# THE EVALUATION OF IN-REACTOR VIBRATORY AND IMPACT MOVEMENTS OF CIVIL AGR FUEL ELEMENTS

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**SUMMARY**

In the development of fuel assembly designs for the Civil Advanced Gas-Cooled Reactors under construction in the United Kingdom, studies of the in-reactor vibration behaviour of the assemblies have been performed in out-of-pile test loops:

— K. A. Sansom and H. K. Edwards "The Development of a stable fuel assembly for the Hinkley Point 'B' nuclear reactor". Paper to be presented at the Conference on Vibration Problems in Industry, Keswick, April 1973.
— D. Bramwell "Dynamic behaviour of Mk 2 CAGR fuel pins in a flow and noise environment". Paper to be presented at the Conference on Vibration Problems in Industry, Keswick, April 1973.

This paper describes the development of procedures for evaluating the criteria which determine the allowable vibration of the fuel, as outlined below.

A Civil AGR fuel element comprises 36 stainless steel clad uranium dioxide fuel pins supported by a grid and located by two braces within a graphite sleeve. A typical fuel assembly is made up of a stack of eight fuel elements surmounted by a plug unit. Vibration of the complete assembly will induce forced vibration of the internal fuel element components. Simple dynamical concepts have been used to deduce the relation between internal component motion and the measured overall fuel assembly motion, and have been supplemented by experiments where necessary. Component vibration excited directly by the flow turbulence or by noise has been measured separately.

The most significant damage modes which can result from these vibrations are fatigue, wear and impact damage due to the fuel assembly tapping against the reactor core bricks.

Conditions under which fatigue of the fuel pin cladding could occur have been assessed from basic materials data allowing for the statistical variation in fuel pin response within the reactor. Wear damage has been assessed from component tests on the wear due to relative axial movement at the pin/brace locations. These basic data provided a theoretical wear model which was adapted to estimate the wear due to lateral movement. Supporting wear tests on fuel elements subjected to lateral vibrations have confirmed these predictions within a factor of 2. Impact damage has been assessed from laboratory tests on the impact strength of fuel components.

This paper presents a new approach in relating the internal motion of components within a complex fuel assembly to the measured vibration response of the complete assembly, and in interpreting the wear and fatigue data from different sources to derive criteria for allowable vibratory motion.

## 1. Introduction

In the development of fuel assembly designs for the Civil Advanced Gas-Cooled Reactors under construction in the United Kingdom, studies of the in-reactor vibration behaviour of the assemblies have been performed in out-of-reactor test loops (Sansom and Edwards [1], Bramwell [2]). The aim of these studies was to show that, during reactor operation, the assembly would not vibrate sufficiently to cause any significant reduction in fuel life. For this purpose it was also necessary to develop criteria relating the possible modes of damage to the fuel with the vibratory and impact movements of the fuel elements. This paper describes the development of procedures for evaluating the criteria which determine the allowable vibration of the fuel.

The fuel assembly is itself a very complex structure and the necessity to provide working clearances between components introduces non-elastic behaviour which affects the linearity of the response. Moreover, allowance must be made for the variations in behaviour due to component tolerances and scatter in material properties, and due to the effects of irradiation. This reduces the ability to make accurate predictions of the fuel assembly vibration. Consequently the vibratory response of the fuel is described in terms of very simple dynamic models. Expressions for fuel damage have been determined from experiments on unirradiated components and have been approximately adjusted for the effects of varying fuel-can contact in-reactor. All these factors limit the value of a more advanced analysis of the vibratory motion. It has been shown that the simple models adopted here describe the vibration and wear behaviour of the fuel elements in out-of-reactor tests with reasonable accuracy. This approach is likely to be as reliable as any more refined technique in providing acceptance criteria for fuel vibration.

## 2. Description of the Fuel Assembly and its Environment

Fig. 1 shows a Civil AGR fuel element. Each element contains 36 stainless steel clad $UO_2$ fuel pins supported at the bottom end by a grid and located at the centre and top of the element by braces, within a graphite sleeve. A typical fuel assembly is made up of a stack of 8 fuel elements surmounted by a plug unit providing shielding and hold-down functions.

The arrangement of the assembly in the reactor is shown schematically in Fig. 2. In-reactor the assembly is seated on a fixed support at the bottom end and is located laterally by a piston seal mounted on the lower half of the plug unit and engaging in the reactor guide tube. The upper section of the plug unit is guided within the reactor stand-pipe. The assembly is cooled by an internal flow of gas through the fuel and by external flow through the core. The level of acoustic noise in the reactor channels is typically of the order of 150 db.

## 3. Nature of Fuel Assembly Motion

Experimental work of the type described by Sansom and Edwards [1] has shown that, in a simulated reactor flow, the fuel assembly undergoes random harmonic vibrations at frequencies of 2.5 and 7 Hz. These correspond to the first and second bending modes as measured in tests using mechanical excitation, and as calculated for the fuel column. This movement will induce a forced vibration of components within the fuel elements which could lead to wear or fatigue problems. Impacts could also occur under conditions where the fuel assembly is sufficiently close to the core bricks to tap against them. Furthermore, the flow turbulence and noise field within the fuel elements gives rise to vibration of the fuel pins and other components at their own natural frequencies (Bramwell [2]).

4. Types of Damage

The following effects on the fuel have been considered:

(a) Compressive failure of graphite sleeves due to bending strains.
(b) The possibility of opening joint gaps between elements in the assembly.
(c) Wear of graphite sleeves due to rubbing against core components.
(d) Fatigue of fuel pins due to bending strains resulting from fuel assembly vibration.
(e) Fatigue of grids and braces resulting from fuel assembly vibration.
(f) Wear due to rubbing between grids and sleeves.
(g) The effect of impacting against the core on the loads experienced by fuel element components and on the fuel pin strains.
(h) Wear due to relative movement between pins and braces.
(i) Fatigue and wear due to pin vibration excited by the gas flow and by noise.

Consideration of these effects has shown cladding and brace fatigue, pin/brace wear and damage due to impacting to be the most significant damage modes. These are considered below in more detail.

5. Vibration Models

5.1 Vibration of the Fuel Assembly

The vibration of the fuel assembly can be described by considering the column up to the piston seal as a beam with end supports and ignoring the effect of the plug unit above the seal. This simplification can be made since in practice the piston seal is found to behave effectively as a clamped support with little coupling between the motion of the plug unit and that of the fuel. The natural frequencies calculated for the fuel column have been found to be in reasonable agreement with the measured vibration frequencies of the assembly.

5.2 Vibration of Fuel Pins Induced by Fuel Assembly Vibration

The oscillatory inertial loading applied to the fuel pins due to whole-body motion of the fuel assembly will cause the pins to deflect at the forcing frequency and in the same direction over the length of the pin. The possible modes of deflection of the pins resulting from fuel assembly vibration will depend on the nature of the support at the grid and brace positions which will vary with manufacturing and assembly tolerances. Some possible deflection modes are illustrated in Fig. 3.

The bending moment applied to the pin due to the inertial loading will be greatest if the pin is held at one end only. However, deflection of the pin in the cantilever mode (Fig. 3a) is limited by the available clearance in the braces. This restricts the maximum bending strain in this mode to a very low level. The same restriction applies to modes with the pin held at the two extreme ends only (as in Fig. 3b), where the maximum deflection is limited by the centre brace. The higher order modes such as Fig. 3c are not subject to this restraint and so are of most concern from the point of view of fatigue damage. For the pinned-clamped-pinned case, the maximum bending moment on each half of the pin is

$$M_{max} = \frac{\alpha m (\frac{\ell}{2})^2}{8} = \frac{\alpha m \ell^2}{32} \qquad \text{eq. (1)}$$

where $\alpha$ is the maximum acceleration, m the mass per unit length, and $\ell$ the length of the pin.

The maximum strain is given by

$$\varepsilon_{max} = \frac{M_{max}\ r\ Q}{E.I} = \frac{\alpha m \ell^2 r\ Q}{32\ E.I} \qquad \text{eq. (2)}$$

where r is the pin radius, EI is the pin bending stiffness, and Q is the magnification factor (assumed to be 1 since the forcing frequency is much smaller than the pin natural frequency). The acceleration $\alpha$ is given by

$$\alpha = 4\pi^2 f^2 y_{max} \qquad \text{eq. (3)}$$

where f and $y_{max}$ are the frequency and amplitude of the fuel assembly vibration. Then from (2) and (3)

$$\epsilon_{max} = \frac{\pi^2 f^2 \, y_{max} \, m\ell^2 r \, Q}{8 \, E \, I} \qquad \text{eq. (4)}$$

which relates the maximum bending strain in the pin to the amplitude and frequency of fuel assembly vibration. Fig. 4 shows the agreement between the calculated values from eq. (4) and experimental values measured on an instrumented fuel assembly under mechanical excitation.

In order to assess the pin vibration in terms of wear at the pin/brace locations it is necessary to know both the rubbing movements and the contact load. Since there is very little information on the contact loads to be expected in fuel assemblies in the reactors, this has been estimated analytically as follows.

The effect of high contact loads will be to restrain the pin due to friction, so that rubbing and consequently wear, would only occur if the excitation forces are sufficient to overcome the frictional forces. So if the contact load at a brace is L, the frictional restraint will be $\mu L$ (where $\mu$ is the coefficient of friction) and this must be exceeded by the excitation force for rubbing to occur. It can be shown that the force required to produce a node in a pin at a brace position lies in the range 0.25-1 kg/g acceleration, depending on the mode of pin vibration. A mean value of 0.5 kg/g has been assumed here. For fuel pin movement excited by vibration of the fuel assembly, the applied acceleration in terms of g is given by

$$\alpha = \frac{4\pi^2 \, y_{max} f^2}{981} \qquad \text{eq. (5)}$$

where f and $y_{max}$ are again the frequency and amplitude of fuel assembly vibration. Hence the reaction force R required between the pin and the brace to prevent movement of the pin at the brace position is

$$R = 0.5 \; \frac{4\pi^2 \, y_{max} f^2}{981} \qquad \text{eq. (6)}$$

For sliding to occur this must be greater than the frictional force

$$\frac{2\pi^2 \, y_{max} \, f^2}{981} \geq \mu L \qquad \text{eq. (7)}$$

Thus the rubbing contact load L is defined by

$$L \leq \frac{2\pi^2 \, y_{max} f^2}{981 \, \mu} \qquad \text{eq. (7a)}$$

if rubbing motion occurs.

5.3 Vibration of Fuel Pins due to Fuel Assembly Impacting

If the fuel assembly impacts against the core, the impulse will be transmitted through the braces and grids to the fuel pins which will then deflect and undergo damped oscillations at their natural frequencies. The impulse will also be transmitted along the length of the fuel assembly and will excite vibration of the assembly in all its normal modes at their natural frequencies. It is possible that the higher harmonics of fuel assembly vibration will coincide with the natural frequencies of some fuel pins, thus coupling the pin response with that of the fuel assembly. The movements involved could contribute both to fatigue and wear damage. Although attempts have been made to calculate the response of the fuel assembly after impact and the consequent pin vibration, it has not been possible to refine these calculations or to specify the input data with sufficient accuracy to determine the conditions under which a problem would exist.

Consequently, the response of the fuel pins to impacting of the fuel assembly has been determined experimentally by mechanically oscillating a fuel assembly containing instrumented fuel pins and measuring the pin strains with the assembly oscillating freely and also impacting against the fuel channel. With free oscillation the fuel pins responded at the driving frequency with amplitudes similar to those predicted in Section 5.2 (see Fig. 4). Under impacting conditions the pins responded at the driven frequency and at higher harmonics of the driven frequency, and also at the fuel pin natural frequencies. The total rms pin strains were enhanced by 25-30% due to impacting.

5.4 Vibration of Fuel Pins due to Flow and Noise

The vibration of fuel pins in an instrumented fuel assembly in response to simulated reactor flow and noise has been determined experimentally (Bramwell [2]). The test pins include a range of fuel/can contact to cover the range of dynamic behaviour of fuel pins in-reactor, and will also have random restraints at the grid and brace positions within the range of manufacturing and assembly tolerances. However, the number of instrumented fuel pins which can be included in these tests is small compared to the total number of fuel pins in the reactor. This demands some attempt at evaluating the statistical significance of the experimental results.

Fig. 5 shows a histogram of the rms strain peaks measured in a typical vibration test. The results can be described by either a half-Gaussian or a Rayleigh distribution. Since both yield similar results in this analysis, only the Gaussian distribution will be considered further here. It is assumed that the distribution of fuel pin strains in the reactor and in the rig is the same and that both are Gaussian. On this basis it is possible to estimate the expectation of exceeding any given strain level. This is illustrated in Fig. 6 which shows the relative strain amplitude for which there is a 1 in 288 expectation of exceeding that strain (this means that on average one fuel pin per assembly will exceed this strain level). Similarly, the strain amplitudes for which there is a 1 in 2880, 1 in 28,800 and 1 in 100,000 expectation of being exceeded are as shown. These ratios correspond to 1 pin per 10 assemblies, 1 pin per 100 assemblies and 1 pin per 350 assemblies. Also for the rig case the strain amplitudes for which there is a 1 in 10 expectation of exceeding that strain (this means that on average 1 pin per 10 instrumented pins in the vibration test will exceed this strain level), a 1 in 20, 1 in 50 and 1 in 100 expectation of exceeding that strain level are shown. The relative amplitudes at which these expectations occur are defined by the probability distribution. Thus, the maximum strain expected to occur on a selected proportion of fuel pins in the reactor can be deduced from the maximum measured strain in the vibration test as a function of the number of instrumented fuel pins in the test, as shown in Fig. 7. This shows that if the number of instrumented pins is more than 20 then the maximum strain to be expected in-reactor should be approximately twice that found in the test. The precise value depends on the acceptance criterion selected.

Some further analysis is required to establish the pin motion in relation to wear. For simplicity, vibration of the pin in the first pinned-pinned mode only is considered. This covers the case of a fuel pin with pinned support at the bottom grid and top brace and rubbing at the centre brace.

It may be shown that, for a fuel pin vibrating at its natural frequency in the first pinned-pinned mode, the energy supplied per cycle to overcome damping is

$$U = \frac{m \ell \pi y_{max}^2 \omega_n^2}{2Q} \qquad \text{eq. (8)}$$

where $m$ is the mass per unit length of pin, $\ell$ is the pin length, $y_{max}$ is the vibration amplitude, $\omega_n$ is the natural frequency and Q is the magnification factor.

The shape of the vibrating pin is

$$y = y_{max} \sin \frac{\pi x}{\ell} \qquad \text{eq. (9)}$$

from which

$$\left[\frac{d^2 y}{dx^2}\right]_{max} = \left(\frac{\pi}{\ell}\right)^2 y_{max} \qquad \text{eq. (10)}$$

The maximum strain in the pin $\epsilon_{max}$ is given by

$$\epsilon_{max} = \frac{\sigma_{max}}{E} = h \left[\frac{d^2 y}{dx^2}\right]_{max} = \frac{h \pi^2 y_{max}}{\ell^2} \qquad \text{eq. (11)}$$

where $\sigma_{max}$ is the maximum pin stress and h is the distance from the neutral axis of the pin to the outer fibre.

Then $$y_{max} = \frac{\epsilon_{max} \ell^2}{h \pi^2} \qquad \text{eq. (12)}$$

and $$U = \frac{m \ell^5 \omega_n^2 \epsilon_{max}^2}{2Q\, h^2 \pi^3} \qquad \text{eq. (13)}$$

This expression defines the energy supplied to the pin per cycle to overcome damping, in terms of the maximum strain in the pin and the magnification factor Q.

If the vibrating pin rubs against a brace, the energy loss per cycle due to friction $U_F$ is

$$\begin{aligned} U_F &= 2 \times \text{friction force} \times \text{stroke length} \\ &= 4\, \mu L\, \bar{y} \end{aligned} \qquad \text{eq. (14)}$$

where $\bar{y}$ is the amplitude of the pin at the brace position.

For a pin rubbing against a brace, the limiting condition is given by assuming that all the input energy to the pin is dissipated in friction due to the rubbing process.

This gives

$$4\ \mu L\bar{y} = \frac{m\ell^5\omega_n^2\ \epsilon_{max}^2}{2Q\ h^2\pi^3} \qquad \text{eq. (15)}$$

Since for the first pinned-pinned mode $\bar{y} = y_{max}$, substitution of eq. (12) in eq. (15) gives

$$L = \frac{m\ell^3\omega_n^2\ \epsilon_{max}}{8Q\ h\ \pi\ \mu} \qquad \text{eq. (16)}$$

which defines the maximum contact load in the rubbing process.

## 6. Damage Models

### 6.1 Fuel Pin Cladding Fatigue

High frequency fatigue tests on AGR cladding material indicate the existence of a fatigue limit for narrow band Rayleigh random vibration at about 600 micro-strain rms (Sumner [3]). For AGR fuel pins this limiting value is reduced by the effects of temperature, strain concentration due to can ribbing and pellet interfaces, and irradiation. Allowance is also made for the fact that vibratory movements of fuel pins are not the sole source of in-service strain. These considerations provide a lower limit for the maximum acceptable strain due to fuel pin vibration in-reactor to ensure freedom from fatigue failure.

### 6.2 Fatigue of Grids and Braces

Vibration tests have been carried out to determine the endurance of CAGR grids and braces. In these tests, individual fuel elements were shaken at reactor temperature on a vibrating table to produce failure of the brace webs (Bryant and Jones [4]). These results have been extrapolated to the total number of cycles expected in-reactor to define the endurance limit for brace fatigue.

### 6.3 Impact Damage

Fuel pin vibration excited by impacting can be evaluated in terms of wear and fatigue damage. However, impacting of fuel elements against a stiff restraint can cause fracture of the graphite sleeves and significant distortion of the brace structure (Jones [5]). At low temperatures the graphite is the weaker component but at 650°C the brace becomes weaker giving a lower limit to the permissible approach velocity of impact. It is assumed that the same limits apply to fuel elements in a complete assembly impacting against the core.

### 6.4 Pin/Brace Wear

Pin/brace wear due to lateral vibration of CAGR fuel elements was first observed during multiple on-load charge/discharge trials on complete fuel assemblies, and subsequently in furnace vibration tests on single fuel elements. Fig. 8 illustrates the form of this wear with the pin ribs being partially worn through at the point at which they contact the brace and with corresponding wear grooves in the brace dimples. These were essentially accelerated tests aimed at establishing limits for fatigue damage so that they provided only very limited data on the wear process. No other information was available on this type of wear from which to estimate its growth in-reactor. However, pin/brace axial rubbing tests covering a wide range of loads and stroke lengths had been carried out in order to assess wear damage due to thermal cycling movements in-pile (Jones and Moore [6]). These tests were made up to $10^6$ cycles with rubbing distances down to 0.075 mm. The wear model developed for this process was adapted to the case of lateral vibration, and shown to be in reasonable agreement with the limited wear results from the furnace tests. This was then used as a basis for assessing pin/brace wear due to in-pile vibration. This procedure, which involves a number of assumptions, is discussed below.

The wear observed in the pin/brace axial rubbing tests was described by the equation

$$W = F\,(L-0.05)N^{\frac{1}{3}} \qquad \text{eq. (16)}$$

where W is the depth of wear in mm, F is a coefficient dependent on stroke length, temperature and material properties, L is the contact load in kg, the observed wear being very small below a threshold load of 0.05 kg, and N is the number of cycles. The process was characterised into four different regions dependent on the stroke length, the effects being described by the coefficient F (as shown in Fig. 9). For stroke lengths less than 0.25-0.40 mm, the movement was less than the area of contact so that debris particles could be trapped between the rubbing surfaces and enhance the wear rate. It has been assumed that over the small clearances available for pin movement at brace positions (0-0.4 mm), the rubbing process due to lateral vibration would be similar to this.

On this basis eq. (16) has been used to estimate the wear in the furnace vibration tests. The contact load at the braces in these tests was estimated in the manner described in Section 5.2, and appropriate values for the wear coefficient F were assigned from Fig. 9. Although only limited wear measurements had been made, these were in reasonable agreement with the estimated wear curves.

There was some evidence in the furnace tests of slight mushrooming of the pin ribs indicating possible wear due to hammering rather than rubbing. Also the limited number of measured wear points in these tests was insufficient to substantiate the $N^{\frac{1}{3}}$ rate of growth. However, the model has been used to estimate pin/brace wear due to fuel pin vibration and subsequent furnace tests have confirmed these predictions within a factor of 2 (see Fig. 10).

## 7. Evaluation of Fatigue, Wear and Impact Limits

The materials data used in specifying the damage models of Section 6 have largely been obtained using sinusoidal excitation, whereas the fuel vibration is random in nature. These data have been used to evaluate the limiting vibratory movements assuming the fuel vibration is sinusoidal at the maximum amplitude observed in the vibration tests, and is therefore pessimistic. Further development of the model to estimate the limiting conditions in terms of random vibration of the fuel is being pursued. However, this is dependent, not only on a more detailed analysis of the vibratory movements, but also on a fuller understanding of the damage processes involved.

The acceptance criteria for allowable vibratory motion of the fuel can now be evaluated by fitting the damage models in Section 6 to the relevant vibration models in Section 5. The results are shown in Figs 11 and 12. Fig. 11 gives the limiting fuel pin strain in relation to acceptance criteria for fatigue and wear due to fuel pin vibration. Fig. 12 gives the limiting conditions of fuel assembly vibration, in terms of frequency and amplitude, for acceptance criteria involving fatigue, wear and impact damage. The conditions defining the threshold for pin/brace wear are also shown in Fig. 12.

These assessments rely on very limited basic data and simple analytical models for which pessimistic assumptions have been made. The results can only be taken to indicate the main areas of uncertainty, rather than defining specific limits for fuel assembly and fuel pin vibration. It is also possible that other damage modes could occur which have not been considered here.

## 8. Conclusions

The analysis given in this paper demonstrates the development of a new approach in evaluating the vibratory motion of components within a complex fuel assembly by means of simple dynamical concepts. By drawing on component and materials fatigue and wear data from various sources it is possible to derive acceptance criteria for allowable vibratory motion of the fuel. This analysis has been applied to the case of a Civil AGR fuel design.

This approach involves a number of simplifying assumptions and makes use of limited basic data so that its value lies in indicating the main areas of risk, rather than in defining in absolute terms the limits for fuel vibration. Supporting experimental work has confirmed the general validity of this approach.

## 9. Acknowledgement

The permission of the Member for Reactors, United Kingdom Atomic Energy Authority, Mr R V Moore for publication is gratefully acknowledged and also the contribution from many people who have been involved in this work.

## References

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[2] BRAMWELL,D., "Dynamic behaviour of Mk 2 CAGR fuel pins in a flow and noise environment". Paper 213 presented at the Conference on Vibration Problems in Industry, Keswick. April 1973.

[3] SUMNER, G., Unpublished work.

[4] BRYANT, R L., JONES, P M., UKAEA Internal Document.

[5] JONES, P M., "Transverse impact of graphite sleeved fuel elements - the vibratory response and damage criterion". Paper D4/6 to the 2nd SMIRT Conference, Berlin. September 1973.

[6] JONES, P M., MOORE, B., UKAEA Internal Document.

FIG.1 SECTIONAL VIEW OF Mk II CIVIL ADVANCED GAS COOLED REACTOR FUEL ELEMENT

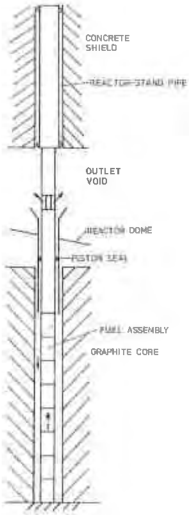


FIG 2 SCHEMATIC ARRANGEMENT OF CIVIL AGR FUEL ASSEMBLY AND CHANNEL

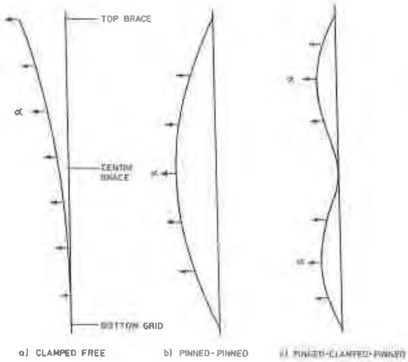


FIG 3 PIN VIBRATION MODES RESULTING FROM OSCILLATORY INERTIAL LOADING DUE TO FUEL ASSEMBLY VIBRATION

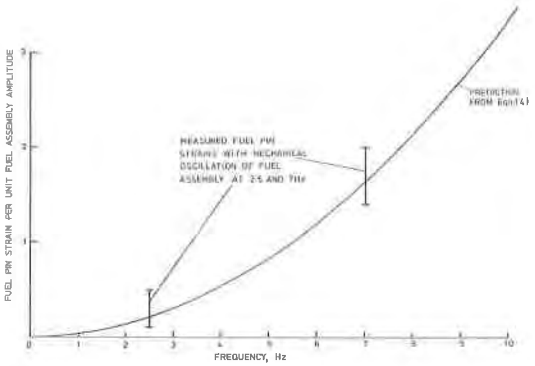


FIG 4 COMPARISON OF PREDICTED AND MEASURED PIN STRAINS DUE TO FUEL ASSEMBLY VIBRATION

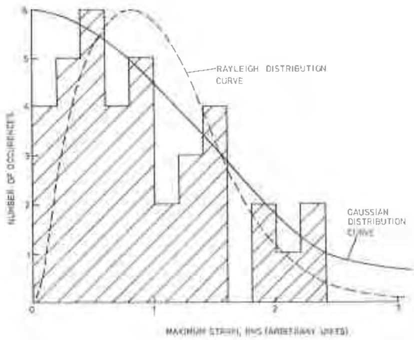


FIG 5 HISTOGRAM OF MAXIMUM R M S FUEL PIN STRAINS OBSERVED IN CAGR FUEL VIBRATION TEST DUE TO FLOW AND NOISE

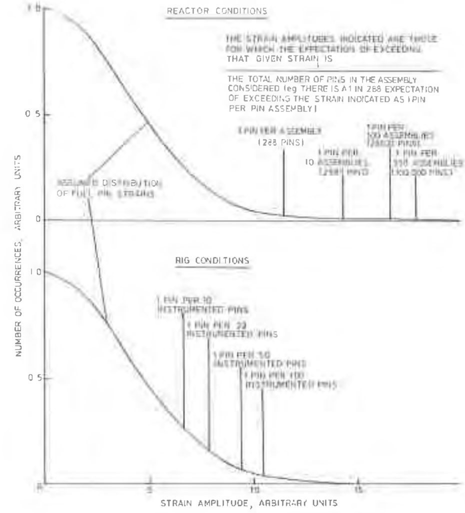


FIG 6 COMPARISON OF EXPECTATION OF MAXIMUM FUEL PIN STRAINS IN THE REACTOR WITH THOSE OBSERVED ON INSTRUMENTED FUEL PINS IN VIBRATION TESTS

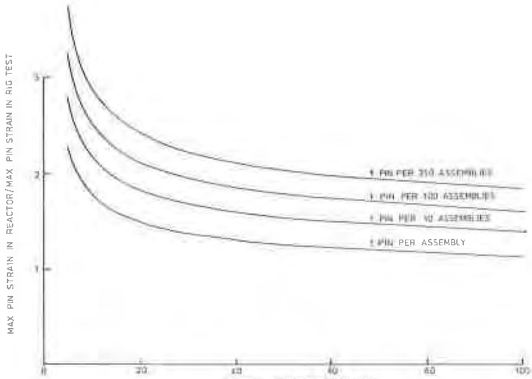


FIG 7 STATISTICAL ASSESSMENT OF IN-REACTOR FUEL PIN STRAINS RELATIVE TO MEASURED STRAINS IN OUT-OF-REACTOR RIG TESTS


FIG 8 Example of brace and pin wear

D 3/

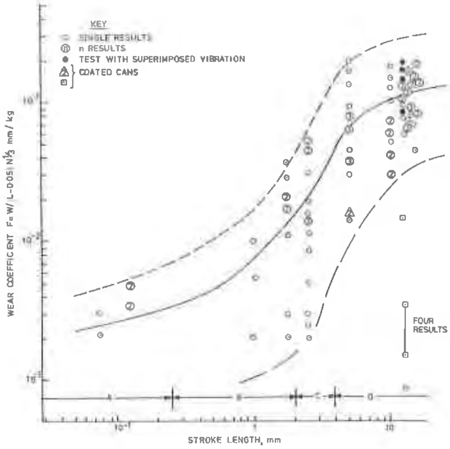


FIG 9. VARIATION OF PIN/BRACE WEAR WITH STROKE LENGTH

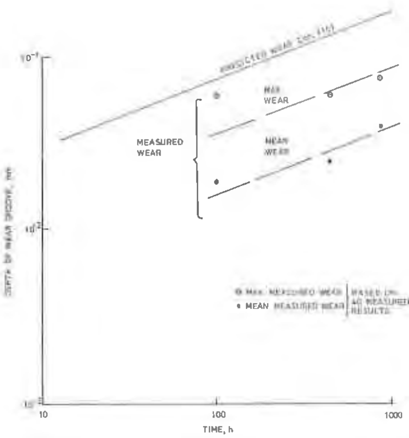


FIG 10. COMPARISON OF PREDICTED AND MEASURED PIN/BRACE WEAR IN FUEL ELEMENT VIBRATION TESTS

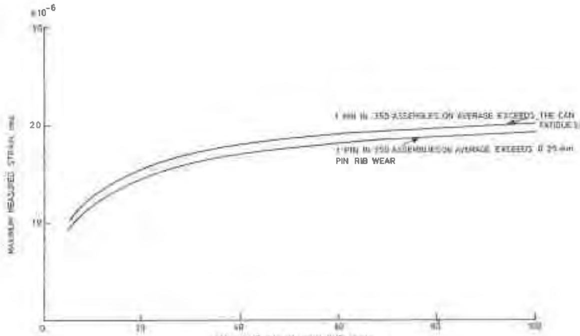


FIG 11 ESTIMATED FATIGUE AND WEAR CRITERIA FOR FUEL PIN STRAIN DUE TO FLOW AND NOISE

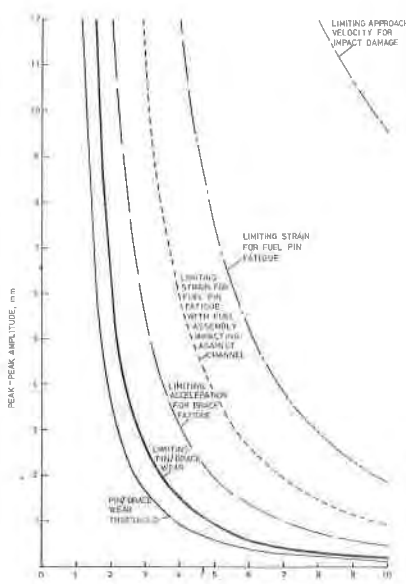


FIG 12 ESTIMATED AMPLITUDE AND FREQUENCY OF FUEL ASSEMBLY VIBRATION FOR VARIOUS LIMITING CRITERIA OF FUEL ELEMENT ENDURANCE