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〈**Technical Paper**〉

Fretting Wear of Fuel Rods

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핵 연료봉의 마모

이 해

초 록

경수로용 핵 연료봉의 동특성과 Archard의 이론을 근거로 마모와 관련되는 기본설계 파라미터 8가지를 규정 제시하였다. 또한 활주임계치와 비접촉임계의 산출방법을 예시했다.

1. Introduction

Fuel rod wear due to flow-induced vibration is a continuing concern in the design of liquid-cooled reactors. Mechanical wear at the contact regions between fuel rods grid dimples and grid springs has been observed in commercial nuclear power plants. In some cases, severe irrepariable damages were reported.

Eventhough the wear problem was discussed by I. Newton as long ago as 1704, our present knowledge of wear is mainly based on research work done during the last twenty to thirty years. Most of the reported work gives experimental data obtained by rubbing one material surface against the other. Very little work is reported on the theoretical aspects. Precise prediction of wear appears to be not possible. However, an estimate of wear to within an order of magnitude is possible. For most combination of materials laboratory test data show appreciable scatter and have to be presented in a statistical form.

Archard presented theory for adhesive wear[2] and general wear theory was reported by Baylor.[4,5] In this paper Archard's theory was used for fretting wear estimate of a fuel rod. Wear producing parameters were identified, and the expected amount of wear occuring at the rod-grid contact regions for any given rod vibration amplitude and spring preload force was presented. Wear analysis for a fuel rod with time-dependent spring preload force was also made.

2. Archard Theory[2] of Adhesive Wear

Wear can be classified either as severe wear or as mild wear depending on the size of the particles removed from the mating surfaces. When the mean diameter of the wear particles is of the order of 10^{-4}m or greater, the wear is called severe, otherwise it is called

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mild[1]. The true area of contact between two surfaces is the sum of all the flattened areas of contact of the touching asperities. The mean diameter of the individual contact areas is of the order of 10^{-4}m. Therefore, we can interpret mild wear as occurring when the damage is confined to the true area of contact and severe wear as occurring when the damage covers the entire contact area. Another way of classifying wear is based on the mechanisms of wear. Rabinowitz[2] uses adhesive, abrasive, corrosive, and surface fatigue wears. In reality, wear is caused by one or a combination of these forms of wear. One example is fretting, which is caused by small amplitude oscillatory motion of the rubbing surfaces. Due to the motion, adhesive wear occurs first. The loose particles formed corrode in time and they in turn produce larger and harder particles. Abrasive wear then follows.

Archard has derived an adhesive-wear equation for unlubricated sliding based on the following assumptions.

1. The worn volume V produced in sliding a distance L is proportional to the true area of contact.
2. The true area of contact is formed by local plastic deformation.
3. Particles removed by the sliding motion are hemispherical and of the same diameter.

The wear equation is

$$V=S\frac{F_w L}{3H} \tag{1}$$

where the wear coefficient S is a proportionality constant, H is the hardness of the softer member of the sliding pair, and F_w is the normal force on the contacting surfaces.

The number 3 in the denominator is the shape factor that applies to hemispherical fragments. This number changes if we assume a different shape for the removed fragment. For example, the shape factor is equal to unity for a cubical fragment. The hardness H is measured by a Brinell or Vickers hardness tester. When Eq. 1 is converted to give the wear volume in cubic inches, we have

$$V=\frac{SF_w L}{4260H} \tag{2}$$

where L is the total sliding distance in inches.

3. Fuel Rod Wear Analysis

Prototype fuel rods are supported along their length by seven grid assemblies. Each grid assembly consists of a 17×17 array of grid cells. Each cell has two springs and four dimples. A typical cell and its mathematical model are shown in Fig. 1.

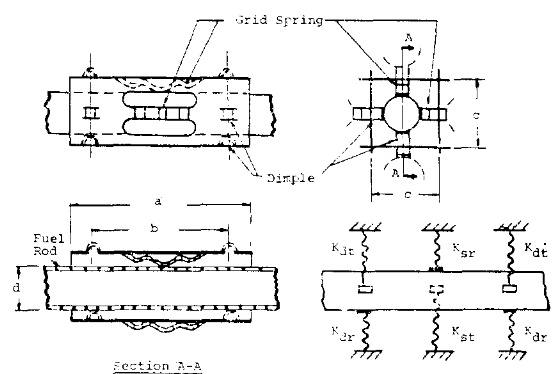

Fig. 1 Grid cell and its mathematical model.

In the normal operating condition of a fuel assembly, the wear producing motion between fuel rod and grid occurs 1) at all grid-rod contact points due to the shortening effect when the rod span deflects, and 2) at the side dimples when the maximum vibration amplitude is greater than the slippage threshold amplitude y_{sn}. Dynamic characteristics of prototype fuel rod are documented in Ref. 7.

3.1. Shortening Effects

For a conservative analysis of the shortening length during a vibration, we will assume simple-supported boundary conditions. Then the shortening Δl of a typical span length l is given by

$$\Delta l = \frac{1}{4}\left(y_1 \frac{\pi}{l}\right)^2 \cdot l \qquad (3)$$

For the prototype six-span fuel rod, we substitute $l = 0.665$m and obtain

$$\Delta l = 3.710 \times y_1^2 \qquad (4)$$

when the maximum single-peak vibration amplitute is $y_1 = 1 \times 10^{-4}$m, then $\Delta l = 3.71 \times 10^{-8}$m. This is much less than the slippage at the side dimples as can be seen in the following section. Therefore, we will neglect shortening effects in our future wear analysis.

3.2. Sliding Distance at Grid Dimples

Relative motion between the fuel rod and side dimples initiates when the tangential force at a side dimple exceeds the friction force. The maximum single-peak vibration amplitude at which slippage starts, the slippage thres hold amplitude(Fig. 2), is given by

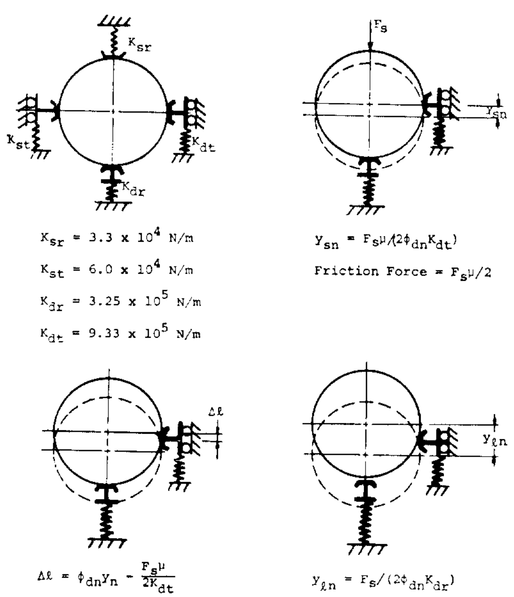

Fig. 2 Schematics of grid cell dynamics.

$$y_{sn}\phi_{dn}K_{dt} = \frac{F_s}{2} \cdot \mu$$

or

$$y_{sn} = \frac{F_s\mu}{2\phi_{dn}K_{dt}} \quad \text{for any mode } n \qquad (5)$$

where y_{sn} is slippage threshold amplitude of the nth mode, ϕ_{dn} is the maximum modal displacement of the rod at any dimple location, K_{dt} is the tangential stiffness of the side dimples, μ is the friction coefficient, and F_s is the spring preload force. For example, the slippage threshold amplitude for the six-span fuel rod vibrating in its fundamental mode is

$$\begin{aligned} y_{s1} &= \frac{F_s\mu}{2\phi_{d1}K_{dt}} \\ &= \frac{F_s\mu}{2 \times 3.145 \times 10^{-2} \times 9.33 \times 10^5} \\ &= 1.704 \times 10^{-5} F_s\mu \qquad (6) \end{aligned}$$

In this calculation the maximum value of the modal displacement of the dimple is $\phi_{d1} = 3.145 \times 10^{-2}$, which occurs at mass point 42, grid No.5.[7]

Once slippage initiates at the side dimples, the natural frequency, mode shapes, and system damping of the rod will change. The modal damping due to the rubbing between the fuel rod and dimples becomes amplitude-dependent and the natural frequencies decrease due to the reduced effective tangential dimple stiffness. As the vibration amplitude increases further, lift-off occurs and severe damage may result. When the alternating radial reaction force acting at the dimples is greater than one-half of the spring preload force, lift-off will occur. The vibration amplitude at which lift-off initiates, the lift-off threshold amplitude, is given by

$$y_{ln} \cdot \phi_{dn} \cdot K_{dr} = F_s/2$$

or

$$y_{ln} = \frac{F_s}{2\phi_{dn}K_{dr}} \quad \text{for any mode } n \qquad (7)$$

where y_{ln} is the lift-off threshold amplitude of the nth mode, and K_{dr} is the radial stiffness of the dimple. The lift-off threshold amplitude of the prototype fuel rod vibrating in its fundamental mode $n=1$ is $y_{l1}=F_s/(2\times3.145\times10^{-2}\times5.25\times10^5)=3.028\times10^{-5}F_s$m. For $F_s=27$ N, $y_{l1}=8.18\times10^{-4}$m. The maximum radial forces and the locations at which they occur are given in Table 1 for the prototype six-span fuel rods.

Table 1 Location and magnitude of the maximum alternating radial and tangential dimple forces. vibration ammplitude$=10^{-5}$m single-peak.

Mode number	Dimple location		Dimple displacement	Radial force	Tangential force
n	Grid Number	Mass* Point	y_{dn},m Single Peak	$y_{dn}K_{dr}$, N.	$y_{dn}K_{dt}$, N.
1	5	42	3.145×10^{-7}	.165	.293
2	3	22	3.271×10^{-7}	.172	.305
3	2	14	2.528×10^{-7}	.133	.236
4	2	14	1.890×10^{-7}	.992	.176
5	1	4	2.876×10^{-7}	.151	.268
6	6	54	4.330×10^{-7}	.227	.404
7	5	42	7.440×10^{-7}	.391	.694

※See Reference 7.

The ratio between the lift-off and slippage threshold amplitudes is

$$\frac{y_{ln}}{y_{sn}}=\left(\frac{K_{dt}}{K_{dr}}\right)\frac{1}{\mu}\cong6 \quad \text{for any mode } n \text{ when } \mu=0.3 \qquad (8)$$

It should be noted that lift-off occurs when a vibration amplitude is 6 times greater than the slippage threshold amplitude for the friction coefficient $\mu=0.3$. For a larger friction coefficient, impacting occurs at lower amplitude (Fig. 2).

The impact sliding may cause a serious damage to the grid and severe wear to the fuel rod. There is no theory available for impact sliding wear at present. In the following fuel-rod wear analysis, we will assume that the dynamic characteristics of the system do not change due to slippage, and that the vibration amplitudes are such that no impacting occurs at the grid support. Then the total wear producing motion L at a side dimple during operation time t is

$$L=4f_nt\left(\phi_{dn}y_n-\frac{F_s\mu}{2K_{dt}}\right) \qquad (9)$$

where the subscripts n denotes the mode number, f_n is the rod vibration frequency, ϕ_{dn} is the maximum normalized modal displacement of the rod at the dimple location, y_n is the maximum modal vibration amplitude of the rod, $F_s\mu/2$ is the friction force acting on the dimple, and K_{dt} is the tangential stiffness of the dimple.

3.3. Wear Analysis

A formula for fuel rod wear is obtained by substituting Eq. 9 and into Eq. 1.

$$V=2f_ntSF_s(\phi_{dn}y_n-F_s\mu/2K_{dt})/3H \qquad (10)$$

The numerical values for a prototype six-span fuel rod vibrating in its fundamental mode for 1000 hours are:

$f_1=29.3$ Hz

$\phi_{d1}=3.145\times10^{-2}$ (Ref.7)

$K_{dt}=9.33\times10^5$ N/m (Ref.7)

$t=3.6\times10^6$sec.

$H=2450$ MN/m^2

$S=2\times10^{-5}$

Substitution of above numerical values into Eq. 10 results

$$V=F_s\ (1.8054\times10^4y_1-.3076F_s\mu)\times10^{-12}\text{m}^3 \qquad (11)$$

During the operating life of fuel rods, the spring preload force F_s changes considerably due to thermal relaxation and other environmental factors. It decreases exponentially with time from initial spring preload force of 27 N to almost zero at the end of life. The friction coefficient for Zircaloy/Inconel combination varies greatly due to environmental changes. Controlled laboratory tests show it varies between .3 and .4.

The amount of wear for a fuel rod with time-dependent spring preload force is given by

$$V=\frac{S}{3H}\int_t F\frac{dL}{dt}\cdot dt \qquad (12)$$

where the integral is to be evaluated over the time intervals of interest. When the variations of F, and of dL/dt with time are known, the integral can be evaluated easily for any given vibration amplitude. It should be noted that L is a function of F as well as a function of t, as shown in Eq. 9.

As an illustration we will now calculate the amount of wear for the prototype fuel rod vibrating in its fundamental mode.

The numerical values that will be used are
vibration amplitude $=2\times10^{-4}$m single-peak
friction coefficient $\mu=.3$
operation time $t=24,000$ hours (13)
spring preload force $F_s=26$ N initially and it decreases 4 N stepwise after each 6000 hours of operation

We will substitute Eq. 13, $F_s=4j+10$ and $\mu=.3$ into Eq. 11. and sum for the total numbers of step changes to find the accumulative wear V_1 for 1000 hours at each load step. For the required amount of wear V we simply multiply V_1 by 6; resulting

$$V=6V_1=\sum_{j=1}^{4}\{6(4j+10)3.611-.0923\times(4j+10)\}\times10^{-12}\text{m}^3$$
$$=8\times10^{-10}\text{m}^3 \quad \text{for } \mu=.3 \qquad (14)$$

The wear depth h can be calculated from the shape of the worn volume. If we assume that the worn volume has the shape of a cylindrical segment of length $=1.27\times10^{-3}$m (dimple width), and diameter $D=1.072\times10^{-2}$m, then

$$h=(V/1.753\times10^{-4})^{2/3}=2.75\times10^{-4}\text{m} \qquad (15)$$

4. Conclusions

When Archard's wear theory is used, the mechanical wear rate of a vibrating fuel rod depends on the spring preload force, natural frequency, vibration amplitude, tangential dimple stiffness, and friction coefficient in addition to the operation time t, Brinell Hardness H, and wear coefficient S. Lift-off at a dimple occurs when the alternating force in the radial direction is greater than one-half of the spring preload force F_s. The minimum vibration amplitude at which lift-off occur for $n=1$ mode is 8.18×10^{-4}m.

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