DISA torsiograph: a test of susceptibility to transverse vibrations near the torsional resonant frequency

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DISA TORSIOGRAPH - A TEST OF SUSCEPTIBILITY TO TRANSVERSE VIBRATIONS NEAR THE TORSIONAL RESONANT FREQUENCY

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1.0 INTRODUCTION

A DISA Model 51D07 Torsional Vibration Transducer was recently used to measure the torsional vibration spectrum on the drive shaft of a fireboat built for the Department of National Defence in Vancouver, B.C. The results of the tests showed a large torsional vibration component at a frequency close to 4 hertz, the torsional resonant frequency of the transducer. Because torsional amplitudes were not expected to be significant at that frequency the possibility of torsional resonant frequency excitation by a forcing transverse oscillation was postulated. In order to verify the transducer's operation under just such a condition the test described in the body of this report was performed.

2.0 EXPERIMENTAL

The test program involved the operation of the DISA torsional transducer at known sinusoidal torsional vibration amplitudes over a shaft speed range equivalent to the speeds which were encountered during the sea trials of the fireboat. At each test point, i.e. at each shaft speed-torsional amplitude pair, measurements were taken both with and without a transverse oscillation applied directly to the transducer mounting. The results were compared for evidence of an induced torsional oscillation.

An overall view of the experimental set-up is shown in Figure 1. A 5 HP motor with a variable speed output transmission (680-4100 rpm) drove a Hooke's couple device (universal joint). A 21 inch diameter flywheel with a polar moment of inertia of approximately 1 lb sec^2ft provided excellent smoothing to the drive system. The torsional amplitude of the Hooke's couple output shaft was varied by means of a swivel plate which altered the angle of intersection of the axes of the input and output shafts of the couple (see Reference 1 for details). The torsiograph was not directly coupled to the output shaft but was driven by a "seamless" steel belt and pulley system, Figure 2, to simulate the actual measurement conditions which had prevailed on the fireboat. (Previous to use on the fireboat the torsiograph had been calibrated on the Hooke's couple using the same belt and pulley system and also with larger pulleys to simulate the actual diameter of the ship's shaft.)

An electromagnetic shaker mounted beneath the torsiograph mounting plate was used to apply transverse oscillations to the torsiograph, Figure 2. The frequency and force applied to the mounting by the shaker were controlled by the force feedback system detailed in Figure 3. For the test results reported in the next section the oscillation frequency was 4 hertz and the torsiograph mounting plate was subjected to a peak-to-peak displacement of 0.02 inches.

The torsiograph output was recorded on an instrumentation f.m. tape recorder operating at a tape speed of 15 i.p.s. and was monitored concurrently on an oscilloscope. The recorded data was then analysed on a 400 line narrow band frequency analyser and comparisons were made between runs with and without vertical oscillations.

3.0 TEST RESULTS

The results of the narrow band analysis of the torsiograph output for tests at zero torsional vibration input are shown in Figure 4 and at a torsional amplitude of ±0.11 degrees (5 degrees intercept angle on the Hooke's couple axes) in Figure 5. The shaft speeds were varied from 630 rpm to 1500 rpm. Since the torsional oscillations occur at twice the shaft frequency this speed range was equivalent to 1260 rpm to 3000 rpm on the actual ship shaft. (Maximum shaft speed was 1800 rpm during sea trials.) In each photograph the upper spectrum is the result without transverse oscillation applied to the torsiograph while the lower spectrum is the result with transverse oscillation.

The primary vibrations evident in the frequency spectra are the Hooke's couple frequency and the shaft frequency. The analyser cursor indicates the Hooke's couple frequency in the figures and the amplitudes have been normalized relative to the amplitude at the Hooke's couple frequency in Figure 5a. (These amplitudes are indicated in the second column of figures from the left above each photograph, 1.00 for both A and B in Figure 5a for example, and 361×10^{-3} and 378×10^{-3} for A and B in Figure 4c.) There is no significant difference between tests with and without vertical oscillations. A torsional component at 4 hertz is discernible in some spectra, Figure 5c for example, but the amplitude is negligible.

4.0 CALCULATION OF BELT-PULLEY SYSTEM RESONANT FREQUENCY

Although there was no evidence on the torsional frequency spectra from these test results to indicate the presence of a resonance in the belt-pulley system during the tests, an approximate calculation of that frequency was made to see if it could have had any influence on either these test results or the results of the sea trials.

According to Reference 2, the natural frequency of the belt-pulley system is given by

$$f_p = \frac{0.195\ r}{(\gamma \cdot J_p \cdot L_S)^{\frac{1}{2}}} \text{ hertz}$$

where r = the pulley radius
γ = $1/E \cdot A$
E = Young's modulus
A = belt cross-sectional area
J_p = polar moment of inertia of the pulley and internal spindle of the torsiograph, but not including the seismic mass
L_S = length of belt between contact points; this is equal to the length between centres when the driver pulley and driven pulley are of the same diameter

The resulting natural frequency for our test system was approximately 200 hertz and therefore was of no influence on the results reported here. A similar resonant frequency applied to the sea trial system because of offsetting changes in pulley radius and polar moment of inertia.

5.0 CONCLUSION

Tests on a DISA Model 51D07 Torsional Vibration Transducer have shown

that transverse oscillations at the torsional resonant frequency of the instrument do not excite significant torsional resonant amplitudes in the instrument.

Belt-pulley system resonant frequencies were also shown to be well above the frequency range of interest and therefore were not the cause of large torsional amplitudes which had been observed during field use of the instrument.

6.0 REFERENCES

1. T.H. Hammell
J.E. Carvish
"A Technique for Measuring, Recording and Analyzing Torsional Oscillations of Crankshafts of Reciprocating Engines"

Report ME-232 National Research Council Division of Mechanical Engineering June 1969

2. W. Ker Wilson
"Practical Solution of Torsional Vibration Problems",

Vol. 5 Vibration Measurement and Analysis 3rd Edition, p. 19, ff., Chapman & Hall Ltd., London, 1969

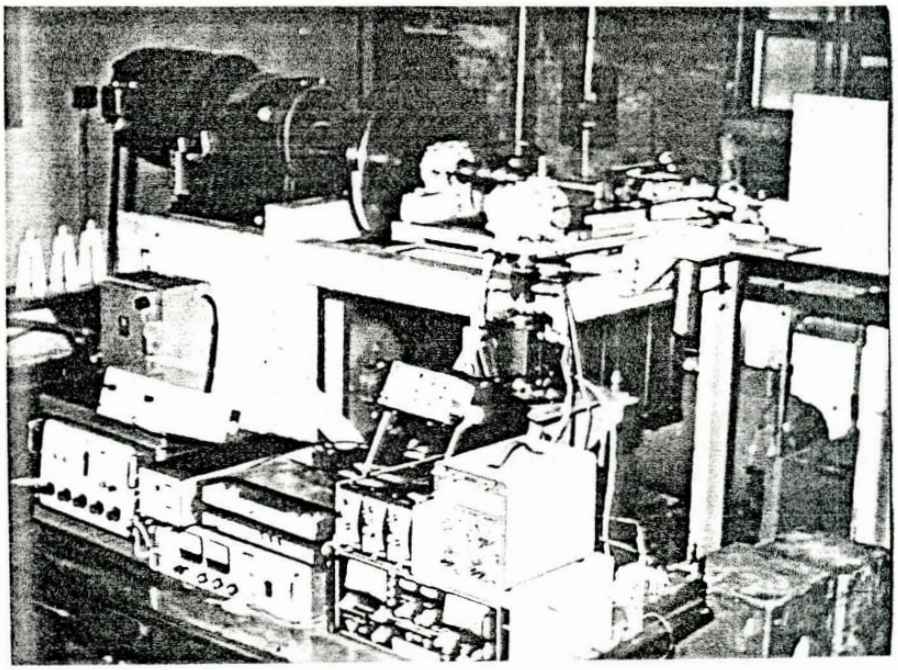


FIGURE 1 TORSIOGRAPH TEST APPARATUS

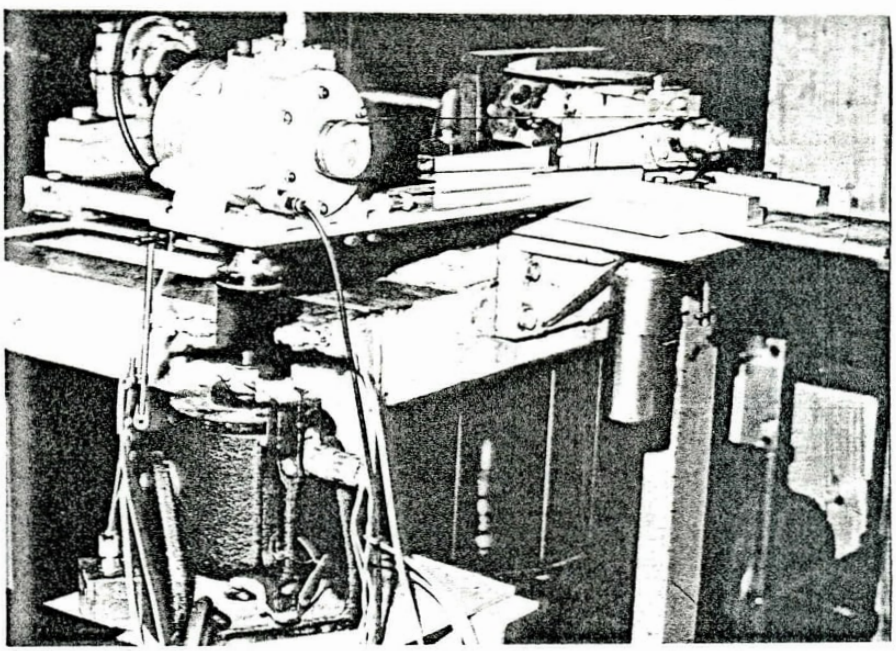


FIGURE 2 BELT-COUPLED TORSIOGRAPH MOUNTED ABOVE VIBRATION EXCITER

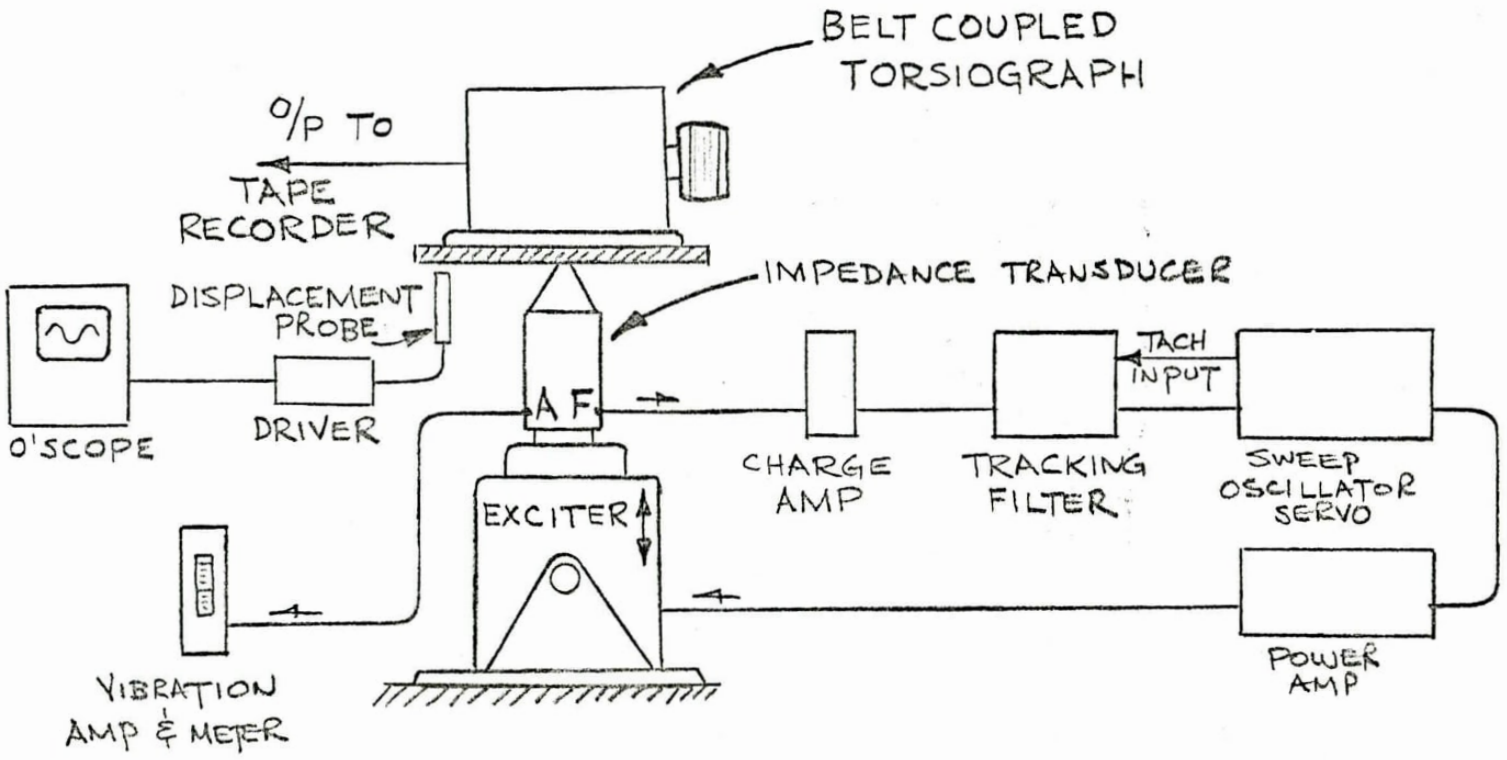
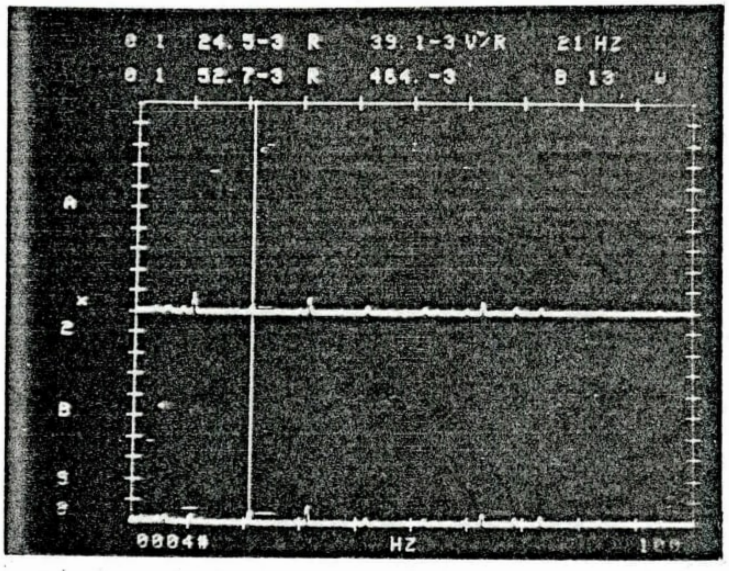


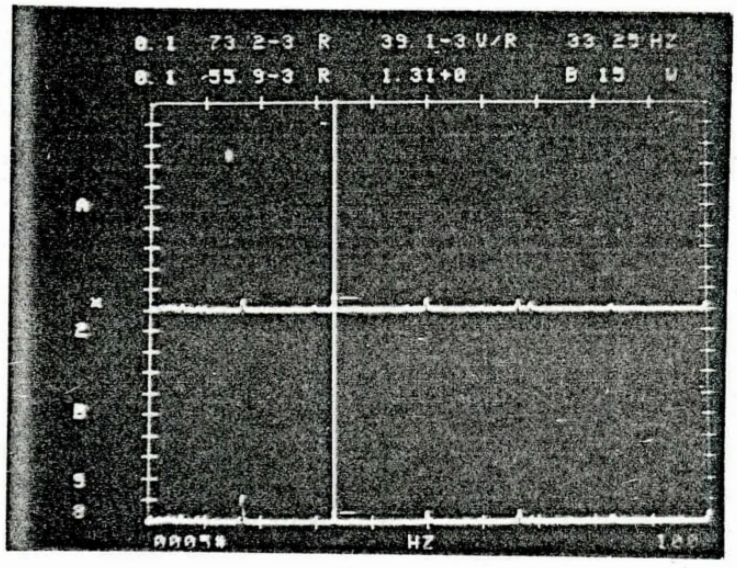
FIGURE 3 INSTRUMENTATION TO STUDY THE EFFECTS OF TRANSVERSE VIBRATION ON THE PERFORMANCE OF A BELT COUPLED TORSIOGRAPH



4a

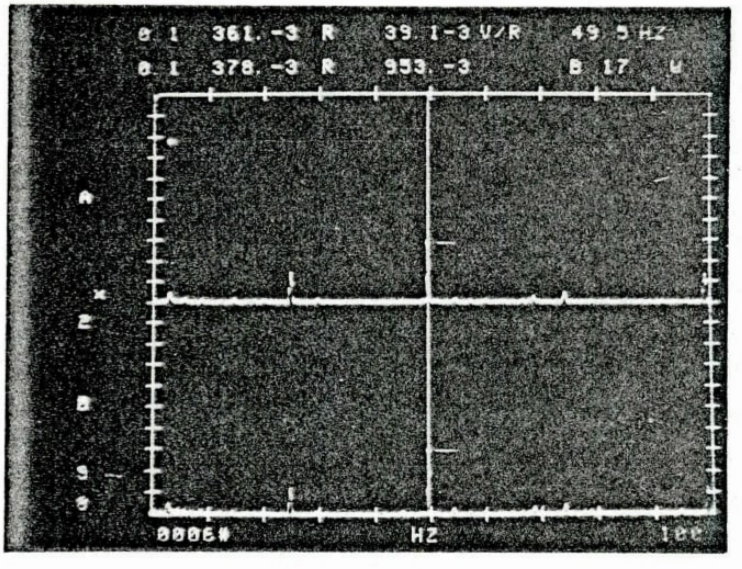
Hooke's Couple Frequency

21 hertz



4b

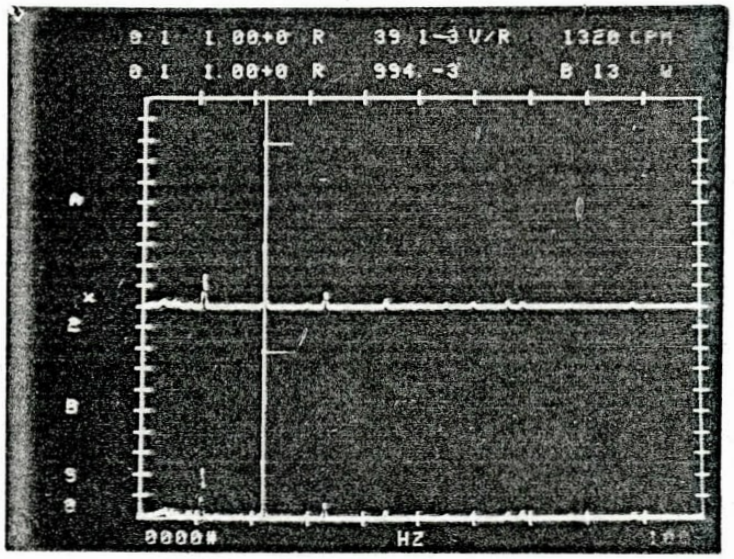
33.3 hertz



4c

49.5 hertz

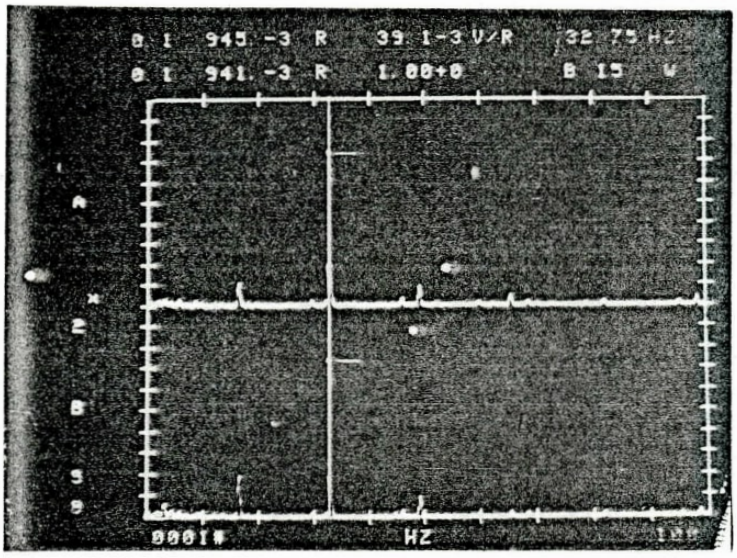
FIGURE 4 Torsiograph Output Spectra with Torsiograph Amplitude Nominally Zero



Hooke's Couple Frequency

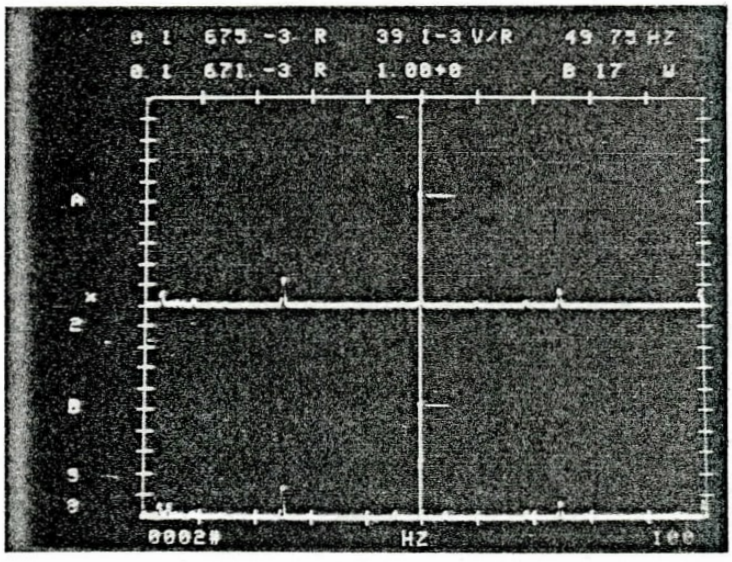
5a

22 hertz



5b

32.8 hertz



5c

49.8 hertz

FIGURE 5 Torsiograph Output Spectra with Torsional Amplitude = ±0.11 degrees