closely spaced rod bundles. Each bundle is 50 cm long and simply rests over the lower one. The coolant conditions (∿ 50 bar pressure) vary between a few °C subcooling at the entrance and ∿25% (by weight) steam quality at the exit.

This paper presents some of the experimental results obtained with a two-phase two-component mixture (simulating the behaviour of steam-water) flowing in an adiabatic out-of-pile circuit. In particular the vibration of pressure tube and the relative motion of the rod bundle with respect to the pressure tube have been measured.

A preliminary analysis of the results is presented. An attempt has been made to correlate the relative motion to the two-phase flow pressure fluctuations. At this purpose fluctuationsof pressure difference along axis and around pressure tube wall have been measured and their relationship with the rod-bundle-pressure tube motion has been analysed.

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1. INTRODUCTION

1.1. For the first core of the CIRENE prototype reactor (a 40 MW(e) pressure tube, heavy water moderated and evaporating light water cooled reactor [1], the envisaged fuel elements consist of bundles of 19 closely spaced rods about 50 cm long assembled together through flexible Zircaloy end plates located at both ends of each bundle. Each power channel has eight bundles resting one over the other, without any mechanical connection. The rods are made of 2.0 cm O.D. Zircaloy thin-walled tubes, filled with uranium oxide pellets so that the tube wall is normally collapsed over the oxide pellets. The rods are spaced at middlength by spacers brazed over the claddings. The whole bundle assembly is spaced from the pressure tube wall by means of wear pads brazed over the peripheral rods (see Fig. 1). The fuel elements are cooled with evaporating light water at a medium high velocity (exit velocity 25 ÷ 30 m/s).

One of the problems which might affect the fuel element behaviour is fretting corrosion between wear pads and pressure tube wall since an excessive corrosion of the latter might lead to its rupture [2]. Fretting corrosion between rod-to-rod spacers is a minor problem because of the limited life of the fuel element (with respect to that of pressure tube) and because of the less severe consequences from its rupture.

Fretting corrosion between wear pads and pressure tube wall may occur as a consequence of the relative impact and friction between the two surfaces, both depending on the vibrational characteristics of bundles, rods and pressure tube, when submitted to random forces due to the coolant flow.

This paper deals with the experimental analysis carried out to investigate the power channel vibrational characteristics. The main aims of the investigation are the following ones:

a) measuring the vibration amplitude and frequency at various positions along the power channel;
b) determining the vibration mechanism in two-phase flow;
c) analysing the influence on the vibrational characteristics of some thermohydraulic parameters such as flowrate, quality, pressure, heat flux, etc.;
d) analysing the influence on vibrational characteristics of the parameters connected with the channel mechanical configuration.

Most of these aspects are covered in the present paper.

1.2. Experimental and theoretical studies on parallel flow induced vibrations started some years ago when this was recognized as a possible major problem for reactor design. In many reactor types, fuel elements consist of cylindrical rod bundles, and generated power is removed by means of parallel coolant flow.

To the authors' knowledge, Burgreen et al. [3] reported the first experimental determination of the type and magnitude of vibration of simulated fuel rods and bundle in a water flow loop. The parameters which were analysed were the hydraulic diameter, the diameter and length of the rods, the type of end constraints, the water flow velocity. The experimental results show that the frequency of the induced vibration of the rod is independent of the water velocity, and close to the natural frequency in water. The paper recognizes that the vibration amplitude follows an irregular time history and maximum amplitude is taken as the significant parameter.

In the same paper, an analysis is made of the possible mechanism of vibration excitation, by interpreting the experimental results as a proof that the vibration is self-excited. The hydrodynamic force of the water flow is assumed as energy source for vibration and by means of dimensional analysis, a correlation is proposed. The experimental data show a fair agreement with the predicted values.

The possibility that the excitation mechanism could be due to oscillatory instabilities was then studied by Paidoussis [4] , who found both theoretically and experimentally that this possibility was restricted to very high flow velocities, out of the range of nuclear reactor applications. Starting from the former analysis, Paidoussis [5] then developed a new model for the subcritical vibrations, assuming as forcing fuction the fluid forces acting on the rod due to cross flow components of the fluid flow itself. This correlation shows a better agreement with all the obtainable experimental data,but it fails in predicting the amplitude of two-phase flow induced vibrations. Moreover, from a physical point of view, the former correlations are not convincing, since they do not take into account the stochastic nature of the response.

Reavis [6] probably first recognized this nature, assuming that the pressure fluctuations acting on the surface of the rod are the cause of vibration. He expressed the mean square value of the response using the equation developed by Thomson [7] , which involved the concept of spatial correlation, and assumed for the pressure fluctuations field an idealized description of boundary layer turbulence.

Wambsganss and Boers [8] used the same approach, but pointed out the need for an experimental description of the actual pressure fluctuations, which could be due also to flow disturbances generated in the experimental circuit. Since in their experiments they lacked information about the spatial correlation, they neglected it and assumed that the forcing function could be proportional to the pressure fluctuations measured in a fixed point of the wall of the test section. This simplification seems to lead to substantial errors, as it will be shown in this paper (par. 5).

Another theory which takes into account the random nature of excitation is the one developed by Griffith et al. [9] , who postulated that vibration is directly connected to the unsteady component of the momentum flux, and to its effect on not perfectly parallel walls of the rods. This model cannot be applied straightforwardly to practical problems because the variation with time of the momentum flux is unknown and its measurement is rather difficult. It must be emphasized that the suggested mechanism is quite different from the one proposed by Reavis although a connection between them can exist if axial and radial momentum fluctuations are each other correlated.

2. EXPERIMENTAL

2.1. The experiments have been carried out in an experimental facility (called IDRA) already installed at CISE for studying two-phase (gas+liquid) hydrodynamics in adiabatic conditions [10] . The two-phase mixture is formed in a tee-mixer with the gas entering the run side; the liquid is injected through an annular slot with an adjustable opening. After passing through a calming section, the mixture enters the vertical test-section from the bottom. At the top, the test-section leads into a high efficiency separator, from which each separated phase returns to its own circuit.

The fluid is a two-phase two-component mixture which simulates the behaviour of steam-water mixture at the reactor operating conditions (pressure: ∼ 50 bar). Nitrogen at ∼21 bar has been selected as the gas phase since at room temperature density - which is one of the most important physical properties for scaling two-phase flow [11] - is equal to that of saturated steam at 50 bar; viscosity is also quite close to that of steam and therefore the behaviour of the gas phase should be accurately reproduced.

As to the liquid phase, cold water has been so far adopted, which has physical properties quite different from that of saturated water at ∼ 50 bar. One of the major difference is that concerning surface tension (73 dyn/cm against 22 dyn/cm for hot water) because this is another physical property which greatly affects the two-phase flow behaviour [11] . In this connection tests will be soonly carried out with another liquid (acetone) having a surface tension closer to that of saturated water. The other physical properties - density and viscosity - are also different from those of hot water but this should not affect the results, since the hydrodynamics of two-phase flow does not depend markedly on viscosity [11] and the density difference is rather low (approximately 20%). In this concern however tests have been carried out with different gas densities (the two-phase flow behaviour depends on the ratio ρ_ℓ/ρ_g rather than on the separate values ρ_ℓ and ρ_g [12]) by varying the system pressure.

The validity of the criteria followed to simulate steam-water flow with a two-component mixture - at least from the point of view of hydrodynamics - has been extensively investigated at CISE in the previous years and the results obtained in these conditions have been used to develop correlations which have been found quite suitable for steam-water flow (for example pressure drop [13] , void fraction [14] , etc.).Thus the only doubts that are left from the viewpoint of the present investigation are relevant to the effect of surface tension.

Another important aspect which is still unknown concerns power generation in the actual fuel channels which makes steam quality vary along length while in the present experiments this parameter remains constant. A special test section is being planned for a through investigation of this problem.

The flowrate and quality investigated range ($G = 100 \div 200$ g/cm^{2}s; $0.02 < X < 0.25$) corresponds to that of the CIRENE reactor although not all the conditions have been reproduced due to limitations of the plant components (namely the gas circulator).

The test-section VIBRO is illustrated in Fig. 2. The pressure tube, made of yellow brass in order to have a modulus of elasticity close to that of Zircaloy-2 at the reactor operating conditions, has the following dimensions (similar to the ones of the CIRENE pressure tube):

thickness	= 2.5	mm
inside diameter (maximum)	= 104.5	mm
ovalization (maximum)	= 0.7	mm

The outside diameter of the VIBRO fuel bundles has been made equal to 103.7 mm.

The other dimensions of the bundles are similar to the ones of the CIRENE Prototype power channel. The fuel bundles have been constructed using rods made up of a brass cladding and lead pellets, which simulate the linear density of UO_2 pellets, and yellow brass sheet for the end plates. Furtherly the end plates are screwed (not welded) to the rods and there are no spacers between the rods, but only wear pads on the peripheral rods.

Eight bundles are located in the pressure tube; they are axially compressed (axial load 200 kg) by an adjustable spring located in the upper end fitting. These fittings are simplified with respect to those for the CIRENE reactor but are fixed to the external structure by means of the same type of constraints: the lower end is clamped, the upper end is simply supported on freely rotating balls.

The constraints are fixed to a concrete wall not directly connected with the facility metallic structure to avoid disturbances coming from the plant rotating components through the constraints. The inlet and outlet of the test-section are connected to the circuit by means of flexible joints in order to remove the disturbances reaching the test-section through the pipes (see par. 2.3).

In order to investigate the spatial correlation of pressure differential fluctuations (see par. 5) a suitable test-section, called IT-35 (Fig. 3) was used. It consists of a yellow brass tube with several diametrally opposite pressure taps distributed both axially and circumferentially, connected to the same IDRA loop.

2.2. The instrumentation developed to measure the channel vibrational characteristics is the following:

- strain gauges bonded longitudinally to the pressure tube wall at various positions to detect the low frequency flexural vibrations of the channel considered as a beam. The positions near the clamped end and the middlength of the tube are chosen because the bending moment relevant to the first mode of vibration is maximum in their proximity. The position near the hinged end is chosen to give a measurement of the constraint efficiency in both static and dynamic conditions. Finally strain gauges have been located at about one third of the pressure tube length because they are particularly suitable to measure the bending moment due to the channel second mode of vibration.
- piezoelectric accelerometers applied to the pressure tube wall to detect the high frequency circumferential vibrations of the tube considered as a thin cylindrical shell.

- Differential transformer displacement transducers, which measure the relative movement of the bundles with respect to pressure tube wall. They consist essentially of a rodlet which penetrates into the pressure tube, and contacts a wear pad of the fuel bundle; the differential transformer floating core is fastened to this rodlet.
- Variable reluctance or eddy current gauges, which measure the relative movement of rods with respect to pressure tube wall. They consist essentially of a coil, which is fastened to the outer side of the tube, whose inductance varies with the distance of a soft iron pellet located inside the sheath tube in a position facing the core.
- Piezoelectric pressure transducers, connected with pressure taps located at the inlet, middlength and outlet of the VIBRO test section and in various positions along the IT-35 test section. (Fig. 3).
- Piezoelectric force transducers, equipped with a rodlet which can penetrate into the pressure tube and contact the fuel bundle through a micrometric regulation of penetration. Eight of them are placed in two different sections at 90° each other to study (see par. 4.3) frequency of impacts, impact forces and contact forces between wear pads and pressure tube.

The signals from the transducers are recorded on a magnetic recorder and then fed into one analog to digital conversion unit of an IBM 1800 process computer. Data analysis is carried out by means of the auto and mutual power spectra and correlation functions technique. The PAFFT and PAMIR codes, developed at CISE [15] , perform in real time statistical analysis of auto and mutual properties of random signals by means of the Fast Fourier Transform method. These codes are also suitable to periodic signals analysis, with particular regard to noise detection by means of correlation analysis. In random vibration problems, the study in the frequency domain, namely the use of auto and cross power spectral densities of the signals, was found especially suitable. In some cases data analysis was made by means of a real time spectrum analyzer and a time domain analyzer.

2.3. The systematic investigation has been preceeded by a detailed analysis of any possible contribution to test section vibration due to the presence of the plant rotating components, such as pumps and circulators. Experimentally this analysis was carried out by looking essentially at the signals coming from the pressure tube strain-gauges and accelerometers for different experimental situations (for example by using at the same flowrate two different circulators or by disconnecting the test section inlet pipes from the circuit, or by operating the circulators and the pumps through a by-pass line with no flow in the test section).

In a preliminary stage of the analysis the test section was fastened to a column of the facility supporting structure. In this condition a marked effect due to plant components on the vibration of the pressure tube was evidenced especially in the low frequency range (2-30 Hz) by observation of the strain-gauge signals. A typical power spectral density curve is given in Fig. 4: the peaks at 8.5, 17 and 34 Hz depend on the periodical pressure oscillation in the gas line due to a reciprocating compressor used as a gas circulator. Such oscillations appeared also when disconnecting inlet and outlet test section pipes and its was inferred therefore that the disturbances were transmitted to the test section through the constraints (for greater details see reference [16]).

A new location was therefore selected with the constraints fixed to a concrete wall and proved to be completely satisfactory. Moreover a suitable pressure oscillation damping device was installed in order to eliminate these external disturbances. A typical power spectral density curve, corresponding to the previous one, is given in Fig. 5 which shows that the peaks have almost completely disappeared.

As other have pointed out recently [8] correct design of suitable systems to undertake significant and well defined experiments in this field should take into account the effects of both plant moving components, such as pumps and other features such as piping and supporting structure. Results could then be correctly used in predicting vibrations in the actual reactor plant. One of the reasons why a general agreement has not yet been found between the various data sources and therefore no general theory has not yet been formulated for flow-induced vibrations, is probably a lack of characterization of each experiment as far as intrinsic disturbances of each experimental facility are concerned.

3. CHARACTERIZATION OF THE SINGLE COMPONENTS

A theoretical and experimental investigation of the natural frequencies and stiffnesses of the whole channel, of the bundle column, and of the single rods has been carried out, in order to have a better understanding of the experimental results of pressure tube-fuel relative motion and contact forces.

- <u>flexural vibration of the channel</u>: the frequency of the channel flexural vibration is influenced by the inertia and elastic properties of the pressure tube and bundle column, and by the presence of an axial tension. For the test section pressure tube alone, the calculated frequency is

$$f_1 = 14 \text{ Hz}$$

If the bundles participate with their mass to the tube movement, the fundamental frequency lowers to

$$f_1' = 4.72 \text{ Hz}$$

By taking into account the effect of the axial tension in the tube, due to the internal pressure, the above calculated frequency becomes:

$$f_1'' = 5.09 \text{ Hz}$$

Fig. 5 is a typical power spectrum of the low frequency range CIRENE pressure tube vibration detected by strain-gauges. The sharp peak at 5 Hz proves that the assumptions made are correct, that is the effect of the bundle column stiffness on channel vibration is negligible.

- Flexural vibration of the bundle column: the eight CIRENE bundles are piled up and axially loaded in an out of channel support and prove to behave as an elastic, continous beam. The natural frequency is 1.25 Hz and the spring stiffness at middlength about 0.6 kg/mm. These values correspond to a clamped-clamped beam with a flexural stiffness equal to 19 times the stiffness of the single empty rod, the contribution of fuel being only in mass. It is reasonable to presume that the bundle column assembled in the pressure tube has some permanent contact points with the tube wall and therefore a greater natural frequency (three equally spaced intermediate supports raise the fundamental frequency to ~7 Hz).
- Flexural vibration of the single rod: the measured natural frequency is about 50 Hz, close to the theoretical value for a hinged-hinged beam. Careful measurements of the damping ratio are made both in air and in different hydraulic conditions. The results are plotted in Fig. 6.

4. RESULTS

The fuel bundle column has been assembled, unloaded and reassembled in the channel four times in order to have reproducibility measurements.

4.1. Pressure tube motion

Fig. 7, where root mean square (r.m.s.) strain of the pressure tube due to the first mode of vibration is plotted for all hydraulic conditions and different assemblies, shows that the absolute motion of the pressure tube in the low frequency range, as detected by strain-gauges, is repeatible.

To the authors' knowledge, no information is so far available in literature about the mechanism of induced vibrations on the pressure tube in such a channel configuration. It would be useful to determine what contribution to pressure tube vibration is directly given by the two phase flow noise and what amount of the vibration is on the other hand substained by the vibrating bundle column.

4.2. Relative motion

The relative motion is characterized by two well separated power bands, as shown in Fig. 8 which is a typical power spectrum of relative motion signals coming from variable reluctance gauge:

a) a low frequency band, from 1 to about 15 Hz, corresponding to the motion of the bundles;
b) a high frequency band, centered at about 50 Hz, corresponding to the rod natural frequency.

R.m.s. displacements in the two frequency bands have been calculated in different hydraulic conditions and the main results are the following (see Fig. 9,10,11,12).

- The r.m.s. displacement in the low frequency band is always much higher than in the high frequency band. In the range of specific mass flowrates and mass qualities explored the fuel rod vibrations show a maximum, whereas the amplitudes of vibration of the fuel bundles monotonously increase with flowrate and quality, although a maximum is likely to occur at higher qualities.

- Pressure has a marked effect on vibration phenomena. For a given volume quality and specific mass flowrate vibration amplitudes increase with decreasing pressure.
- The static equilibrium position of the bundle column, i.e. the number of contact points with the pressure tube wall and the elastic response of the structure change in the different assemblings, so that a large scattering of the experimental results is observed. A good agreement is found between the results of the first and second assembling. In the third the number of permanent contacts between wear pads and pressure tube is probably higher because the low frequency band spreads to about 30 Hz and the amplitudes of vibration are considerably reduced. In the forth assembling, however, amplitudes of relative motion are considerably higher.

The consideration that pressure tube absolute motion is repeatible while fuel bundles relative motion is not, leads to the hypothesis that the former may have little influence on the latter. The two following experiments have been carried out to verify this assumption:

a) the pressure tube has been clamped in several points but no marked effect on fuel relative motion has been observed;
b) the pressure tube has been excited by an electrodynamic vibration machine in a suitable hydraulic condition ($G = 10\ g/cm^2.s$, $X = 0.40$) in order to have very little coolant noise and reproduce the correct fuel column damping characteristics. Absolute motion of the pressure tube necessary to reproduce the correct fuel relative motion has turned out to be considerably greater than in the actual case. The conclusion that fuel motion is practically indipendent of the tube motion may be of great importance: for instance, experimental results which do not take into account the effect of calandria tubes are still accurate in predicting fuel vibration in the actual reactor plants.

4.3. Interaction forces

The interaction forces between fuel bundles and pressure tube wall are studied by means of the above mentioned force transducers. Due to the bundle column lability, significant data could be obtained only using couples of diametrally opposite transducers and averaging the results. Starting from rest position, with all transducers extracted, and exploring the vibrating field for different penetrations in the different hydraulic conditions, the following significant information could be obtained:

- when the transducers enter the vibrating field of the bundle column, interaction begins with sharp impacts of the wear pads on the force transducer exploring rod. Times of contact are 20 ms and impact forces of the order of one kilogram.

- The frequency of such impacts is a function of penetration as well as of hydraulic conditions. By increasing the penetration, frequency of impacts increases up to a point when no more detachments are detected. In such condition a permanent contact between fuel rod and pressure tube is simulated. This relative penetration is a mesure of peak amplitude of vibration.

- A typical power spectrum of contact forces is shown in Fig. 13. Peaks at about 5 and 20 Hz are in the range of the low frequency vibrations, while the ~100 Hz peak corresponds to the natural frequency of the fuel rod with the additional support due to the force transducer.

- Contact forces are made up of a continuous component and a oscillating one. The continuous component is due to the elasticity of the bundle column, taking into account the number of its permanent contact points due to its obliquous configuration. The oscillating component must be somehow related to the forcing field and therefore to random vibrations. In Fig. 14 peak values of the oscillating reaction force are plotted versus quality and flowrate. They follow closely the behaviour of peak amplitudes of vibration reported on the same figure.

The existence of a correlation between vibration amplitude and contact force seems therefore extremely likely.

5. ANALYSIS

Prediction of the actual vibration characteristics induced by the two-phase coolant flow in a channel geometry as the one considered in this paper appears as a formidable task because of the simultaneous presence of motions of different components that may influence each other.

Also the existing information for such a geometry is negligible. Thus in order to get a preliminary idea of the vibration mechanism in two-phase flow, an attempt has been made to utilize the existing models for predicting vibration amplitudes at least for the bundle column, assuming the random nature of excitation and the relative movement of the bundle column be unaffected by the pressure tube movement.

Of the various models, the empirical ones such as those proposed by Paidoussis [5] have proved to be completely inadequate. The one proposed by Griffith [9] could not be used because the unsteady momentum flux was not measured and its correlation to pressure fluctuation, which instead was measured, is quite unknown.

The model proposed by Reavis [6] has been considered, according to the approach suggested by Gorman [17] .

The original approach of Gorman to the analytical model proposed by Reavis was to consider the driving force acting on the rod in a given direction as resulting from the differential pressure fluctuations on diametrally opposite sides of the rod. The equation developed by Gorman is

$$\sqrt{< y^2(t) >} = \frac{L D \psi_L \psi_D \sqrt{< p^2(t) >'} \sqrt{\pi f_1}}{M \omega_1^2 \sqrt{4 \xi}} \qquad (1)$$

where

$\sqrt{< y^2(t) >}$ = r.m.s. amplitude of rod vibration

$\sqrt{< p^2(t) >'}$ = r.m.s. pressure differential oscillation per unit band width

$\omega_1 = 2 \pi f_1$ = circular fundamental frequency of rod

$M = \frac{W}{2g}$

W = weight of rod

g = gravitational acceleration

L = length of rod

D = diameter of rod

ξ = damping ratio

ψ_L = effective length ratio

ψ_D = effective diameter ratio

The equation was derived from Thomson's [7] complete theoretical solution for the first mode response of a pinned beam to distributed random forces, under the following assumptions:

a) vibration is substained by random pressure fluctuation in a statistically homogeneous pressure field;
b) the rod is simply supported and the system is lightly damped;
c) the general spatial cross correlation coefficient of the fluctuating pressure differentials may be expressed as a function of the difference of the respective space variables and furthermore the variables are separable. This means that:

$$\frac{< p(x, \theta, t)\, p(x', \theta', t) >'}{< p^2(t) >'} = f(x-x')\, g(\theta - \theta') \qquad (2)$$

Setting $x = x'$ and $\theta = \theta'$ one obtains

$$g(\theta - \theta') = \frac{< p(\theta, t)\, p(\theta', t) >'}{< p^2(t) >'} \qquad (3)$$

$$f(x - x') = \frac{< p(x,t)\, p(x',t) >'}{< p^2(t) >'} \qquad (4)$$

The terms ψ_L and ψ_D in equation (1) can be evaluated knowing the longitudinal and peripheral cross-correlations of the driving force, $f(x-x')$ and $g(\theta - \theta')$.

For this purpose, pressure differential signals at various longitudinal and angle separations on the IT-35 test section were recorded on magnetic tape. Signals were processed using a more sophisticated digital technique than the analogic method used by Gorman.

The method was developed noticing that the numerator of the right terms in equations (3) and (4) is described in the frequency domain by the co-spectral density function, that is the real part of the cross spectral density function. In this way, pressure differentials recorded at different axial or angle positions, were processed in order to evaluate the cross spectrum and the autospectra; then the power in the 40 to 60 Hz range (centered at the fuel rod natural frequency) of the co-spectrum and of either autospectrum constituted respectively the numerator and denominator of (3) and (4). Fig. 15 is a typical power spectrum of differential pressure fluctuation. It must be said incidentally that it shows a good agreement with the shape of power density curves of momentum flux measured by Griffith et al. [9] .

Root mean square differential pressure in the 40 to 60 Hz band is plotted versus hydraulic conditions in Fig. 16. It follows closely the behaviour of r.m.s. rod amplitude of vibration.

Plots of longitudinal and peripheral correlation coefficient are given in Fig. 17 and 18. The longitudinal correlation is of the same damped cosine type found by Gorman on the single rod, whereas the peripheral correlation, referring to a bundle of 19 rods, is of course somewhat different.

By using the values of ψ_L deriving from our longitudinal correlation data, a mean value of $\psi_D = 0.75$ [18] and the damping ratios plotted in Fig. 6, the reliability of the method was checked for the hydraulic conditions $G = 110\ g/cm^2 s$ with $X = 0.24$, and $G = 200\ g/cm^2.s$, with $X = 0.023$ and $X = 0.091$. Analytical results superimposed on the average experimental results over the four VIBRO assemblings are given in Fig. 19.

The average discrepancy is about 20%, which must be considered as a significant result owing to the approximation involved in applying to the single rod the pressure fluctuation differentials measured between opposite sides of the pressure tube wall.

Attempts have been made to extrapolate the method for predicting r.m.s. amplitudes of vibration of the fuel bundle, in the low frequency range. No significant results have so far been obtained, owing to the following reasons:

a) r.m.s. differential pressure in the low frequency (1 ÷ 15 Hz) band is very low and practically indipendent of flowrate and quality;
b) the model applies to a hinged-hinged beam, whereas the bundle column is clamped-clamped and its natural frequency varies according to the number of permanent contacts with the pressure tube wall.

A different approach is therefore necessary to explain low frequency vibrations.

Absolute pressure oscillations have also been investigated according to Wambsganss and Boers [7] assumption that the system forcing function is proportional to pressure fluctuations in a fixed point of the test section wall. A typical power spectrum of a signal coming from a pressure transducer at middlength of the VIBRO test section is shown in Fig. 20. It shows a completely different frequency range and shape from differential pressure spectra. R.m.s. pressure fluctuation in the 1-8 Hz band is plotted versus flowrate and quality in Fig. 21. The behaviour is completely different from r.m.s. vibration amplitude. It is therefore evident that spatial correlation of pressure fluctuations cannot be disregarded without leading to substantial errors.

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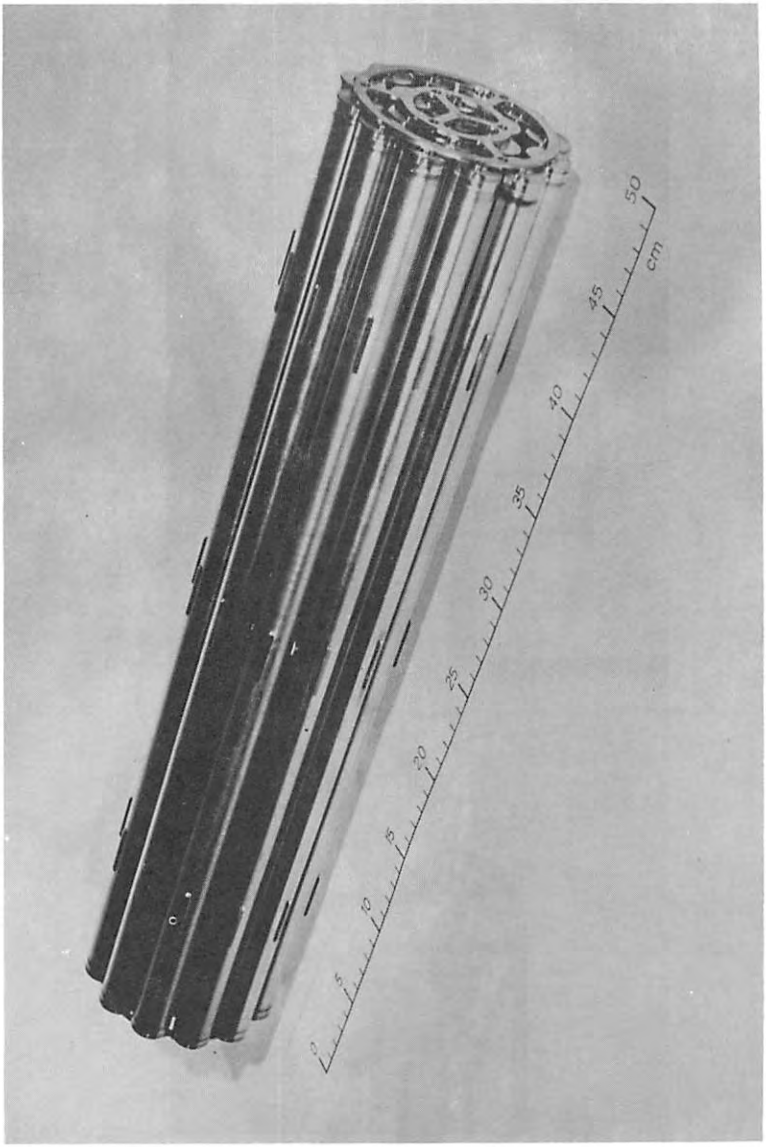

Fig. 1 CIRENE fuel element

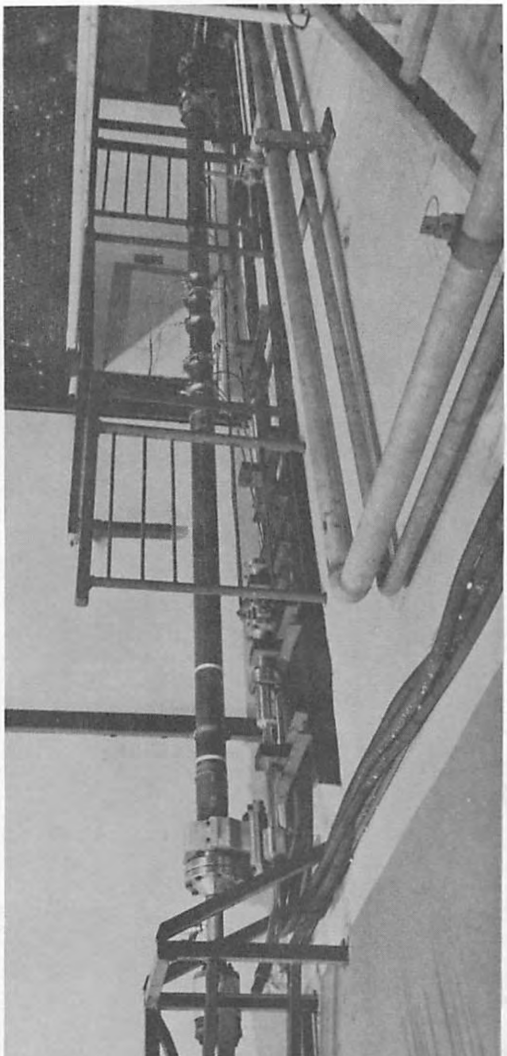

Fig. 2 VIBRO test section

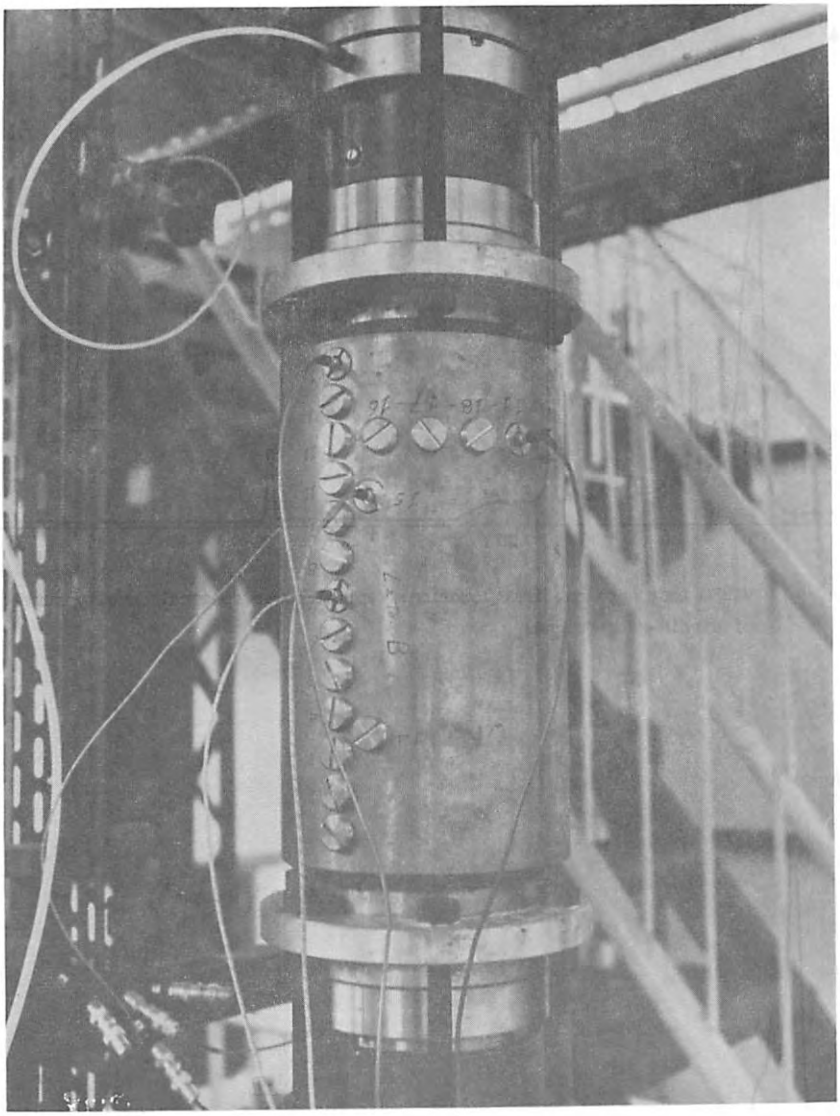

Fig. 3   IT-35 test section

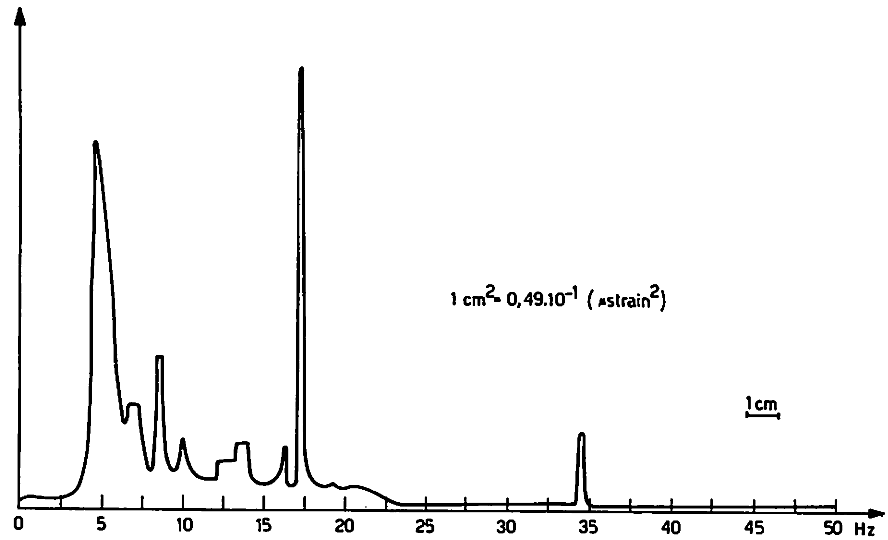

Fig. 4 VIBRO test section,first location: typical power spectral density curve of strain-gauge signal

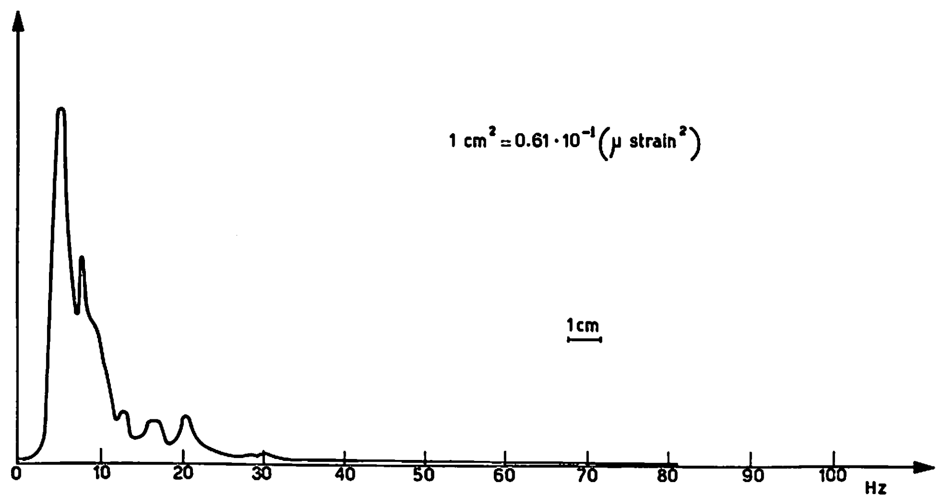

Fig. 5 VIBRO test section,second location: typical power spectral density curve of strain-gauge signal

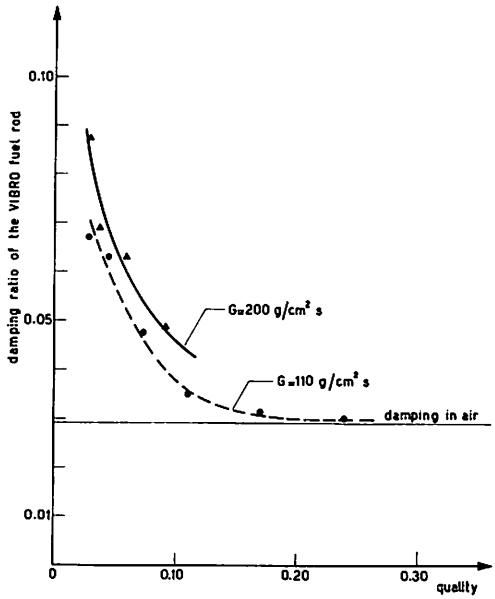

Fig. 6 Fuel rod damping ratio versus flowrate and mass quality

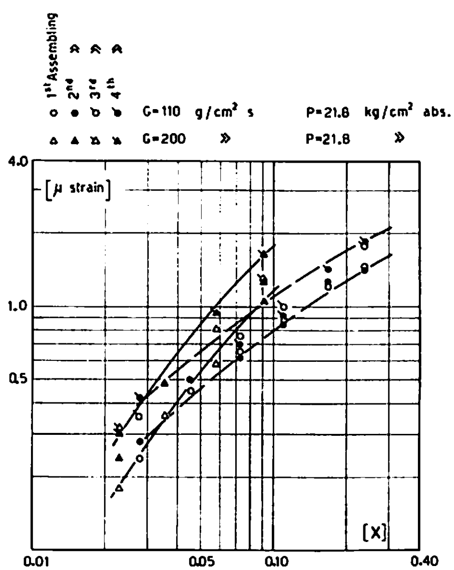

Fig. 7 R.M.S. pressure tube first mode vibration versus flowrate and mass quality

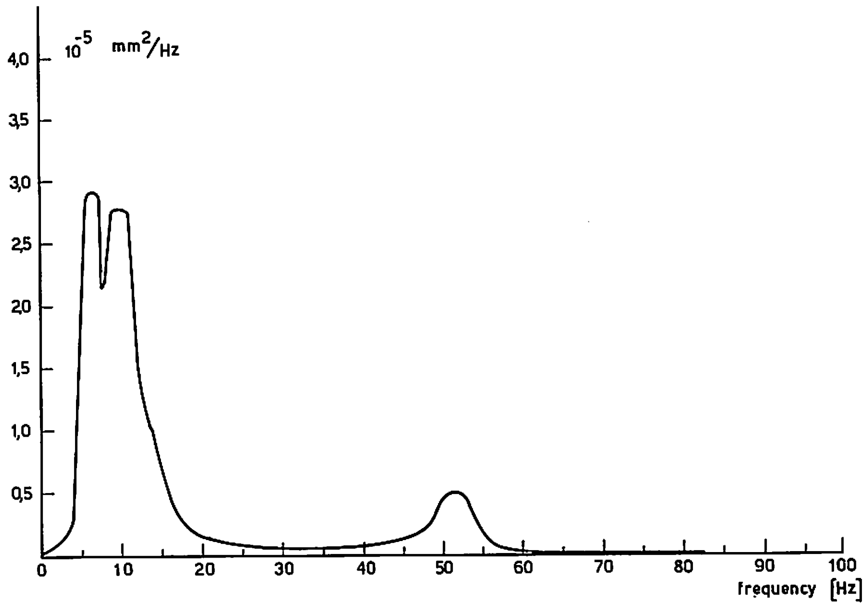

Fig. 8 Typical power spectral density curve of variable reluctance gauge signal

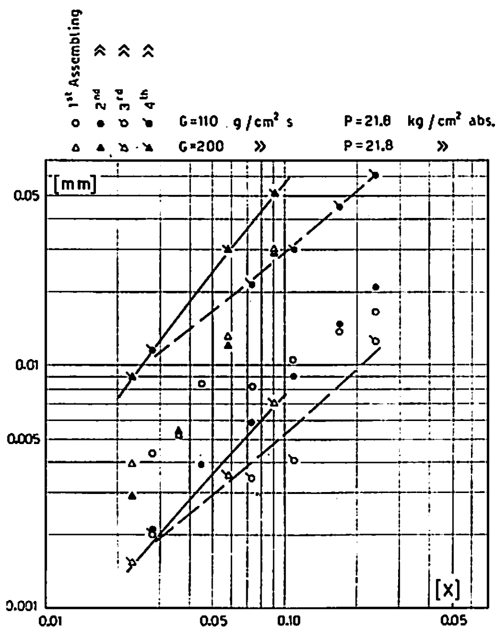

Fig. 9 R.M.S. fuel bundle displacement versus flowrate and mass quality

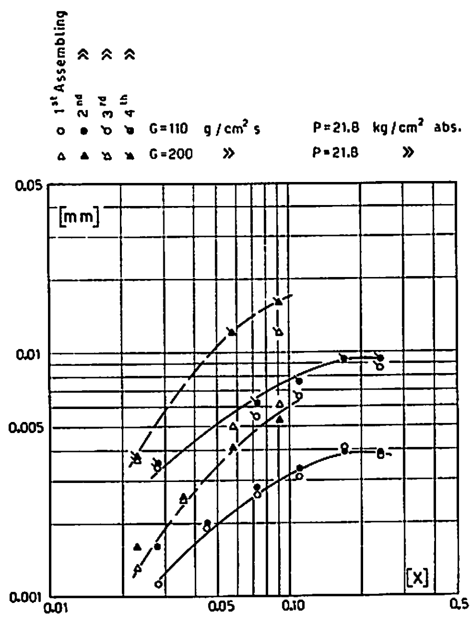

Fig. 10 R.M.S. fuel rod displacement versus flowrate and mass quality

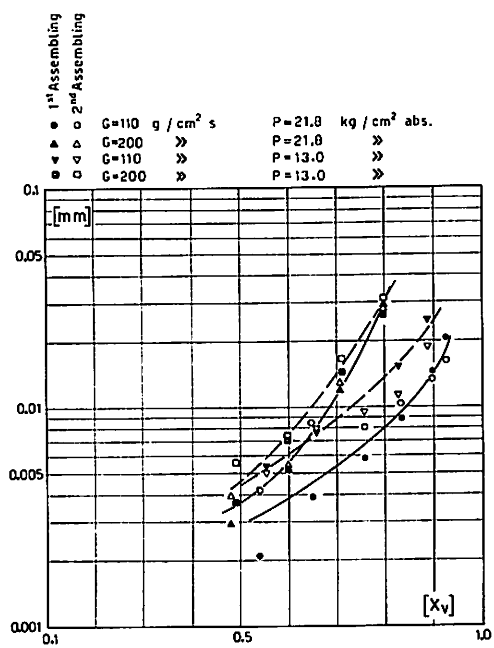

Fig. 11 R.M.S. fuel bundle displacement versus pressure, flowrate and volume quality

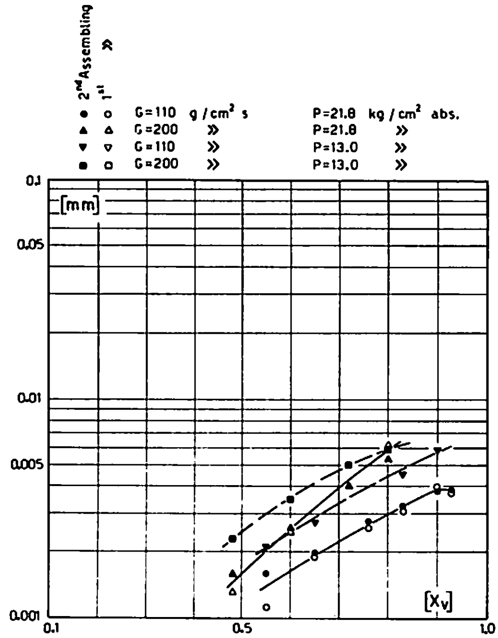

Fig. 12 R.M.S. fuel rod displacement versus pressure, flowrate and volume quality

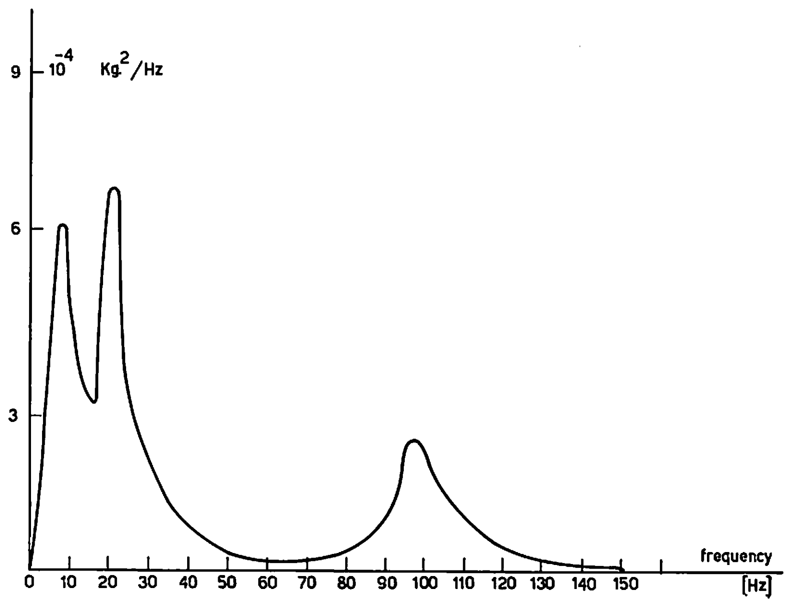

Fig. 13 Typical power spectral density curve of contact force transducer signal

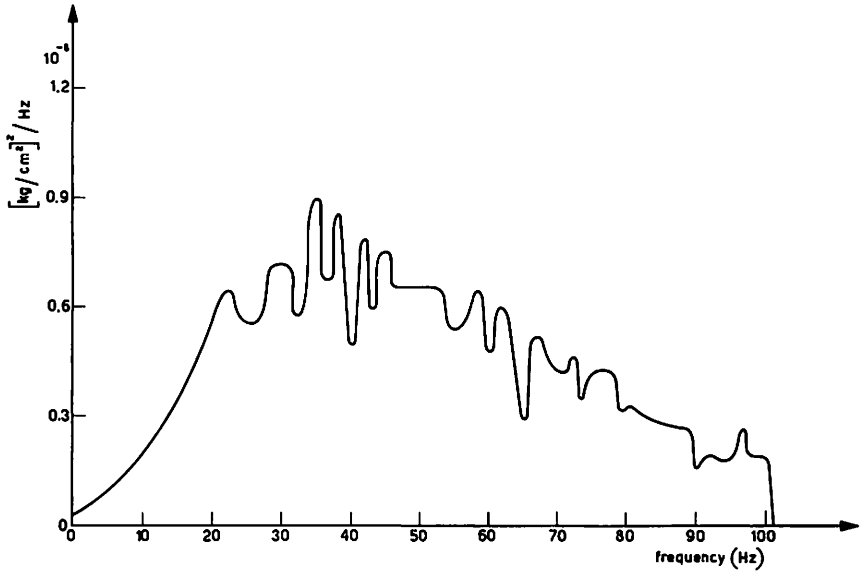

Fig. 14   Peak values of oscillating reaction force and vibration amplitude versus flowrate and mass quality

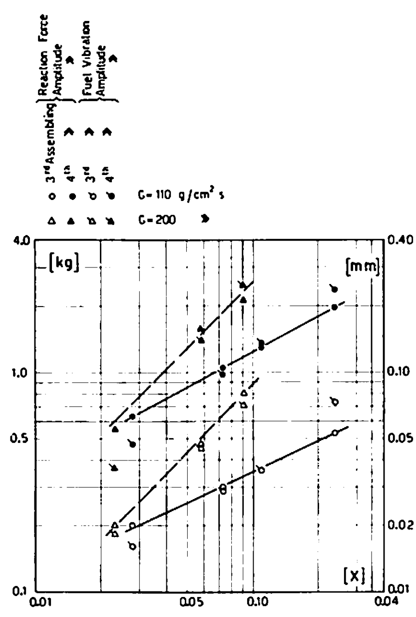

Fig. 15   Typical power spectral density curve of differential pressure fluctuation

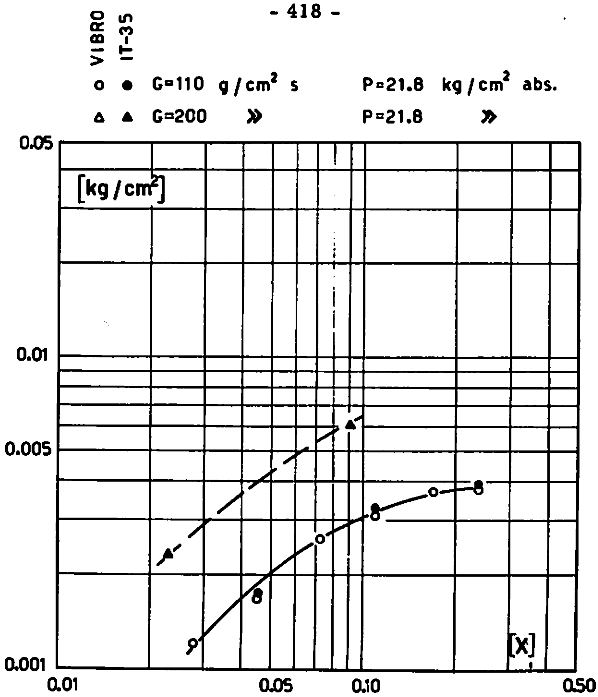

Fig. 16 R.M.S. differential pressude in the 40-60 Hz band versus flowrate and mass quality

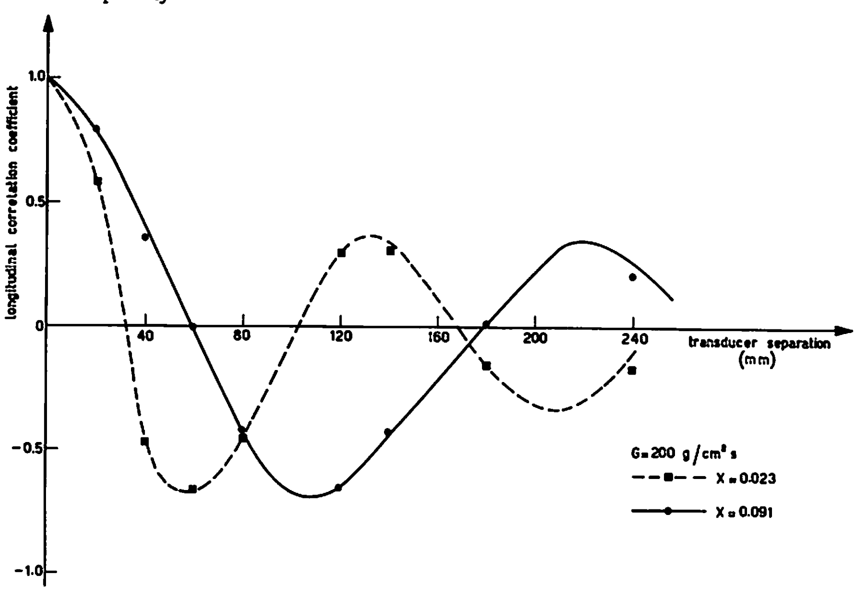

Fig. 17 Typical pressure differential longitudinal correlation coefficient

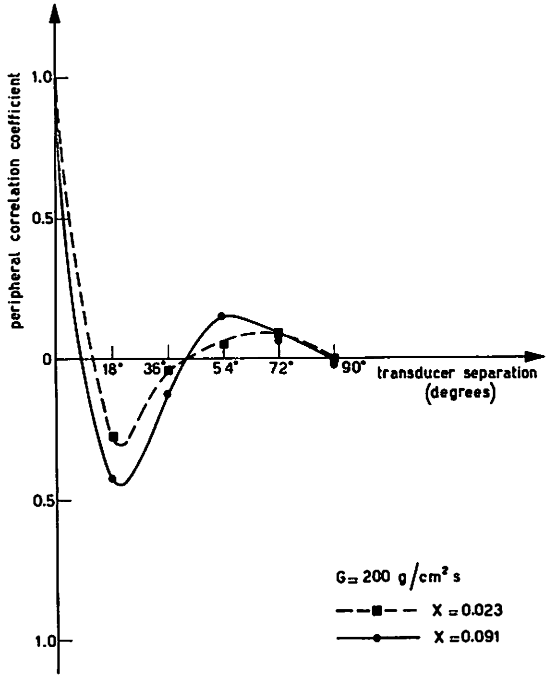

Fig. 18 Typical pressure differential peripheral correlation coefficient

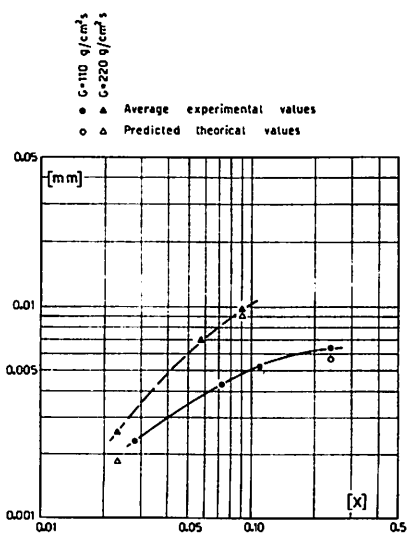

Fig. 19 Comparison between experimental and theoretical values of fuel rod r.m.s. displacement

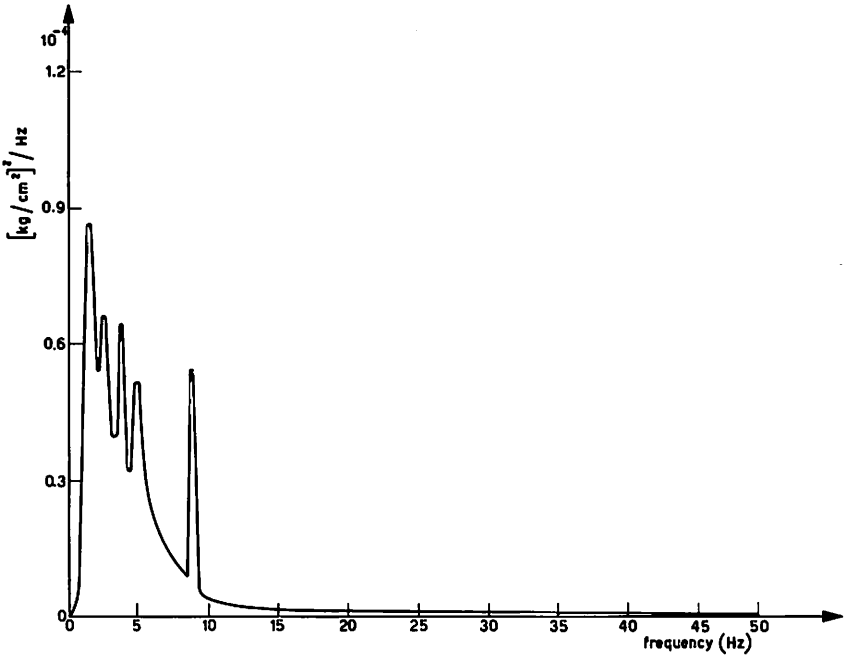

Fig. 20 Typical power spectral density curve of pressure fluctuation

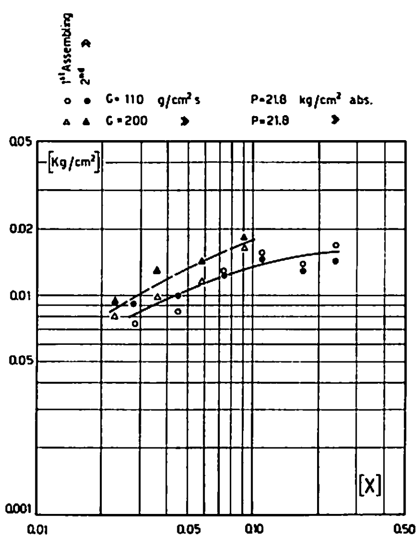

Fig. 21 R.M.S. pressure fluctuation versus flowrate and mass quality.

DISCUSSION

Q T. J. LEDWIDGE, Australia

1. How are the results taken with a non-condensable gas reconciled with the situation in which vapour generation and collapse take place ?
2. Could you explain the shape of the spatial correlation function if the fluid is perfectly random and homogeneous then it can be thought of as a Markov process and hence one would expect a correlation of an exponential form ?

A L. CEDOLIN, Italy

1. In the thermohydraulic conditions characterized by annular dispersed flow, the disturbances caused by the generation and collapse of vapour bubbles are of minor importance with respect to the disturbances caused by the annular dispersed flow itself (due to momentum fluctuation, local and time variation of the phases' velocity and concentration, long disturbance waves and so on). The principal object is then the simulation of the main physical properties of the mixture (such as density, viscosity, surface tension and so on). In our experiments we simulated the density of the steam-water mixture, and we will investigate the influence of surface tension using acetone as a liquid phase. We think that the influence of vapour generation may only be related to the presence of a variable quality along the channel, and we have planned experiments on it.
However, the model we used to predict the fuel rod vibration can be utilized for any thermohydraulic conditions, provided that the pressure differential oscillations are investigated.
2. The shape of the longitudinal correlation function can be explained with the presence of a convective velocity in the sense of the fluid flow for the pressure disturbances.

Q E. OHLMER, JRC Ispra, Italy

1. I am surprised to see your PSD-plots for pressure fluctuations in two-phase-flow showing very small and decreasing powers for low frequencies. We have with many measurements in single-phase (water) flow never found such a behaviour in the pressure fluctuations. Have you an interpretation of this strange behaviour of the results from our measurements ? Could it be a filtering-influence of the analysis apparatus ?
2. The results for impact forces as you have given here, are very interesting. In fact the wear and fretting corrosion problem is one of the most important aspects of fuel element vibrations. The impact forces are strongly influenced by the elasticity of the two impacting bodies. I think, in your experiments the impact transducers will have a different material than the liner tube. Consequently the peaks of impact forces will be different in the measurement than in reality, when the bundle will impact the tube. Certainly your results will be qualitatively all right, but not also quantitatively. Have you considered this point in your results ?

A L. CEDOLIN, Italy

1. The question clearly regards the PSD of the pressure differential between diametrally opposite taps, since the absolute pressure fluctuations (see Fig. 20) have been found to contain most of the power in the low frequency band. We think that the difference pointed out in single and two phase flow may be related to the extremely turbulent nature of the latter, which shows much higher pressure fluctuations. In any case, the difference is not due to the signal's processing, since we used a high pass filter 0.1 Hz.

2. Of the two impacting surfaces - the fuel rod with its wear pad and the pressure tube wall - the former is of course the most flexible one, and the characteristics of the impact phenomenon should be influenced by the elasticity of the pressure tube surface only to a very limited extent. A reconfirmation of this hypothesis is given by the fact that by increasing the penetration of the force transducers, the peaks of the impact forces have been found to increase up to the amplitude of the oscillating component of the impact forces in conditions of permanent contact. This means that contact forces are governed only by the exciting forces and rod inertia, i. e. therefore flexibility prevents from the appearing of forces due to the impact velocity.