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DYNAMIC STRAIN MEASUREMENTS IN THE CRANKSHAFT OF A DIESEL ENGINE

by

W.F. Curtis

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PERSONNEL

The tests were conducted by W.F. Curtis, F.B. Bryant, and L.E. Wedding, of the David Taylor Model Basin staff, with the assistance of the Internal Combustion Engine Laboratory of the Engineering Experiment Station. The report was written by W.F. Curtis.

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ABSTRACT

Alternating strains in the crankshaft of a 2000-HP Fairbanks-Morse diesel engine were measured by six electrical-resistance strain gages attached to the web of the crankshaft which was nearest the load and hence was subjected to the highest stress. Measurements were made at a series of speeds and loads up to rated load, both with and without torsional dampers on the crankshaft. The rated load without dampers was 1200 HP at 720 RPM; with dampers it was 2000 HP at 900 RPM.

The maximum measured strain was 318×10^{-6} inches per inch peak single amplitude; it occurred at 720 RPM with full load, while operating without dampers. The maximum strain measured while operating with dampers was 115×10^{-6}, expressed in the same units. This occurred at 400 RPM, the ninth-order torsional critical speed, which was not compensated by the dampers used. The maximum available power at this speed corresponded to only about 10 per cent of full-load torque. The alternating strain measured at full load with dampers was 48.7×10^{-6} inches per inch peak single amplitude.

INTRODUCTION

Because of failures in the cast-iron crankshafts of 2000-HP Fairbanks-Morse 9-cylinder diesel engines in service, the Bureau of Ships directed the Engineering Experiment Station at Annapolis to make a study of the circumstances surrounding these failures and an analysis of the engineering features involved.

A metallurgical examination of the fractured shaft of one engine, as shown in Figure 1, indicated (1)* that the initial fracture occurred at a point on the side of one web of Crank 9, about halfway between the crankpin center and the shaft center, on the face of the web toward the crank.

Following this examination, the Engineering Experiment Station undertook a vibration analysis of an intact engine of the same size and type. It was found that the rotating system had a principal natural frequency of 3600 cycles per minute** and that the torsional oscillations of largest amplitude were the ninth-order oscillation, 400 RPM, the fifth-order oscillation, 720 RPM, and the fourth-order oscillation, 900 RPM. The ninth-order oscillation, though quite strong, was of no practical importance because it is below the useful speed range. The eighth, seventh, and sixth orders were of insignificant amplitude. The Engineering Experiment Station therefore designed and installed pendulum dampers to neutralize the fourth- and fifth-order torsional oscillations.

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* Numbers in parentheses indicate references on page 12 of this report.

** The figures given here are for the engine system after installation of the spider for supporting the damper pendulums. This spider remained in place throughout the test.

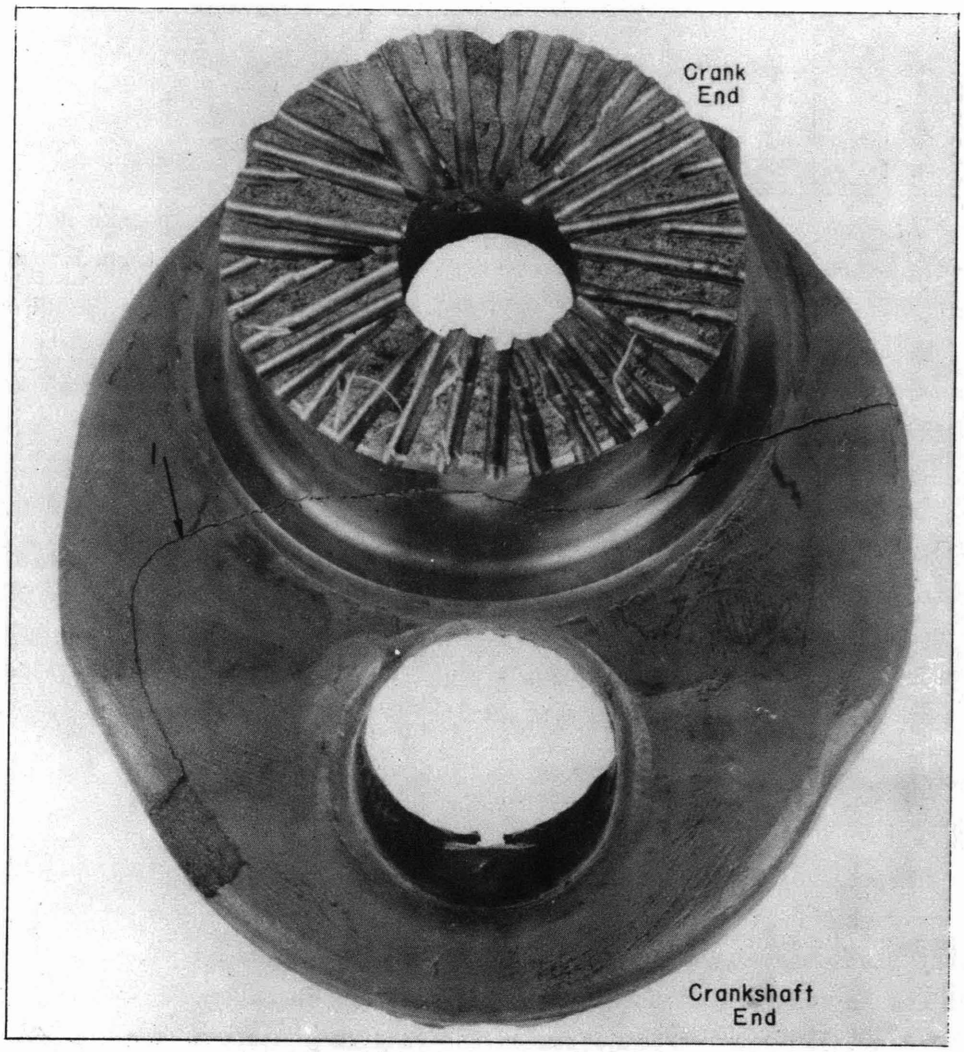

Figure 1 - Photograph of Break in Engine End Web of Crank 9 on 2000-HP Diesel Engine Generator Set, Looking Away from Generator

The initial fracture occurred at the point marked 1; this was assumed to be the region of maximum stress.

In accordance with a directive from the Bureau of Ships (2) the David Taylor Model Basin measured dynamic strains in the crankshaft during operation of the intact engine at the Engineering Experiment Station. The purpose of the test was to make a further study of the possible causes of crankshaft failures in similar engines.

As indicated in Figure 2, the engine had 9 cylinders, and was connected directly to an electric generator adjacent to Cylinder 9.

The main bearing between Crank 9 and the generator takes all the end thrust on the crankshaft.

GENERAL CONSIDERATIONS

A brief study of the forces involved will indicate that Crank 9, next to the load, is the most heavily loaded of all the cranks. A further study will indicate

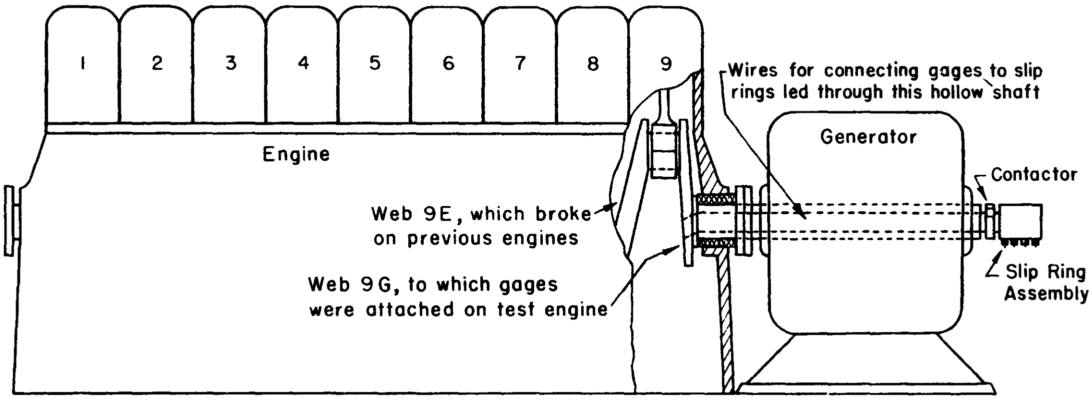

Figure 2 - Schematic Diagram of Engine, showing Location of Crank Webs

The web on the engine end of Crank 9 is designated as Web 9E, and that on the generator end as 9G. The photograph in Figure 1 is of the inboard or crank side of Web 9E on a similar engine.

that Web 9G, next to the generator, is the most heavily loaded of the two webs of Crank 9.

The fact that the fractures in service had all occurred in Web 9E could be accounted for by systematic defects in the crankshaft castings or by the fact that the cross-sectional area of Web 9E was slightly less than that of Web 9G.

Because of practical difficulties strain gages could not be mounted on the Web 9E of the engine under test, in the regions where the service fractures occurred, so all further consideration of forces and stresses had to be devoted to Web 9G, next to the generator.

As shown in the schematic diagram of Figure 3, Web 9G is loaded as follows:

1. A bending moment $T_9 \cdot S/2$ in the web, bending the web about an axis parallel to the centerline of the crankshaft. This moment is caused by the cumulative effect of the torques of all 9 cylinders, i.e., the circumferential components of the forces exerted by the 9 connecting rods.

2. A variable compression $A \cdot P$ in the web, due to the radial component of the force exerted by Connecting Rod 9.

3. A bending moment $t \cdot S/2$ in the web, bending the web about an axis perpendicular to the shaft centerline. This

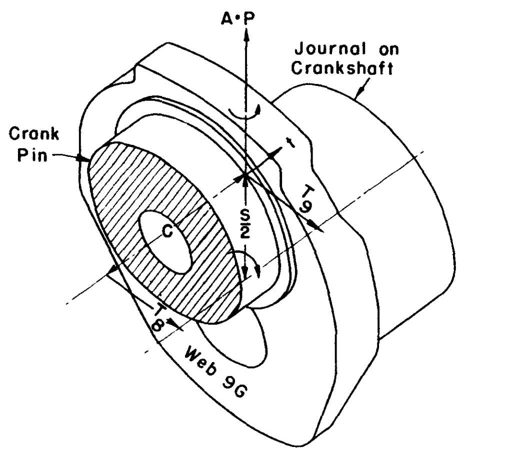

Figure 3 - Schematic Diagram of Web 9G

Here T_9 may be taken as the force on Web 9G causing rotation of the load, $S/2$ as half of the stroke S, A as the area of Cylinder 9, P as the unit pressure in Cylinder 9, T_8 as the torque transmitted from Cylinders 1 to 8 inclusive, t as the end thrust on the crankshaft, and C as the length of the crank pin.

moment is the resultant of two moments, one arising from any misalignment of the crankshaft and the other from the endwise or thrust components of the forces of all 9 connecting rods.

4. A twisting moment $T_8 \cdot C$ in the web, deforming it about an axis perpendicular to the shaft centerline and intersecting both the shaft centerline and the crankpin centerline, due to the cumulative circumferential force from Cylinders 1 to 8 inclusive, applied to the engine end of Crankpin 9.

The web resists the torsional or $T_9 \cdot S/2$-type stress as a cantilever beam with the line joining the centers of the journal and the crankpin as a neutral axis. Stress of this type is greater at increasing distances from the line of centers.

The direct compression or $A \cdot P$-type stress is distributed rather more evenly, but there is probably a maximum stress in the area directly between the journal and the crankpin. The stress concentration is particularly high close to the journal and to the crankpin.

The web also resists the bending-type stress, $t \cdot S/2$, as a cantilever beam, but the direction of flexure is at right angles to that in the torsional case. The bending stresses are compressional on the available or inboard face of Web 9G.

The twisting moment $T_8 \cdot C$ produces quite complicated shearing stresses in the web. These in turn produce tensile and compressive stresses in the face which are not parallel to the line of centers. These stresses increase with increasing distance from the line of centers.

Of course concentration of all types of stress occurs in the fillets, but the surface there was too rough to permit installation of strain gages.

TEST APPARATUS

The conditions of the test required a remote-reading strain gage of small dimensions and good frequency response, capable of rotating with the shaft and adaptable to reading at a distant point. The only known gage which satisfied these requirements was the electrical wire-resistance gage.

Previous to this test it was not certain that the cements ordinarily used for securing the wire-resistance gage to the surface under test would withstand the operating temperature and the exposure to oil, so preliminary experiments were conducted* to settle this point.

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* Four types of cement had previously been used for attaching these gages: Bakelite cement, Duco Household Cement, air-drying Glyptol, and oven-drying Glyptol. Duco Household Cement is a commonly used plastic-base adhesive manufactured by DuPont. Glyptol is a trade name for a class of oil-resistant, insulating, adhesive varnishes manufactured by the General Electric Company. Bakelite cement was discarded at the outset as it was known to be attacked by mineral oil. Protective coatings of both Duco and Glyptol were also considered. The range of temperatures investigated was from 75 degrees to 200 degrees Fahrenheit. The temperature anticipated at the gage location in the engine was 150 degrees Fahrenheit. No evidence of actual failure of any cement was found; evidence of plastic flow in the cement was inconclusive although the strain cycle employed was long, lasting 2 to 3 minutes. Strain measurements at high temperatures can be in error due to changes in the gage constant with temperature or to some other cause.

The most convenient combination employed Duco cement to secure the gage, and a coating of Glyptol to protect it. With this combination the range of uncertainty in the measured strain, under the rather difficult conditions encountered, was plus 10 per cent and minus 20 per cent, and the most probable error was minus 10 per cent of the alternating strains measured. Results with other combinations were the same within the limits of experimental error.

This degree of uncertainty in the alternating stresses was felt to be tolerable in view of the purpose of the measurement and the fact that no superior method was available. All other sources of error were negligible by comparison with this one.

LOCATION OF GAGES

It was at first proposed to mount several strain gages at selected points on the inside surface of Web 9E; the location of the crack, Figure 1, gave some indication of the location of the maximum stresses. However, this scheme involved running connecting wires from Web 9E to Web 9G of Crank 9, and the connecting wires would have had to run unsupported for a distance equal to the width of the connecting-rod bearing. Windage and vibration would have been very likely to break the wires or damage the gages to which they were connected.

Clearances did not permit installing gages on the web face nearest the generator, consequently the gages were installed on the inside face of the web nearest the crank pin. Two gages were also installed on the edge of the web; see Figure 4.

It was intended to locate the gages at points corresponding to points along the fracture; however, an error in transferring the points from Web 9E to Web 9G displaced some of them from their best positions.

As the direction of maximum stress at each point was unknown, the gages were mounted with their axes parallel to the line joining the crankpin and journal centers.

ELECTRICAL DISTANT-READING GEAR FOR THE DYNAMIC STRAIN GAGE

A schematic diagram of the basic strain gage circuit used is shown in Figure 5. Here the upper resistor $a \times R$ represents an inactive load resistor and the lower resistor R represents the strain gage. When the gage is not subjected to any applied strain,

$$E' = E \frac{R}{R + aR} = \frac{E}{1 + a}$$

When a strain $\frac{\Delta l}{l}$ is applied to the gage, the resistance of the gage becomes $R(1 + S\frac{\Delta l}{l})$ where S is the "gage constant." This results in a new value of E', namely,

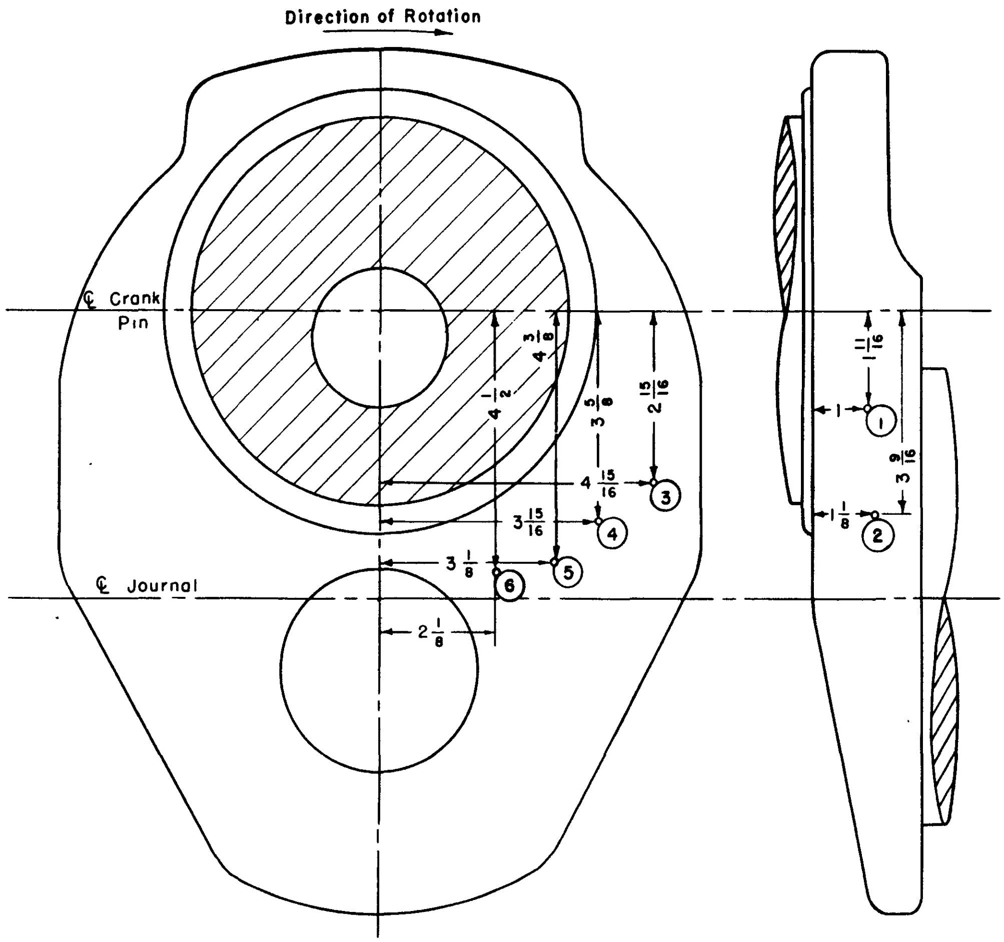

Figure 4 - Location of Wire-Resistance Gages on Crankshaft Web

The left-hand view is an elevation of the inside surface of Web 9G, looking toward the generator. The strain axes of all gages were parallel to the line joining the center of the crankpin and the center of the journal.

$$E_1' = E\frac{R\left(1 + S\frac{\Delta l}{l}\right)}{R\left(1 + S\frac{\Delta l}{l}\right) + aR} = E\frac{1 + S\frac{\Delta l}{l}}{1 + a + S\frac{\Delta l}{l}}$$

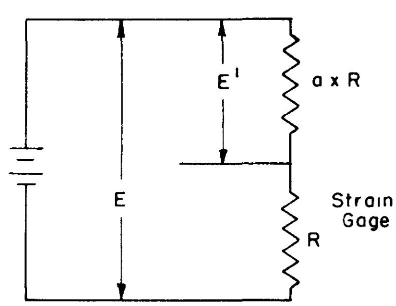

Figure 5 - Basic Circuit for Dynamic Strain Gage

In the case under consideration, the applied strain was cyclic, and the device used to measure E' was an amplifier and oscilloscope; the amplifier employed resistance-capacity coupling and hence was sensitive only to the alternating component of E'. If now $\frac{\Delta l}{l}$ represents the maximum departure of the strain from its average value, the peak value of the alternating component of E' is then

$$e = E_1' - E' = E\left[\frac{1 + S\frac{\Delta l}{l}}{1 + a + S\frac{\Delta l}{l}} - \frac{1}{1 + a}\right] =$$

$$E\,\frac{1 + a + S\frac{\Delta l}{l} + aS\frac{\Delta l}{l} - 1 - a - S\frac{\Delta l}{l}}{\left(1 + a + S\frac{\Delta l}{l}\right)\left(1 + a\right)} \doteqdot E\frac{aS\frac{\Delta l}{l}}{\left(1 + a\right)^2}$$

The approximation involved in the last expression is the result of neglecting $S\frac{\Delta l}{l}$ in comparison with $1 + a$ and is amply justified in practical cases.

Calibration was effected by a standard signal generator,* which, in accordance with usual practice, was calibrated in RMS volts. If e_0 RMS volts are required to produce a deflection of 1 inch on the oscillograph, and if the gage circuit produces y inches single deflection, then, remembering that e_0 RMS volts = 1.414 e_0 peak volts,

$$e = 1.414\ e_0 y_1 = E\frac{aS\frac{\Delta l}{l}}{(1 + a)^2} \qquad \frac{\Delta l}{l} = \frac{1.414\ e_0(1 + a)^2}{aSE}\ y_1$$

The constants used were E = 6 volts, a = 1, S = 2.0, whence $\frac{\Delta l}{l}$ = 0.472 $e_0\ y_1$.

A schematic diagram of the circuit arrangements used is given in Figure 6; this diagram is largely self-explanatory.

The mercury slip rings shown in this diagram and indicated in Figure 2 consisted of stainless steel rings mounted on an insulating sleeve at the end of the

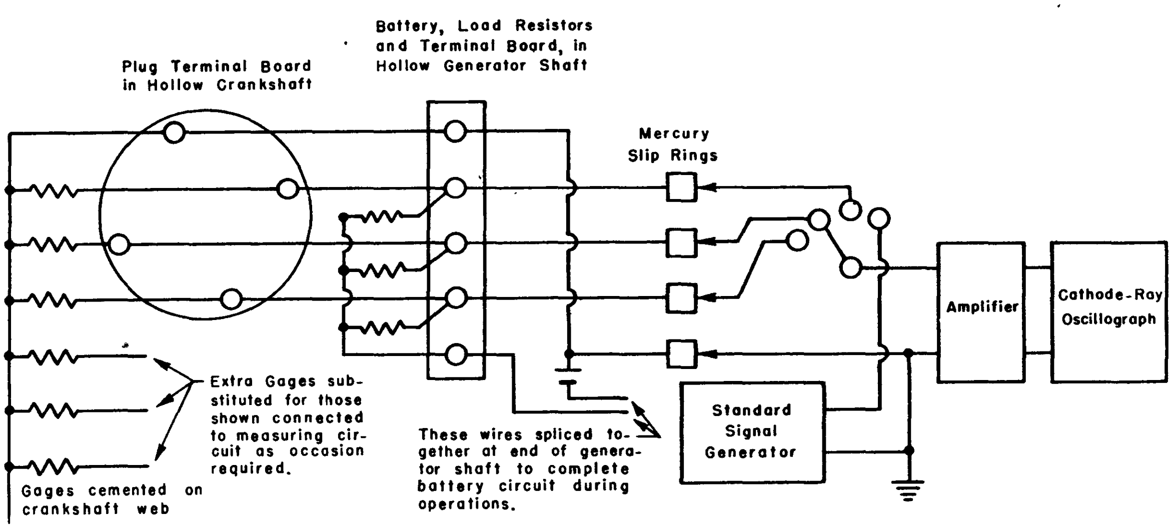

Figure 6 - Circuit for Observations on Three Strain Gages as used on Diesel Engine Crankshaft

* The standard signal generator used in this test consisted of a low-power, audio-frequency oscillator manufactured by the Taylor Model Basin and a "microvolter," i.e., a rectifier-type voltmeter and calibrated attenuator manufactured by the General Radio Company.

generator shaft. These rings dipped into pools of mercury contained in deep grooves in a block of insulating material which was held fixed with reference to the generator frame. Stainless steel electrodes in the mercury pools were connected to the amplifier as shown in Figure 6. The slip-ring assembly used was originally built for use on a different project (3) and contained four rings. This made it possible to observe only three of the six gages during any one test; to change to other gages it was necessary to stop the engine and change connections on the crankshaft.

In addition to the apparatus shown in Figure 6, a contactor mounted on the generator shaft was arranged to short-circuit the amplifier input momentarily at the instant Piston 9 was at its top dead center; see Figure 2. This produced a sharp excursion of the oscilloscope beam which indicated the phase of the alternating strains relative to the position of the crankshaft. This excursion was very prominent on the oscilloscope screen when it was observed visually, but the rate of travel of the beam was so great that the excursion did not register on the film when the oscilloscope pattern was photographed. The contactor did produce a slight break in the recorded trace, however, which was located with the help of notes made during the test. As an aid to reading the oscillograms, the positions thus located have been marked on the prints manually; see Figures 7 to 12. The recorded trace on the remaining oscillograms was so faint that the location of the dead-center point was uncertain.

The oscillograms were made by photographing the oscilloscope screen with a moving-film camera made by removing the intermittent-motion mechanism and shutter from a commercial moving-picture camera. Since the photographs lacked sufficient contrast, they were traced in ink for reproduction in this report.

TEST RESULTS

All measured values of strain and the conditions under which they were obtained are given in Table 1. The principal results are shown graphically in Figures 13 to 15.

DISCUSSION OF RESULTS

Bearing in mind that the engine system has torsional critical speeds at 400, 720, and 900 RPM, and that the 720 and 900 RPM vibrations are neutralized by the dampers, the experimental results can be readily explained. It is obvious from Figures 13 and 14 that strains of large amplitude are primarily the result of torsional oscillations. This is most strikingly shown by the fact that at 720 RPM, the strain under 10 per cent load with no dampers is about five times the strain under full load with the dampers in place. At 400 RPM high strains were found in all tests; this is in accordance with expectations as the dampers did not neutralize the ninth-order oscillation. Figure 15 shows that the strain decreases with decreasing distance from the torsional neutral axis, which at once suggests that the principal strain arises

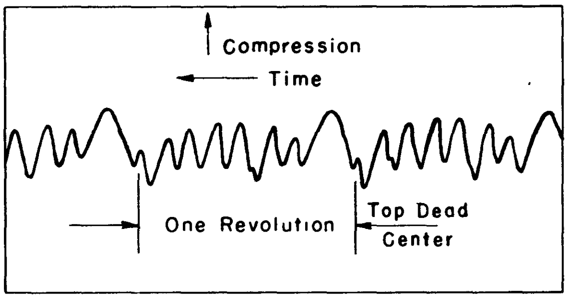

Figure 7 - Record from Gage 3 at 450 RPM, Light Load and No Damping

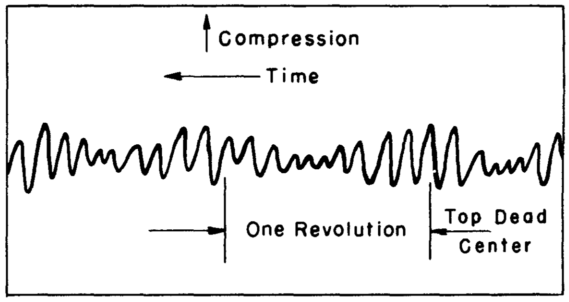

Figure 8 - Record from Gage 2 at 450 RPM, Light Load and No Damping

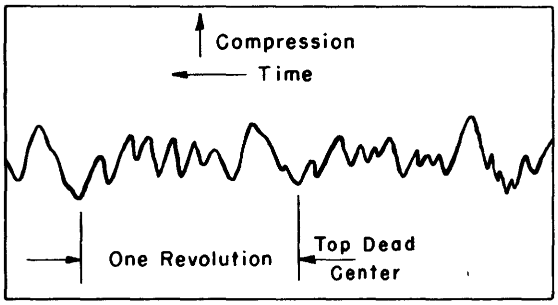

Figure 9 - Record from Gage 6 at 450 RPM, Light Load and No Damping

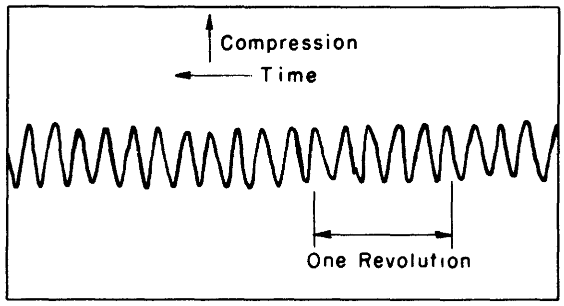

Figure 10 - Record from Gage 2 at 720 RPM, Full Load and No Damping

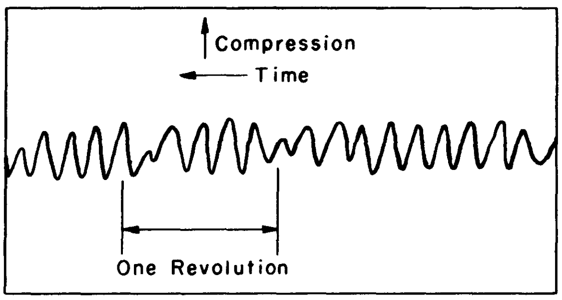

Figure 11 - Record from Gage 3 at 720 RPM, Full Load and No Damping

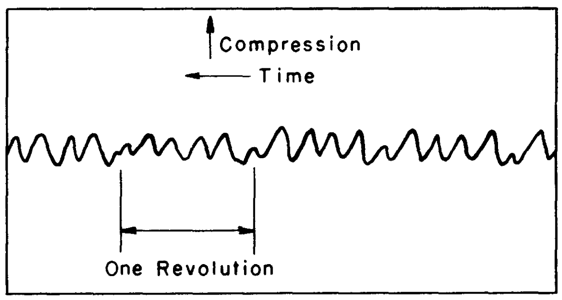

Figure 12 - Record from Gage 6 at 720 RPM, Full Load and No Damping

TABLE 1

Alternating Strain Data from Test of Fairbanks-Morse 2000-HP Engine

RPM	External Load HP	Strain in parts per million, Peak Single Amplitude					
		Gage 1	Gage 2	Gage 3	Gage 4	Gage 5	Gage 6
Damper weights removed - light load							
340	0	22	23	29	28	14	29
400	77		162	162			117
450	95		40	47			36
500	104		20	54			54
550	111		26	40			40
600	120		34	53			53
650	131		44	68			68
700	155		131	137			108
720	160 approx		252	238			140
Damper weights removed - load simulating service conditions							
400	333	146	162	162	146	122	*
600	1195	45	71	52	58	41	*
650	1320	63	78	104	105	73	*
700	1552	302	255	255	318	255	228
720	1595	282	318	285	257	205	142
Damper weights replaced - full load							
340	0		23	26	27		
400	266		104	116	99		
500	870		28	41	37		
600	1200		19	42	38		
700	1580		31	40	47		
810	1600		40	55	61		
900	2020		28	45	49		

* The gage could not be read because of poor contact.

from torsion. While the correlation between strain and distance from the axis is not perfect, the observed tendency supports the view that the strain due to torsion exceeds the strain due to the web-twisting moment $T_8 \times C$ except at full load with dampers. Both torsion and the web-twisting moment are affected by torsional oscillation.

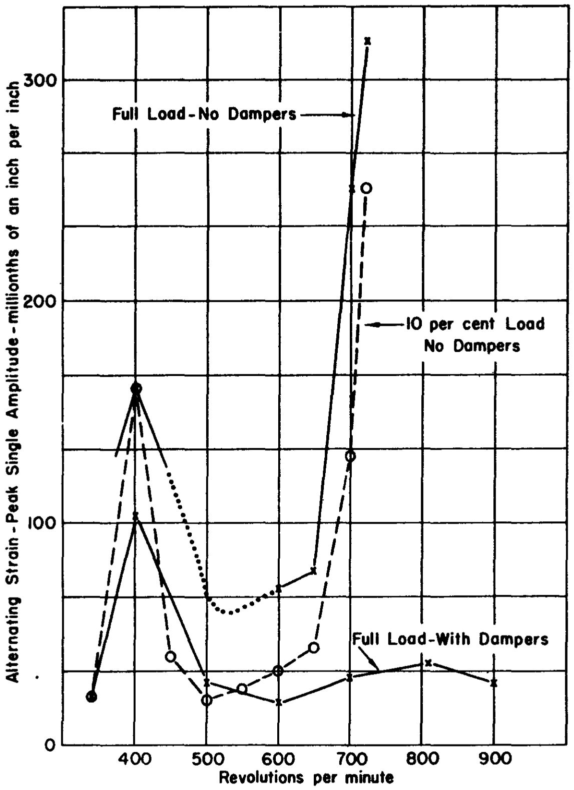

Figure 13 - Alternating Strain - Gage 2

This gage was located on the edge of Web 9G of Crank 9.

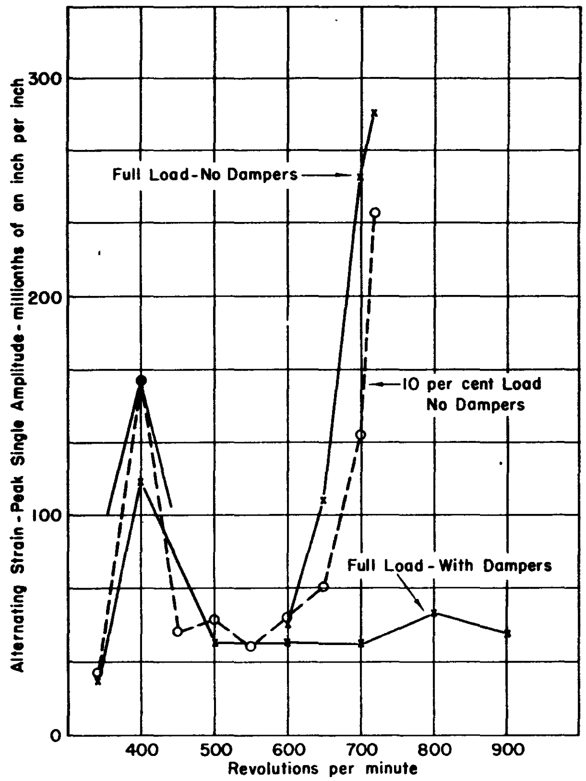

Figure 14 - Alternating Strain - Gage 3

This gage was located on the face of Web 9G of Crank 9.

The distribution of strain as shown in Figure 15 does not indicate any great stress concentration in the area investigated. In view of the uncertainty as to the stress distribution, it is probable that strains somewhat higher than the measured strains do exist elsewhere in the crank web investigated, but the orderly distribution of strain exhibited makes it rather unlikely that any great concentration exists in the main portion of the web. Stress concentration at the fillet is somewhat more probable, but as previously stated strains in this vicinity could not be measured. However, the results of the metallurgical examination (1) militate against the hypothesis that stress concentration in the fillet is a cause of failure.

In the absence of information on the fatigue limits of the material and the steady strain which is superimposed on the dynamic strains here measured, it is not certain that the material of the crankshaft is overstressed; in fact, the magnitude of the observed strains is quite moderate. However, the fact that strains existing in the presence of torsional oscillations are 5 to 6 times the strains found when the oscillations are damped very strongly suggests that the torsional oscillations are responsible for the failure of the crankshaft. It is noteworthy that the strain can be reduced to a small fraction of its present value by two methods:

1. By restricting the speed to the range between 500 and 620 RPM;

2. By equipping the engine with pendulum dampers.

Neutralizing both the fourth- and fifth-order oscillations makes it possible to operate the engine up to at least 900 RPM with relatively little strain in the crankshaft.

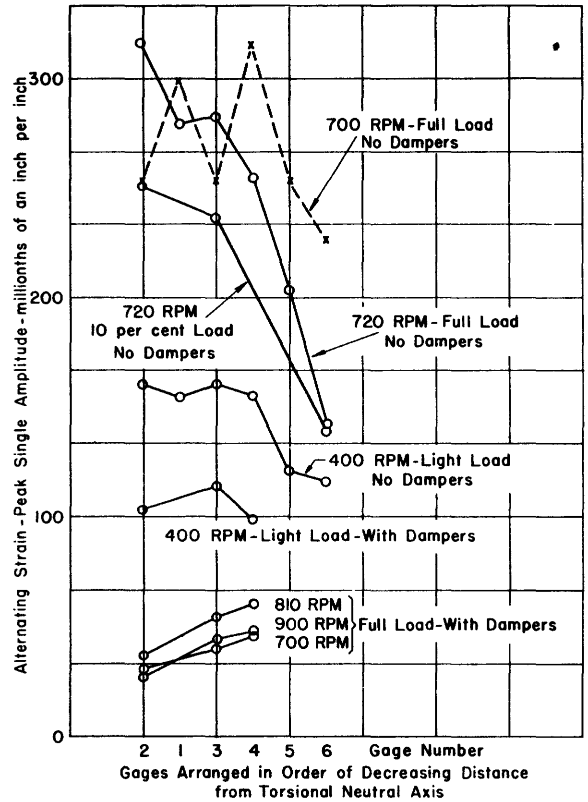

Figure 15 - Effect of Gage Location on Strain in Crankshaft

For the location of the strain gages, see Figure 4 on page 6.

CONCLUSIONS

The strains in the crankshaft of this engine are greatly magnified by torsional resonance. With the engine operated under conditions simulating service conditions, the maximum strain found in the crankshaft was 320×10^{-6} inches per inch peak single amplitude. The smallest stress which could have produced this strain is 8000 pounds per square inch. This occurred at 720 RPM, which is the maximum rated speed for this engine and is near a torsional critical speed.

When the engine was operated with torsional dampers, the maximum measured strain in the crankshaft at speeds within the operating range of 500 to 900 RPM was 60×10^{-6} inches per inch. The corresponding equivalent stress is 1500 pounds per square inch.

Taking into account the facts that alternating strains somewhat in excess of the measured values could have occurred, and that a large steady stress was also present, it is very probable that the strains occurring when operating without torsional dampers were excessive. However, further tests, including fatigue tests of the crankshaft material, would be necessary to prove this point definitely.

REFERENCES

(1) "Testing of Fairbanks-Morse Diesel Engine with Cast Iron Crankshaft," Report B-7985, U.S. Engineering Experimental Station, Annapolis, Maryland.

(2) BuShips letter S41(350,457) of 25 November 1942 to TMB.

(3) "Alternating Torque Tests on a Turbine Shaft, U.S. Destroyer MOFFETT," by W.F. Curtis, TMB Report R-85, April 1943.