Damping and excitation in the torsional vibrations calculation of ship propulsion systems

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Abstract. Calculation of torsional vibrations is essential in the early phase of the design of any ship propulsion system, after selection of shafting diameters in accordance with the Classification Rules. Later on, during ship trials, the calculation shall be validated by measurements on board. The calculation results depend upon inertial moments of actual masses, stiffness of shafting components, damping in the actual shafting components, as well as the excitation forces and moments exerted by the propulsion engine(s) and the propeller. Inertial moments and stiffness can be determined with no ambiguities. However, this is not the case with either the damping, or the engine excitation. Actually, during validation on board the calculation supposed damping is the main influential factor to be verified. The aim of this paper is to present and compare several models to define and compare damping definitions in the the torsional vibrations calculation (Frahm's model, Archer's model, physical damping, magnification factor, etc.) in a systematic way, to enable designers to correctly apply the selected damping model. Further on the engine excitation may be expressed by means of cylinder pressures or tangential forces to the cranks. The essentials of the two models and the procedure to convert one excitation model into another one are also presented. The application of the presented damping and excitation models is presented on an actual ship propulsion system and conclusions drawn.

Key words: *marine shafting design, steady-state torsional vibration analysis, SimulationX, validation by on-board measurements, acceptance criteria*

1. Introduction

Designer of any ship propulsion system has the main goal from the very beginning: to select the propeller that enables ship to achieve contracted speed for the given ship hull form, as well as to select the proper main propulsion prime mover (e.g. Diesel engine, steam or gas turbine plant and reduction gearbox) able to produce and transmit power to the main propulsion shafting.

The next step is determination of design form, dimensions, material and service loading of the shafting itself. Preliminary dimensions, i.e. external and internal diameters of particular shafts may be easily determined on the basis of MCR power, relevant rotational speed and mechanical properties of the selected material by implementing classification Rules. These classification Rules are generally based upon IACS Unified Requirement UR M68, comprising simple formulae applied to the calculation of these diameters.

However, in this very first design phase it is very important do determine the shafting steady state response to the engine and propeller variable torque excitation around the shafting axis, i.e. torsional vibrations response. It is a difficult task in this phase, because the entire shafting system has not been completely defined yet. Unfortunately, in case of improper design in this phase, there is not much that can be done in later phases, other than providing and installing the torsional vibration damper. For this reason, proper calculation of torsional vibration is necessary in the initial phase of the marine shafting design.

In general, torsional vibrational response of shafting depends also upon its design form, dimensions, material and service loading. The most appropriate model for the analysis of shafting system torsional vibrations is the model with lumped masses (represented by their mass moments of inertia around the shafting axes), massless shafts (representing stiffness and damping of parts of the system) and engine loading.

Considering steady-state response in terms of angular, torque and stress amplitudes for various shafting rotational speeds in the operational speed range the particular necessary data to be provided are the mass moments of inertia for each concentrated mass, the torsional stiffness of shafts, structural damping in the shafts, damping of propellers, flexible couplings and torques due to cylinder pressures and inertial forces of the reciprocating parts of engine systems for a single engine cycle (two-stroke or four stroke).

Evaluation of either mass moments of inertia for each lumped mass in the system, or stiffness of every particular shaft element is not difficult and can be performed in a rather straightforward non-ambiguous manner. Unfortunately this is not the case with the estimation of damping for particular elements of the system. The damping may be defined in various terms. In addition to this, the excitation engine torques may be defined in several different ways: cylinder pressures or crank tangential forces, both in closed form or expressed by the Fourier series coefficients.

For these reason this paper focusses on presenting several definitions of damping in the torsional vibrations shafting model, the methods to convert among them, as well as the definitions of excitation torques in the model and their origin. In addition to this, classification societies (e.g. in [1] and [2]) require validation of the torsional response of the shafting system by measurements on-board the first in the series of newly built ships. In case the results do not match the 5 per cent margin difference, the calculations shall be run again. So, it is necessary to have the proper reliable methodology for defining of element damping properties and excitation forces readily available in the shafting numerical model.

2. Modelling of torsional vibrational damping

The torsional damping estimation is the most ambiguous for the marine shafting designers. No designer can be completely confident whether the damping data introduced and implemented in the torsional vibration calculations are correct, unless the calculation results are validated by means of measurement on-board [3].

The damping is the effect tending to reduce the vibratory amplitude of any oscillating system. Energy dissipation always accompanies damping itself. For the calculation of marine shafting torsional vibrations the four main types of damping are important [3]:

– viscous damping;

– fluid damping;

– internal damping; and

– structural damping.

The cause of viscous damping is the energy loss occurring in lubricating liquid between the system parts in relative motion. Viscous damping force is directly proportional to the relative velocity between the moving parts of the vibrating system. Viscous damping may be considered as absolute (between the moving part and the non-moving environment) or relative (between the two parts in relative motion) [3].

The cause of fluid damping is the dynamic interaction of propeller and surrounding water. The cause of internal (material) damping is the mechanical energy dissipation within the material of the shafting, material of flexible couplings, as well as within the torsional vibration dampers. The cause of structural damping is the relative friction between the shafting system elements that are in mutual contact [3].

Owing to the fact that only the linear viscous damping model enables a simplified analytical calculation approach, all the remaining types of damping models are in practice transformed to the equivalent viscous damping, as follows: fluid damping as absolute viscous damping, internal and structural damping as relative viscous damping [3].

For the above stated reasons there exist several possibilities to define and enter damping data in various torsional vibration calculation computer programs. The definitions of damping used in the most important programs will therefore be presented here.

Program SimulationX, developed by ITI GmbH [4], Dresden uses the following approach. Viscous damping torque amounts to:

$$T_D = b\omega \,[\mathrm{Nm}] \tag{1}$$

and the damping approach factor, B (in the expression: $b = B\sqrt{k}$) uses the following "rule of thumb" to estimate the damping:

B = 0,005 ... 0,01 - damping in metallic materials (e.g., shafts)
B = 0,10 ... 0,25 - damping in highly elastic materials (e.g., rubber coupling elements)
B = 0,05 ... 0,15 - structural and contact damping (e.g., gear teeth contacts / toothing)

Relative damping (ratio of damping energy), ψ (nonlinear, frequency dependent)

$$\psi = 2\pi \frac{\omega}{k} b \quad \rightarrow \quad b = \frac{\psi}{2\pi} \cdot \frac{k}{\omega} \tag{2}$$

Lehr's damping factor, D

$$D = \frac{\psi}{4\pi} = \frac{b\omega}{2k} \rightarrow \quad b = 2D \cdot \frac{k}{\omega} \tag{3}$$

where:

k – element linear stiffness, Nm/rad

b – element linear viscous damping, Nms/rad

ω – phase velocity of vibration, rad/s

In the program ShaftDesigner, developed by prof. Y. Batrak, the following damping definitions are used (the denotations for the k, b and ω as specified above):

Ratio of damping energy, ψ

$$\psi = \frac{2\pi \cdot b\omega}{k} = 2\pi\kappa = \frac{2\pi}{\sqrt{Q^2 - 1}} = 4\pi\varepsilon = \frac{2\pi}{M} \tag{4}$$

where:

κ – non-dimensional damping factor

Q – vibration magnifier

$$Q = \frac{\sqrt{k^2 + b^2\omega^2}}{b\omega} = \frac{\sqrt{1+\kappa^2}}{\kappa} = \frac{\sqrt{4\pi^2 + \psi^2}}{\psi} = \frac{\sqrt{1+4\varepsilon^2}}{2\varepsilon} = \sqrt{1+M^2} \quad (5)$$

ε – percent of critical damping, %

$$\varepsilon = \frac{b\omega}{2k} = \frac{\kappa}{2} = \frac{\psi}{4\pi} = \frac{1}{2\sqrt{Q^2 - 1}} = \frac{1}{2M} \quad (6)$$

M – dynamic magnifier

$$M = \frac{k}{b\omega} = \frac{1}{\kappa} = \frac{2\pi}{\psi} = \sqrt{Q^2 - 1} = \frac{1}{2\varepsilon} \quad (7)$$

Program GTORSI, developed by MAN Diesel & Turbo, Copenhagen, uses the following definitions:

- absolute torsional damping (in % of critical damping), ρ_θ [%]
- physical damping (between the actual and previous inertia), b_θ [Nms/rad]
- percentage modal damping wrt. stiffness, ρ_{inner}
- resulting physical damping, $b = \frac{2\rho_{inner}}{\omega} k$ [Nms/rad] (8)

For its importance, the damping of the propeller deserves to be presented separately, regardless of the calculation program. Propeller damping may be presented by means of the equivalent absolute viscous damping. Dimensional equations are generally implemented, so the user is to take care about the units used.

Frahm's propeller damping factor, D_F (in practice: 2,9…3,7), is used to define the propeller absolute damping (Frahm), b_{Ap} as follows:

$$b_{Ap} = D_F \cdot \frac{30}{\pi} \cdot \frac{T_p}{n_p} = D_F \cdot \left(\frac{30}{\pi}\right)^2 \cdot \frac{P_0}{n_{p0}^3} \cdot n_p = D_F \cdot \left(\frac{30}{\pi}\right)^3 \cdot \frac{P_0}{n_{p0}^3} \cdot \omega_p \ \text{kNms/rad} \quad (9)$$

Archer's propeller damping factor D_A (in practice: 25…35, based upon the open water characteristics of the Wageningen B-propeller series) implements a similar approach as for the Fram's factor [3]:

$$b_{Ap} = D_A \cdot \frac{T_p}{n_p} = D_A \cdot \left(\frac{30}{\pi}\right) \cdot \frac{P_0}{n_{p0}^3} \cdot n_p = D_A \cdot \left(\frac{30}{\pi}\right)^2 \cdot \frac{P_0}{n_{p0}^3} \cdot \omega_p \qquad \text{kNms/rad} \quad (10)$$

Obviously:

$$D_A \cdot = D_F \cdot \frac{30}{\pi} \quad (11)$$

In the above equations the following denotations have been used:

P_0 – engine nominal rating power (MCR), kW
n_0 – engine nominal speed at MCR, rpm
i – gearbox transmission ratio

$n_{p0} = \frac{n_0}{i}$ – propeller nominal speed, rpm (12)

$\omega_p = \frac{\pi \cdot n_p}{30}$ – propeller angular velocity, rad/s (13)

n_p – propeller speed, rpm

$$T_0 = \frac{30}{\pi} \cdot \frac{P_0}{n_0} \text{ – engine nominal torque, kNm} \tag{14}$$

$$T_{p0} = i \cdot T_0 \text{ – propeller nominal torque, kNm} \tag{15}$$

$$P = P_0 \left(\frac{n_p}{n_{p0}} \right)^3 \text{ – propeller power curve, kW} \tag{16}$$

$$T_p = T_0 \left(\frac{n_p}{n_{p0}} \right)^2 \text{ – propeller torque curve, kNm} \tag{17}$$

Other propeller damping definitions, such as Ker Wilson's formula, Dien-Schwanecke's formula, as well as MAN Diesel & Turbo's recommendation to set propeller damping as 5% of the critical have been presented in detail in [3].

3. Modelling of engine excitation loading

The Diesel engine cylinder may be provided by the engine manufacturer expressed in various forms:

a) actual cylinder pressures vs. crank angle in the range of either ±180° for two-stroke engines, or ±360° for four-stroke engines;
b) crank forces in tangential (circumferential) direction vs. crank angle, originating also from the combustion pressure in engine cylinders
c) crank forces in tangential (circumferential) direction in terms of Fourier series coefficients (precisely trigonometric approximation coefficients for the orders of 1; 2; 3; … in case of two-stroke engines and orders of 0,5; 1,0; 1,5; 2; 2,5; … for four stroke engines.

In practice it is often necessary to provide simple means to convert among these forms. Harmonic analysis, i.e. expressing of cylinder pressure/crank force vs. crank angle in the terms of trigonometric approximation coefficients, from (a) to (b) or to (c) above, is rather easy, following the procedure for the approximate calculation of Fourier series coefficients by e.g. their numerical integration. However, the reverse procedure, from (c) to (b) or to (a) above, may be a tricky one. For this reason the Excel/VBA program S06HarmSynt has been developed and will be shortly presented hereafter.

Program *S06HarmSynt* calculates tangential force, cylinder pressure and crank torque, all vs. crank angle, for the two cases: case of gas normal firing and gas compression only (misfiring) for 2-stroke and 4-stroke internal combustion engines, from the following input data: cylinder bore diameter, ratio of crank radius and connecting rod length, crank radius and harmonic cosine and sine components of Fourier series expansion of gas normal firing and gas compression only tangential pressure values, given for orders 0,5; 1; 1,5; 2; … for 4-stroke engines or orders 1; 2; 3; … for 2-stroke engines.

Program input data comprise the gas normal firing and misfiring N harmonic (cosine and sine) components F_{TC} and F_{TS} expressed as: $p = F_T / A_{cyl}$, where

F_T – force in tangential direction, N

$A_{cyl} = \pi\, d^2/4$ – cylinder area, mm² (18)

The calculation procedure can briefly be described as follows:
Crank angle range α in 2-stroke engines

$$-360^0 \le \alpha \le +360^0 \tag{19}$$

Crank angle range α in 4-stroke engines

$$-180^0 \le \alpha \le +180^0 \tag{20}$$

Ratio of crank radius to the connecting rod length

$$\lambda = r / l \tag{21}$$

Connecting rod angle

$$\sin\beta = \lambda \cdot \sin\alpha \tag{22}$$

Gas force (positive downwards)

$$F_{gas} = p \cdot \frac{\pi\, d^2}{4} \tag{23}$$

Tangential force on the crank journal due to gas forces

$$F_T = F_{gas} \cdot \frac{\sin(\alpha+\beta)}{\cos\beta} \tag{24}$$

Cylinder pressure from tangential force

$$p = \frac{4F_T}{\pi\, d^2} \cdot \frac{\cos\beta}{\sin(\alpha+\beta)}; \sin(\alpha+\beta) \ne 0 \tag{25}$$

(valid under condition: $\sin(\alpha+\beta) \ne 0$, otherwise: linear interpolation for nearby values)
Trig. approximation for tangential forces

$$F_T = \frac{F_{T0}}{2} + \sum_{k=1}^{N} F_{TC,k} \cos\frac{k\alpha}{2} + F_{TS,k} \sin\frac{k\alpha}{2} \tag{26}$$

for zero crank angle

$$F_{T0} = -2\sum_{k=1}^{N} F_{TC,k} \cos k\pi \tag{27}$$

Mean indicated pressure (numerical integration)

$$p_{m,i} \frac{1}{2} \int_{\alpha_{\min}}^{\alpha_{\max}} p(\alpha) \left(\sin\alpha + \lambda \frac{\sin\alpha \cdot \cos\alpha}{\sqrt{1-\lambda^2 \sin^2\alpha}} \right) \cdot d\alpha \tag{28}$$

The calculation example to illustrate the presented methodology for a two-stroke engine cylinder excitation, where inertial forces are to be considered separately begins from the data presented in the following table.

Table 1 Input data for an actual engine excitation loading calculations

Engine licence:	**MAN B&W**	Type:	**6S50MC-C, 9180 kW / 123 rpm**

engine working cycle (two stroke-2, four stroke-4)	*cycle*=	2	
cylinder bore	*D*=	500	mm
ratio of crank radius and connecting rod length	$\lambda = r/l =$	0,4878	
crank radius (half of piston stroke)	*r*=	1000	mm

	Inertia	Gas normal firing		Total		Gas misfiring only			[Nmm/mm³]
Order	SIN	COS	SIN	Ampl		COS	SIN	Ampl	
0	A_0=	**1,2624064**			A_{0comp}=	**0**			
1	0	0,762603	1,459809			-0,0003	0,1566		
2	0	0,0092984	1,727643			-0,0016	0,2393		
3	0	-0,234102	1,315507			-0,0009	0,2222		
4	0	-0,277969	0,930338			0,0011	0,1672		
5	0	-0,297135	0,600769			0,0006	0,1235		
6	0	-0,230167	0,362201			0	0,0899		
7	0	-0,1787	0,224368			0,0003	0,0644		
8	0	-0,137667	0,1123			0,0004	0,0454		
9	0	-0,085266	0,048867			0,0001	0,0319		
10	0	-0,0569	0,0176			-0,0002	0,0226		
11	0	-0,0326	-0,00943			0,0003	0,0154		
12	0	-0,010833	-0,0175			0	0,0108		

The calculated results have been presented in Figure 1.

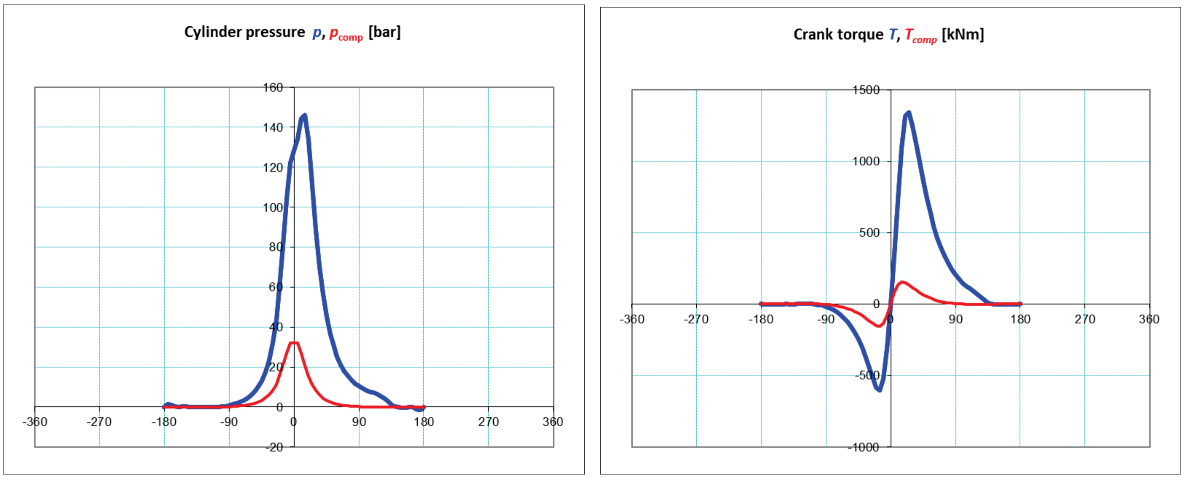

Figure 1 Calculation results, cylinder pressure and crank torque vs. crank angle

4. Validation of calculation on-board

Torsional vibrations calculation shall be verified on-board the first ship in the series. The most practical way is measurements by strain-gauges, connected into the full Wheatstone-bridge, that are glued on the surface of the shafting part which can be easily accessible from the machinery space (e.g. intermediate shaft). The strain-gauges measure strain, for the various levels of shafting rpm. This strain is converted into torsional stress and finally the torsional stress vs. shafting curve is plotted. Looking into this graph easily reveals critical speeds and maximal stress levels. In accordance with class Rules, measured critical speed shall not differ to the calculated ones by more than 5%. A more detailed presentation of measurement methods and interpretation of the results would be beyond the scope of this paper.

5. Illustrative example of the torsional vibration calculation

For the illustration of the proposed methodology, the two-stroke propulsion engine system has been selected to be briefly presented hereafter. These calculations have been performed with and the results obtained by the SimulationX program [4].

The main propulsion system of the oil-tanker consists of the 5-cylinder two-stroke slow speed main propulsion engine connected to the fixed-pitch four bladed propeller by means of the intermediate shaft and the propeller shaft.

The absolute damping in the engine cylinders, as well as the absolute damping of the marine propeller in the system, is modelled by means of dynamic magnification elements specially developed for this purpose. This possibility to develop and implement self-developed elements is an important advantage of the SimulationX software.

Figure 2 shows the calculation model for the shafting system.

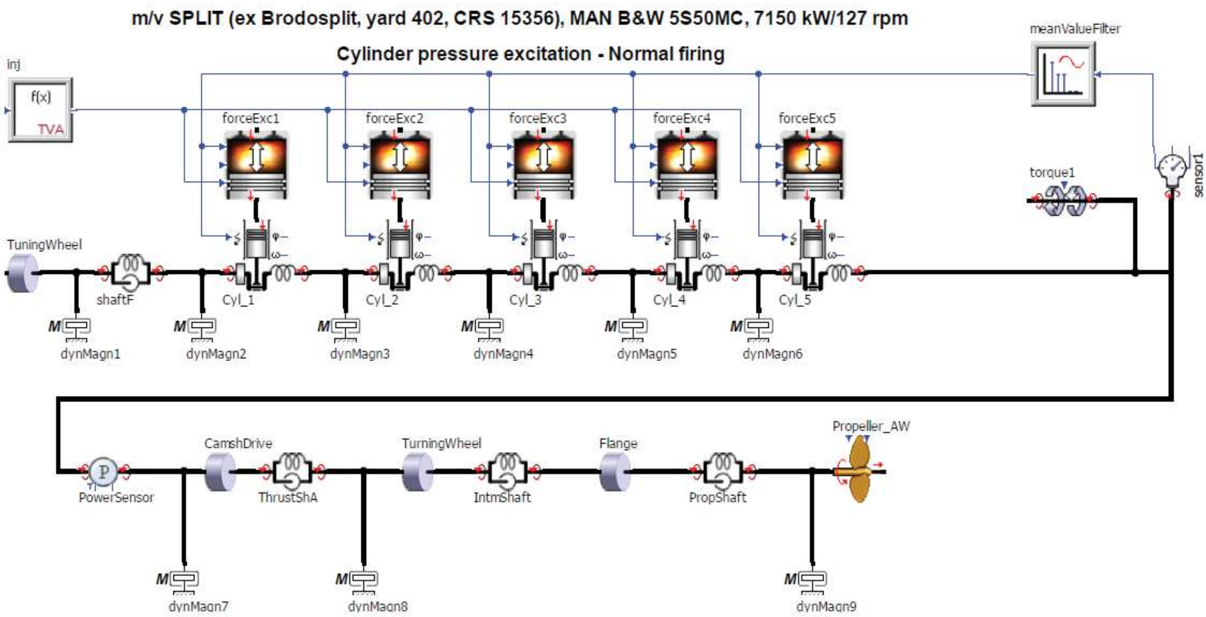

Figure 2 SimulationX shafting model for the calculation of torsional vibrations

Figures 3 and 4 present the steady-state calculation results for the torsional stress (MPa) in the intermediate and propeller shaft vs. the shafting speed (rpm) for each excitation order separately, as well as their sum and mean value. The allowable stress levels are also shown.

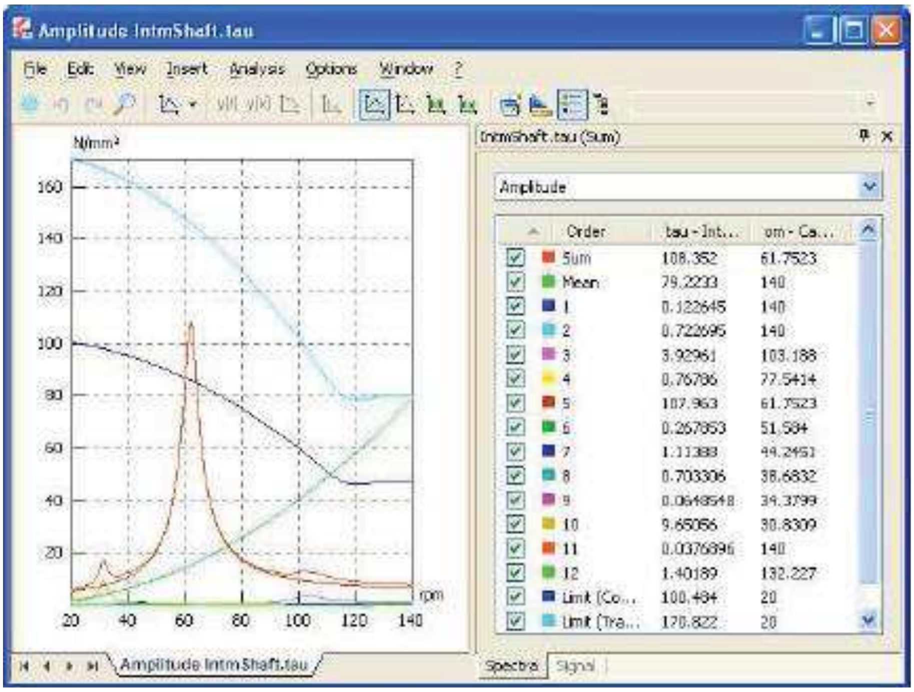

Figure 3 Calculation results: torsional stress in the intermediate shaft

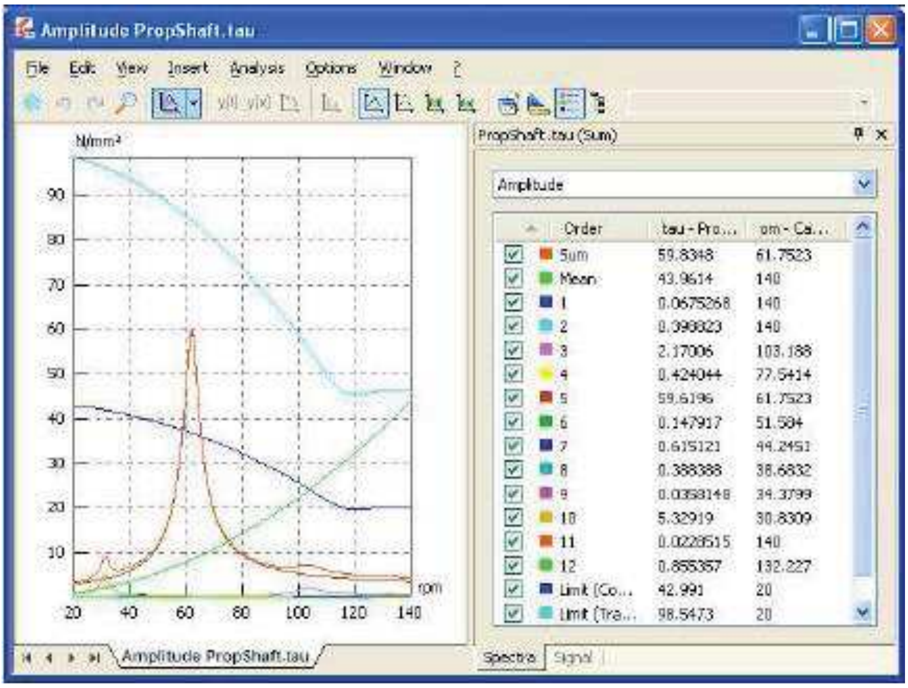

Figure 4 Calculation results: torsional stress in the propeller shaft

An important outcome of these analyses is that, for the particular ship in question, the propulsion system continuous operation within the range of shafting rotational speed of about 60 rpm should be avoided (barred speed range), in order to decrease the possible damage to the system due to resonance caused by excitation.

6. Conclusion

Torsional vibrations calculations are essential calculations which have to be performed in a very early stage of the shafting design process, by means of an appropriate software program.

The most difficult part in preparing data for these calculations, i.e. steady state response of the system modelled by lumped mass and massless stiffness and damping elements is to define damping and engine excitation in a proper way.

For this reason the methodology of definition of the damping implemented by several modern software programs has been presented in such a way that particular values can be easily converted from one to another and the results compared. This was the primary goal: to enable user to select the damping model best fitted for the purpose of modelling the real system.

An additional goal was to present the approach to the calculation of engine excitation in other forms (cylinder pressure vs. crank angle, or crank tangential force vs. crank angle), when these are given in terms of trigonometric approximation (Fourier's coefficients) for various excitation orders. There are some tricky points in this approach, to which the attention has been drawn.

Validation of the calculation results is essential, by measurements on-board, being the only way to check out whether the damping and engine excitation has been correctly taken in the calculations.

An illustrative example, showing the system and the obtained shafting torsional stress results has been presented in the end, just to show the powerful possibilities of one of the calculation programs intended for torsional vibration calculations (such as SimulationX).

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