ENHANCED TORSIONAL VIBRATION MODEL VERIFICATION BY MEANS OF CYLINDER PRESSURE MEASUREMENTS

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Abstract

The accuracy of a torsional vibration calculation (TVC) is mainly influenced by the correctness of input parameters such as inertia, stiffness, gas excitations and damping. Due to the uncertainty of the TVC, classification societies require a torsional vibration measurement (TVM) on sea trial for verification. On two-stroke direct driven propulsion systems, the TVM is normally done by measuring the crankshaft angular velocity or by strain gauge measurement on the intermediate shaft. All new two-stroke engines from MAN Energy Solutions (MAN ES) are equipped with PMI, a system capable of recording the angular velocity and the cylinder pressure for all cylinders. In this paper we have studied different methods for improving the TVC model by removing the uncertainty of the major excitation source, the gas excitations. One of the ideas we have studied is normalization of the measured angular amplitude harmonics by the corresponding measured tangential pressure excitations to eliminate the excitation variation. The normalized measured figures are compared with calculations of the torsional vibration for unity harmonic excitations. The normalized measured torsional vibration amplitudes appear much smoother as a function of speed and are therefore easier to compare with calculations. Furthermore, the modal analysis methods applied on the normalized data can extract the natural frequency and system damping with good precision. This may potentially be an enhanced method for torsional vibration model verification.

INTRODUCTION

Classification societies require TVM on direct coupled two-stroke engines to be carried out on the sea trial of the first vessel in a series to verify the safety of the propulsion shafting. It is common to base the measurement on the angular crankshaft vibration only rather than measuring stress directly at the propulsion shafting by using strain gauges. The stress amplitudes can be determined indirectly by means of the TVC. However, the quality of the method is influenced by how well the TVC matches the measurement in terms of natural frequencies, damping, propeller curve and so on.

In this paper we describe a method that removes the uncertainty of the gas excitations and facilitates the identification of natural frequencies and system damping, which are essential for the estimation of the propulsion shafting torsional stress. Furthermore, all that is needed for measuring is the PMI-system installed on all new MAN B&W two-stroke diesel engines. The main purpose of the PMI-system is cylinder pressure monitoring and Adaptive Cylinder Control (ACCo).

BACKGROUND

It is challenging to correlate the measured crankshaft angular vibration to 1-node torsional stress in the shafting in the barred speed range (BSR). Usually, this is done in the TVC, possibly by adjusting propeller inertia and damping to fit the TVM. In some cases system tuning may introduce an uncertainty since the exact natural frequency and damping may be difficult to extract from the TVM. Furthermore, the actual gas excitation in the cylinders may deviate as a consequence of variations in engine load caused by for instance sea condition, navigation, draft and propeller light running. In addition the parameters influencing the gas excitations can be defined differently within the BSR. Finally, the engine speed may be instable at the main critical frequency.

The measurements presented in the paper are based on a continuous PMI recording during a slowly and continuously run-up or run-down of engine revolutions. A true steady-state ship speed is consequently not obtained. The PMI-system measures the cylinder pressure together with the corresponding crank angle time stamps with 0.5 degree resolution. These measurements are converted to torque excitation on the crankshaft for each cylinder and to angular velocity (including torsional vibration) of the crankshaft free end. Then the data is frequency analysed using an order tracking technique. Initially the measured cylinder pressures were used to verify the predicted gas excitations used for the TVC. Furthermore, it is also possible to generate a gas-harmonic table based on the measurement for improved correlation between TVC and TVM.

ANALYSIS

One of the ideas we have studied is normalization of the measured angular amplitude harmonics by the corresponding measured tangential pressure to eliminate the excitation variation. We experienced that the measured torsional vibration amplitudes appear much smoother as a function of speed when normalized, see figures 1 and 2.

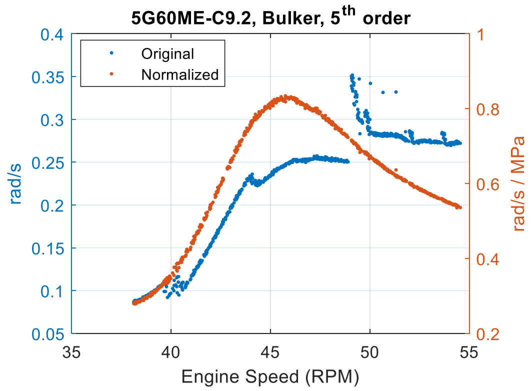

Figure 1 – Measured 5th order angular velocity, original and normalized data.

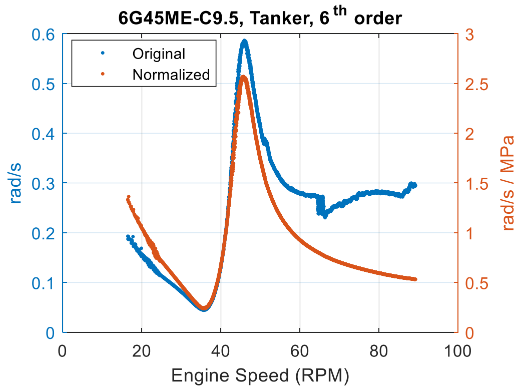

Figure 2 – Measured 6th order angular velocity, original and normalized data.

To compare the normalized measured data with TVC, unity harmonics must be applied. In figure 3, a comparison is shown with unity gas-harmonics consisting of 1 MPa tangential excitation for all orders at all engine loads.

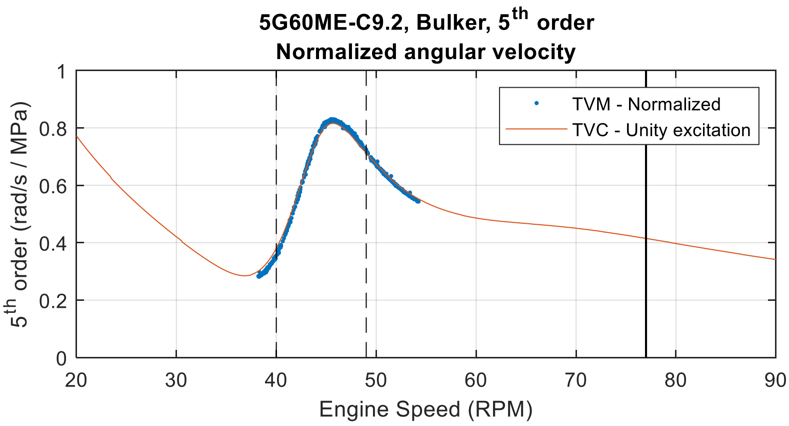

Figure 3 – 5G60ME-C with a spring type TVD. Comparison of the normalized measured crankshaft free end 5th order angular vibration with the TVC.

The tangential pressure used for normalization is based on the average tangential pressure of all cylinders. The pressure is measured on each cylinder during one full revolution. Hereby the mean excitation correlates with the mean angular vibration of each revolution. Depending on the vibration mode, the cylinder pressure unbalance changes the excitation pattern but the effect on the 1-node is limited. In any case, the previously mentioned disturbances, such as variation in engine load, are nearly eliminated and the variation of the excitation is removed. Thus, the natural frequency and damping can be evaluated more easily. The BSR is characterized by high resonant torsional stress in the propulsion shafting, which is of special interest.

MODAL ANALYSIS

When having both torsional excitation and response it is obvious to consider modal analysis to extract the natural frequency and damping figures. However, the Multi-Input-Single-Output (MISO) of our Multi-Degree-Of-Freedom (MDOF) system has no immediate modal solution. Our idea was to study if modal analysis solvers could estimate the modal parameters from the normalized angular vibration, which in modal terms is a transfer Frequency Response Function (FRF). The FRF is the measured angular vibration (rad) divided by the excitation torque (Nm) in the frequency domain, also denoted as the receptance. The excitation torque is obtained from the cylinder gas pressure and the oscillating mass with the crankshaft mechanism factor applied. The mass forces diminish with harmonic number. The MISO can be approximated as a Single-Input-Single-Output system (SISO) by adding together the excitations taking the MDOF modes into account. When only considering the natural frequency and the damping, and not the absolute modal constant, the effects of unequal gas excitations and MDOF modes can be ignored. The lumped excitation we use for the normalization is then simplified to the sum of the cylinder excitations.

Presently, we have studied 0- and 1-node torsional vibrations of 5- and 6-cylinder propulsion systems for 5th and 6th order, respectively, with the goal to estimate the torsional stress in the BSR. The study covers engines equipped with either a spring type torsional vibration damper (TVD), a viscous type TVD or a tuning wheel. As proof of concept we have validated the method on data from TVC and confirmed a very good agreement even with application of different propeller damping models. A modal analysis of a few real measurement data is presented in figures 4-6. In two of the cases the corresponding strain gauge measurements on the shafting were available, see the next chapter.

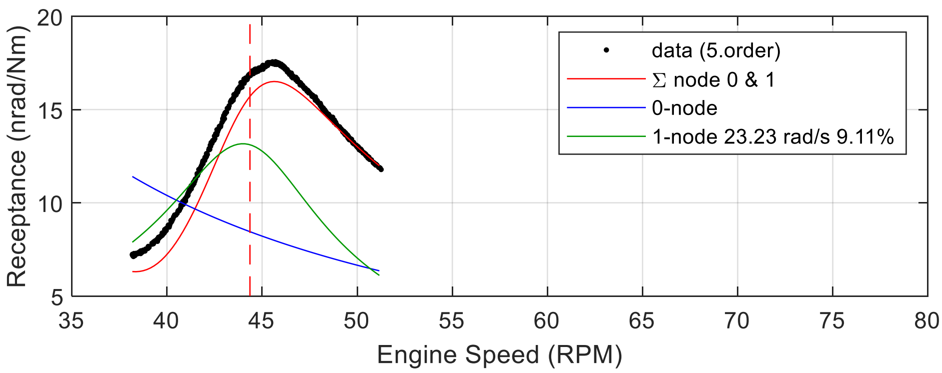

Figure 4 – 5G60ME-C with a spring type TVD. Comparison of the measured FRF for the crankshaft free end 5th order angular vibration with the result of modal analysis. BSR 40~49 RPM

The example in figure 4 is an analysis of measured data from an engine with a spring type TVD. The 1-node undamped natural frequency is estimated to 23.23 rad/s (red dashed line) and the viscous system damping coefficient to 9.11%. The superposition of the estimated 0- and 1-nodes is seen to deviate a little from the original measured data, perhaps because the damper mode expected at 35 rad/s is not taken into account. An improvement of the measurement could be to measure in the full speed range instead of just covering the BSR.

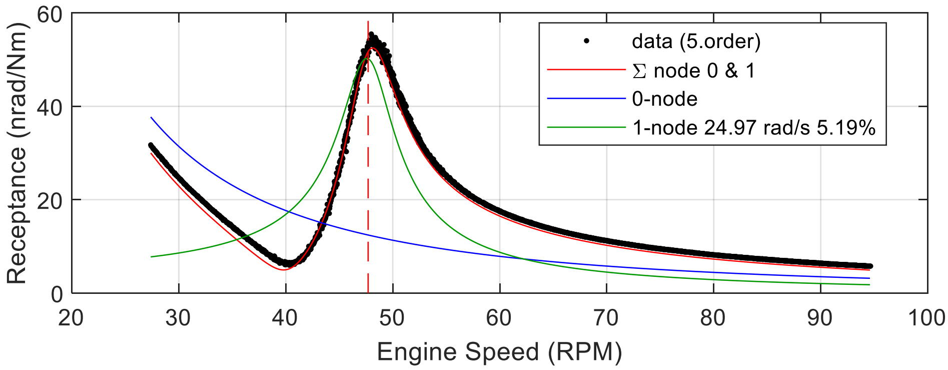

Figure 5 – 5G50ME-C with a viscous type TVD. Comparison of the measured FRF for the crankshaft free end 5th order angular vibration with the result of modal analysis. BSR 41-54 RPM.

The example in figure 5 is an analysis of measured data from an engine with a viscous type TVD. The 1-node undamped natural frequency is estimated to 24.97 rad/s (red dashed line) and the viscous system damping coefficient to 5.19%. The superposition of the estimated 0- and 1-nodes is seen to comply well with the original measured data. The 2-node is expected at a much higher frequency.

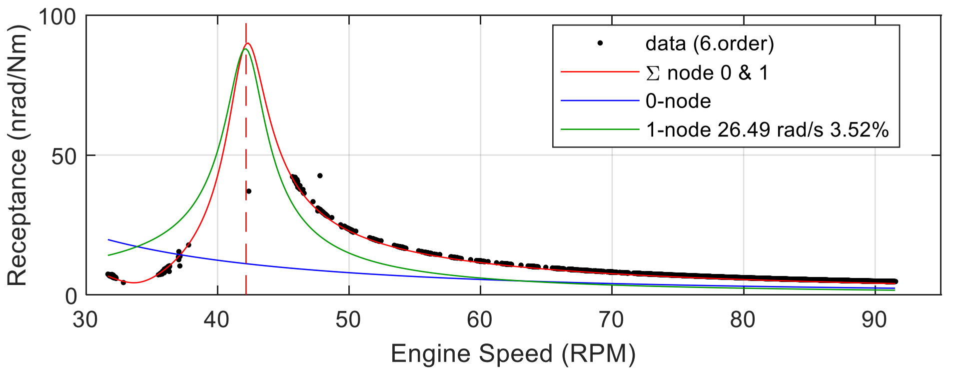

Figure 6 – 6G50ME-C with a tuning wheel. Comparison of the measured FRF for the crankshaft free end 6th order angular vibration with the result of modal analysis. Even without data in the BSR 37.5-46 RPM a reasonable estimation is obtained.

The example in figure 6 is an analysis of measured data from an engine with tuning wheel. The 1-node undamped natural frequency is estimated to 26.49 rad/s (red dashed line) and the viscous system damping coefficient to 3.52%. The superposition of the estimated 0- and 1-nodes is seen to comply well with the original measured data even though there were no measured data in the BSR that was passed quickly. The 2-node is expected at a much higher frequency.

The derived modal model can be applied to other measurements on the same system or to a simulation, provided the conditions are not changing the propeller damping and the position of the resonances. As an example the modal model must be evaluated separately for a CP-propeller in full and zero pitch.

The described modal analysis can also be used for studies of the in general uncertain internal engine damping in combination with propeller damping measurements [1].

ESTIMATION OF TORSIONAL STRESS IN SHAFTING

A single factor relationship between the displacement amplitude and the stress or torque anywhere in the lumped mass-elastic system can be found in the Holzer tabulation of each vibration mode, see figure 7. We use the ratio between the torque amplitude in the intermediate shaft and the angular displacement amplitude at the crankshaft fore end. In the paper this ratio is denoted as the Holzer factor.

Modal form no. 1,   Cyclic frequency =   26.028 rad/s ,

ID no	Mass name	Angular deflection (rad)	Shaft torque (kNm)
33	Tuning wheel	1.0211E+00	
38	Flange	1.0211E+00	-2.2481E+04
1	Cylinder	1.0000E+00	-2.2661E+04
2	Cylinder	9.7019E-01	-2.8018E+04
3	Cylinder	9.3535E-01	-3.3215E+04
4	Cylinder	8.9380E-01	-3.8226E+04
5	Cylinder	8.4846E-01	-4.3013E+04
6	Cylinder	8.0071E-01	-4.7558E+04
27	Camdrive+Thrust	7.6525E-01	-5.1848E+04
32	Turning Wheel	7.3768E-01	-5.2707E+04
44	Flange	-1.5070E-01	-5.4606E+04
45	Flange	-1.0389E+00	-5.4597E+04
73	Main Propeller	-1.7009E+00	-5.4473E+04

Figure 7 – Example from a GTORSI TVC of a 1-node mode where the Holzer factor is -5.4606E+04 kNm divided by 1.0211 radians.

Theoretically, the shaft torque is given by a superposition of modes found by multiplying the Holzer factor for each mode with the corresponding Single-Degree-Of-Freedom system (SDOF) angular amplitude. In the analysis, the SDOF amplitude is found by the FRF multiplied with the excitation torque we used for the normalization. The frequency separation between the propulsion shafting 1- and 2-nodes is usually sufficiently large to disregard the 2-node. In addition the damper mode is usually highly damped if a spring type TVD is used and can also be disregarded. Furthermore, the influence from these modes in the shafting is low. Thus, only the 1-node SDOF needs to be considered. This presumption can be verified with TVC data. The estimation is expected to be good in the vicinity of the 1-node resonance where the influence from higher modes is insignificant. The advantage of estimating the shafting stress in this way is that disturbances such as excitation variation are taken into account, and the result can in principle be verified directly by the corresponding strain gauge measurements. The best possible Holzer factor should be taken from an updated TVC considering the findings.

The two examples in figures 8 and 9 show reasonable good agreement with the corresponding strain gauge measurements on the intermediate shaft. The accuracy of the torque measured with strain gauges is influenced by the accuracy of measuring strain and the G-modulus shaft material property. Since it is not practically possible to calibrate a torque measurement on a propulsion shafting, the actual precision is unknown but assumed to be within few percent.

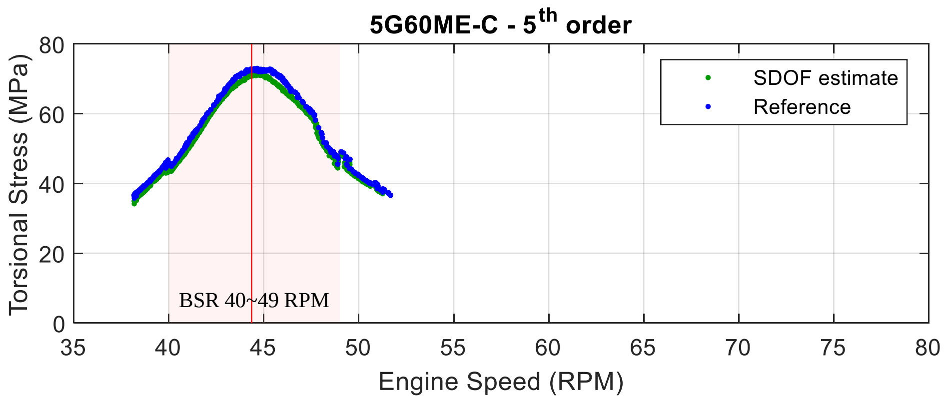

Figure 8 – 5G60ME-C with a spring type TVD. 5th order torsional stress in the intermediate shaft. The SDOF estimate shows good agreement (<2%) with the reference data. Reference data is measured on the intermediate shaft with strain gauges.

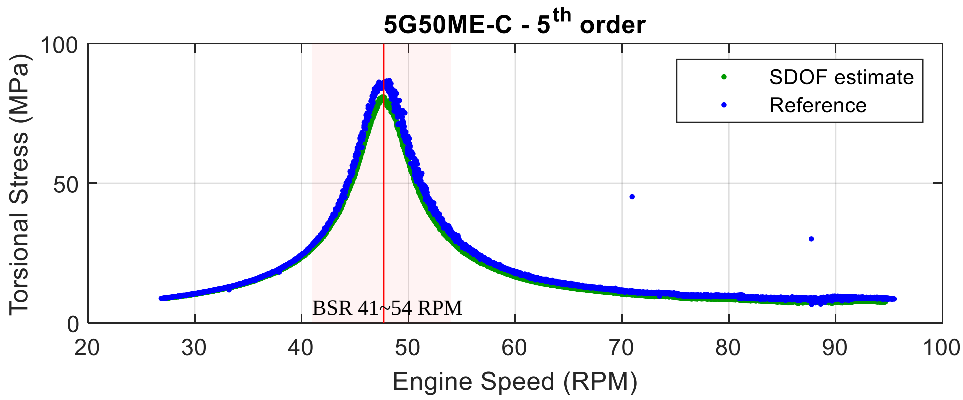

Figure 9 – 5G50ME-C with a viscous type TVD. 5th order torsional stress in the intermediate shaft. The SDOF estimate shows acceptable agreement (~6%) with the reference data. Reference data is measured on the intermediate shaft with strain gauges

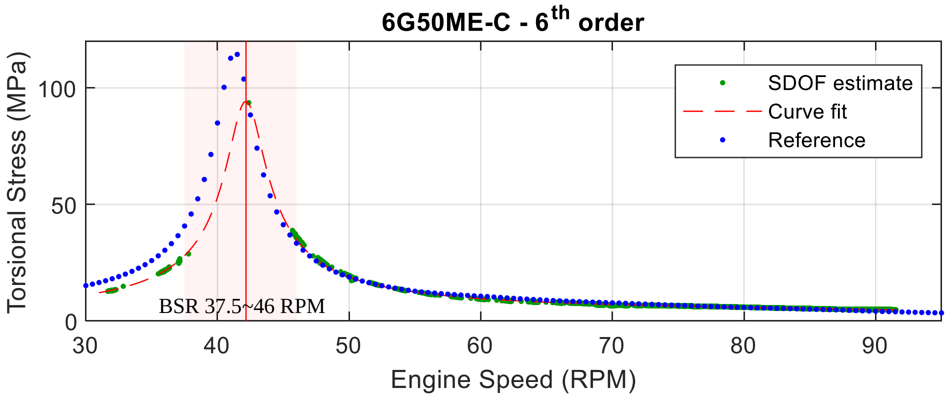

Figure 10 – 6G50ME-C with tuning wheel. 6th order torsional stress in the intermediate shaft. The reference is the original TVC. There is no measured data in the BSR, but the SDOF estimate indicates that the 1-node natural frequency is higher than originally calculated. Furthermore, the TVC should be tuned to improve the Holzer factor before the final SDOF estimate. The curve fit is the SDOF response based on the curve fitted measured excitation.

Even without measured data in the BSR, it appears to be possible to estimate the probable stress in the BSR, see figures 6 and 10.

SUMMARY

The paper proposes a method to enhance the torsional vibration model verification using a normalization of the measured torsional vibration by figures obtained from the cylinder pressure. The normalized amplitudes are nearly uninfluenced by disturbances from the gas excitation and thus simply the iterative process of correcting the TVC.

Modal analysis can also be applied to obtain the natural frequencies and the system damping for at least the 1-node mode. Furthermore, from the 1-node SDOF model and the Holtzer factor from the TVC, the torsional stress in the propulsion shafting can be estimated with good accuracy within the important BSR. Measurements can easily be obtained even from ships in service and that could be useful for obtaining knowledge for the torsional vibration layout of future newbuildings.

REFERENCES

[1] Orthmann P., *Measurement of Hydrodynamic Moment of Inertia and Damping of Propellers,* Torsional Vibration Symposium (2017).