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FRETTING WEAR OF HEAT EXCHANGER TUBES

by
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ABSTRACT

PART I - EXPERIMENTS

The results of a series of measurements made on the fretting wear of heat exchanger tubes and support plates at room temperature in a nitrogen/air atmosphere are presented. The fretting wear is shown to be a function of the amplitude and frequency of tube vibration as well as the gap between the tube and the support plate and the mean load supported by the tube. An empirical model is developed in Part II for predicting the fretting wear.

PART II - MODELS

Conceptual and empirical models are developed for the fretting wear of heat exchanger tubes. The models are based on the experimental data of Part I of this series and on the concept that fretting wear is the result of relative motion between the tube and the support plate.

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PART I - EXPERIMENTS

INTRODUCTION

Many heat exchangers are fabricated from tubes which convey fluid between tubesheets and are supported at regular intervals by passing through oversized holes in supporting plates, A fluid flows over the tubes and exchanges heat with the fluid in the tubes. This flow can induce tube vibration [1].[1] As the tubes vibrate against the support plates, material can be fretting away from both the tube and the support plate. If the fretting wear is sufficiently severe, the tubes will eventually wear through and the heat exchanger will leak. Vibration-induced fretting wear is second only to corrosion as a cause of heat exchanger failures [2].

It is reasonable to assume that the fretting wear of a vibrating heat exchanger tube due to transverse tube vibration is a function of the following parameters:

1. Amplitude of tube vibration.
2. Mode shape of tube vibration.
3. Frequency of tube vibration.
4. Geometry of tube (length, diameter, wall thickness, and boundary conditions),
5. Geometry of support plates (thickness, hole diameter, spacing between plates).
6. Mean reaction (preload) between tube and support plate.
7. Tube material (alloy, modulus of elasticity, hardness, surface finish or coating).

[1]Numbers in brackets denote References at the end of paper,

8. Plate material (modulus of elasticity, hardness, surface finish, or coating).

9. Temperature

10. Environment (surrounding atmosphere).

Additional parameters and interaction between parameters may be important as well. It is not generally known what effect these parameters have on fretting wear and in particular if change in one of these variables can be exploited to produce a heat exchanger design that is substantially free of fretting wear. Therefore, an experimental study was made to explore the effect of variables 1 through 8 on the fretting wear induced by a vibrating heat exchanger tube.

DESCRIPTION OF THE TEST APPARATUS

The basic requirements of the test apparatus are that (1) it closely simulates the interaction of the tube and the support plate, of interest, and (2) must permit accurate measurements of very small amounts of fretting wear so that significant test results can be obtained in a short period of time. The test rig used in this test is shown in Fig. 1. The fretting wear is produced between test specimens (a) held to the tube, and (b) held to the plate. Since these specimens each weigh no more than 50 grams, they can be weighed to an accuracy of ± 0.00002 gram using conventional balances. This permits small amounts of fretting wear to be measured by weighing the specimens before and after a test and subtracting to obtain the weight loss due to wear on the tube or on the support plate. Since the test rig is symmetric about the midspan of the tube, each test results in wear on two identical sets of tube and plate specimens. The

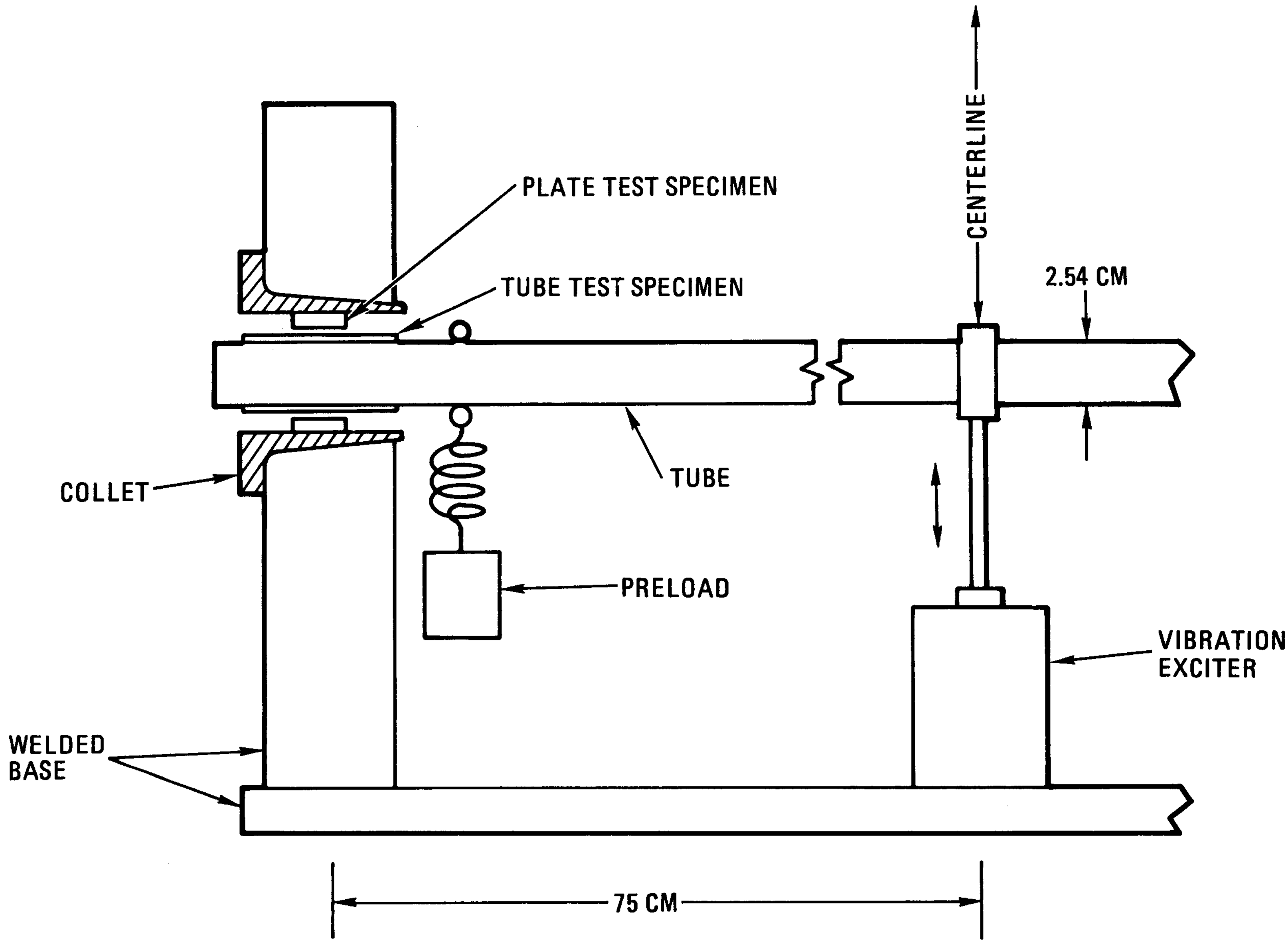

Fig. 1. Test apparatus. The apparatus is symmetric about the center line of the shaker

difference in wear rates measured at each end of the tube gives an estimate of the experimental scatter.

The shaker provides a unidirectional sinusoidal excitation to the tube at midspan. However, it was observed that the ends of the tube move in oval orbits as the ends of the tube impact and rebound off the support plates. Ko [3] has found that the fretting wear of a tube is not substantially reduced as the excitation approaches unidirectional motion.

The tube amplitude was measured as peak-to-peak amplitude at midspan of the tube using an optical technique. It was found that an accelerometer at midspan did not provide an accurate measure of tube amplitude. Shock waves traveling along the tube greatly distorted the accelerometer signal. These shock waves were produced by impacting of the tube and the support plate.

TEST PROCEDURE

Tests were performed in the following sequence:

1. Tube and plate test specimen materials were chosen and the test specimens (2 each for tube and plate) were machined.
2. Specimens were ultrasonically cleaned in acetone and alcohol baths.
3. Specimens were weighed on a precision balance to an accuracy of ± 0.00002 gram.
4. Test specimens were installed in the test rig and vibration initiated at desired amplitude, and frequency. Preload, if any, was applied to the ends of the tube.

5. Tests were stopped after continuous running for a number of hours or days.
6. Specimens recleaned and reweighed.
7. The weight loss of the specimens was determined and normalized by dividing by the total number of cycles of vibration during the test.

Typically, the test parameters were chosen for each test so that only a single parameter was varied from those in the following base test.

1. Amplitude of vibration (peak-to-peak at midspan), $A_{p-p}/D=0.033$ (D = tube diameter = 2.54 cm).
2. Frequency of vibration, f = 50 Hz.
3. Preload, P = 0
4. Gap on diameter between tube and hole in plate, G/D = 0.035.
5. Plate thickness, $t_p/D = 0.6$.
6. Tube wall thickness, $t_T/D = 0.120$.
7. Tube specimen material: 410 Stainless Steel; hardness $R_B = 87$.
8. Plate specimen material: 2-1/4 Cr, 1 Mo hardness, $R_c = 30$.

Typically, tests were run for 24 hours; however, some tests where wear rates were very low were run for more than 72 hours. Other tests which produced very severe wear rates were stopped after 8 hours. All tests were conducted at room temperature in a nitrogen/air mixture which was employed to minimize the effect of atmospheric moisture.

While the weighing technique produced an accuracy in measurement as high as ± 0.00002 gram, or $\pm 4.6 \times 10^{-12}$ gram/cycle for a 24 hour, 50 Hz test, the scatter between measurements made on identical specimens at each end of the tube suggests that unavoidable scatter in the tests produces experimental uncertainty as high as $\pm 15\%$ with the scatter increasing as the wear rate decreases. This scatter was ameliorated somewhat by averaging the sample data over the two test specimens at each end of the test tube.

APPEARANCE OF THE TEST SPECIMEN

The general appearance of a tube test specimen after severe fretting wear (8 hours at 50 Hz and an amplitude of 0.1 tube diameters) is shown in Fig. 2. In this case, the tube support plate was 0.2 inch (0.51 cm) in thickness. The test specimens after other tests had a similar appearance, but in general the wear groove was much shallower. In all cases the wear pattern was that of a smooth textured groove whose depth varied around the tube circumference and had the general appearance of being cut with a sand blaster. All the wear tests resulted in a fine powdered debris in the vicinity of the test specimens which ranged in size from very fine powder to particles as large as 1 mm across.

Fig. 3 is an electron microphotograph of the surface of an Alloy 800 H (Ni-Fe-Cr) tube test specimen after a wear test. The surface flakes on the specimen are apparently the result of local work hardening due to impacting between the tube and the plate which delaminates the work hardened material. The wear process of material removal seems to be

Fig. 2. Tube test specimen after a severe fretting wear test

Fig. 3. Surface of an Alloy 800H tube test specimen after a wear test

confined to a very thin surface layer which is less than 0.01 mm (0.0001 inch) thick. Fig. 4 is an electron microphotograph of the powdered debris collected after a test using Alloy 800H test specimens. Note the thin flake appearance. Test specimens of other alloys produced similar debris but that produced by Alloy 800 resulted in particularly large thin, flat flakes.

RESULTS

The test results can be grouped according to the effect of various parameters on the wear rate.

Material

The effect of the material and its hardness on the wear rate is summarized in Tables 1 and 2.

Table 1 suggests that the wear at the plate and at the tube are generally comparable if the tube and plate are the same material and that differences in wear between the wear rate at the tube and the plate resulted more from a change of materials than location of the test specimen. It was generally found during the tests that tube and plate wear rates were nearly equal when the same materials were employed for both the tube and the plate. If different materials were used then the ratio between the tube and plate wear varied between 1/3 and 3 (2/3 to 4/3 for 2-1/4 Cr-1 Mo and Alloy 800H, Table 1). This ratio seemed much more dependent on the materials chosen than on the tube amplitude or frequency of vibration or other mechanical variables. The effect of material hardness on plate wear

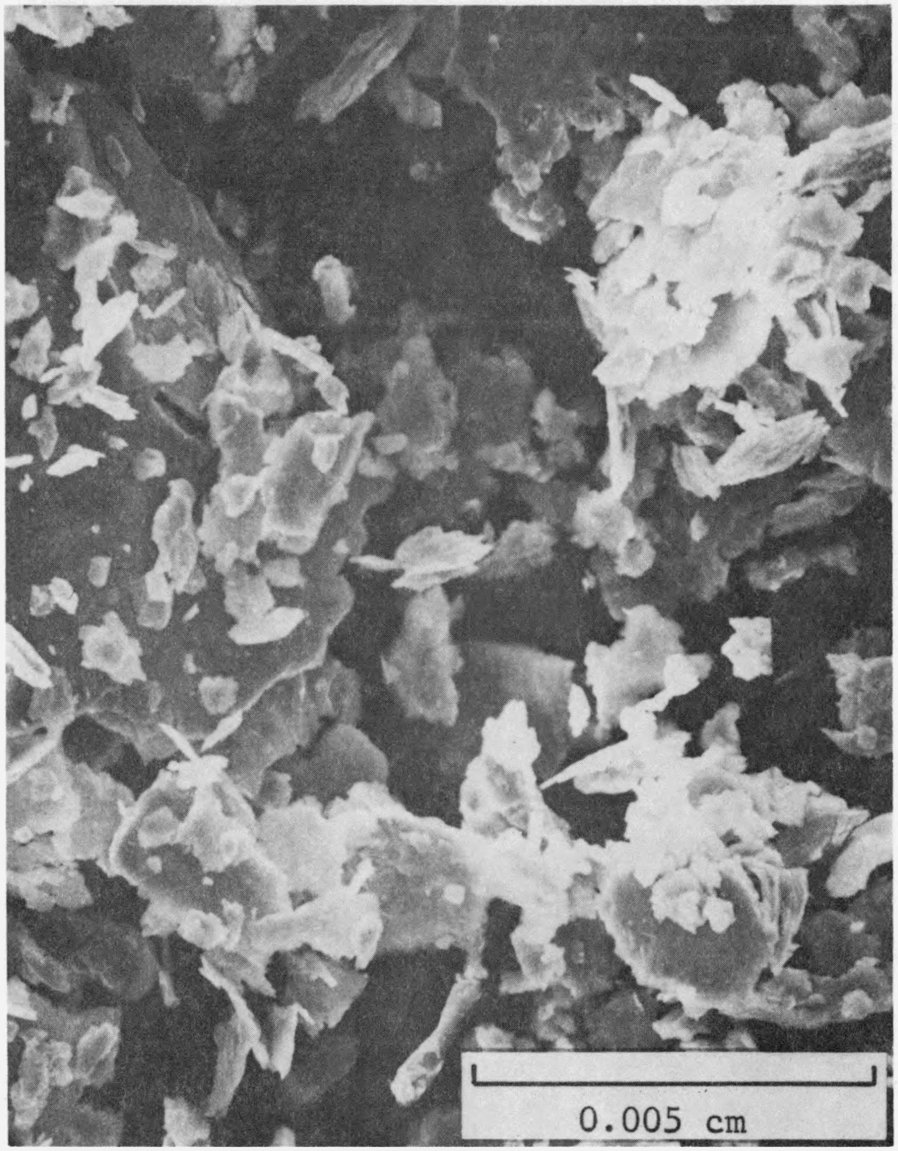

Fig. 4. Debris produced by fretting wear between Alloy 800H test specimens

can be seen in Table 2. While wear generally decreases with increasing hardness (compare 2-1/4 Cr-1 Mo in the annealed, tempered, and hardened conditions) certain materials such as 304 and 410 stainless steel had wear rates much below that which would be expected from their hardness alone.

Duration of the Test

The rate of fretting wear at the plate as a function of time is shown in Fig. 5. The data for this figure was obtained as follows: A single set of samples (2 tube specimens and 2 plate specimens) were machined and tested for 24 hours. The test parameters were identical to those of the base test case. The wear rate per cycle at the plate during this test gave the data point for day 1. The samples were then cleaned, reweighed, and reinstalled in the test rig in the same orientation as before, and the test was repeated. The wear per cycle of this second test gave the data point for day 2, and so on. It can be seen from Fig. 5 that the rate of fretting wear is accelerating with time. During these tests, the specimens accumulated significant amounts of fretting wear. At the start of the tests, the gap between the tube and the support plate was 0.035 tube diameters (2.54 cm). By the day 10 test, the gap had increased to a maximum of 0.057 tube diameters.

Gap Between Tube and Support

Fig. 6 shows the effect of increasing the gap between the tube outer diameter and the inner diameter of the hole in the support plate on the wear rate at the plate. The gap was progressively increased by

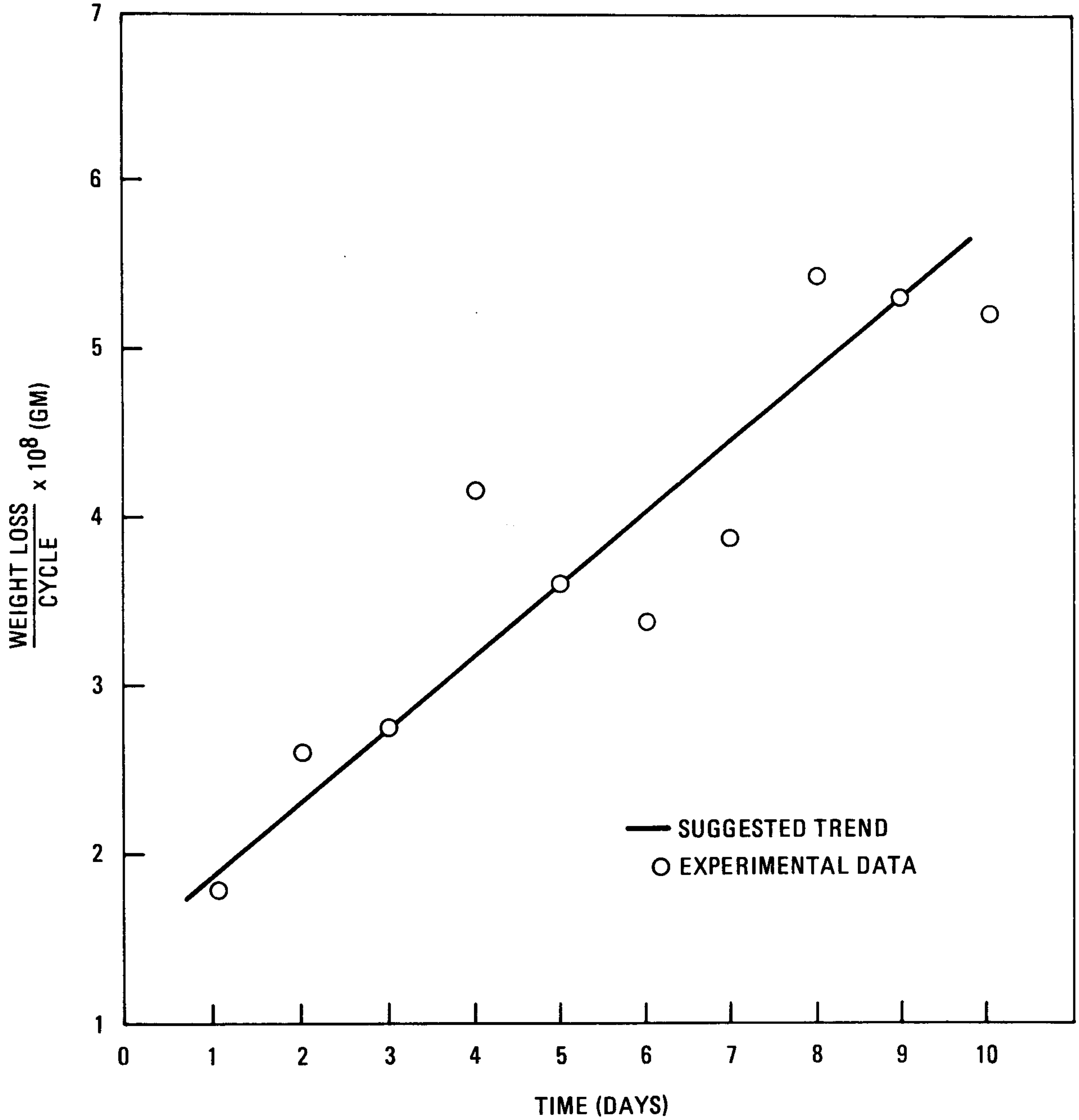

Fig. 5. Rate of fretting wear as a function of time. All parameters conform to base test case.

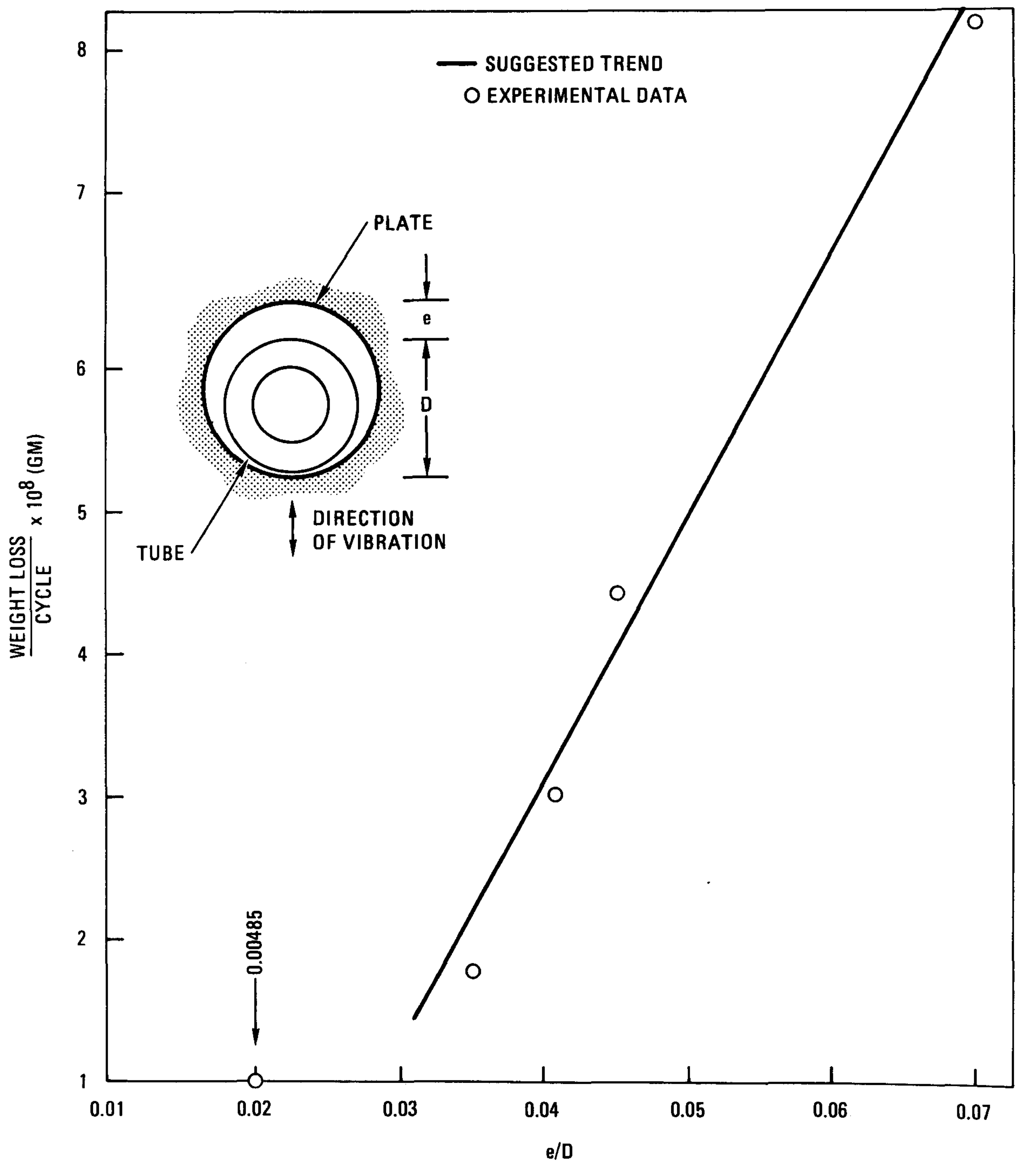

Fig. 6. Rate of fretting wear as a function of the gap between the tube and the support plate. All parameters except gap conform to the base test case.

machining plate specimens with various inner diameters. The gaps in Fig. 6 refer to the gaps at the start of the tests. By the end of the 24 hour tests, these gaps had increased slightly. All other test parameters conform to the base test. Note the sharp increase in wear rate with increasing gap in Fig. 6. This implies that by minimizing the gap between the tube and the support plate, one can minimize fretting wear. Welding the tube to the support plate will reduce the gap and fretting wear to zero, but this can also result in high thermal stresses and reduce tube damping to the point that the tubes become very susceptible to flow-induced vibration.

It is reasonable to hypothesize that the rate of wear increase with time, shown in Fig. 5, is due to the increase in gap between the tube and the plate as material is worn away. For example, by the day 10 test in Fig. 5 the gap had increased from the initial 0.035 tube diameters to a maximum of 0.057 tube diameters. Fig. 6 suggests that a wear rate of 6.2×10^{-8} gm/cycle is appropriate for a 0.052 gap. This is in reasonable agreement with the observed wear rate of 5.2×10^{-8} gm/cycle observed at day 10 in Fig. 5. The discrepancy is most likely due to the unevenness of the wear pattern worn by testing in Fig. 5 as compared with the uniform diametrical gaps of Fig. 6.

Amplitude and Frequency of Tube Vibration

Figs. 7 and 8 show the effect of amplitude and frequency of vibration on fretting wear at the plate. All the remaining test parameters corresponded to those in the base test case. The increase in the fretting wear

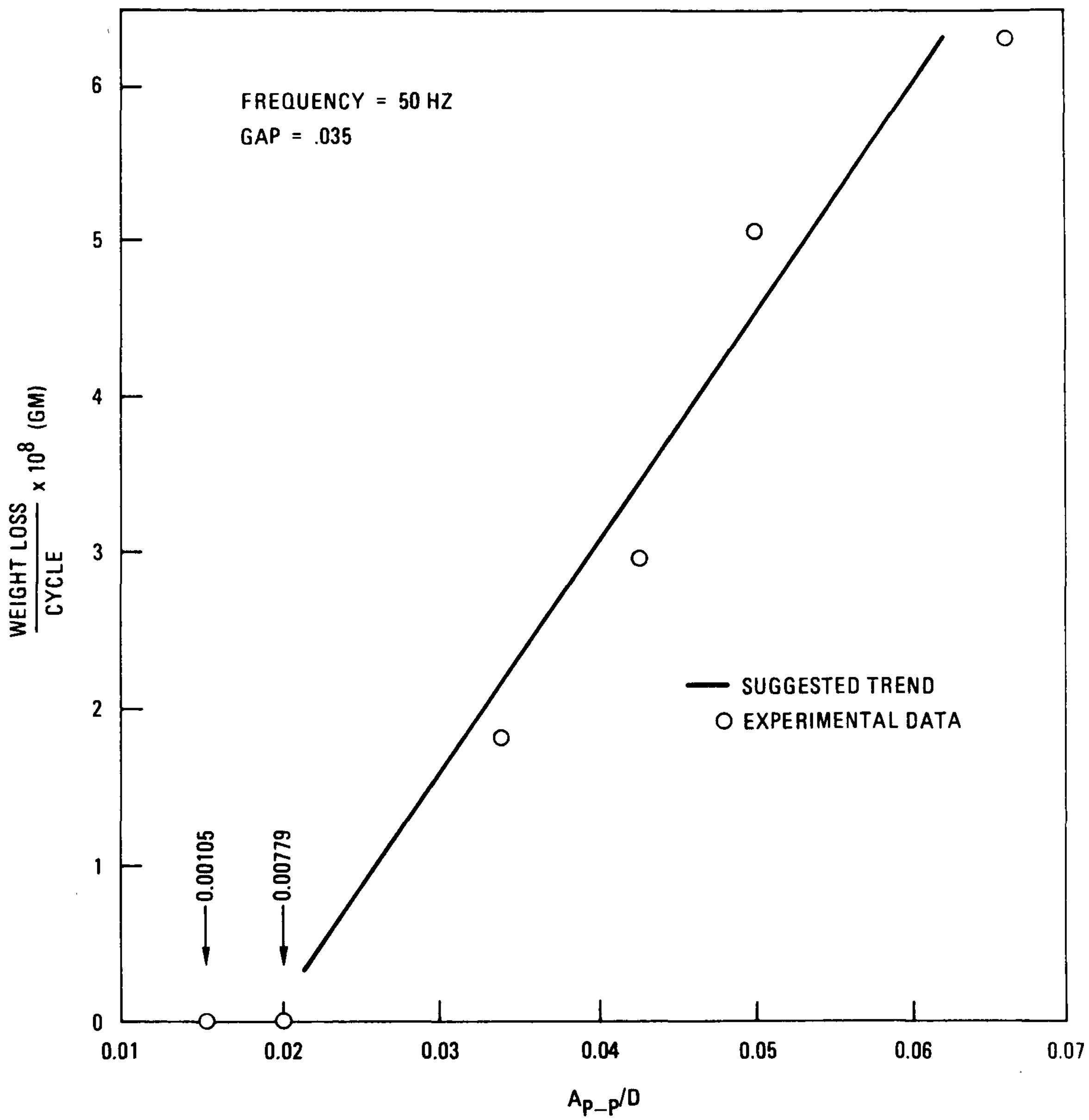

Fig. 7. Rate of fretting wear as a function of the peak-to-peak tube amplitude (Ap-p) measured at midspan. D = tube diameter = 2.54 cm. All parameters except amplitude conform to base test case.

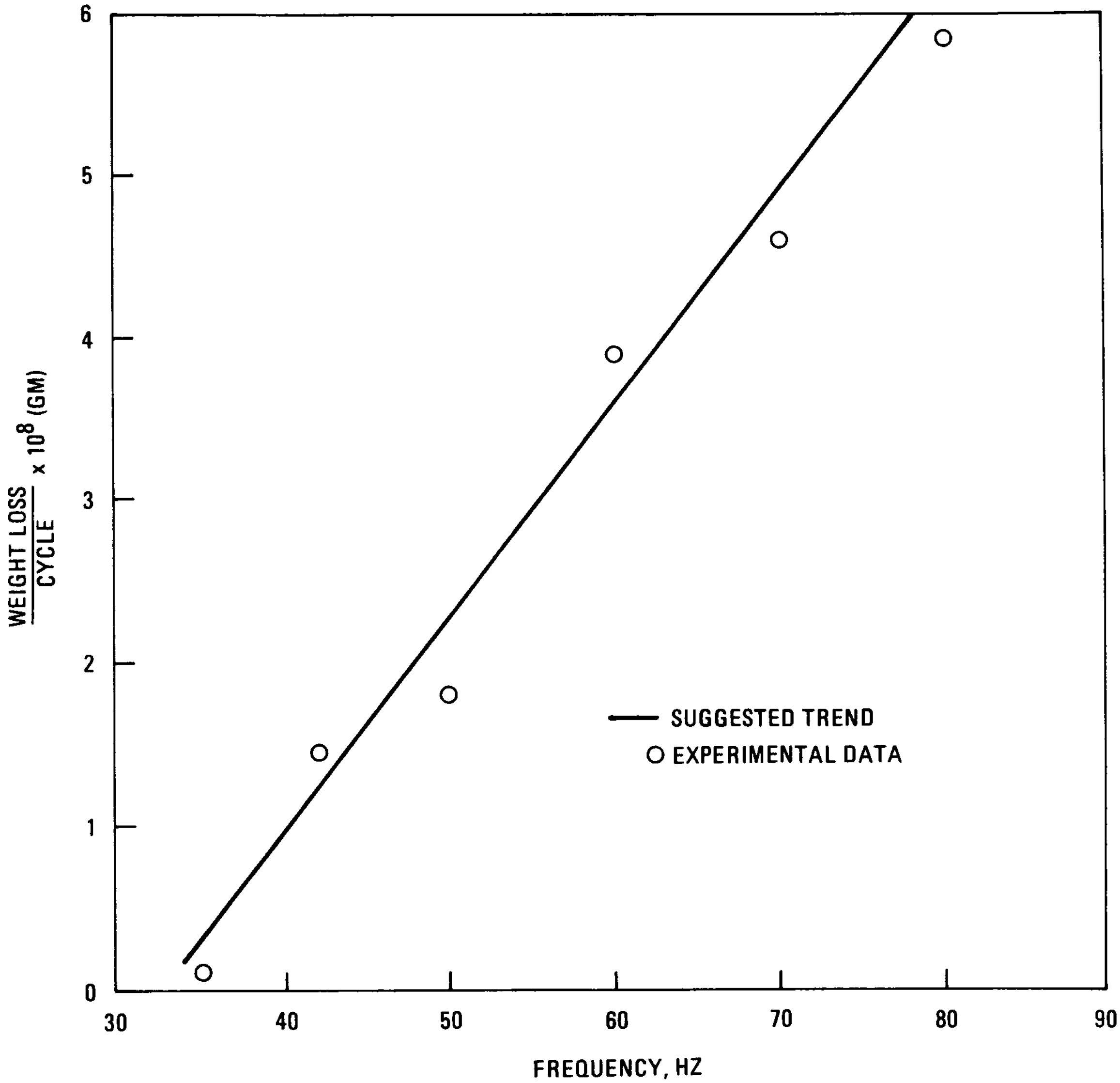

Fig. 8. Rate of fretting wear as a function of the frequency of tube vibration. All parameters except frequency conform to base test case.

per cycle with increasing amplitude or frequency can probably be attributed to the increase in the kinetic energy of the tube with amplitude and frequency of vibration.

Preload

Fig. 9 shows the effect of preloading the tube on the fretting wear at the plate. The preloads were applied by suspending two equal weights from the tubes near the tube/support plate junction by means of an extensible tie to reduce shock (Fig. 1). A zero preload corresponds to the tube support plates just supporting the weight of the tube. Note that increasing preload sharply reduces fretting wear once the preload exceeds a critical preload. This critical preload increases with the amplitude of tube vibration. The preload effect is apparently due to the decrease in tube motion at the support plates with increasing preload.

Other Variables

Increasing wall thickness was found to increase the wear slightly. A factor of two change in tube wall produced about a 15% change in the wear rate. This change in the wear rate is comparable to the experimental uncertainty. Increasing tube wall results in both an increase in the stiffness and the weight (preload) of the tube. These effects are apparently self-cancelling to a degree.

Roughening the surface of the test specimens had little effect on fretting wear. When one set of plate test specimens was roughened by machining grooves 5×10^{-4} cm (2×10^{-4} in) deep, the wear rates which

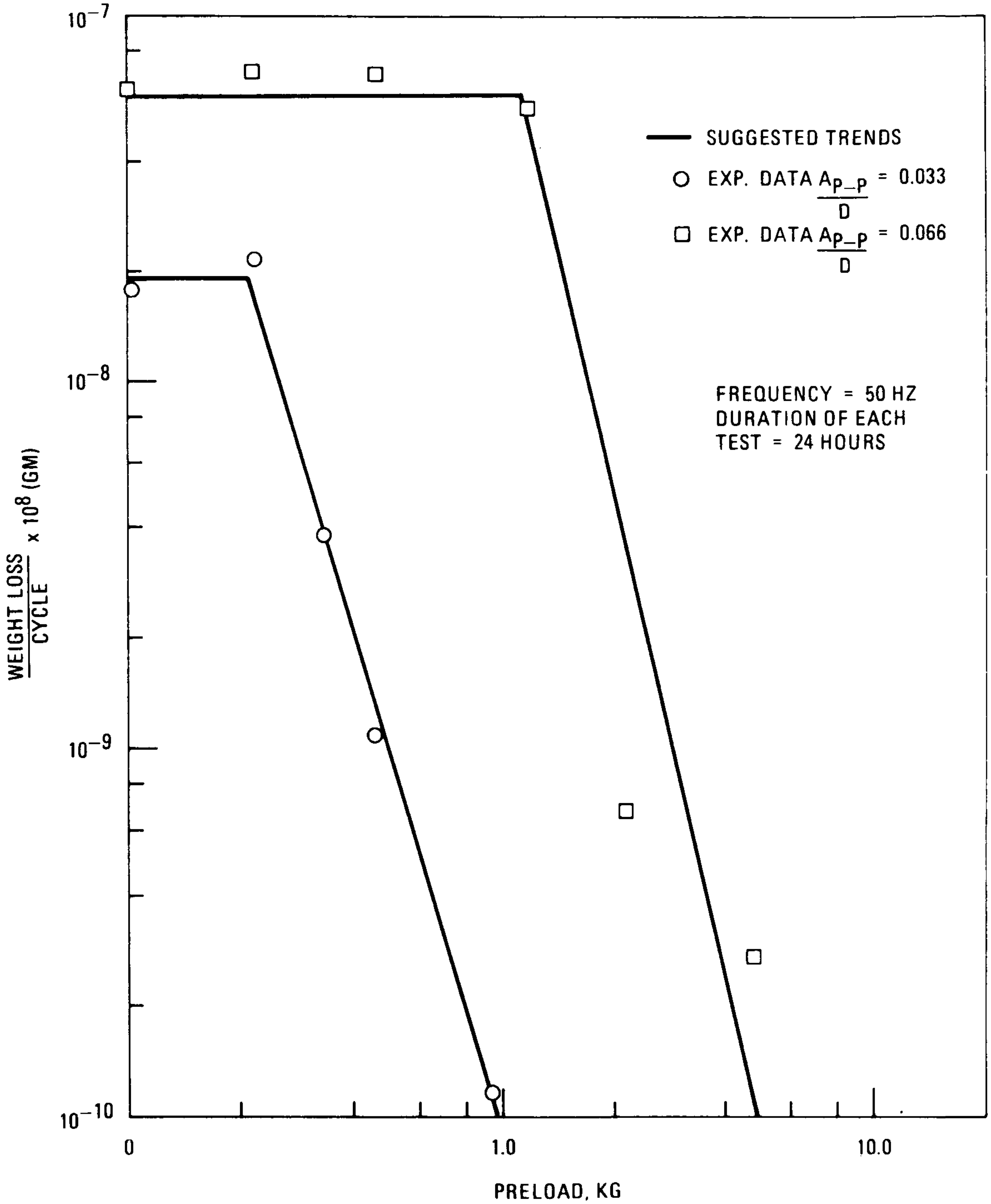

Fig. 9. Rate of fretting wear as a function of preload on the tube. All parameters, except preload and amplitude for the Ap-p/D=0.066 case.

resulted were virtually identical to those produced by the nominal, unroughened specimens. The test parameters were identical to that in the base test case.

Decreasing the thickness of the tube support plate was found to increase the fretting wear. When the thickness of the tube support plate was decreased by a factor of three from that in the base case (0.60 in, 1.52 cm), the rate of fretting wear at the plate increased by a factor of 1.87. When the thickness was increased by a factor of three, the fretting wear at the plate decreased by a factor of 69. The rate of fretting wear at the tube showed similar but less pronounced changes with variation in the support plate thickness.

During one test, conforming to the base test, powered graphite was injected every two hours into the tube-plate interface. It was found that the resultant wear rates were a factor of 30 below those observed without the graphite. However, it is not known whether the reduction of wear is produced by the lubricating action of the graphite or the ability of the graphite particles to fill the gap between the tube and the support plate.

Although no attempt was made to quantitatively measure acoustic noise, it became obvious during the course of the tests that the chattering sound radiated by the test rig was at least roughly correlated with the wear rate. The sound emitted during a test increased with the amplitude and frequency of vibration, as did the wear rate. The sound decreased sharply when preloads were applied to the ends of the tube, as did the wear rate. In fact, if a test did not produce an audible rattling sound, then the wear rate was found to be very small. This implies that the wear rate is directly tied to the local motion and resultant impacting of the ends of the tube with the support plate.

CONCLUSIONS

A series of tests have been made on the fretting wear of a heat exchanger tube vibrating between supporting plates. All tests were performed in a nitrogen/air atmosphere at room temperature with the fretting wear measured by the weight loss techniques. The principal results of the tests were:

1. The fretting/impacting wear produced by this test was the result of surface delamination which was confined to a very thin layer on the surface of the wearing specimens.
2. The wear rate (weight loss per cycle of vibration) of the tube and support plate are approximately equal if both the tube and the support plate are made of the same material.
3. The wear rate is a function of the material and its hardness. It was found that increasing the hardness of 2-1/4 Cr-1 Mo specimens significantly decreased wear. However, other alloys of comparable hardness could produce greater or lesser wear rates.
4. Increasing the gap between the tube and the support plate was found to sharply increase the fretting wear. One result of this is that the rate of fretting wear (weight loss per cycle) will increase with time since as material is worn away, the gap between the tube and the support plate will increase.

5. The rate of fretting wear increases with both the frequency and amplitude of tube vibration. The amplitude was measured at midspan.

6. The rate of fretting wear decreases sharply with increasing preload applied to the tube at the tube/support plate junction. Preloads as small as 1 kilogram resulted in order of magnitude decreases in fretting wear. This effect is apparently the result of decreased tube motion at the support plate with increasing preload.

7. The rate of fretting wear is positively correlated with the noise produced by the tube. If the tube was heard to rattle strongly during a test, a high wear rate was obtained. If no rattling was detected, the wear rate was very small.

In Part II, these data are incorporated in a predictive model for fretting wear.

TABLE 1

Tube Material	Plate Material	Tube Wear (gm/cycle x 10^8)	Plate Wear (gm/cycle x 10^8)
Alloy 800H	Alloy 800H	4.09	4.11
	2-1/4 Cr-1 Mo	2.72	2.14
2-1/4 Cr-1 Mo	Alloy 800H	2.60	3.95
	2-1/4 Cr-1 Mo	2.14	2.00

TABLE 1. Tube wear and plate wear for various materials. The amplitude, frequency, and preload of the tests are the same as those of the base test. Tube wall thickness = 0.188 in (0.477 cm).

TABLE 2

Plate Material	Hardness R_B	R_C	$\frac{\text{Weight-Loss}}{\text{Cycle}} \times 10^8$ (GM)
Alloy 800H (Ni-Fe-Cr)	80	-	3.81
2-1/4 Cr-1 Mo Annealed	87	-	3.72
304 Stainless Steel	89	-	0.709
2-1/4 Cr-1 Mo Quenched & Tempered	-	29	1.77
Mild Steel	-	32	1.59
2-1/4 Cr-1 Mo Hardened	-	40	1.64

TABLE 2. Effect of alloy and hardness on the rate of fretting wear of the plate. R_B = Rockwell B scale, R_C = Rockwell C scale. All other test parameters conform to the base test case. Hardness increases from top of Table to bottom.

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PART II - MODELS

INTRODUCTION

Fretting wear is produced by relative motion at the interface between two structures. The classical form of the equation for the fretting wear is that the material loss due to wear is proportional to the area of contact times the distance of travel [1].[1] The proportionality constant is a function of the materials involved, the environment, the normal load applied at the surfaces and other variables. This wear model has proved very valuable in evaluating wear due to sliding. Unfortunately, the model does not lend itself to the fretting wear produced by vibrating heat exchanger tubes because the motion of a heat exchanger tube in a support plate is very complex and is not accurately modeled by simple sliding [2]. The vibrating tube can repeatedly make and break contact with the support plate and there can be considerable impacting as well as sliding. These considerations make it very difficult to characterize the motion of a vibrating tube in a support plate in a deterministic sense or even to obtain meaningful statistical measures of the tube and support plate interaction.

In this paper, conceptual and empirical models are developed for the fretting wear induced by a transverse vibration of a heat exchanger tube. The models are designed to bypass the difficulties of the tube and support plate interaction by relating the fretting wear directly to the major mechanical parameters describing the overall dynamics of the tube and support plate system. These parameters, such as tube amplitude and frequency of vibration,

[1]Numbers in brackets denote References at the end of paper.

materials and mean load on the tube can be obtained from the system description and an analytical or experimental vibration analysis. Thus, the models presented here are designed to make a first step to closing the loop by making it possible to predict the fretting wear as a function of the mechanical parameters which an engineer or analyst can reasonably hope to obtain.

CONCEPTUAL MODEL

The fundamental assumption of the models which will be presented in this paper is that fretting wear is the result of relative motion between a tube and a support plate (Fig. 1 of Part I of this series). The relative motion between the tube and support plate can be considered to be the sum of longitudinal tube motion, tube motion parallel to the tube axis, and transverse tube motion, motion perpendicular to the tube axis, at the support plate.

The longitudinal tube motion can be approximately analyzed by considering the longitudinal motion induced by the transverse vibration of a pinned-pinned tube in the fundamental mode. The transverse displacement of the pinned-pinned tube is:

$$\mathcal{Y}(x,t) = A \sin \frac{\pi x}{L} \sin\omega t \qquad (1)$$

$$0<x<L$$

$\mathcal{Y}(x,t)$ is the transverse deformation which is a function of the longitudinal parameter, x, and time, t. L is the span of the tube, ω is circular frequency of the vibration and A is the midspan amplitude of the tube displacement. If the neutral axis of the tube is not stressed axially during vibration, then the longitudinal contraction at each end of the tube during vibration is:

$$\Delta_L = \frac{1}{2}\int_0^L \left[1 + \left(\frac{\partial Y}{\partial x}\right)^2\right]^{1/2} dx \qquad - \frac{L}{2}$$

$$\approx \frac{\pi^2}{8}\ \frac{A^2}{L}\ \sin^2 \omega t \quad \text{for } A \ll L. \qquad (2)$$

The longitudinal deformation is second order in the tube amplitude, A, and thus is neglected in linear vibration theory. The sum of the absolute values of the longitudinal deformations during each quarter cycle is the total longitudinal travel of the tube.

$$\sum |\Delta_L| = \frac{N}{2}\frac{\pi^2 A^2}{L} \qquad (3)$$

where N is the total number of cycles of vibration. For the base test case described in Part I of this series,

A = 0.042 cm (half peak-to-peak amplitude).

L = 150 cm

N = 4.32×10^6 cycles (24 hours at 50 Hz)

thus

$$\sum |\Delta_L| = 250 \text{ cm}$$

in the 24 hour test period. This is a significant amount of travel, however, it will be shown in the following paragraph that the travel due to transverse vibration can be far greater.

Consider the transverse vibration of a single span beam whose ends are supported by passing through oversized holes in support plates. If the vibration amplitude is sufficiently large, then the ends of the tube will transverse the gap (as shown in Fig. 1 of Part I in this series) during vibration because the beam mode will approach that of a free-free beam. During one cycle of vibration the tube has the potential for crossing the gap between the tube and the support plate twice. Thus the total maximum transverse travel of the tube across the gap during N cycles of vibration is:

$$\Delta_T = 2NG \tag{4}$$

where G is the diametral gap between the tube and the support plate. For the base test case

$$G = 0.089 \text{ cm}$$

$$N = 4.32 \times 10^6 \text{ cycles}$$

thus

$$\Delta_T = 768000 \text{ cm}$$

which is over 3000 times as large as the longitudinal travel. Moreover, the transverse motion is associated with large exchanges of kinetic energy as the tube impacts with the support plate two times per cycle of vibration. Thus it is reasonable to believe that if the amplitude of vibration is large enough to permit the tube to transverse the gap at the support plates, then this transverse motion will dominate the fretting wear. More simply stated, unless the tube rattles in the support plate, the fretting wear is expected to be small.

The ends of a single span tube can vibrate transversely in the gap between the tube and the support plate during vibration only if the mean load applied to the tube ends is less than the shear load required to hold the ends of the tube stationary. The shear load required to prevent transverse vibration of the ends of a tube is easily found from elementary beam theory:

$$R_s = EI\frac{\partial^3 Y}{\partial x^3}\bigg|_{x=0,L} \tag{5}$$

E is the modulus of elasticity and I is the area moment of inertia of the cross section. For a pinned-pinned tube span in the fundamental mode, this shear reaction is simply:

$$R_s = EI\frac{\pi^3}{L^3}\ A \sin \omega t$$

The shear reaction to maintain the stationary ends of a clamped-clamped single span tube is 3.35 times greater than that for a pinned tube for the same midspan amplitude and a moment must be supplied as well to prevent rotation of the ends of a clamped-clamped tube.

For example, for the base test case of Part I of this series, the required shear load to maintain the ends of the tube stationary is

$$|R_s| = \begin{cases} 1.07 \text{ kg} & \text{pinned-pinned} \\ 3.58 \text{ kg} & \text{clamped-clamped} \end{cases}$$

The weight of the tube supported by the support plate at each end of the tube was

$$W = 1.28 \text{ kg},$$

less a small amount of weight supported by the shaker at midspan. Thus the shear load required to maintain the ends of the tube stationary is comparable or less than the available load due to the weight of the tube. As a result, it would be expected that the ends of the tube would move during vibration. This motion was observed. Moreover, it would be expected that if preloads on the order of 2 kilograms were applied to the ends of the tube, then the weight of the tube plus the preload would be sufficient to prevent significant transverse motion of the ends of the tube and the fretting wear would be greatly reduced. This was also observed as noted in Part I of this series.

These concepts can be applied to a multispan tube such as shown in Fig. 1. Fig. 1a shows a stationary tube with clamped-clamped ends and a support at midspan with a gap. If sufficient load is applied at midspan to enforce a node, then the tube can vibrate either in the second mode of a clamped-clamped tube (Fig. 1b) or in a mode corresponding to half-span clamped-clamped tubes (Fig. 1c) or in a higher mode with a node at midspan. In the antisymmetric mode shown in Fig. 1b, no shear load is required at the midspan support to maintain the node at midspan if the tube is perfectly symmetric about midspan. If the tube is slightly asymmetric, then some shear load will be required to hold the tube midspan stationary. However, the required shear load is apt to be small and it is unlikely that significant transverse motion will occur at midspan in this mode. However, the mode shown in Fig. 1c will require a substantial shear load at midspan to maintain the tube midspan stationary since the mode is symmetric about the midspan. If the tube begins to vibrate slightly in this mode, then unless sufficient load is maintained on the tube by the midspan support, the tube midspan will begin to vibrate transversely with the mode shape shown in Fig. 1d. This mode corresponds to the fundamental mode of a clamped-clamped tube and has a natural frequency substantially below that of the modes of Figs. 1b and 1c. Thus, if the tube can overcome the load applied at the midspan, it can vibrate in a mode which has substantially lower frequency than other modes and which will be strongly encouraged by flow-induced vibration. The mode of Fig. 1c will also produce large transverse motion of the tube in the support plate and substantial fretting wear.

The conclusion that can be drawn from the conceptual model is that of the various modes available for vibration of a heat exchanger tube, those modes which have the potential for transverse motion at the support plates are most likely to result in fretting wear.

EMPIRICAL MODEL

An empirical model for fretting wear has been developed for the fretting wear based on the experimental data of Part I in this series and the conclusions drawn from the conceptual model. The model accounts for the variation in fretting wear due to transverse vibration with the following parameters.

1. Frequency of tube vibration.
2. Amplitude of tube vibration.
3. Gap between the tube and the support plate.
4. Tube wall thickness.
5. Mean load on the tube.

The model is designed to predict the fretting wear at the support plate for a 2-1/4 Cr-1 Mo, R_c=29 material. As noted in Part I, the wear at

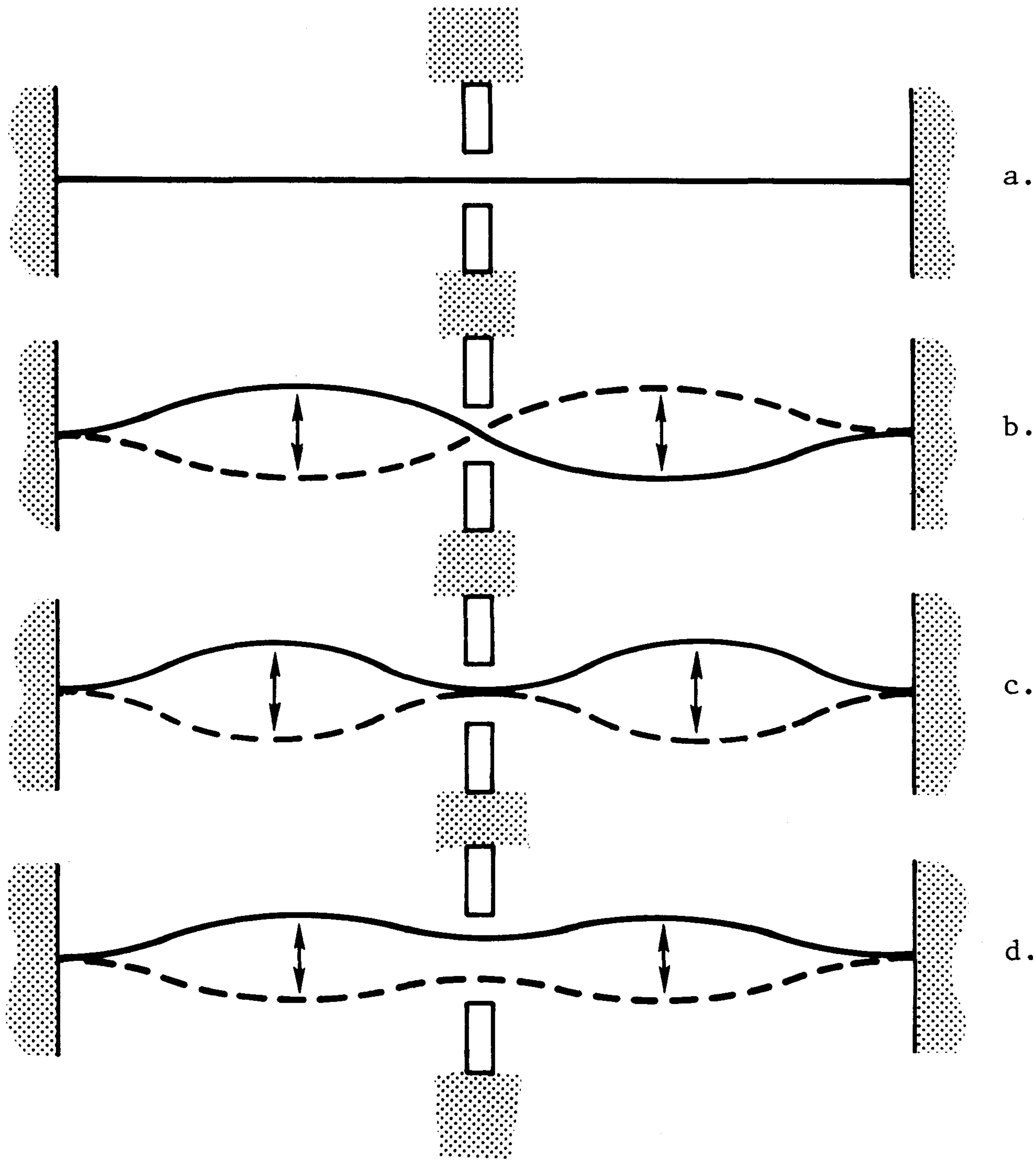

Fig. 1. Vibration modes of a clamped-clamped tube with a midspan support. (a) equilibrium, (b) antisymmetric mode (2nd mode of clamped-clamped tube), (c) half span clamped-clamped symmetric mode, (d) symmetric mode with transverse motion at midspan.

the tube is expected to be nearly identical to the wear at the plate if both are made of the same material. If different materials are employed, then adjustments can be made using the data presented in Part I of this series as are discussed below.

The following form was chosen for an empirical fit to the experimental data presented in Part I of this series:

$$w = \alpha_1 f^{\alpha_2} \left(\frac{A_c}{D}\right)^{\alpha_3} \left(\frac{A_G}{D}\right)^{\alpha_4} \left(\frac{t}{D}\right)^{\alpha_5} e^{\alpha_6 (R_S - W - P_L)} \tag{6}$$

A_c = Peak-to-Peak transverse amplitude of the tube which would be developed at the support if the gap dimensions were very large and no load was applied to the tube at the support.

A_G = The smaller of either A_c or the diametral gap at the support plate (the support plate hole inside diameter less the tube outside diameter).

D = Tube diameter

f = Frequency of vibration, Hz.

t = Tube wall thickness.

R_S = Amplitude of the minimum shear load required to maintain the tube stationary, (i.e., reduce A_c to zero) at the support plate when vibrating in a mode with characteristic model amplitude A_c, kg.

w = Weight loss in grams per cycle of vibration.

W = Mean weight of tube supported at the support plate, kg. ($W = 0$ if tube is vertical)

P_L = Preload applied to the tube at the support plate, kg.

α_1 thru α_6 = Constants

The form of Eq. 6 was based on the following assumptions: (1) The tube wear depends not only on the individual parameters listed in the introduction to Part I but also on their interrelationship. For example, if the amplitude of tube vibration approaches zero, then the wear must approach zero regardless of the gap size or frequency. Thus, the form of an empirical wear law must be multiplicative rather than additive. (2) The wear must approach zero as the gap, amplitude and frequency approach zero. (3) The wear rate will be greatest when the preload, P_L, is zero. Moreover, the effect of preload should be scaled relative to the preload required to hold the end of the tube stationary.

A_c is used to characterize the potential for transverse vibration at the support plate. For the single span tube of Part I, A_c was determined from the mode shape of a free-free tube whose midspan amplitude is known. For example, if the midspan amplitude was 0.033 diameter, then the amplitude of the ends of the tube in the fundamental free-free mode must be 0.109 diameters.

A_G is used to characterize the maximum transverse amplitude which can occur at the support plate in the absence of loads on the tube. Clearly this amplitude can never exceed the dimensions of the gap.

The shear reaction R_s is used to compare the shear load available to suppress transverse vibration at the gap, $W+P_L$, with the shear load required to suppress the vibration. In the present case, R_s was estimated as the shear load required to maintain the beam in a pinned-pinned mode with the same midspan amplitude as that which resulted in A_c.

Using standard least squares curve fitting techniques, the following constants were obtained in fitting the experimental data of Part I.

$\alpha_1 = 6.30 \times 10^{-5}$ grams-sec$^{5.61}$/cycle

$\alpha_2 = 5.61$

$\alpha_3 = 1.74$

$\alpha_4 = 6.68$

$\alpha_5 = 1.74$/kg

$\alpha_6 = 1.80$/kg

This curve fit is compared with the experimental data in Fig. 2. While significant scatter is present, the curve fit provides substantial correlation of the data over nearly four orders of magnitude. While the parameters α_1 - α_6 are based on experimentally measured wear at the support plate for a 2-1/4 Cr-1 Mo, R_c = 29 plate, it is reasonable to assume that the parameters α_2 through α_6 are nearly independent of the plate material. α_1 can be adjusted for other materials using the data presented in Tables 1 and 2 of Part 1. Similarily since it has been noted in Part 1 that the tube tube and plate wear at approximately the same rate if both are of the same material, Eq. 6 can also be applied to predict tube wear for a 2-1/4 Cr-1 Mo tube. As for plates, adjustments in α_1 for different materials can be made using Tables 1 and 2 of Part 1.

Two effects are notable in the curve fit. First, reducing vibration amplitude at the support plate results in a large decrease in the fretting wear. The tube amplitude can be reduced either by (1) reducing overall tube vibration, or (2) reducing the gap between the tube and the support plate. For example, if one reduces the gap between the tube and the support plate by a factor of 2, perhaps by placing a device between the tube and the support plate and no other variables are changed, then the fretting wear could be reduced by a factor of $2^{6.68}$ or 103. Second, increasing preload on the tube can also substantially reduce wear. If a tube which is loaded just enough so that $R_S=P_L+W$ is then loaded an additional 2 kilograms, it would be expected that the fretting wear would be reduced by a factor of $e^{3.6}$ or 36.6.

The principal difficulty in applying Eq. 6 to multispan tubes comes in determining the characteristic tube amplitude at the gap, A_c. If the gaps are small then one tends to assume a pinned joint model for the plate and as a result that the tube amplitude at the gap is assumed to be zero, $A_c = 0$. The Experience of Part I and the conceptual model of Fig. 1 show that this is incorrect. In fact, if a multispan tube with small gaps is vibrated, then the tube can be heard to rattle strongly at certain vibration amplitudes implying that the tube does indeed move in the gap and a pinned point is an inappropriate model for the plate as far as wear is concerned (although the pinned joint model may provide an appropriate model for estimating the natural frequencies). A very conservative approach for estimating the characteristic amplitude at the gap in a plate of a multispan tube is to assume that the plate of interest does not exist, determine a mode shape and tube amplitude at the now missing plate, and use this amplitude in Eq. 6. Since the tube amplitude will be over estimated by this approach, the wear estimate should be substantially conservative. A second, simplier, estimate of A_c is to assume that $A_c = A_g$, the diametral gap. This will be conservative provided the amplitude of tube vibration is less than the diametrical gap but, of course, it does not allow one to take credit for a low level of vibration.

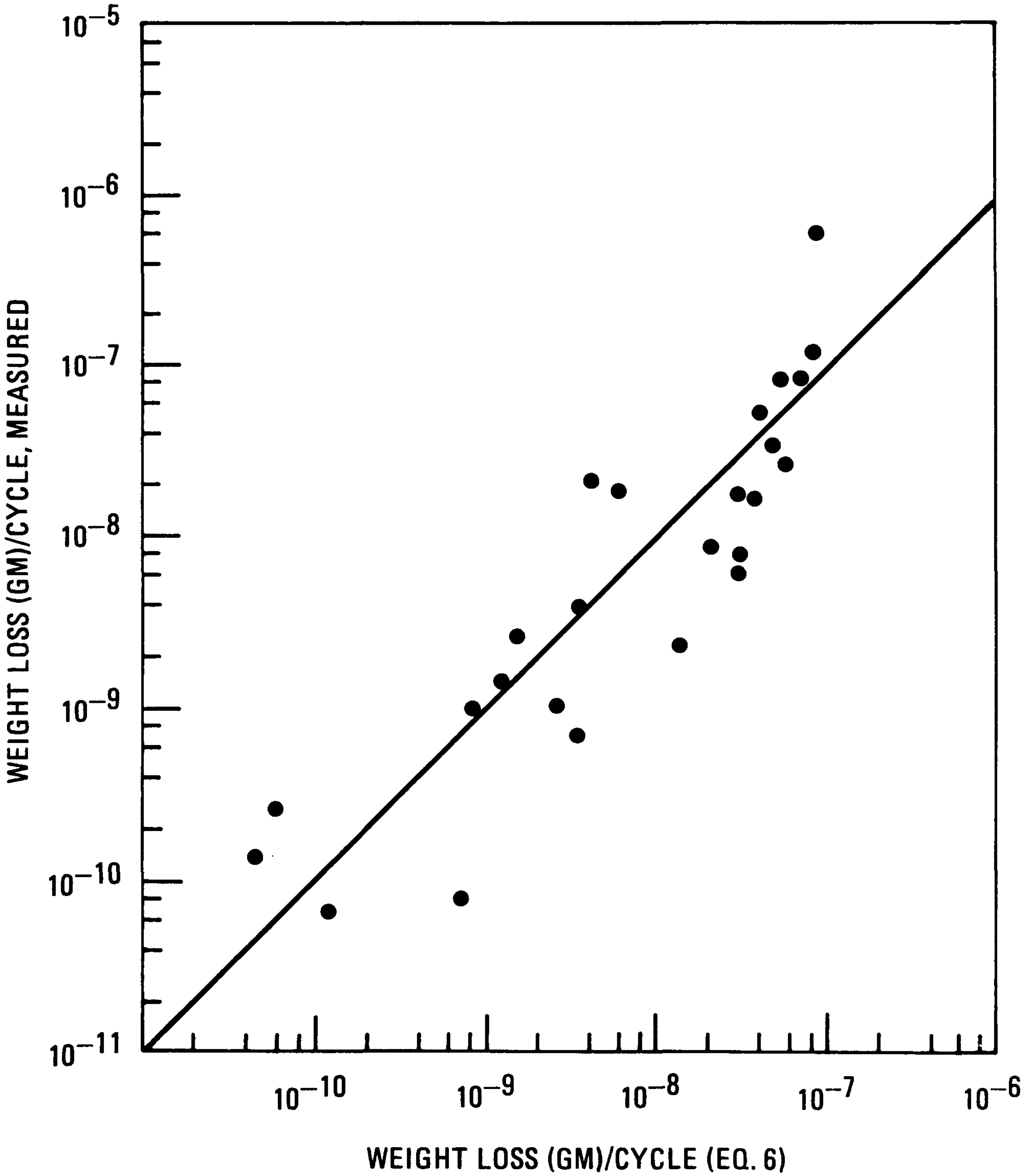

Fig. 2. Comparison of experimental data and the empirical curvefit of Eq. 6.

CONCLUSIONS

Conceptual and empirical models have been formulated for the fretting wear induced by a transverse vibration of a heat exchanger tube. The following conclusions have been obtained:

1. In many cases the fretting wear of heat exchanger tubes will be dominated by transverse, rather than longitudinal, tube motion at the support plate.

2. Significant transverse tube motion at a support plate can arise only if the shear load required to maintain the tube stationary at the support plate in a given vibration mode is significantly less than the reactions generated by the weight of the tube plus preloads.

3. An empirical model suggests that fretting wear can be substantially reduced by reducing transverse tube vibration at the support plate. This can be accomplished by either (1) reducing overall tube vibration amplitude, (2) reducing the gap between the tube outside diameter and the support plate inside diameter, perhaps incorporating a device to accomplish this, or (3) increasing the mean preload applied to the tube at the support plate.

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