

Application Note

Marine Vehicles and Offshore Installations: Laser Diagnostics of Machinery Health*

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Abstract

For reasons of both safety and economy it is desirable to avoid the breakdown at sea of heavy machinery in marine transport and offshore installations. Unscheduled loss of production or power during 'downtime' enhances vulnerability and often carries severe financial penalties. This has led to an increasing interest in the concept of monitoring machinery health to detect impending failure. This Application Note demonstrates how the Torsional Vibration Meter Type 2523 can be used to address the problem of diagnosing failure in viscous shear torsional dampers which are normally fitted to the crankshaft of large marine propulsion diesels and generator sets. The method allows condition diagnosis to be achieved while the engine is running and avoids the need for 'downtime' associated with traditional fluid sampling methods.



Torsional Vibration Meter Type 2523

Introduction

The majority of large marine diesel engines require some form of damping or detuning device to be fitted to the crankshaft in order to prevent the build-up of large vibration amplitudes and stresses at torsional resonances. For higher power output engines, the viscous shear damper is the usual choice because of its excellent heat dissipation capability. This damper consists of a light metal casing which is attached to the free end of the engine crankshaft and in which a relatively heavy internal seismic mass is enclosed. Small clearances exist between the sides and circumference of the seismic mass and the enclosing casing which are filled with a highly viscous

fluid, typically silicone oil. During normal engine operation, the viscous drag exerted by the damper fluid is sufficient to cause the internal mass to move with the casing and both members rotate at essentially the same speed. At a 'critical' speed, however, a harmonic of the torque exerted on the crankshaft system coincides with a natural torsional mode with increased amplitude. In this case, relative motion occurs between the seismic mass and the casing causing viscous shearing of the damper fluid which tends to damp out the torsional vibrations.

A decline in the performance of the damper is usually associated with a change in the operation viscosity of

the damper fluid. The normal practice for health diagnosis is thus to periodically sample the damper fluid for laboratory analysis and subsequent determination of its viscosity. This method suffers from severe practical disadvantages which are:

- (i) In order to obtain a fluid sample the engine must be stopped.
- (ii) Often, ancillary equipment must be removed in order to gain access to the damper fluid sampling points. During this time, the temperature and viscosity of the fluid may change considerably from its true operational value.
- (iii) In general, the fluid sample is re-

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turned to the damper manufacturer for analysis thereby, further extending the period of 'down-time'.

(iv) The total number of possible samples is restricted to such an extent that condition assessment on a regular basis is precluded.

(v) Degradation of the viscous fluid does not usually occur in a linear fashion and, thus, the results from infrequent sampling can rarely supply sufficient trend information to allow scheduled maintenance/overhaul procedures to be confidently extended.

Clearly an in-situ method of assessing viscous damper health offers a significant advantage. This has recently been provided by the application of optics and laser technology to the problem and relies on the in-situ measurement of crankshaft torsional vibration.

The Torsional Vibration Meter Type 2523

The in-situ measurement of torsional vibration of rotating components is a notoriously difficult problem. Conventional measurement systems employ fixed optical or magnetic transducers which monitor the passage of a slotted disc or toothed wheel connected to the rotating component. Torsional vibration data are obtained via demodulation of the observed fluctuations in slot or tooth passing frequency. These types of system have a limited dynamic range and are subject to 'noise' problems if the rotating target component or structure to which the proximity transducer is attached vibrate as a solid body. The advent of the Torsional Vibration Meter Type 2523 has solved these problems. A brief description of its physics of operation is as follows. More information can be found in the Product Data Sheet for Type 2523.

With reference to Fig. 1, two low-powered parallel laser beams are directed at the rotating component which can have arbitrary cross section. Light is backscattered from the two points A and B and is collected onto the surface of a photodetector. The current output of the photodetector is modulated at the difference in light frequency between the two scattered beams which is caused by the Doppler effect when scattering occurs from the moving surface at A and B. Any solid body oscillation of the target, denoted by the velocity V in the figure, is ignored by this geometry since equal Doppler shifts are produced from this form of motion.

The difference in Doppler shifts from A and B is due to the rotational speed of the target N and the corresponding velocities V_1 and V_2 respectively. This frequency difference f_D can be shown to be given by [2,3]

$$f_D = (4\pi/\lambda) Nd \sin \alpha \quad (1)$$

where λ is the wavelength of the laser light, d is the beam separation distance and α is the angle which the

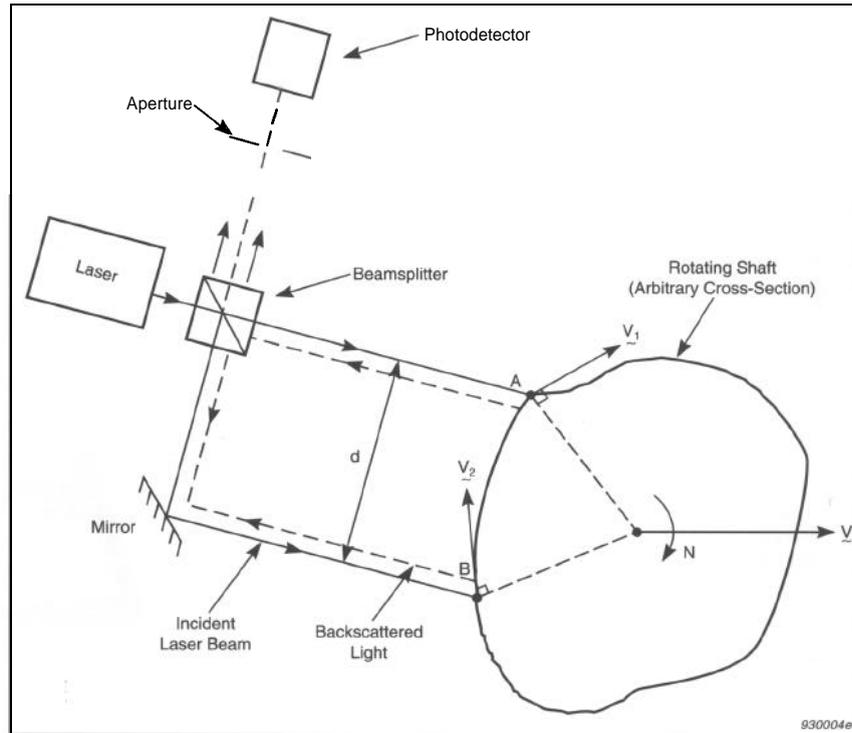


Fig. 1. Torsional Vibration Meter Type 2523: optical geometry

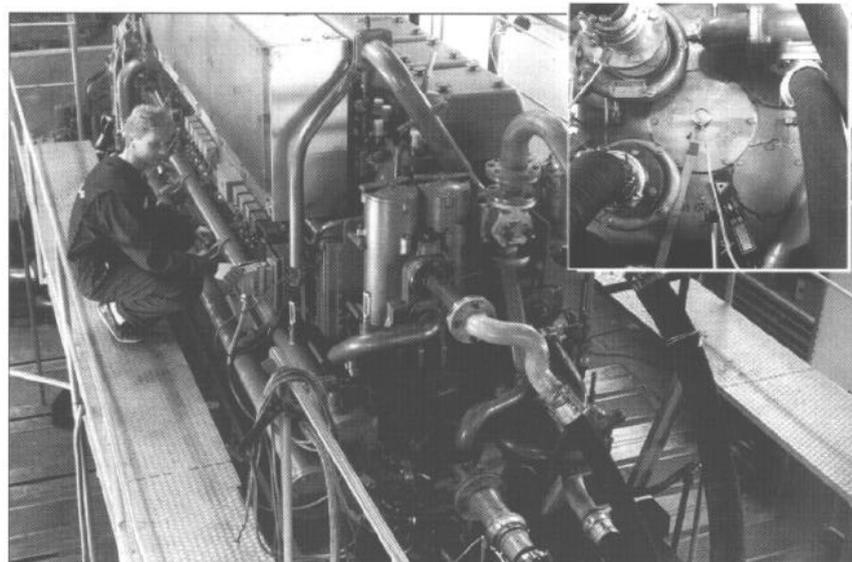


Fig. 2. Torsional vibration measurement on a marine diesel engine using Type 2523

plane of the incident laser beams makes with the rotational axis of the component. With reference to eqn (1), frequency tracking, the detector output f_D produces a time resolved analogue of the target rotational speed N ,

the fluctuating part of which is the required torsional vibration velocity.

It is now possible, therefore, for the marine engineer to quickly assess the torsional vibration level of an engine crankshaft [1] by simply pointing the

parallel laser beams at its side or end surface. Fig. 2 shows the instrument in operation on the crankshaft of a marine diesel engine.

Damper Health Diagnosis

Theoretical considerations

Fig. 3 shows the two-mass model used to describe the torsional vibration response of an engine fitted with a viscous shear damper. The inertia I_d represents the seismic mass of the damper and $\theta_d(t)$ its instantaneous angular displacement. The engine shafting system is represented by a single degree of freedom system comprising a mass of inertia I coupled to an arbitrary fixed point through a massless shaft of torsional stiffness K . Viscous damping torque exerted on the damper casing by the seismic mass is modelled by a viscous dashpot of damping coefficient C .

The vibration response characteristics of the two-mass model in Fig. 3 are well documented [4,5]. The steady state vibration amplitude, θ , of inertia, I , to excitation by a harmonic torque $T(t) = T \cos(\omega t - \phi)$ is given in non-dimensional form as a dynamic magnifier M where

$$M = \frac{\theta}{\theta_{st}} = \left\{ \frac{r^2 + 4\zeta^2}{r^2 (1-r^2)^2 + 4\zeta^2 [\mu r^2 - (1-r^2)]^2} \right\}^{1/2} \quad (2)$$

where

$\theta_{st} = T/K =$ angular deflection at the damper casing if the harmonic torque was applied statically,

$\mu = I_d/T =$ ratio of the inertia of the damper's seismic mass to the inertia of the shafting system,

$\omega_n = (K/I)^{1/2} =$ undamped natural frequency of the main shafting system,

$r = \omega/\omega_n =$ forced frequency ratio (engine speed term),

$\zeta = C/C_c =$ damping ratio where $C_c = 2I_d\omega_n$.

Fig. 4 shows the response of the dynamic magnifier, M , versus the forced frequency ratio, F , for different values of the damping ratio, ζ .

The results are calculated for a typical inertia ratio of $\mu = 1/3$. For zero damping the peak vibratory response (i.e. resonance) occurs at an excitation frequency that coincides with the undamped natural frequency

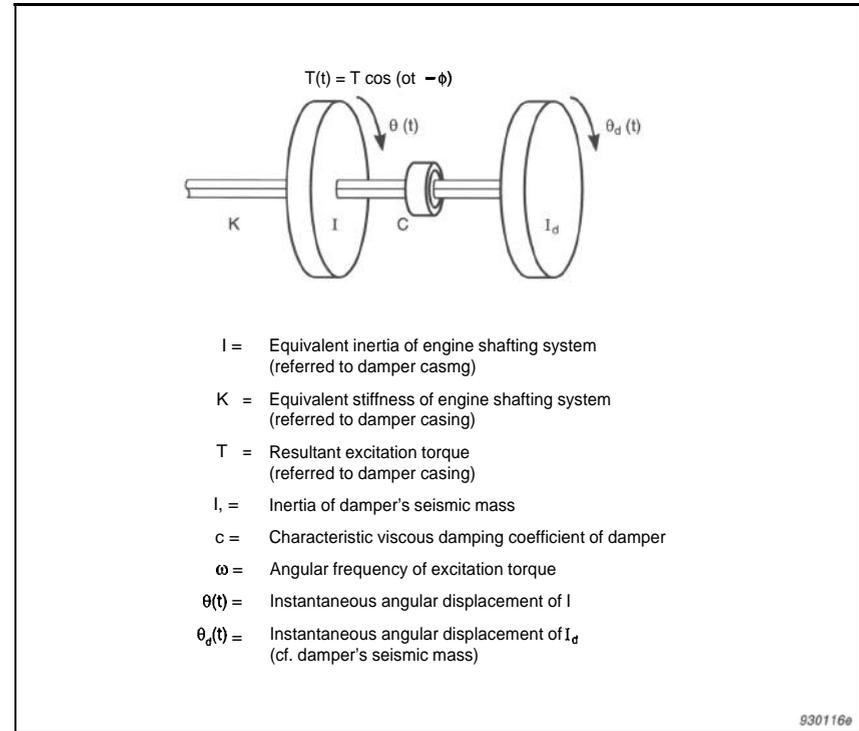


Fig. 3. A two-mass dynamic model of an engine installation fitted with a viscous shear torsional vibration damper

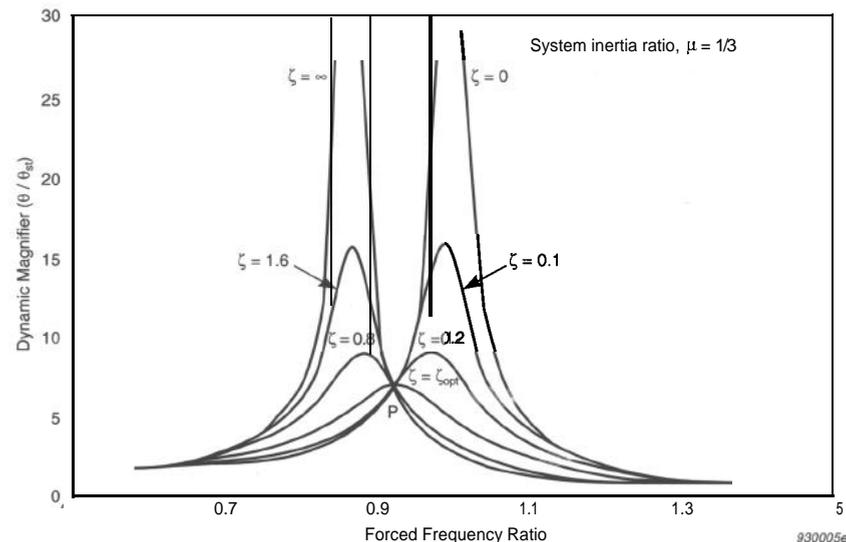


Fig. 4. Viscous damper: Dynamic magnifier curves for various values of damping ratio (ζ)

ω_n (i.e. $r = 1$). If the damping is infinite so that the two masses I and I_1 are locked together then resonance occurs at a lower excitation frequency ω_n given by.

$$\omega_n' = \omega_n (1/(1 + \mu))^{1/2} \quad (3)$$

Also shown in Fig. 4 are other response curves for different values of the damping ratio, ζ , between these extremes. All curves pass through a common point P regardless of ζ . Optimum damping is achieved with a response curve whose maximum coincides with P . In this condition the torsional vibration amplitude at resonance is minimised. Given that the damper fluid viscosity is the primary controller of the damping ratio ζ in practice, it is usual to select a fluid whose viscosity initially provides a value of ζ close to ζ_{opt} . Whatever the value of ζ , however, the resonant frequency of the system will lie between ω_n and ω_n' and have a finite amplitude in practice.

An immediate possibility for assessing damper health in situ is to use the Torsional Vibration Meter Type 2523 to produce a response curve similar to that shown in Fig.4. This curve should be compared with a baseline measurement taken, for example, at pass off for the engine where near to optimum damping conditions will have been achieved. Shifts in the resonant frequency and/or associated resonant vibration amplitude could be used as performance assessment criteria. It is now useful to predict the magnitude of the changes in these parameters produced by changes in damping ratio ζ using the two-mass model.

With reference to eqn (2) it is straightforward to predict how the resonant frequency and amplitude for a given system will vary as a function of the damping ratio ζ . Results calculated for a system with an inertia ratio of $\mu = 1/3$ are shown in Figs. 5 and 6. With reference to Fig. 5 the points X' and X'' indicate the resonant amplitudes that correspond to changes in the optimum damping ratio of 50% and 200%, respectively. In both cases, the resonant amplitudes are approximately 30% higher than those predicted for an optimum damping condition.

With reference to Fig. 6 the points Y' and Y'' indicate how the resonant frequency of the system is affected by similar percentage changes in damping. In this instance, the value of the resonant frequency changes by approximately 5%.

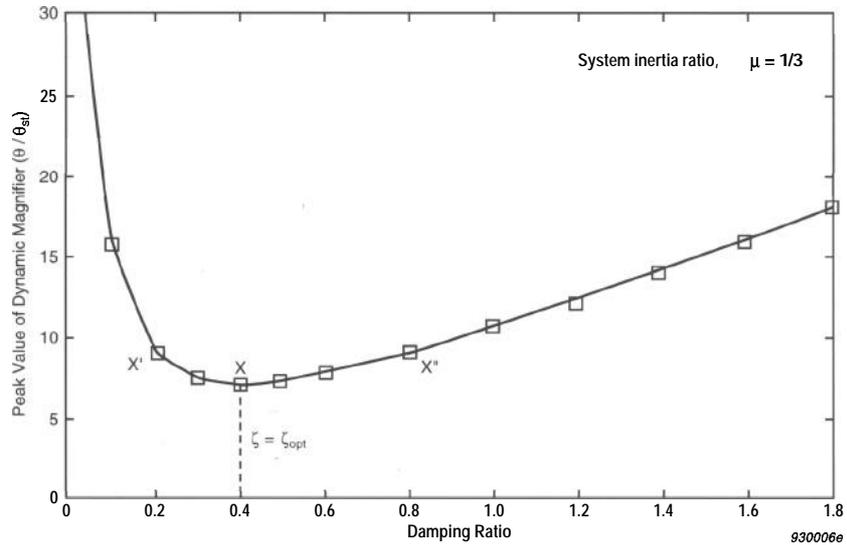


Fig. 5. Viscous damper: effect of damping ratio on peak value of dynamic magnifier (i.e. dynamic magnifier at resonance)

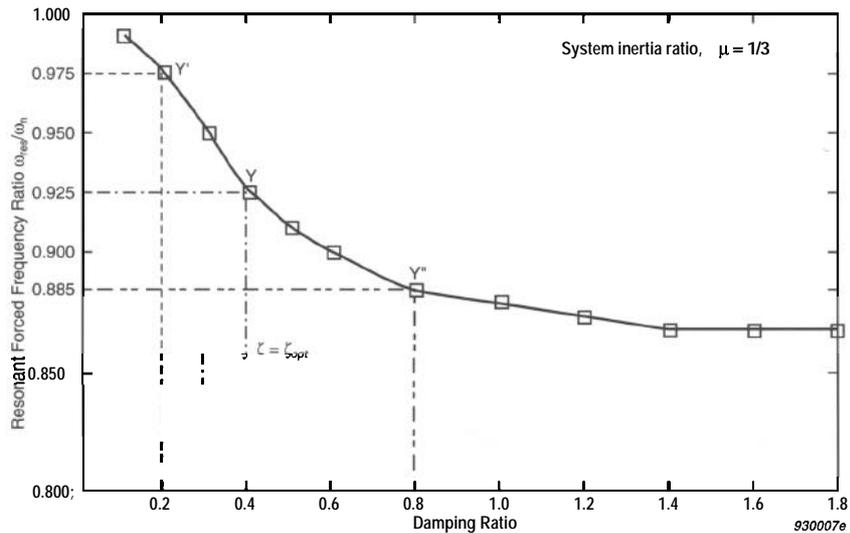


Fig. 6. Viscous damper: effect of damping ratio (ζ) on resonant forced frequency ratio

Examination of Fig. 4 shows that relatively larger changes in the values of the dynamic magnifier with respect to the damping ratio ζ occur in the region of the limiting resonant frequencies of the system ω_n and ω_n' defined earlier. Fig. 7 shows this result. With reference to this figure it is clear that a reduction in damping ratio from the optimum value causes a steep rise in torsional vibration amplitude at an excitation frequency (engine speed) corresponding to ω_n and similarly at ω_n' if an increase in damping ratio occurs. For a 50% decrease and 200% increase in damping ratio, the dynamic magnifier at ω_n and ω_n' increases by 63% and 54% respectively. These results suggest that by using

the Torsional Vibration Meter Type 2523 to measure torsional vibration amplitudes at engine speeds corresponding to ω_n and ω_n' it is possible to provide an effective assessment of damper health. Further to this, the values of ω_n and ω_n' are easily calculated from manufacturers data.

Practical considerations

It is possible to use the Torsional Vibration Meter Type 2523 to produce the complete torsional response curves shown in Fig. 4. This could be achieved by:

- (i) recording torsional vibration response signatures at a number of different engine speeds-Fourier analysing these signatures to pro-

duce the corresponding response spectra and then plotting the amplitude of an isolated order (or orders) from these spectra as a function of engine speed;

- (ii) using a commercially-available tracking filter to "track" a specified torsional order (or orders) over a range of engine speeds automatically. This method is inherently quicker than (i) and produces data at all engine speeds. In order to tune the tracking filter, however, a trigger signal proportional to the average engine speed is required.

In either of the above cases the assessment of condition would rely upon obtaining baseline torsional vibration data at engine "pass off" or some other point in time when the damper was known to be in good condition.

The theoretical considerations have suggested that the time-consuming task of (i) or the necessity for the specialised equipment of (ii) is not necessary and that monitoring the torsional vibration amplitude of a harmonic order at a single engine speed should provide a reliable method of damper health assessment. In practice,

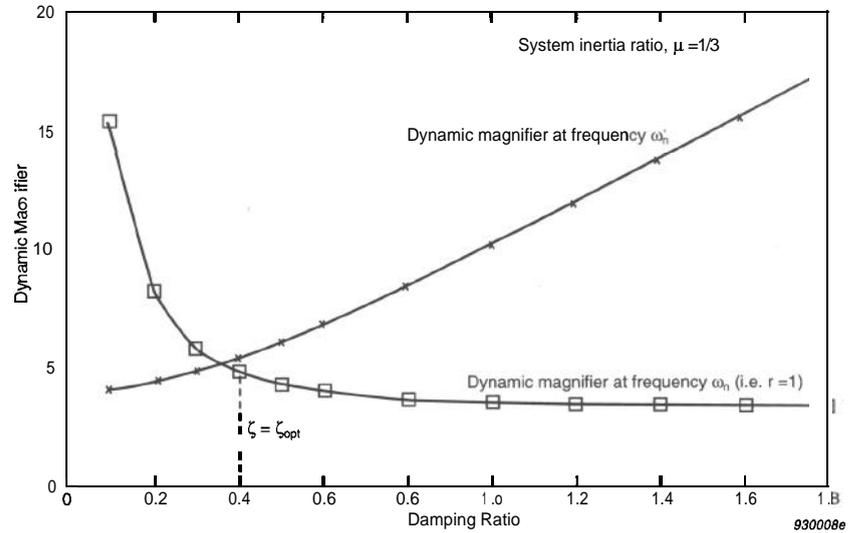


Fig. 7. Viscous damper: effect of damping ratio on dynamic magnifier at frequencies ω_n and ω'_n

the onset of damper degradation is usually caused by a decrease in fluid viscosity and in this case the engine speed of choice is that corresponding to ω_n . If the operating speed range of the engine allows it, however, it is recommended that vibration levels at both ω_n and ω'_n be monitored in order

to confirm the suspected trend in the response curve. In this way the engineer can diagnose damper health by comparing vibration levels of a harmonic order *in situ* with those that were previously taken at engine pass off when the damper was operating optimally.

Experimental Tests

The practicability of the monitoring technique was examined on a 16-cylinder turbocharged diesel installation. It was not possible to conduct a simulated fault analysis in the test since the engine damage risk was too high. However, service fault data for other similar installations was available for comparison. The tests were used to establish the validity of the two-mass model and the accuracy in predicting trends in the response curves.

The accuracy of the two mass model depends on the two following conditions:

- (i) The torsional response of the engine shafting system at the speeds of interest is dominated by the contribution of a single vibration mode.
- (ii) The damping associated with the engine shafting system at the speeds of interest is provided primarily by the viscous damper.

In practice, the engine speeds which are of interest are those defined by the operating speed range of the en-

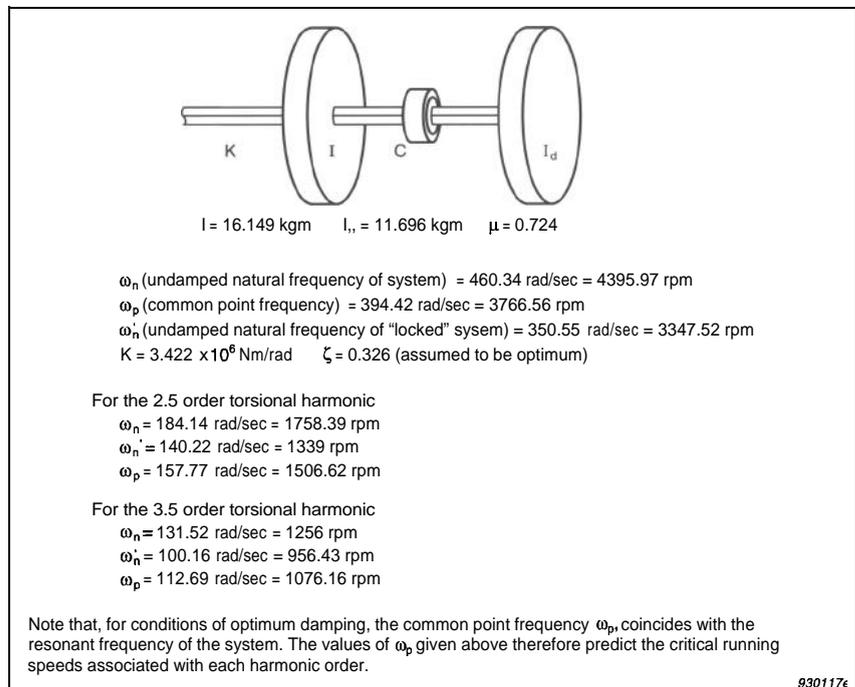


Fig. 8. A two-mass dynamic model of the diesel engine under test

gine. The presence of a viscous damper would therefore suggest that the above requirements will, in general, be satisfied. The torsional damper is introduced to suppress the vibrations of a resonant critical speed lying within or close to the operating speed range and by definition, the torsional resonance is associated with a specific mode of vibration.

Data pertaining to the two-mass model for the engine installation under test are summarised in Fig. 8. Fig. 9 shows the amplitude of the 2.5 and 3.5 torsional orders measured by the Laser Torsional Vibrometer at the damper casing. Also included in Fig. 9 are the corresponding amplitudes as predicted by the two mass model assuming optimum damping. The model correctly predicts the trends in the response curve and identifies the resonance of the 2.5 order critical at an engine speed of approximately 1500 rpm. The predicted resonance of the 3.5 order critical at 1076 rpm was outside the full load operating speed range but is, however, still evidenced in the experimental results taken. The discrepancy between predicted and measured amplitudes is expected and can be attributed to a theoretical over-estimation of harmonic torque magnitudes and to the exclusion of any other sources of damping except the viscous damper.

With reference to Fig. 8, it is clear that a decrease in damping is best detected by measuring the vibration amplitude of the 3.5 order critical at 1256 rpm (ω_n). Similarly an increase in damping is best detected by measuring the vibrational amplitude of the 2.5 order critical at 1339 rpm (ω_n'). The measured levels would be compared against historical baseline data.

It should be noted that, in practice, the advantage of the in situ measurement provided by the Torsional Vibration Meter Type 2523 will allow rapid assessment at other engine

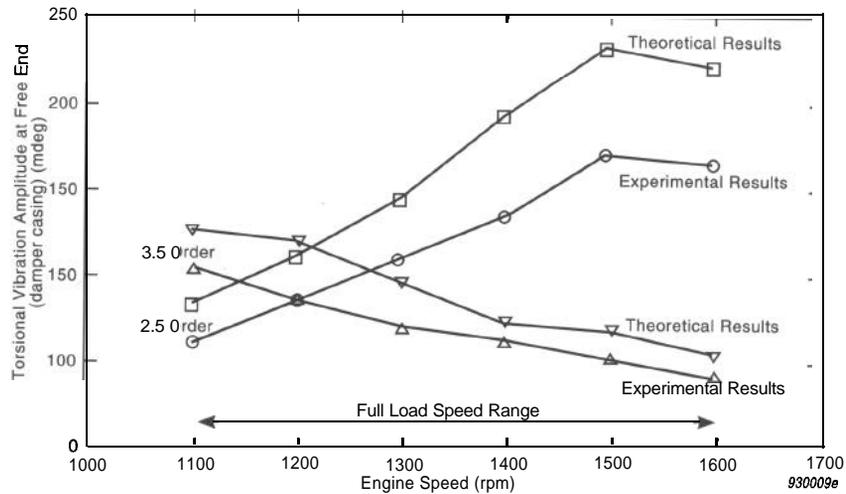


Fig. 9. Comparison of theoretical and measured 2.5 and 3.5 order response curves

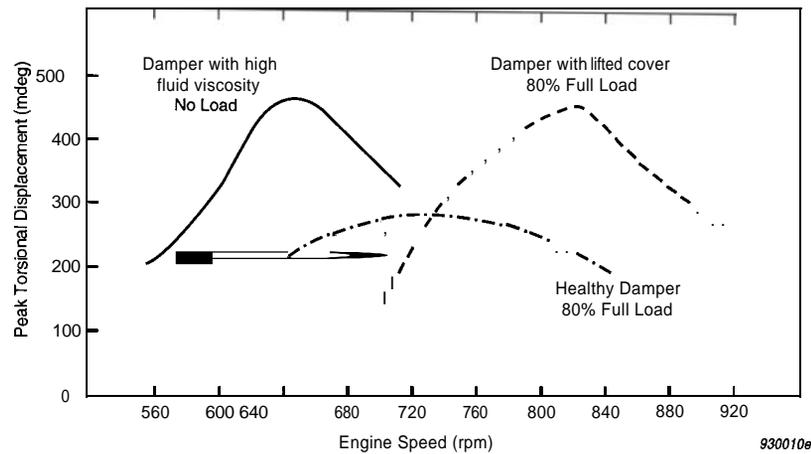


Fig. 10. The effect of viscous damper faults on the 2.5 order response curve of a 16-cylinder "V" engine driving a generator (derived from service failure records)

speeds within the operating range in order to confirm suspected trends in the response curve.

Fig. 10 shows the changes in the 2.5 order response curves taken from available service failure data for a 16-cylinder "V" engine driving a generator set. The trends in behaviour due to

the changes in damping provided by the torsional damper are correctly predicted by the two-mass model. Further to this the amplitude of vibration measured is well within the dynamic range of the Torsional Vibration Meter Type 2523.

Conclusions

"Critical" engine speeds occur in large marine propulsion diesels where a harmonic of the torque exerted on the crankshaft system coincides with a torsional natural frequency. Large torsional vibration amplitudes of the crankshaft, which would lead to fatigue failure under such conditions, are suppressed by the use of torsional dampers. Viscous shear torsional

dampers are fitted to larger marine diesels and their performance is dictated by the viscosity of the fluid used in the device. Current monitoring methods for damper performance require sampling the fluid for laboratory tests. This is laborious, expensive, and requires a long period of engine downtime.

The application of modern optics

and laser technology has produced a novel means of assessing damper health without these problems. This has been achieved by using the Torsional Vibration Meter Type 2523 to measure, in situ, torsional vibrations of the damper casing. Theoretical modelling of torsional damper behaviour has suggested several means of assessing damper performance based

on these measurements. One possible method is to monitor the response amplitude of an isolated torsional order over a sufficient range of engine speeds to resolve the location and amplitude of a critical speed (cf. resonance). This response curve would then be compared with the equivalent which was taken at engine pass-off when the damper was performing optimally. The method, however, re-

quired tracking the specified order during engine shutdown and involved additional dedicated hardware.

A superior cost effective method is to monitor the amplitude of the torsional order at either of the two limiting values of its critical speed. These engine speeds correspond to resonances for zero and infinite damping and produce a more sensitive measure of performance. Further to

this they are easily calculated from engine manufacturers data. In this way torsional damper health can be assessed by comparison to two values of torsional vibration amplitude of a specified harmonic order: one taken in situ and the other recorded at engine pass-off when the damper is performing optimally.

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