Application Note

Run-up Order Analysis of Axial Vibrations in a 2190 kW, MAN B&W, Marine Engine

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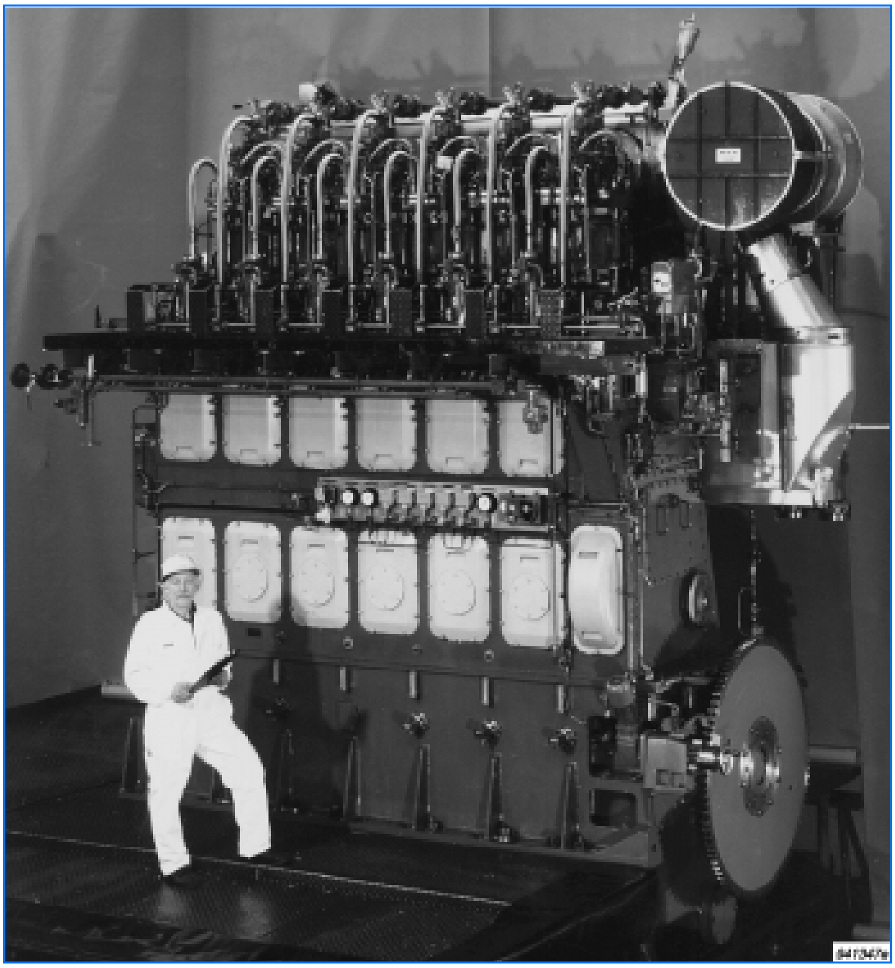
In operation, the propulsion shafting of a ship is deformed axially and torsionally. Both deformations cause axial vibrations, which at critical RPM may become unacceptable. The vibrations are damped by an integral axial vibration damper. The critical RPM are calculated based on a model of the shafting, and measurements are performed to verify the calculated natural frequencies and the effect of the damper. The verification is required by some of the shipbuilding classification societies. It is demonstrated that Type 3555 fitted with Extended Analysis Software Type 7639 is suited for the measurements.



Introduction

When designing marine engines, vibration control is of great importance. Excessive vibration levels may damage the engine/ship and create an annoying human environment.

The propulsion shafting, i.e. the crankshaft, the propeller shaft and the propeller are subject to dynamic torsional deformation as well as dynamic axial displacement, both causing axial vibration. In the design of the propulsion shafting, the vibration caused by the torsional deformation is controlled by tuning the torsional natural frequency (1-node) to an appropriate cpm (cycles per minute). The vibration caused by the dynamic displacement of the propulsion shafting is controlled by the presence of a viscous axial vibration damper (A/V damper). Apart from damping the displacement vibration completely, the A/V damper also dampens the axial vibration caused by the torsional deformation.

The engine under test was designed by MAN B&W Diesel A/S, Denmark. The verification test of the design has also been conducted by MAN B&W using an analysis system developed by them. By courtesy of MAN B&W the test data, recorded on an FM tape recorder, was made available to Brüel & Kjær to show how the analysis can be carried out using the Brüel & Kjær Dual-channel Analysis System Type 3555. The verification test comprises axial as well as torsional vibration measurements. Only the axial vibration measurements are presented here.

Propulsion Shafting Model and Measurement Set-up

The engine is a MAN B&W 6S26MC, a six cylinder, two stroke, 2190 kW engine with a maximum continuous RPM of 250. Due to the alternating compression/power strokes and the load of the propeller, the propulsion shafting is subject to dynamic torsional deformation. The torsional deformation causes changes in the length of the crankshaft which is seen as axial vibration at the free end of the crankshaft. See Fig. 1. The torsional vibration also causes the propeller to rotate with varying speed, which in turn gives a varying thrust. The varying thrust excites the propulsion shafting axially, which also causes axial vibration to be seen at the free end of the crankshaft. The modal model of the propulsion shafting is a rather complex model comprising torsional and axial DOFs. The model predicts that the coupled 1-node torsional mode and the 0-node axial mode, when excited, are the major contributors to the axial vibration.

The torsional 1-node natural frequency was tuned to 847.5 cpm by proper selection of length and diameter of the propeller shaft, and by introduction of a tuning wheel. The resulting mass made the calculated axial, 0-node, natural frequency appear at 992.6 cpm.

Since the engine is a 6 cylinder, 2 stroke engine, the critical speeds are expected to be the speeds where the 6th order of the speed coincides with the natural frequencies. Other orders excite the natural frequencies at other speeds, but the 6th order excitations are expected to be predominant. Table 1 presents the calculated and measured critical speeds and displacements for the two important natural frequencies.

The free end of the crankshaft is fitted with a coupling flange. The deflection signal together with a tacho signal (1 per revolution) were measured on this flange. The measurement was performed in situ, i.e. the engine was in full operation at sea. Two measurements were made. One with the damper disabled and another with the damper active. Over a 25 minute period of time the speed was gradually increased from 100 RPM to the maximum continuous 250 RPM, at all times with maximal propeller pitch corresponding to maximal thrust. The tacho and displacement signals were recorded on an FM tape recorder for later analysis.

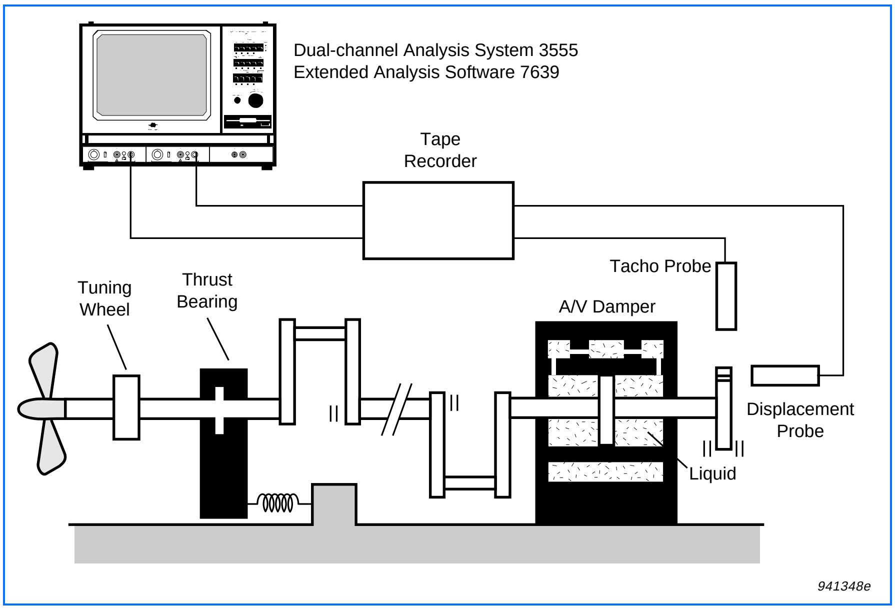


Fig. 1 Propulsion shafting model and measurement set-up

A/V-damper	Passive		Active		
	Calculated	**Measured**	**Calculated**	**Measured**	**Units**
1-node torsional nat. freq. 6th order critical speed 6th order A/V amplitude	847.5 141.3 2.19	861.0* 143.5 1.53	847.5 141.3 0.40	852.0* 142.8 0.52	cpm RPM mm
0-node axial natural freq. 6th order critical speed 6th order A/V amplitude	992.6 165.4 1.99	1009.2 168.2 1.23	no no no	no no no	cpm RPM mm
Max. continuous speed 6th order A/V amplitude	250.0 0.13	250.0 0.13	250.0 0.10	250.0 0.12	RPM mm

*The measurement is not a direct measurement of the torsional natural frequency, but a measurement of the axial deformation caused by the torsional deformation. This can explain the deviations in the results

Table 1 Calculated and measured critical RPM and A/V amplitude

Analysis using Dual-channel Analysis System Type 3555

The subject of the analysis is to determine the critical speeds of the engine, and to measure the corresponding worst case deflection of the free end of the crankshaft. Since the excitation of the propulsion shafting is a function of the rotating shaft itself, order analysis (no smearing and no leakage) is the tool that gives the best determination of the critical frequencies and the max. deflection. The critical frequencies are found by inspecting the dynamic displacement pictured as a function of RPM for the various orders, also called order slices. Adding Extended Analysis Software Type 7639 to the 3555 system, the system becomes capable of performing multifunction order analysis.

Referring to Fig. 2 showing the measurement set-up of the 3555, the analysis is carried out as described below.

From the tape recorder, the tacho signal is connected to channel A and the displacement signal to channel B. The sensitivity of the displacement transducer is entered in the transducer set-up, 1.03 mm/V.

The MULTIfunction is selected to be AUTOSPEC CH.B, and from 100 RPM to 253 RPM a record is registered in the multibuffer whenever the RPM has increased by 0.162 RPM. Since critical speed and maximum displacement are the objectives, the records should be as short and as many as possible. The update criteria 0.162 RPM is the smallest possible increment for the wanted RPM range corresponding to the maximum number of records in the multibuffer. The selection of 50 lines spectra and 1 revolution per record give the shortest record obtainable. The analysis results in a multibuffer containing 947 order spectra as a function of RPM, each spectrum being the maximum displacement as a function of orders from 1 to 50. The averaging option PEAK is selected to register the maximum value seen by the individual records in the multibuffer.

Displaying the Results

Using the 3555 display option M-MAP, it is possible to create a map (waterfall) covering up to 80 records of the multibuffer. By panning through the whole multibuffer it is possible to get an overview of the complete analysis.

Looking first at the analysis of the run-up with the A/V-damper disabled: As expected, the 6th order excites the natural frequencies most seriously. Fig. 2 shows the 6th order SLICE where two critical speeds are identified. 143.5 RPM excites the 1-node torsional natural frequency, and 168.2 RPM excites the 0-node axial natural frequency. The deflection is scaled to RMS. By defining a very simple User-defined Auxiliary Information (UDAI) named AV_AMPLITUDE, the cursor reading is also given as peak millimetres. The measured values are given in Table 1 for comparison with the calculated values.

Fig. 2 also shows other order slices. The 4th order excites the two natural frequencies at 215 RPM and 253 RPM, respectively, and the 7th order excites the axial natural frequency at 144 RPM. The corresponding resonance displacements are considerably smaller than those caused by the 6th order excitation. The remaining orders, not shown, hardly excite the natural frequencies.

Fig. 3 shows the result of the analysis with the A/V damper activated. The figure does not show a particular order slice but a Δ-slice containing the PEAK value of all the orders from 1 to 50. It is seen that maximum deflection still is at the 1-node torsional natural frequency, but that has been considerably damped. Inspecting the orders one by one, it can be seen that the axial resonance has been effectively damped. The maximum deflection is acceptable and the damper works as expected.

It is worth mentioning, that the peak in the order slice does not necessarily coincide with the critical speed. A ship at sea is subject to various influences. The experienced analyst determines the critical speeds by performing some kind of curve fitting to the order slices.

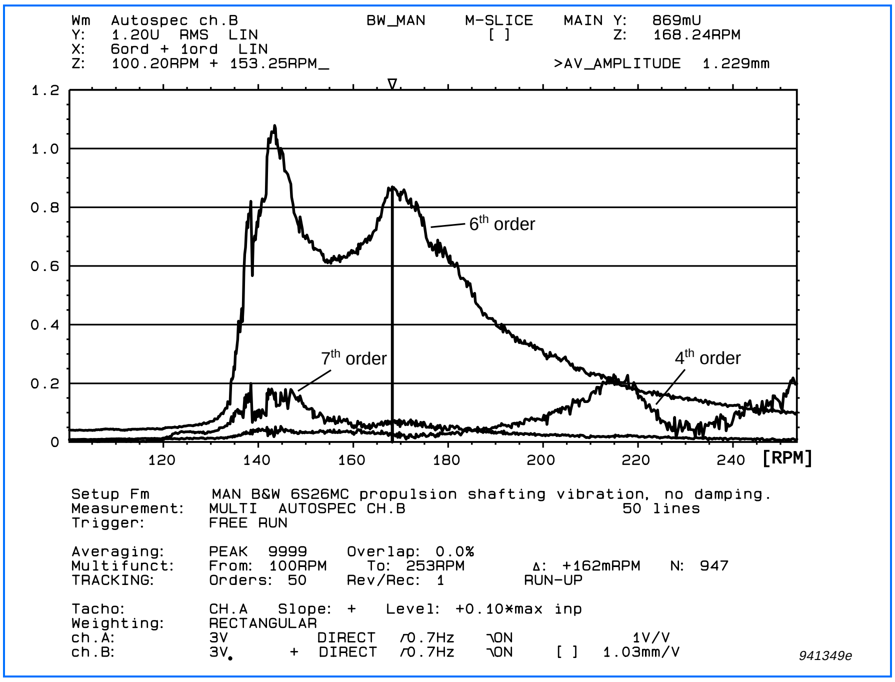


Fig. 2 Measurement set-up and analysis of undamped vibrations

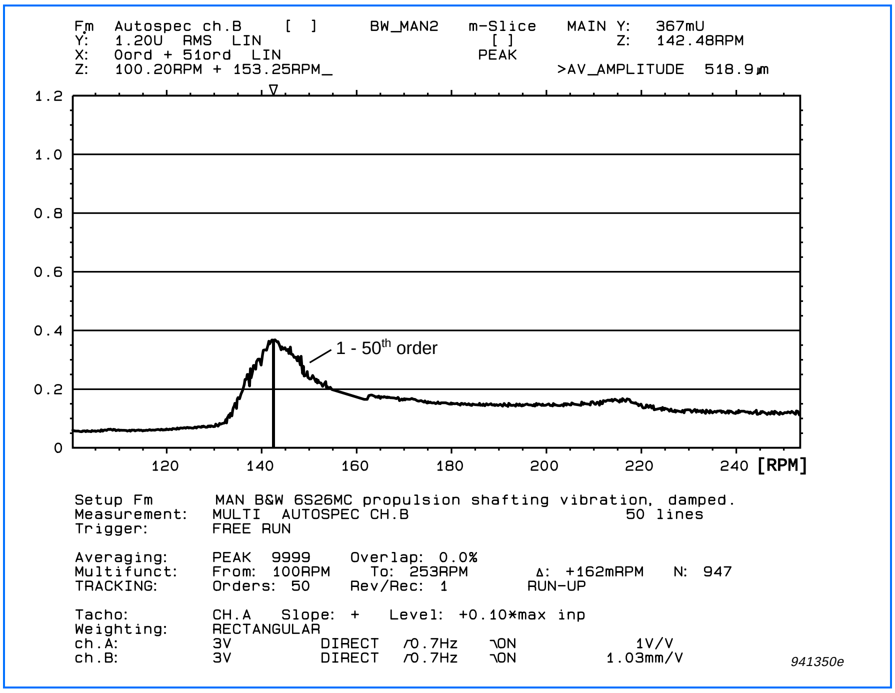


Fig. 3 Analysis of damped vibrations

Related Analysis

In the test described above, the PEAK value of the AUTOSPECTRUM CH. B was addressed. Another option is the COMPLEX SPECTRUM CH.B. This spectrum is the autospectrum of channel B with the phase between the tacho signal and the signal at channel B assigned to it. This option gives the possibility of showing the order slice as a Bode Plot. The Bode Plot offers a very accurate determination of natural frequencies (if the natural frequencies are adequately separated). Also this option opens for determination of Operational Deflection Modes (ODS), order or frequency related.

Using a 3550 multichannel configuration it is possible to measure order slices of two channel functions, for example the Frequency Response Function (FRF).

When PEAK is not the wanted value, it may be of interest to perform linear or exponential averaging to improve the signal/noise ratio. The 3550 system alone offers averaging in the Multifunction Mode.

The verification test as performed by MAN B & W also comprised the time domain function (max. deflection – min. deflection)/2 as a function of RPM. Using UDMF (User-defined MultiFunction) this function or similar functions may be implemented.

Brüel & Kjær Equipment

For tests of this kind the following Brüel & Kjær equipment is suitable:

Tacho Probe:
Photoelectric Probe MM 0024

Transducers:
Any suitable accelerometer, or
Magnetic Transducer MM 0002 (velocity), or
Laser Transducer Set Type 3544 (velocity), or
Capacitive Transducer (displacement).

At low frequencies velocity and displacement transducers are preferred due to better S/N, though large displacements may compromise the use of magnetic and capacitive transducers.

Tape recorder:
Sony PC 208 equipped with Type 5963 Eight-channel DeltaTron™ Accelerometer Supply. Accommodates the tacho probe MM 0024.

Analyzer:
Dual-channel Analysis System Type 3555 with Extended Analysis Software Type 7639. Or higher grade of Multichannel Analysis System Type 3550.

Conclusion

To ensure safe and comfortable operation, the design of the propulsion shafting of a ship must be verified by measurements. Among other properties the natural frequencies and the critical speeds of the shafting should be verified. Also the associated maximal vibration levels, and the effect of an axial vibration damper should be verified. The tool for these verifications is order analysis of the axial vibration of the shafting. The verification test was made by MAN B & W Diesel A/S, and it is shown that Dual-channel Analyzer System Type 3555 fitted with Extended Analysis Software Type 7639 can accomplish the test. By adding User-defined Functions, Type 3555 can accomplish other tests which have a bearing on run-up/down measurements.

Bibliography

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Type 3550 Technical Documentation Vol. 1: Guided tours
— Tour 5: Multifunction
— Tour 6: Tracking

Brüel & Kjær

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