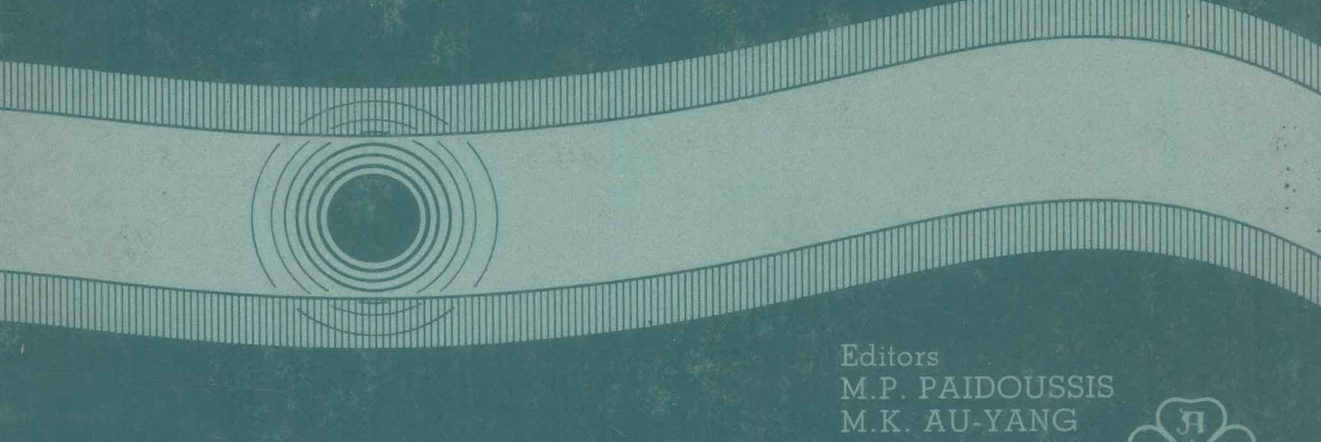
SYMPOSIUM
on
FLOW-INDUCED VIBRATIONS

VOLUME 4
VIBRATION INDUCED BY AXIAL AND ANNULAR FLOWS



Editors
M.P. PAIDOUSSIS
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VIBRATION INDUCED BY AXIAL AND ANNULAR FLOWS

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Applied Mechanics, Fluids Engineering, Heat Transfer, Noise Control and Acoustics, Nuclear Engineering, and Pressure Vessels and Piping Divisions

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PREFACE

The 1984 ASME Symposium on Flow-Induced Vibration is a unique event in the annals of technical meetings organized by ASME. Apart from promising to be one of the most important symposia anywhere on this topic in recent memory (only time will tell exactly how important), it is the first time that such a large symposium on the subject has been organized by ASME. Furthermore, it is the first time that no less than six Divisions of the ASME have cooperated in co-sponsoring a symposium on any given subject, which surely bespeaks of the importance of the subject matter of this particular Symposium. The participating Divisions are:

Applied Mechanics, Fluids Engineering, Heat Transfer, Noise Control and Acoustics, Nuclear Engineering, and Pressure Vessels and Piping.

I should like to thank them all, for without their support this Symposium would not have been the success that it is promising to be.

The Proceedings of the Symposium are published in six bound volumes, containing sixty-eight papers in all, as follows:

Volume 1 Excitation and Vibration of Bluff Bodies in Cross Flow
Volume 2 Vibration of Arrays of Cylinders in Cross Flow
Volume 3 Vibration in Heat Exchangers
Volume 4 Vibration Induced by Axial and Annular Flows
Volume 5 Turbulence-Induced Noise and Vibration of Rigid and Compliant Surfaces
Volume 6 Computational Aspects of Flow-Induced Vibration

The organization of a Symposium of this size, with world-wide participation (from 12 countries), has been both a challenging and rewarding experience. It entailed a great deal of work by many people: the session developers, the reviewers, ASME Headquarters' staff, the 1984 WAM Organizers and, of course, the authors. Of the many people involved, too numerous to mention by name here, I am specially indebted to the session developers and co-editors (O. M. Griffin, M. Sevik, M. K. Au-Yang, S. -S. Chen, J. M. Chenoweth, M. D. Bernstein and A. J. Kalinowski), and would like to single out two: Dr. M. K. Au-Yang and Dr. S. -S. Chen, whom I would like to thank for their unswerving support from the very beginning, when the possibility of a "multidivisional symposium" looked like a pie in the sky! I would also like to thank my secretary, Ruth Gray, for efficiently handling the enormous amount of paperwork involved in several passes of sixty-eight-plus papers across my desk.

Michael P. Paidoussis
Principal Symposium Coordinator
and Principal Editor

FOREWORD

In contrast to most of the papers in this Symposium, the nine which may be found in this volume, Vol. 4 of the Symposium Proceedings, all deal with physical situations in which the flow is parallel, rather than normal, to the long axis of symmetry of the vibrating structure: flow in a pipe, axial flow in clusters of cylinders, and flow in the annulus between two coaxial cylinders (or quasi-cylindrical bodies). From the analytical point of view, this distinction is very important: in most such cases, the separated, rotational flow regions are either minimized or totally absent, which makes the generation of analytical models for prediction of vibration much more feasible than in the case of cross-flow, where the rotational flow regions are more widespread and more important, so far as flow-induced forces and vibration are concerned.

Seven of the nine papers in this volume deal with various aspects of annular flow-induced vibration, which is certainly a welcome development, as it is well known that annular or leakage flows have been responsible for many serious problems in nuclear reactors (thermal shield, core barrel, fuel stringer and control-rod), in jet pumps and in certain types of pressure-reducing valves. The reader here is referred to the paper by M. P. Paidoussis "Flow-Induced Vibrations in Nuclear Reactors and Heat Exchangers: Practical Experiences and State of Knowledge", in Practical Experiences with Flow-Induced Vibrations (eds.: Naudascher, E., and Rockwell, D.), Springer-Verlag, 1980, pp. 1-81. Until recently, the state of the art and the analytical tools available to the designer have been rather rudimentary. In 1982, at the Third International Conference on Vibration in Nuclear Plants, held in Keswick, we saw significant advances – both in terms of understanding and analytical modeling – by Parkin and his associates of U.K.A.E.A. and Hobson of the C.E.G.B. in the U.K. in both cases. These same authors and others carry these advances further in this volume, and it may confidently be predicted that development of this technologically important field will henceforth be very rapid.

In this volume we also have a paper on the effect of axial flow on the dynamics of coaxial cylindrical shells, a successful analytical model for predicting vibration of clusters of cylinders in axial flow, and an interesting paper on parametric resonance of piping containing pulsatile flow.

We would like to thank the authors for their cooperation in submitting papers of high quality to this Symposium, and specifically on the topic of this volume (Volume 4) of the Proceedings, as well as for their willingness to participate and share their experience with others in this Symposium. We would also like to thank the reviewers for their thoughtful comments and for the experience they have brought to bear in the review process, which has ensured the selection of only worthy papers for the Symposium and contributed to the improvement of those finally accepted.

M. P. Paidoussis

M. K. Au-Yang

CONTENTS

REDUCTION OF VIBRATION CAUSED BY FLOW IN AN ANNULAR DIFFUSER

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ABSTRACT

This paper describes a physical model explaining flow induced vibration observed in 6^{o} and 30^{o} annular diffusers where the centre body forming the diffuser has freedom to move radially. The model is based on experimental evidence, and accounts for both fluid behaviour and structural response. The paper concludes by describing two vibration suppression devices both of which have performed successfully in realistic demonstration tests. This work is of application to the on-load refuelling of the UK Advanced Gas Cooled Reactors.

1. INTRODUCTION

On-load refuelling of the UK Advanced Gas Cooled Reactors (AGRs) is an important feature with proven economic benefit[1,2]. The process of on-load refuelling requires the withdrawal of a spent fuel assembly from the reactor core, followed by insertion of a replacement fuel assembly, whilst the reactor continues to produce power.

A fuel assembly is a long slender articulated unit of l/d = 92. It is made up of eight cylindrical fuel elements, together with service components such as neutron, gamma and thermal shields and flow control valve (Fig 1). During on-load refuelling some vibration of the suspended fuel assembly occurs as it is withdrawn from, or inserted into, the matching channel in the reactor core. This is caused by the high density gaseous coolant (CO_2 at 38.5 kg/m^3) flowing up the annulus formed between the channel and fuel assembly.

Currently refuelling is limited to 30% power. This is achieved by reducing reactor power, replacing a batch of 6 - 12 fuel assemblies, then returning to full power for 2 - 4 weeks depending on the batch size. The restriction on refuelling power level is dictated by a number of considerations, of which fuel assembly vibration is one[1]. At present a programme of R & D work is being conducted with the object of increasing the refuelling power level. The vibration studies described here are a UKAEA funded contribution to this programme.

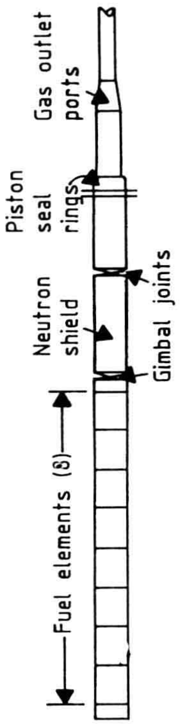
The work described here follows from earlier published work which reported identification of the source of vibration and describes early studies of the vibration mechanism and means of suppressing it.

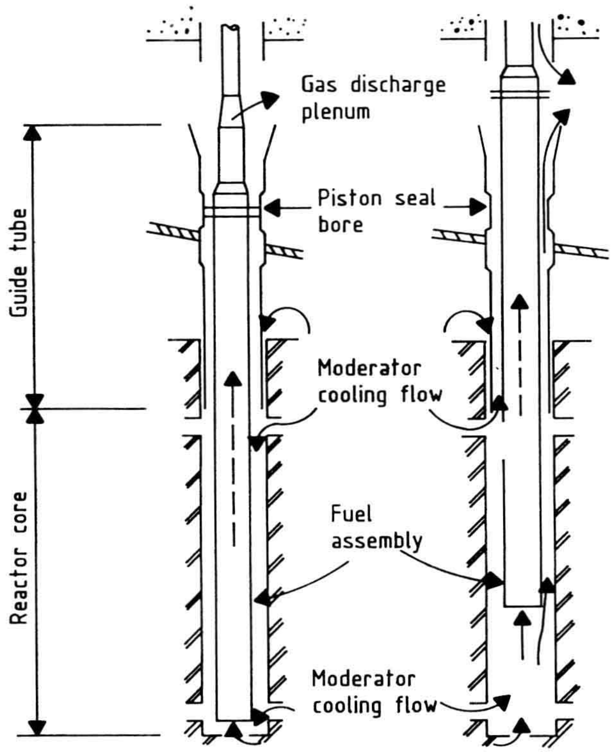
2. DESCRIPTION OF THE PROBLEM

2.1 The reactor background

The source of vibration has been identified as being in a reduced diameter section of the channel known as the piston seal bore (Fig 2).

When the fuel assembly resides in-core, piston ring seals on the fuel assembly are located in the piston seal bore. The object of these is to direct moderator cooling flow downwards through the fuel assembly/channel annulus to the bottom of the core, where it joins the fuel inlet flow (Fig 2(a)). When the fuel assembly is lifted during refuelling this seal is opened, allowing flow upwards through the fuel assembly/channel annulus to the gas outlet ports (Fig 2(b)).





(a) Normal operation (b) Refuelling

FIG 1: AGR fuel assembly.

FIG 2: Hinkley Point-Hunterston fuel assembly and reactor fuel channel configuration.

2.2 The problem - general

The earlier reported work which identified the source of vibration was in two stages. First of all the results of full scale tests at the correct fluid-dynamic conditions were examined. From the frequency and displacement data, the flows, and the fuel assembly/channel relationships including varying axial position of the fuel assembly, it was deduced that the source of excitation was in the region of the piston seal bore[3,5]. This was then confirmed by the application of a vibration suppression device which, when applied to the region of the piston seal bore, proved to be most effective in stabilising the fuel assembly[3,4].

The operational function of the piston seal bore, and its behaviour as a source of fuel assembly excitation, applies equally well to two reactor designs, which include detail differences in both the fuel assembly and the channel. One of these, the Hinkley Point-Hunterston design has already been outlined (Figs 1 and 2). The other, Hartlepool-Heysham 1, has a channel design as shown in Fig 3; and a notable fuel assembly feature, unique to this reactor design, in the form of a large axial compressive spring load (1800 kgs) to prevent gaps opening between fuel elements during on-load refuelling.

The fuel assembly vibrational characteristics in the two reactors are very different, even though the source of excitation is the same for both. Initially it was not clear whether this was a function of the flow interacting with the piston seal bore, the fuel and channel diameters being the same for both; or whether it was a function of the fuel assembly response. A test was conducted interchanging the fuel assemblies and channels. The resulting vibrational characteristics were not significantly altered, so it is argued that the problem is essentially a function of the interaction between the flow and the seal bore. The only significant detail which could cause this is the piston seal bore diffuser angle. This is 6° for Hinkley Point-Hunterston, and 30° for Hartlepool-Heysham 1. In view of the importance of this feature the two designs will be referred to in this paper by the diffuser angle, rather than reactor design. This definition incorporates differences in annular gap size downstream of the diffuser (Fig 4). The significance of this dimension as an independent variable has not been investigated.

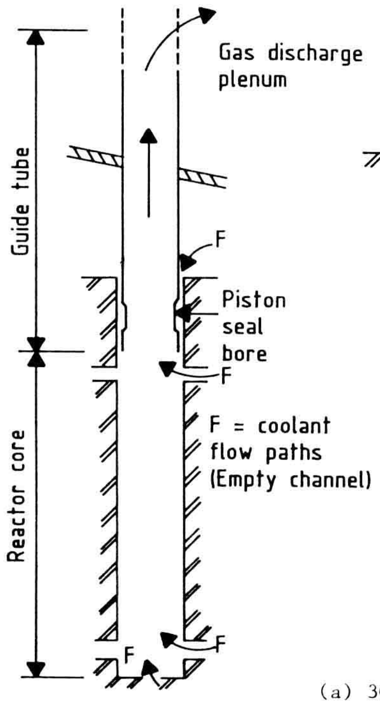
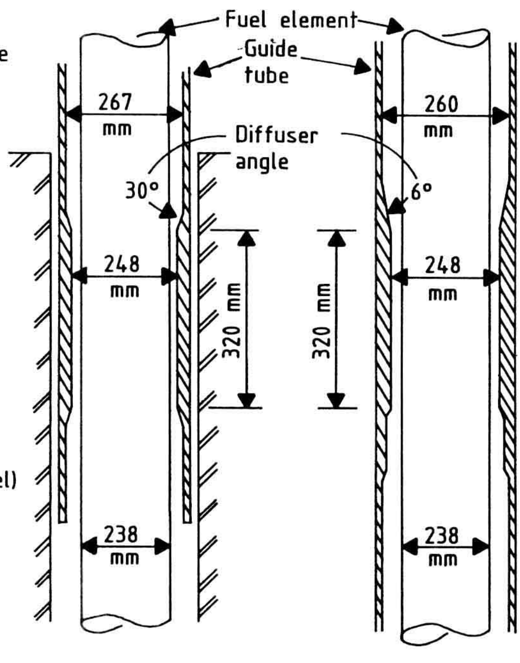


FIG 3: Hartlepool-Heysham 1 reactor fuel channel



(a) 30° Diffuser (Hartlepool-Heysham 1)

(b) 6° Diffuser (Hinkley Point-Hunterston)

FIG 4: Comparison of piston seal bore detail.

2.3 The problem - definition

The dimensions of the two piston seal bores are given in Fig 4. A fuel element is shown in place as is the case during the vibration of concern. It must be emphasised that the fuel element is free to move radially, and to tilt, subject only to the restraints imposed by the fuel assembly in which it is located, the high density flowing gas and the channel boundary.

The essential differences in the vibration characteristics associated with the two piston seal bores can be illustrated in terms of frequency and displacement. The frequency characteristics are shown in Fig 5. The 30^{o} diffuser case results in a linear frequency-flow velocity relationship (Fig 5) suggestive of vortex shedding and characterised by a Strouhal relationship where the Strouhal Number is 0.23[4]. The 6^{o} diffuser results in lower frequencies with no simple relationship to flow (Fig 5).

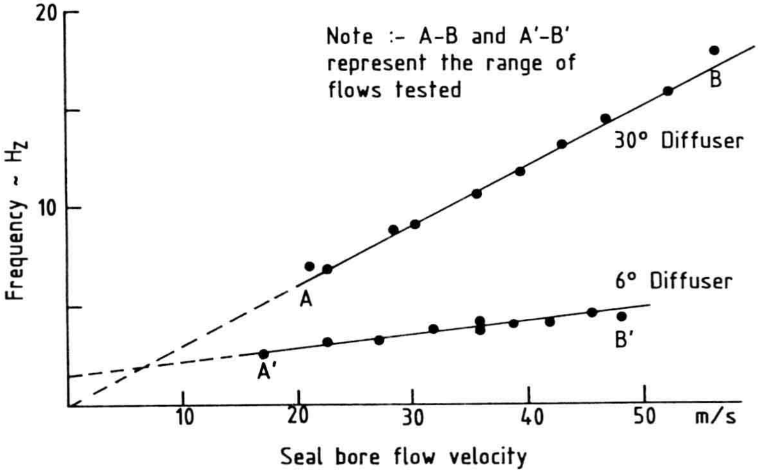


FIG 5: Frequency data at the piston seal bore from full scale tests of the two reactor designs.

The displacement data also reveal well defined differences, the 30^{o} diffuser causing an oscillatory precessional motion (Fig 6(a)), whilst the 6^{o} diffuser results in motion which ranges from diametral to precessional (Fig 6(b) - (d)). The radial displacement of the centre body in the 30^{o} diffuser case, Fig 6(a), is believed to be a function of the circumferential static pressure distributions which develop with eccentricity of the centre body.

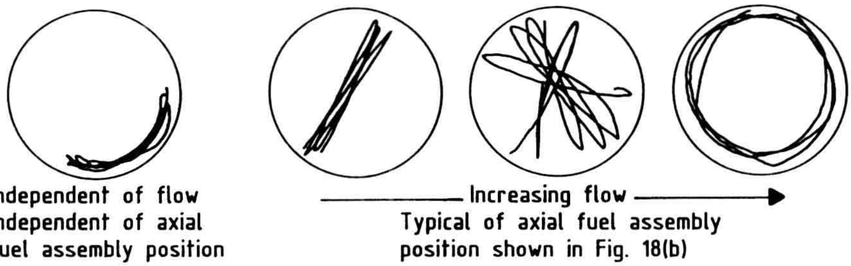
In the case of the 6^{o} diffuser the variations in the displacement diagrams are mainly a function of the axial disposition of the fuel assembly, with a weaker tendency to move from diametral to a more orbital mode on increasing flow.

3. THE PHYSICAL MODEL

3.1 30^{o} diffuser

3.1.1 Vortex shedding

Some important information on the source of vibration came from rig tests of only the piston seal bore section. The test sections of these rigs



(a) (b) (c) (d)

30° Diffuser Examples of motion in the 6° Diffuser

FIG 6: Displacement data at the piston seal bore from full scale tests of the two reactor designs.

were very much as in Fig 4, with the addition of suitable inlet and outlet sections. One study used a full scale model of this sort, with a fixed centre body and air as the circulating fluid. Vortex shedding was observed immediately downstream of the diffuser (as discussed in 31.3) and the resultant pressure pulsations were measured[4]. From this it is argued that the 30° diffuser results in an intrinsic forcing function. This evidence was strengthened by full-scale on-load refuelling tests both in a large dynamic test facility[4] and in a reactor[5]. Both tests series confirmed fuel assembly response frequencies which agreed with those derived from the fixed centre body air model.

3.1.2 Structural response

Two observations should be made of the results from on-load refuelling tests which include the response characteristics of the fully-suspended fuel assembly. First, there is no evidence of resonant response of the fuel assembly over the continuous range of frequencies (7 - 18 Hz) at which it was excited (Fig 5). (It should be noted that vibration is observed to develop continuously at flows up to A in Fig 5, though there are no measured data to quantify this). This at first seems surprising since mechanical resonance tests of this fuel assembly in air[6], indicate 3 modes in this frequency range of which one (at 10.5 Hz) should be particularly responsive because an anti-node coincides with the source of excitation, the piston seal bore. This response frequency may well be significantly modified by the closely constraining channel, and the high density, high velocity gas flowing up the annulus between the fuel assembly and the channel. It may just be possible for added stiffness effects to shift the "in-air" 10.5 Hz mode above the observed range, yet not bring the next most responsive mode (3.9 Hz in air) up to the lower end of the observed range.

3.1.3 Response frequency bandwidth

The second observation is the difference in the frequency bandwidth between pressure pulsations in the air rig tests and the fuel assembly response in the full scale, on-load refuelling tests (Fig 7).

It is worthwhile considering first the nature of the vortex shedding. In earlier work [4] wool tuft flow visualisation was reported which indicated two vortices in the plane of the annulus. More recently further flow visualisation tests have been conducted, this time using air bubbles in water. There are differences between the two sets of observations which are summarised in Fig 8.

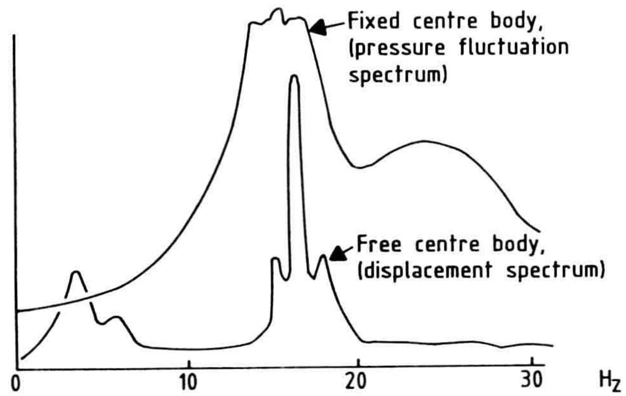
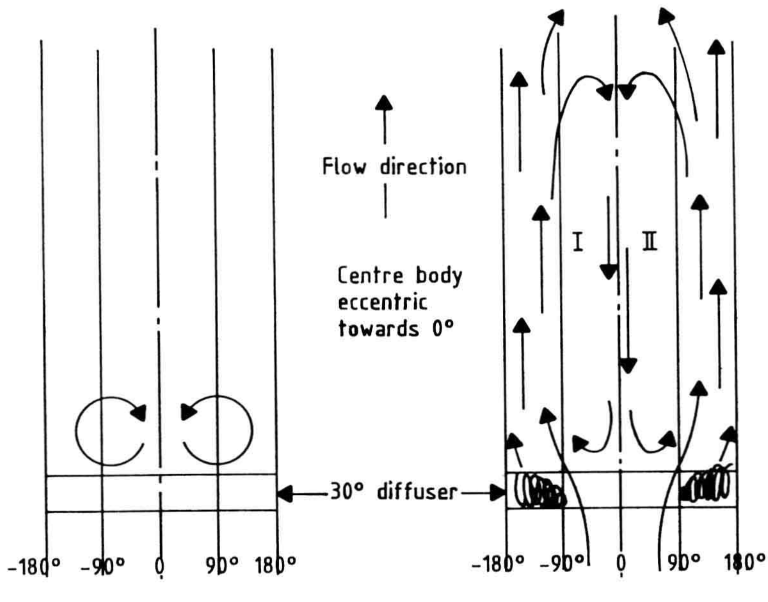


FIG 7: Comparison between fixed and free centre body.



(a) Wool tufts in air
Re = 3.5 x 10^4

(b) Air bubbles in water
Re = 1.3 x 10^4

(Reactor Re = 10^6)

FIG 8: Comparison of flow visualisation results - the annulus unwrapped.

The most complete flow visualisation was obtained from the water rig (Fig 8(b)). For a fixed centre body two features were apparent; one, the spiral vortices held in the diffuser section, and two, the recirculation flows I and II. These features were very stable giving no visual indication of flow pulsation. If the centre body was allowed to be free, though pin jointed well upstream of the seal bore, oscillation of the centre body occurred in the form shown in Fig 6(a). This was consistent with the observed out-of-phase development of recirculation flows I and II. The wool tuft flow visualisation could not indicate the phasing of the vortices shown in Fig 8(a), however they were well defined, had the same rotation as the recirculating flows in Fig 8(b) and so it is assumed that they are a separation effect developing with the higher Reynolds' number flow in the air rig. Because of the agreement between the frequency of pressure pulsations measured in the fixed centre body air rig, and the frequency of vibration of the fuel assembly in the full scale tests (Fig 7) it is deduced that the vortices shown in Fig 8(a) were out-of-phase.

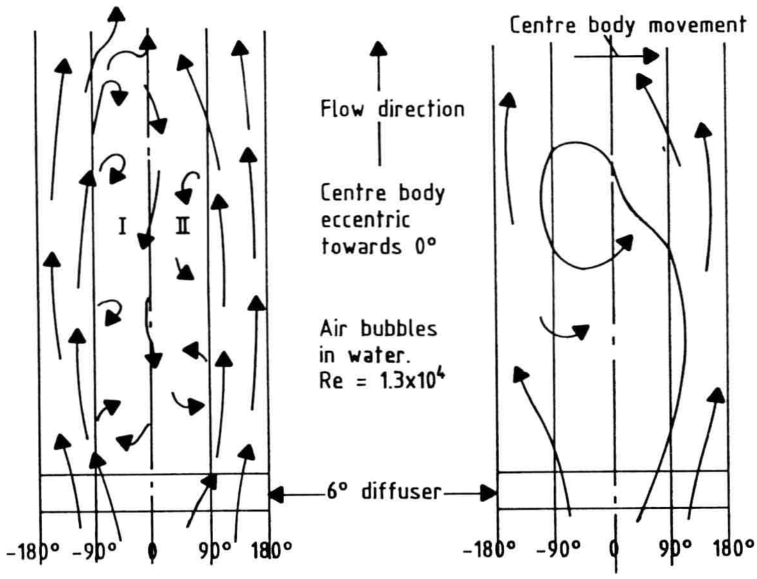
Returning now to the change in frequency bandwidth between air rig tests and the full scale tests (Fig 7) it is argued that this is a consequence of a tuning of vortex shedding by the moving centre body.

The spiral vortices shown in Fig 8(b) were very stable with no sign of systematic detatchment, even in the moving centre body case; it is argued therefore that these were not a part of the destabilising forces but could in fact be regarded as beneficial. This will be considered further in discussing the 6° diffuser.

3.2 6° Diffuser

3.2.1 Flow field

The behaviour of the piston seal bore with the 6° diffuser is totally different as illustrated in Figs 5 and 6. Flow visualisation information from the water rig shows a greater random instability than that exhibited by the 30° diffuser, together with little or no separation at the diffuser. This is illustrated in Fig 9.



(a) Fixed centre body (b) Free centre body

FIG 9: Flow visualisation of 6° diffuser.

The greater random instability reflects the measurements of Fricke[7] who observed greater destabilising forces with a 2° diffuser than 30°. He concluded that this was a result of the less determinate recirculating flow in the 2° diffuser. It is argued that this indeterminate nature of the small angled diffuser makes a systematic recirculation unlikely, as was the case with the large angled diffuser (Fig 8).

When the centre body is allowed to move freely, though pin-supported well upstream of the seal bore, it vibrates in a manner very like that of a fuel assembly, as illustrated in Fig 6(b). This movement is accompanied by gross flow effects as illustrated in Fig 9(b), which represent an out-of-phase and more organised form of the recirculation flows I and II in Fig 9(a).

The conclusion drawn from these observations is that there is no systematic intrinsic forcing function due to fluid flow characteristics as for the 30° diffuser, but that there appears to be broad band turbulent energy available in the fluid stream downstream of the diffuser. This energy is derived predominantly from shear effects at the boundary between the stagnant zones in the centre of the recirculating flow and the upward flow in the wide part of the annulus. It does become organised when the centre body is allowed to move freely and this does result in a seemingly Strouhal-like vortex shedding relationship (Fig 10).

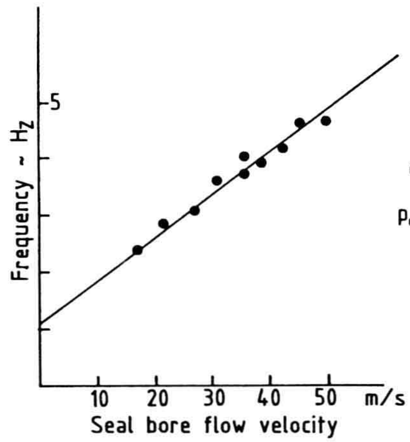


FIG 10: 6° diffuser, variation of frequency with flow velocity

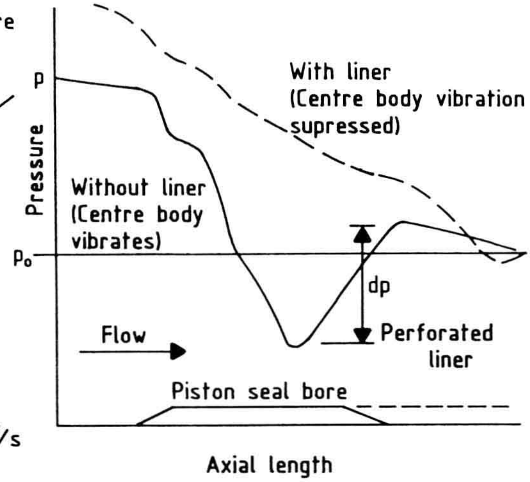


FIG 11: Pressure recovery, with and without vibration suppression

3.2.2 Piston seal bore pressure recovery (dp/p)

This rather more complex picture is supported by observations from a half-scale free centre body air rig, which found a relationship between pressure recovery in the seal bore and vibration of the centre body; the greater the magnitude of the pressure recovery, the more severe the vibration. This is illustrated in Fig 11 which shows axial pressure measurements for a free centre body, with and without a vibration-suppressing perforated liner[8]. The action of this liner will be discussed later.

3.2.3 Pressure recovery stiffness and negative damping

This relationship between vibration and presure recovery is examined in more detail in the following, outlining a feed-back mechanism which can cause vibration. The essential information was derived from axial pressure measurements on a half-scale fixed-centre body air rig. These showed that axial pressure recovery varied around the annulus of a fixed centre body in such a way that there is a resulting centralising force (F) on the centre body (Fig 12).

These measurements were repeated for a series of eccentricities of the centre body, and the implied restoring force derived as a function of eccentricity, (Fig 13), equivalent to a stiffness term.

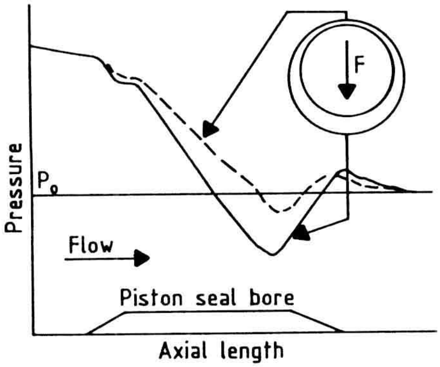


FIG 12: Pressure recovery on either side of an eccentric centre body

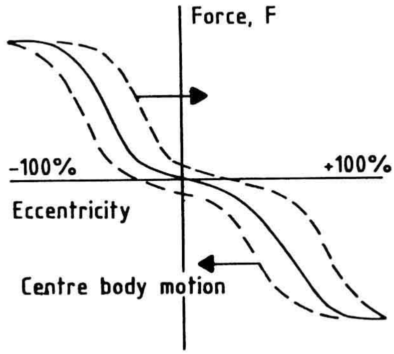


FIG 13: Derived force-centre body displacement relation

Now by making the simplification of ignoring flow re-distribution it can be argued that if the centre body moves as a consequence of centralising force the instantaneous fluid forces will lag in phase (compared with the static counterpart) as shown in Fig 13 by the dotted lines. This velocity dependent modification to the force, together with the momentum of the centre body, will result in side to side motion of the centre body. Measurements of this effect, which amounts to a negative damping, have not been made in this case. They were observed in a very similar case, that of the flow control valve or gag located above the fuel elements in the fuel assembly (Fig 1) and described in [9].

The above hypothesis considers only eccentricity of the centre body. In practise tilt can also occur up to a maximum of 1.7°. It is believed that this has only a modifying effect similar to that illustrated in [4].

3.3 Comparison of the instability mechanisms

In the foregoing it has been argued that the two diffusers behave in two totally different ways. It must be said that this is probably not strictly so. For example the 30° diffuser does exhibit some pressure recovery characteristics on either side of the eccentric centre body as illustrated in Fig 12. In this case these characteristics are however much weaker. Conversely the 6° diffuser does result in vortex shedding when the centre body is allowed to move freely (Fig 9(b)), though not when it is fixed. A parallel could be drawn between this and the tuning of vortex shedding which occurs in the 30° diffuser, when the centre body is allowed to move freely (Fig 7).

It is therefore proposed that the reality of the situation is complex, that the 30° diffuser instability mechanism is predominantly a forced vibration derived from vortex shedding, that the 6° diffuser instability mechanism is predominantly a feed-back mechanism originating in pressure recovery characteristics, but that each is subject to secondary effects characteristic of the other.

3.4 Fluid stiffness and structural response

In a preceding section (3.2.3) discussing the 6° diffuser, it is argued that the fluid forces in the seal bore result in a force on the centre body varying with eccentricity, that is a fluid stiffness term (Fig 13). This will vary with flow rate offering an explanation for the variation of frequency with flow illustrated in Fig 10. It is also argued that there will be a similar, though weaker, fluid stiffness term in the case of the 30° diffuser. Whilst the effect of this fluid stiffness term upon the structural response cannot be calculated, it does seem that it will have a significant effect in such a closely constrained channel with high density fluid. This then makes possible the resonant modification discussed for the 30° diffuser in Section 3.1.2. It also can explain Fig 10, where the finite zero-flow frequency of 1.1 Hz is close to a natural in-air mode of 0.73 Hz, and the trend from this determined from the fluid stiffness term which increases with flow.

4. VIBRATION SUPPRESSION

4.1 General objectives

Two approaches have been explored. One was to seek a modification to the reactor channel especially in the seal bore diffuser region. The other was modification to the fuel assembly, in particular the fuel elements themselves. The merit of the latter is that it can be applied to all reactors, operational or new; the former solution can only be implemented during construction of a new reactor. Solutions of both types have been found and will be described in the following sections.

4.2 30° Diffuser - fuel element modification

Vibration suppression in this case evolved from the concept of destroying or tripping vortices as employed by Fricke[7]. He showed that a single circumferential discontinuity located on the surface of the diffuser, would destroy vortices generated in a configuration similar to that under consideration here. This idea was transferred to the fuel element with two modifications: one, that the discontinuity was recessed (a groove) rather than protruding as used by Fricke. (This was necessary to avoid snagging during refuelling); two, there should be an axial series of these discontinuities to cater for axial movement of the fuel assembly through the seal bore during refuelling. The profile shape chosen (No. 11 in Fig 14) eliminated pressure pulsations as measured in air rigs. This was confirmed by full scale refuelling tests with a fuel assembly (Fig 15).

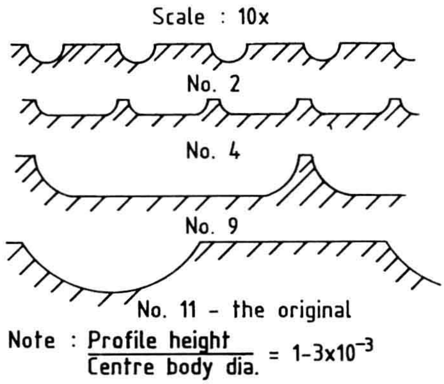


FIG 14: Some tested surface profiles

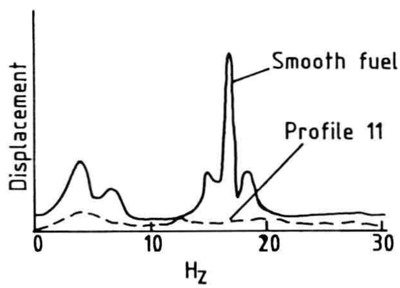


FIG 15: 30° diffuser, original grooves

4.3 6° Diffuser - channel modification

Work with the perforated liner showed that marked pressure loss and subsequent pressure recovery through the seal bore was largely eliminated by inserting the liner in and downstream of the diffuser (Fig 11). Flow visualisation in a full scale two-dimensional air rig indicated that this was a result of the liner controlling the diffusion rate into the part of the passage behind the diffuser (x in Fig 16).

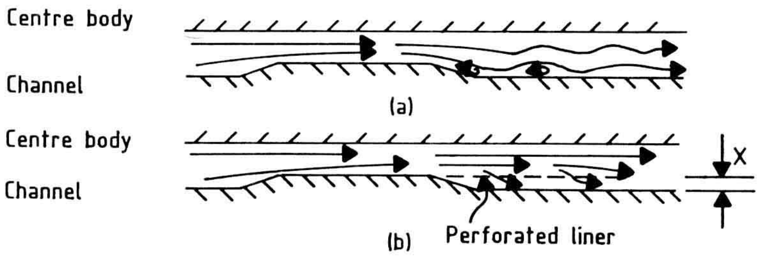


FIG 16: Influence of perforated liner on diffuser flow

A sufficient reduction of the magnitude of the pressure recovery means that because of the lower rate of change of force with eccentricity the hysteresis effects illustrated in Fig 13 become very small and the negative damping term falls to such a value that total damping remains positive, hence leading to a stable situation.

Tests of this device mounted in a channel during full scale fuel assembly refuelling tests showed complete suppression of vibration. These results fitted the air rig observations described above.

4.4 6° Diffuser - fuel element modifications

4.4.1 Initial fuel assembly tests

Following the success of the grooved surface in the 30° diffuser the same surface was tried in the 6° diffuser. It had a marked effect (Fig 17) but was not as successful as in the 30° diffuser. The results shown highlight another feature of the reactor containing the 6° diffuser, which is a structural response aspect not present in the 30° diffuser case. This is illustrated in Fig 18 which relates two different response characteristics to two different fuel assembly axial positions having different freedoms as shown in the figure.

The implication is that the forcing or negative damping term described in Section 3.2.2 is the same for both cases, but that system damping is lower for Fig 18(b) than Fig 18(a), and so the total damping is positive in 18(a) but remains negative in 18(b).

4.4.2 Diffuser flow characteristic and grooving

From the preceding observations it was concluded that the grooving first tested did alter the diffuser flow characteristic, and thereby reduced the magnitude of seal bore pressure recovery, but to an extent which allowed a net residual negative damping in the less restrained fuel assembly (Fig 18(b)). The alteration in diffuser flow characteristic due to grooving is illustrated in Fig 19 using computed data for an axisymmetric two-dimensional representation of the diffuser.