TORSIONAL VIBRATION CALCULATION ISSUES WITH PROPULSION SYSTEMS

Dr. Yuriy Batrak

1. Introduction

Torsional vibration problems arose simultaneously with intensive use of mechanical engines for ship propulsion. But the stories about ship shafts snapping became regularly printed on the newspapers pages since 1870.
Steam paddle steamer GREAT REPUBLIC (Pacific Mail Steamship Company) had three cases of paddle wheel shaft snapping in 1872.
The list of ships with snapped shafts started to rise continuously since transoceanic shipping of the steamers became regular.
1883 – GERMANIC (The White Star Line)
1883 – HELLENIC (Cunard Line)
1890 – UMBRIA (Cunard Line), Fig.1.1

Fig.1.1 SS Umbria

1893 – IONIC (The White Star Line)
1900 – ETURIA (Cunard Line), Fig.1.2

Fig.1.2 SS Eturia

1906 – POLAND (The White Star Line)
This list can be enlarged considerably. According to the statistics of years 1882-1885 shaft lines were damaged 228 times.
Since 1912 when the first ocean-going diesel motor ship SELANDIA (East Asiatic Company) was launched (Fig. 1.3) the number of casualties increased very fast.

Fig.1.3 MS Selandia

Main cause of accidents was a shaft material fatigue.

FATIGUE FACTS

Fatigue is the progressive and localized structural damage that occurs when a material is subjected to cyclic loading. The nominal maximum stress values are less than the ultimate tensile stress limit, and may be below the yield stress limit of the material.

1837 – German mining administrator Wilhelm Albert records observations of metal fatigue.

1842 – The Versailles rail disaster. One of the first accidents due to material fatigue.

1860 – Systematic fatigue testing undertaken by Sir William Fairbairn and August Wöhler.

1924 – A. Palmgren formulates linear damage hypothesis.

1945 – A. M. Miner popularises A. Palmgren's linear damage hypothesis as a practical design tool.

At the beginning of the 20th century a lot of facts were accumulated to start scientific research of the problem. A book of H. Lorenz concerning crankshaft dynamics (1901), papers of H. Frahm, devoted to the steamers shaft lines cracking problem (1902), and G. W. Melville (1903), S. P. Timoshenko (1905), (Fig.1.4) opened the way to wide stream of the publications concerning of propulsion shafting torsional vibration problem.

Fig.1.4 Prof. S.P. Timoshenko

Among the first studies propulsion system torsional vibration the works of Hermann Frahm are most substantial. He did the torsional vibration measurements on the BESOCKI and RADAMES steamers to find the cause of shaft lines snapping. He had the possibility to measure the twisting angle and shaft section twisting velocities very precisely. As a result Frahm found that the reason of shaft snapping is the torsional vibration.
Starting from this moment it was not enough to provide shaft torsional strength calculation only. Every ship propulsion system, equipped with a reciprocating main engine, had to be checked for the torsional vibration resonances appearance.
To reveal torsional vibration resonances torsional vibration excitation frequencies are to be compared with propulsion system torsional vibration natural frequencies. Historically the calculation of the torsional vibration natural frequencies was a first step to the solution of the propulsion shaft snapping problem.
Currently conventional torsional vibration analyses (TVA) comprise free vibration calculation and steady vibration calculation caused by a harmonic excitation.

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In propulsion systems strength estimation, as for any other mechanical object, three problems should be solved:

1. The problem of permissible stresses.
2. The internal forces problem.
3. The external forces problem.

The first problem is a problem of standards and regulations. Solution of the problem is to be based on a practical experience and comprehensive analysis. Finally it is a Classification Societies and other regulation authorities' duty and we will not discuss it.
The second problem is a problem of system structure modelling and selection of appropriate mathematical tools to find the internal forces. It is most easy formalized problem of three mentioned above.
The third problem is the most complicate because its solution lies beyond the scope of shaft mechanics and Rules requirements. It concerns of determination of the environmental effects on a propulsion system that are to be solved within propeller hydrodynamics and diesel engine operation domain. Very often field tests are required to capture the environmental parameters.
For the further discussion of some torsional vibration calculation issues we should turn to the TVA mathematics.

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2. TVA Mathematics

2.1 Torsional vibration equation

Differential equation for torsional vibration calculation of n-degree-of-freedom mechanical system, in matrix form is as follows:

$\mathbf{M}\ddot{\mathbf{X}}(t)+\mathbf{C}\dot{\mathbf{X}}(t)+\mathbf{K}\mathbf{X}(t)=\mathbf{F}(t)$,

where: $\mathbf{X}(t)$ – vector of the twisting angles at the system nodes (solution of the equation);

- $\mathbf{M}$ – mass matrix;
- $\mathbf{C}$ – damping matrix;
- $\mathbf{K}$ – stiffness matrix;
- $\mathbf{F}$ – excitation torque vector.

2.2 Vibration glossary

Free vibration – occurs when $\mathbf{F}(t)=\mathbf{0}$ i.e. a mechanical system vibrates freely after an initial motion was applied.

Forced vibration – occurs where an alternating force $\mathbf{F}(t)\neq\mathbf{0}$ is applied to a mechanical system. In forced vibration the alternating force $F(t)$ does not disappear when the excited motion is prevented.

Self-excited vibration – occurs where the alternating force $\mathbf{F}(t)$ that sustains the vibration motion is created or controlled by the vibration motion itself. When the motion stops the alternating force $\mathbf{F}(t)$ disappears.

Steady vibration – vibration of a mechanical system caused by a periodic excitation when free vibration oscillations have decayed.

Harmonic excitation – occurs when the periodic excitation force alternates according to the harmonic law: $f(t)=A\sin(\omega t+\psi)$.

Transient vibration – occurs when a non periodic alternating force $\mathbf{F}(t)$ is applied.

Parametric vibration – occurs when mass $\mathbf{M}$ and/or damping $\mathbf{C}$ and/or stiffness $\mathbf{K}$ of a mechanical system are variable ($\mathbf{M}(t)$, $\mathbf{C}(t)$, $\mathbf{K}(t)$), but not depend on vibration motion.

Non liner vibration – occurs when mass $\mathbf{M}$ and/or damping $\mathbf{C}$ and/or stiffness $\mathbf{K}$ of a mechanical system depend on vibration motion $\mathbf{X}(t)$.

Non damped vibration – occurs when damping in vibrating system is equal to zero $\mathbf{C}=\mathbf{0}$.

If damping matrix is a liner combination of mass and stiffness matrix $\mathbf{C}=\alpha\mathbf{M}+\beta\mathbf{K}$ (Rayleigh damping case, 1877) the matrix equation for *n*-degree-of-freedom mechanical system can be substituted by the system of *n* differential equations for one-degree-of-freedom oscillator:

$\ddot{x}_i+2\zeta_i\lambda_i\dot{x}_i+\lambda_i^2x_i=Q_i(t) \qquad i=1,2,3,\ldots$

where:

- x_i – *i*-th generalized coordinate;

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λ_i – *i*-th natural frequency;

ζ_i – *i*-th mode damping ratio (modal damping);

$Q_i(t)$ – *i*-th generalized force;

Final solution of torsional vibration problem in this case will be find according to the formula:

$$\Theta(x,t)=\sum_{i=1}^{n}x_i(t)V_i(x),$$

where:

$\Theta(x,t)$ – twisting angles;

$x_i(t)$ – solution of *i*-th equation;

$V(x)$ – *i*-th mode shape.

This approach is known as a mode superposition method.
The simplification is permissible also where damping is not really the Rayleigh damping but is low enough. In such cases we can avoid the need to form a damping matrix based on the physical properties of individual structure elements. It is enough to know mode damping ratios ζ_i.

Torsional vibration calculation issues that will be discussed hereafter concern the second and the third structural mechanics problems: methods used to solve the torsional vibration equation and setting of propulsion system parameters (matrices' elements).

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3. Free vibration calculation issues

The simplest way to check the propulsion system with fixed mass and stiffness parameters for resonance appearance is to perform calculation of a free non damped torsional vibration i.e. to solve the equation:

$$\mathbf{M}\ddot{\mathbf{X}}(t)+\mathbf{K}\mathbf{X}(t)=\mathbf{0},$$

and build a resonance table or plot a Campbell diagram (1924), like shown on Fig. 3.1.

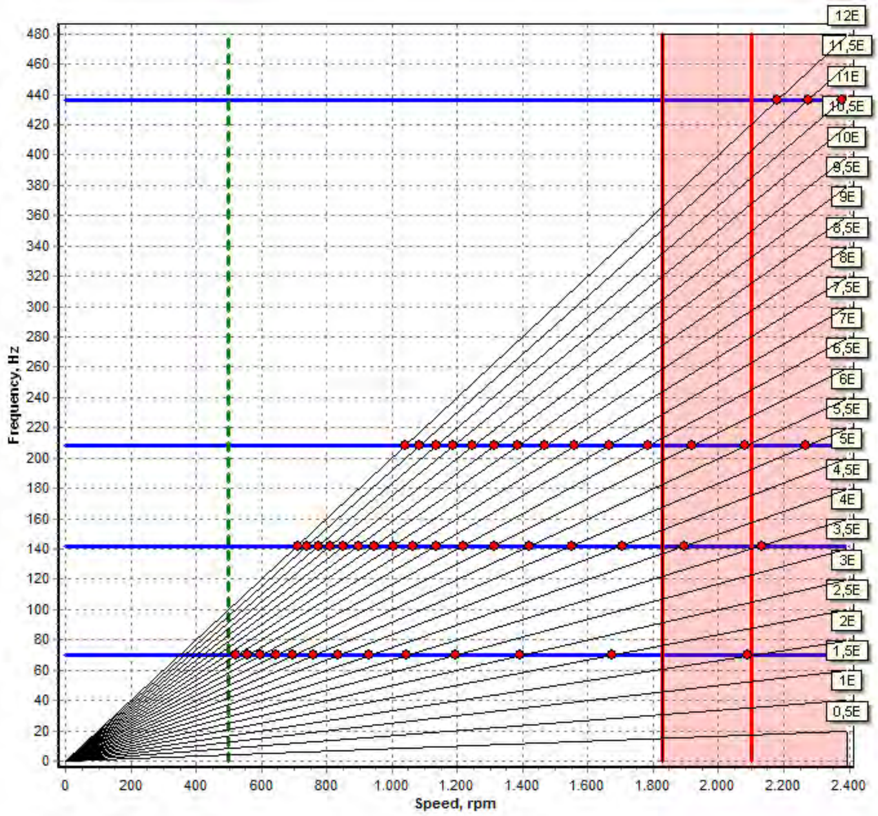

Fig.3.1 Campbell diagram for generator set vibration

The red points show possible resonances i.e. the speeds where the harmonic excitation torque frequencies are equal to the natural torsional vibration frequencies. It does not mean that all resonances will happen but where the excitation energy of certain frequencies is great enough the resonances are inevitable. Thereby free vibration calculation results give an initial idea about the possible resonances in operating propulsion system.

Free torsional vibration calculation for system with fixed mass and stiffness parameters is a purely mathematical problem. Methods of non damped natural frequency calculation are well developed and not discussed in this presentation. The most widely used methods where introduced by M. Tolle and H. Holzer in the period from 1905 to 1921. The matrix methods, that are more suitable for computer based calculations, were developed later.

It should be pointed out that the accuracy of frequencies and mode shapes determination by these methods for the installations with a low damping is sufficient.

When the damping is high, calculated non damped frequencies and mode shapes may be used for the reference only. The actual resonance speed and the calculated non damped resonance speed can differ significantly. It can be seen from the vibration stress diagram (Fig 3.3) calculated for the propulsion train, equipped with viscose torsional vibration damper (Fig 3.2).

Fig.3.2 Propulsion train with the torsional vibration damper

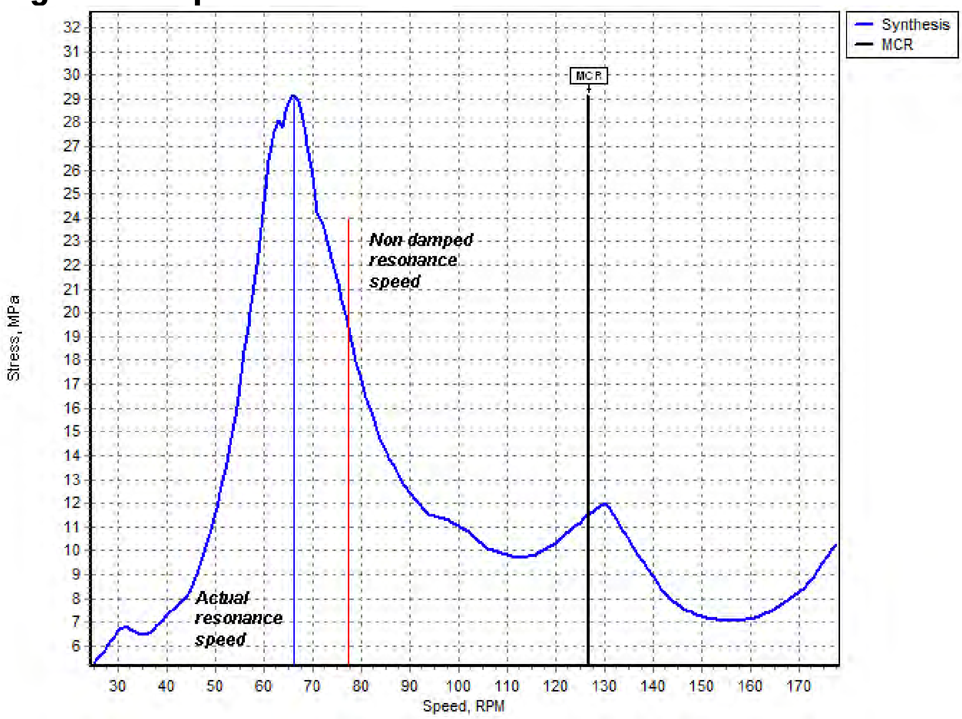

Fig. 3.3 Shift of the resonance speed for high damped system

If the installation comprises the element, dynamic stiffness of which depends on vibration frequency, as we have for Geislinger flexible coupling (Fig.3.4) or on transmitted torque, the natural frequencies must be calculated for the whole range of operating speed.

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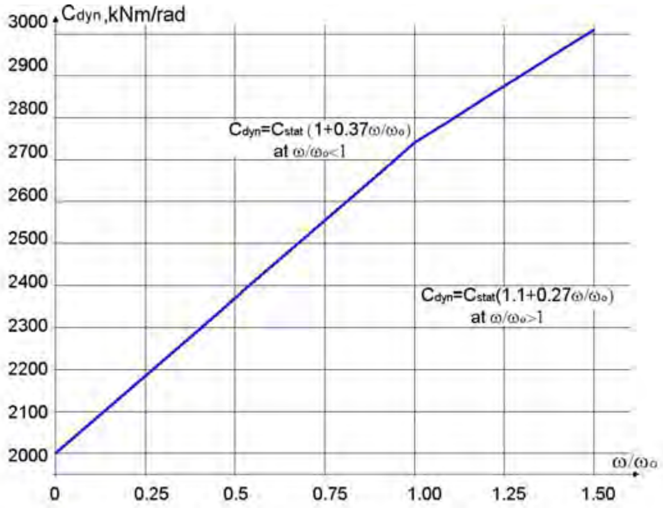

Fig. 3.4 Variable dynamic stiffness of the flexible coupling

In this connection DNV Rules for Classification of Ships recommended to calculate natural frequencies with at least the maximum and minimum values of flexible element stiffness. If the propulsion system comprises several frequency dependent flexible elements, combinatorial approach should be used.

The torsional vibration problem becomes non linear if the dynamic stiffness depends on vibratory torque (Fig.3.5) and natural frequency idea in this case is not applicable at all.

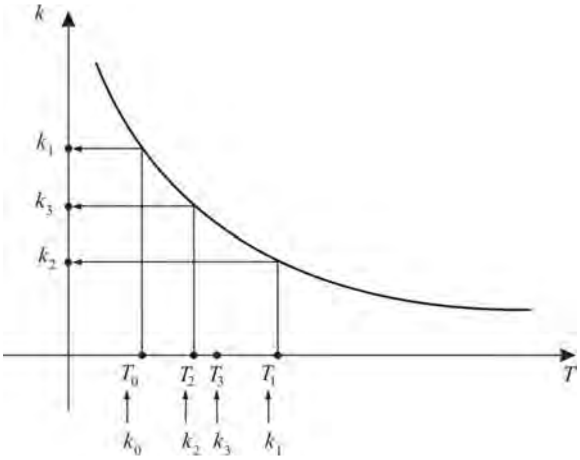

Fig. 3.5 Non linear dynamic stiffness of the flexible element

These circumstances bring down the importance of a free vibration calculation for realistic estimation of the resonance speeds of modern propulsion systems.

4. Forced vibration calculation methods

The goal of forced torsional vibration calculation of a propulsion system is to satisfy the acceptance criteria formulated in Classification Societies Rules or in other regulations or standards.

Irrespective of what the regulation is used to check the acceptance criteria following parameters must be calculated:

1. Vibration angles at the nodes.
2. Vibration torque in the elements.

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3. Vibration shear stresses in the elements.
4. Dissipated power in the flexible elements.
5. Gear hammering for geared propulsion systems

There are two classes of methods for forced vibration calculation: time domain class and frequency domain class. Algorithm for selection of forced vibration calculation method is shown in Fig.4.1.

Most universal method is the time domain method of direct numerical integration of torsional vibration equation. It may be used at any condition regardless of equation coefficients properties and excitation torque type.

To calculate propulsion shafting forced torsional vibration more often frequency domain methods are used: mode superposition method and full solution method. Both methods are applicable only if the excitation torque has a harmonic character. It should be noted that full solution method is always applicable, but the mode superposition may be applicable for low damped propulsion systems only.

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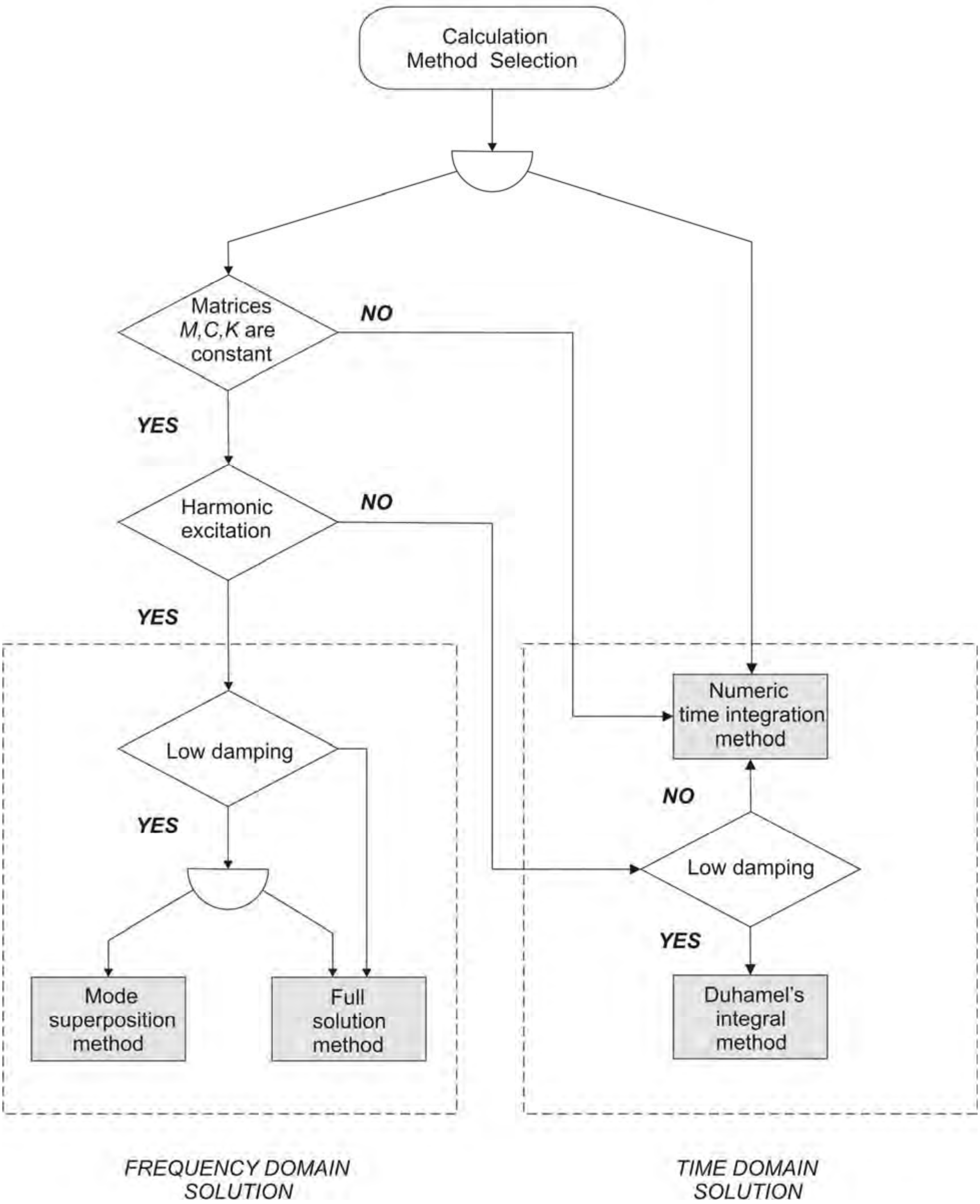

Fig.4.1. Algorithm for selection of forced vibration calculation method

The stress diagrams (Fig.4.2 and Fig.4.3) calculated for intermediate shaft of the system Fig.3.2 show how the mode superposition method results for high damped system differ from the results obtained with the full solution method.

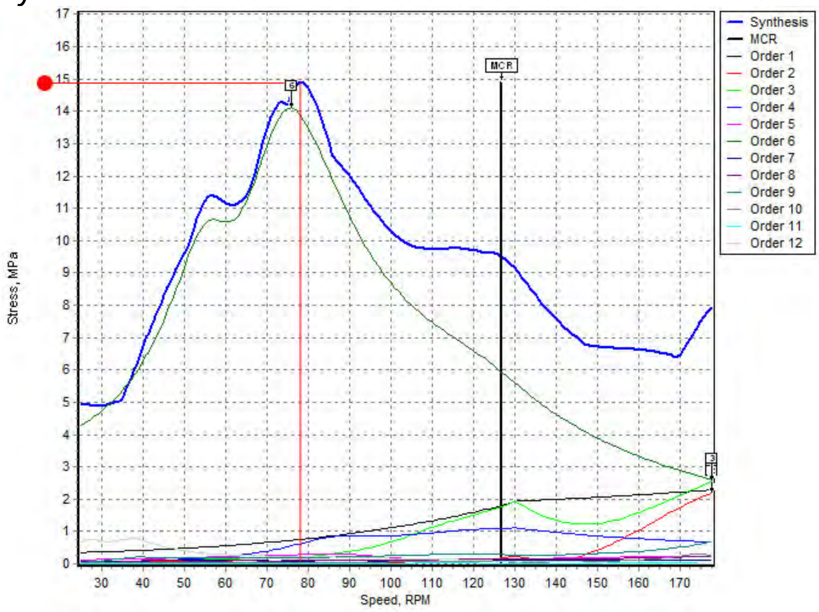

Fig.4.2 Mode superposition method result

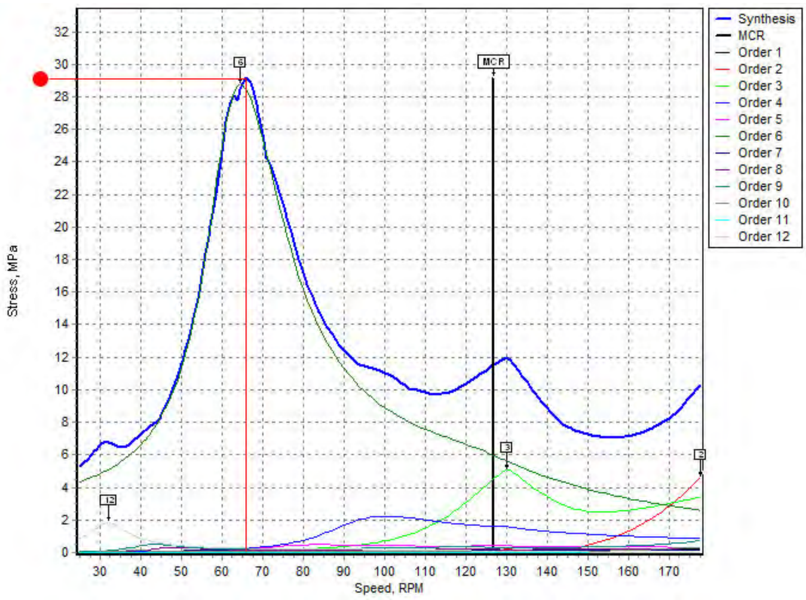

Fig.4.3 Full solution method result

The excitation torque produced by a diesel engine cylinder is periodic but non harmonic (Fig.4.4). Torque period is equal 2π for two stroke engines and 4π for four stroke engines. For such excitation type frequency domain methods for forced vibration calculation of the propulsion systems driven by diesel engines cannot be applied directly.

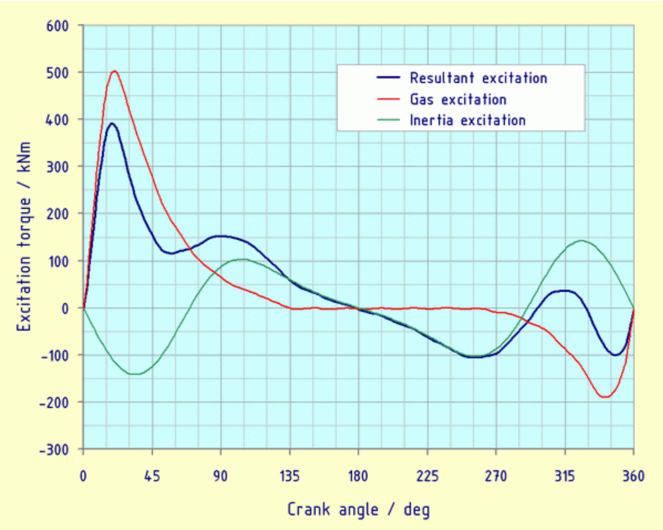

Fig.4.4 The excitation torque of two stoke diesel engine cylinder
(www.cadea.hr/ Torsional.htm)

Provided excitation torque expansion in a Fourier series is used

$$f_i(t) = \sum_{k=1}^{N} A_{ik} \sin(\omega_{ik} t + \psi_{ik}) ,$$

torsional vibration calculation can be performed in the frequency domain i.e. for each excitation order separately.

There are a lot of in house developed and commercially supplied software which implement all or some of above mentioned methods to calculate free and forced vibration of propulsion system.

5. Propulsion train modelling issues

5.1 Shaft discretization

A propulsion shaft line (Fig.5.1) is a continuous body consisting of cylindrical and conical shaft elements and may includes the elements such as diesel engine cranks, torsional vibration dampers and other equipment. Bearings do not influence a torsional vibration significantly and are therefore ignored in TVA calculations.

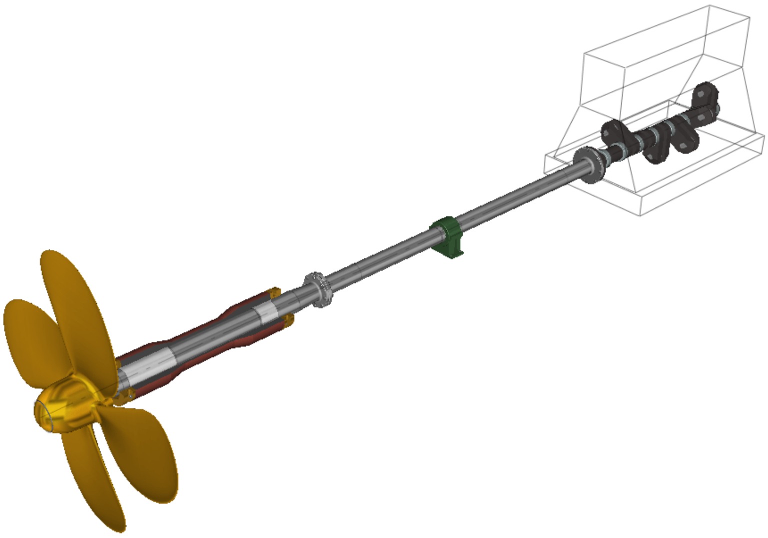

Fig.5.1 Propulsion shafting of the directly driven propulsion plant

While a propulsion shaft line is a continuous body its torsional vibration calculation model usually is presented as a discrete system consisting of lumped masses connected by a weightless flexible elements, Fig.5.2. Propeller shafts and intermediate shafts are long but in the mass elastic system each of them is presented as a single element.

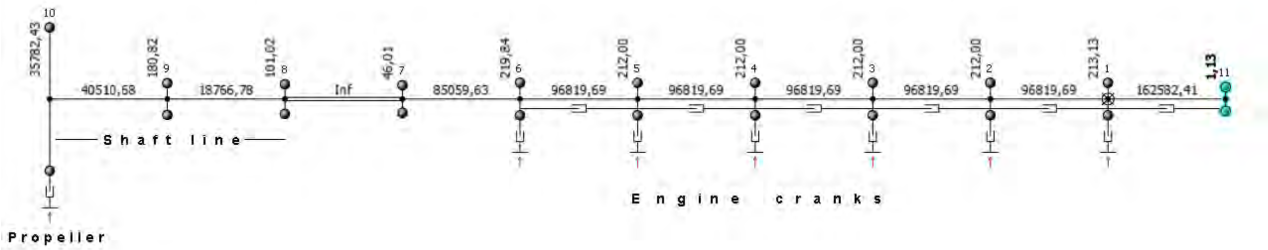

Fig.5.2 Discrete mass-elastic system of propulsion shafting

Currently for the TVA calculation computer programs are used and so simple representation of the continuous structure looks as a heritage from the of manual calculation times when a mass-elastic system with large number of lumped masses was practically impossible to calculate.
But such an approach is valid, even today, when for torsional vibration calculation powerful tool such as the Finite Element Method is used.

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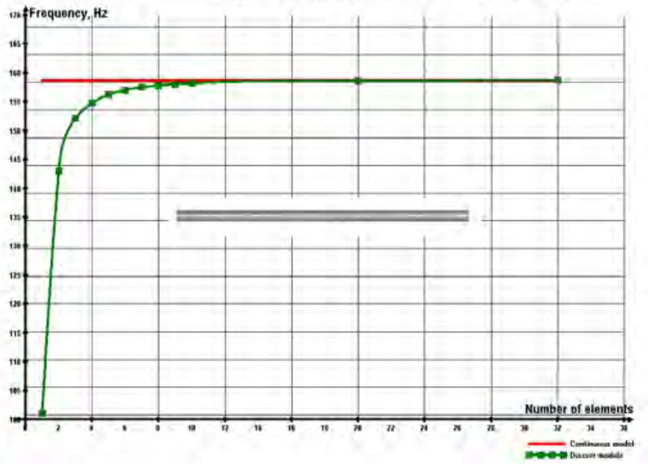
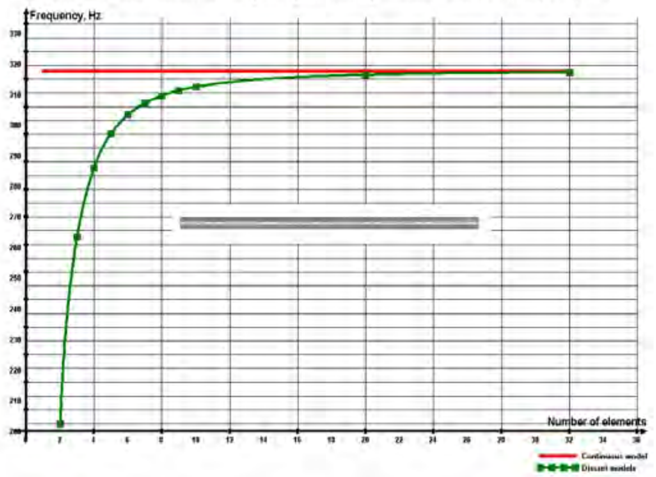
Actually, for the simple shafts, the natural frequencies of torsional vibration depend heavily on the amount of discretization, especially the higher frequencies. It can be seen in Fig. 5.3, where three first mode frequencies are compared with the calculated ones, using discrete models.

First mode torsional frequency for the cylindrical shaft L=10 m, D=0,4 m



Second mode torsional frequency for the cylindrical shaft L=10 m, D=0,4 m



Third mode torsional frequency for the cylindrical shaft L=10 m, D=0,4 m

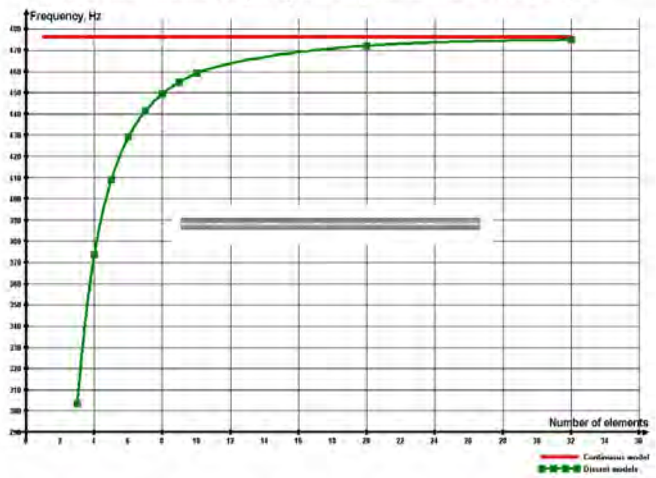


Fig.5.3 The influence of the shaft discretization on 1-st, 2-nd and 3-rd natural frequencies

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The influence of discretization remains considerable when the shaft has a propeller at the end, (Fig. 5.4). Propeller inertia to shaft inertia ratio (Ip/Is) for most of ships is within the range 1 to 40.

As can be seen from the picture, in order to have correct values of the first natural torsional vibration frequency, the shaft model must consist of minimum 5-6 elements. Higher natural frequencies will require more detailed discretization of the shaft.

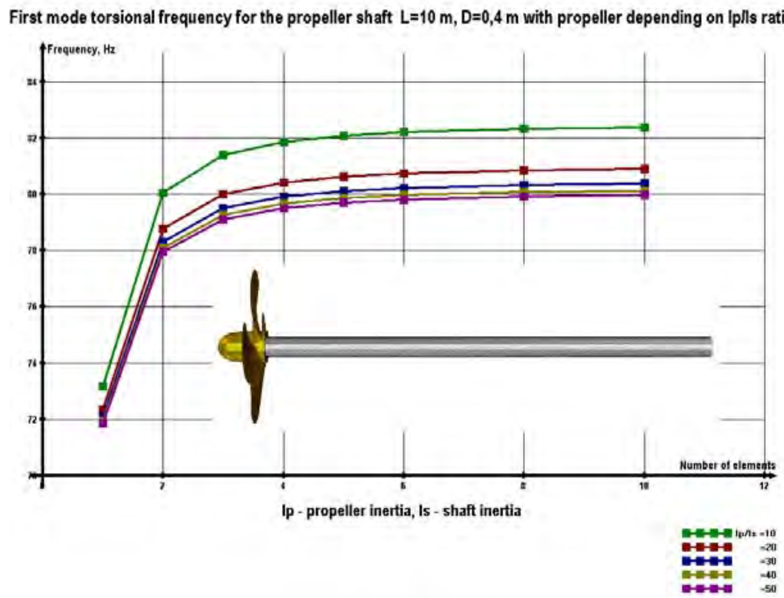

Fig. 5.4 The 1-st natural frequency dependence on shaft discretization and propeller inertia

The influence of discretization on natural frequencies of torsional vibration decreased considerably after the shaft is connected to the engine (Fig.5.5). For the highest frequencies the difference is not greater than 10 % (see Table 5.1).

Fig.5.5 Propulsion shaft line

Table 5.1

Natural Frequency	Number of elements			
	2	4	8	16
1	22,57	22,62	22,64	22,66
2	69,09	69,33	69,39	69,40
3	118,94	119,47	119,55	119,52
4	145,27	149,02	149,58	149,65

5	180,40	184,42	184,96	185,06
6	209,50	214,92	215,31	215,37
7	222,34	237,61	243,42	244,70

The uniqueness of a propulsion shaft line structure allows do not split propeller shafts and intermediate shafts on a short elements when the torsional vibration mass-elastic model is composed.

5.2 Crank mechanism modelling

A piston-conrod-crank mechanism of the engine Fig.5.6 usually is modelled by an equivalent lumped mass which is assumed to excite the same torsional vibration as the actual complicated crank mechanism.

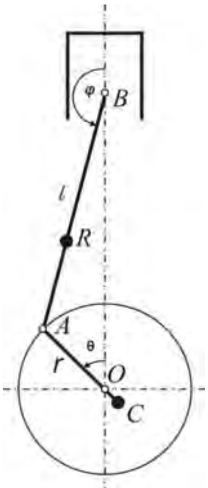

Fig.5.6 Piston-conrod-crank mechanism

The simplest formula for crank inertia moment I is as follows [1]:

$$I = I_{rot} + \frac{1}{2} m_{rec} r^2 ,$$

where:

I_{rot} – the moment of inertia of purely rotating parts;

m_{rec} – the sum of the mass of the piston and of a part of the connecting rod.

Thus, the inertia moment of the masses assumed to be constant i.e. independent of rotation angle.
This assumption leads to satisfactory results for small reciprocated machines but it is also still used in conventional TVA calculations of large marine two stroke diesel engines too.

It is known from mid-1920's (Goldsbrough G.R. [2]) that in reality a piston-conrod-crank mechanism inertia moment varies with the crankshaft rotation twice per revolution. In 1954 Gregory, R.W. investigated non-linear oscillations of a simplified system having variable inertia [3].
The inertia moment function $I(\theta)$ is as follows (Hesterman & Stone [4]):

$$I(\theta) = I_C + m_C h^2 r^2 + I_R \lambda^2 \frac{\cos\theta}{\cos\varphi} + m_P r^2 (\cos\theta \tan\varphi - \sin\theta)^2 +$$

$$+ m_R r^2 \left[(1-j)^2 \cos^2\theta + (j\cos\theta\tan\varphi - \sin\theta)^2 \right],$$

where:

I_C – constant inertia of crank;

I_R – inertia of conrod about it's centre of mass;

m_C, m_R, m_P – masses of crank, conrod and piston (including gudgeon pin);

r, l – crank and conrod lengths;

$$\lambda = \frac{r}{l}, \quad h = \frac{OC}{r}, \quad j = \frac{AR}{l}.$$

As can be seen from Fig. 5.7 (W. Schiffer [5]) the inertia moment of 8RTA/RT-flex96C/CB diesel engine varies significantly; maximum inertia moment is more than twice bigger of minimum one.
More exact model of the piston-conrod-crank mechanism requires special methods to solve a differential equation of torsional vibration with variable coefficients.

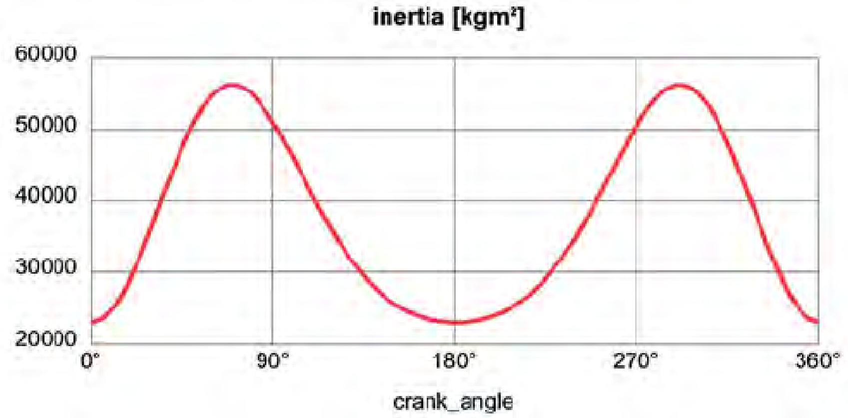

Fig. 5.7 Piston-conrod-crank mechanism inertia moment function $I(\theta)$

Most engineers dismissed these results as unimportant until large diesel engines with heavy pistons were built. A number of torsional failures of crankshafts in large two-stroke marine diesel engines have occurred in cases where no excessive resonance stress was revealed by the TVA (Archer [6]). The piston-conrod-crank mechanisms of these diesel engines were so massive that their reciprocation gave inertia variations substantial enough to excite problematic torsional vibrations. The first who studied the increase of dynamic stress in crankshafts due to this phenomenon was Draminsky P. [7]. He nominated it as a secondary resonance. The secondary resonance results from parametric vibration.
W. Schiffer [5] showed Draminsky's effect for 8RTA/RT-flex96C/CB diesel engine comparing measured and calculated stress using conventional and advanced method, with alternate inertia moment of the crank (Fig. 5.8 and Fig. 5.9).

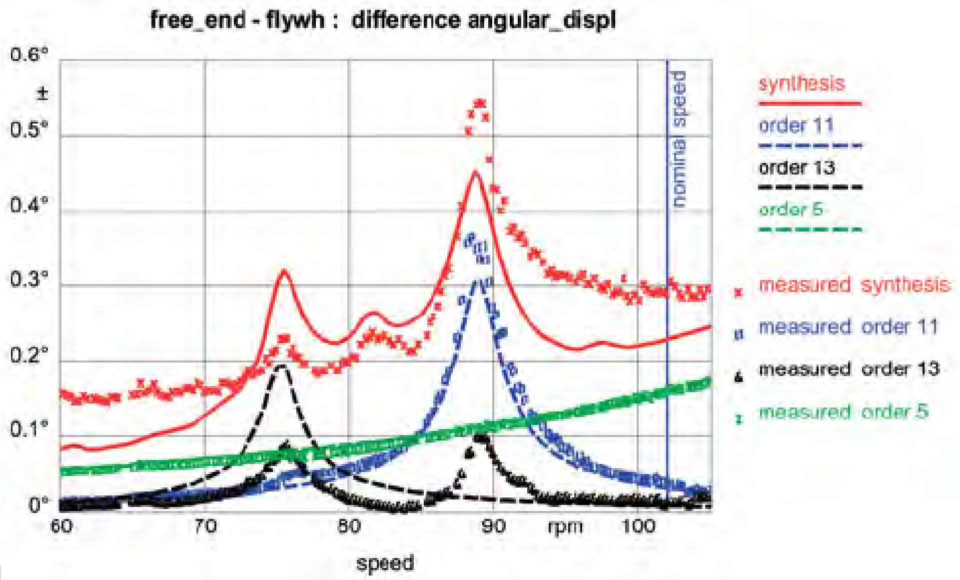

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Fig 5.8 Relative crankshaft angular displacement. (Constant inertia moment of the cranks)

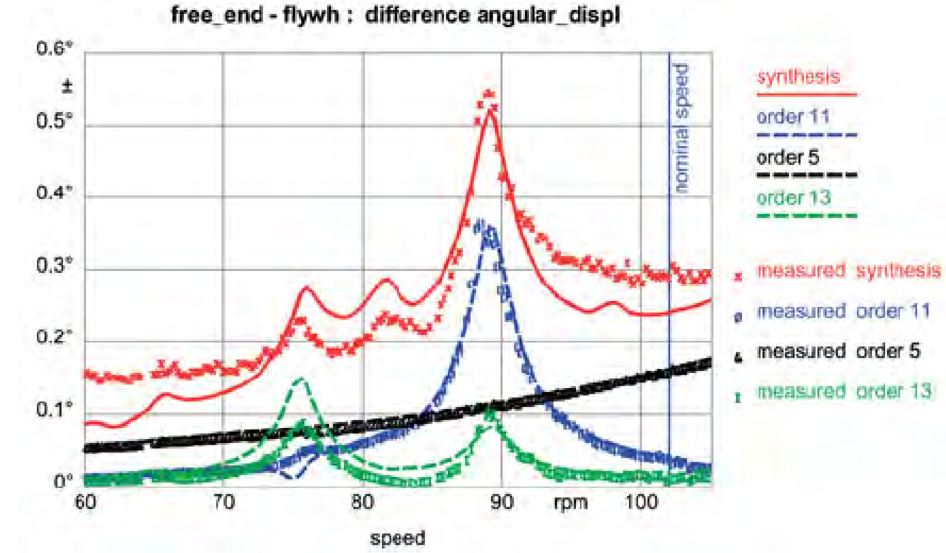

Fig 5.9 Relative crankshaft angular displacement. (Alternate inertia moment of the cranks)

As can be seen large variations in inertia torques gave prove of the secondary resonance in torsional vibrations, which cannot be explained by conventional theories incorporating only the mean values of the varying inertias. The secondary resonance can be extremely dangerous for the crankshafts.

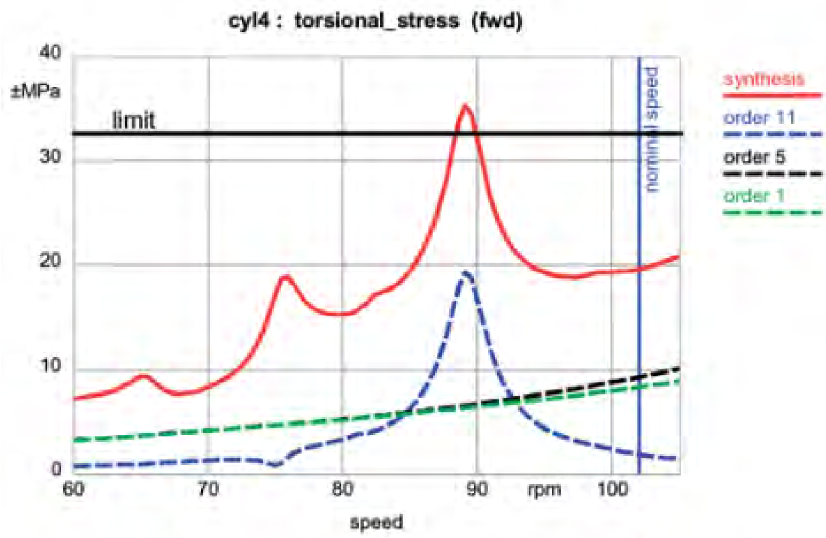

Fig 5.10 Torsional stress calculated with alternate crank inertia moment

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6. Environmental effects issues

6.1. DAQ problem

"Mathematics may be compared to a mill of exquisite workmanship, which grinds your stuff to any degree of fineness; but, nevertheless, what you get out depends on what you put in…"
Thomas Henry Huxley

Possession of best possible software for torsional vibration calculation by no means guarantees the realistic forced torsional vibration calculation results. Reliability of forced torsional vibration calculation completely depends on how accurately propulsion system parameters are set.
Some issues of shaft modelling were discussed above. In addition the shaft designer should collect all the specific data from engine manufacturer, propeller designer, coupling, damper and other equipment suppliers. From the shaft designer point of view this process may be designated as a data acquisition (DAQ).
Very often DAQ problem arises. The data may be not available for different reasons, especially when propulsion system of repaired or renovated ship is calculated (equipment documentation is lost, supplier ignores the inquiry or does not exist at all).
We will not discuss the equipment data acquisition and methods of their presentation in the software. Very specific questions of torsional vibration dampers tuning we will not discuss too. The data that we will discuss are concerned with environmental effects only.
There are two main domains where propulsion system interacts with the environment in the torsional vibration: operating engine, rotating propeller and bearings.

6.2 Excitation data

6.2.1 Diesel engine gas pressure harmonic coefficients

The sources of diesel exciting vibratory torques are gas pressure pulses in cylinders, inertia and weight of a piston-conrod-crank mechanism. Inertia and weight harmonics coefficients can be calculated from engine particulars. As to the gas pressure pulses for the specific diesel engine, harmonic coefficients A_k and phases ψ_k, calculated from measured pressure data at relevant running conditions, are most preferable. On the Fig.6.1 and Fig.6.2 excitation tables for MAN & BW and Caterpillar diesel engines are shown. As can be seen from the tables the diesel engines manufacturers use different approach for gas pressure harmonic coefficients presentation.

Tangential harmonic components

Order BMIP Recalculate... Clear... Save... Load...

Sin | Cos | Resultant | Phase

Order \ BMIP [Bar]	0,0000	2,0000	4,0000	6,0000	8,0000	10,0000	12,0000	14,0000	16,0000
1,0	0,168800	0,257100	0,345400	0,433700	0,522000	0,644800	0,778400	0,922000	1,068000
2,0	0,246600	0,342200	0,437900	0,533500	0,629200	0,769600	0,923000	1,088000	1,255900
3,0	0,222800	0,290900	0,358900	0,426900	0,495000	0,599600	0,713800	0,836900	0,961800
4,0	0,165400	0,215900	0,266500	0,317100	0,367600	0,441800	0,521700	0,607000	0,693600
5,0	0,120900	0,154000	0,187100	0,220300	0,253400	0,299600	0,348600	0,401300	0,454900
6,0	0,087400	0,106800	0,126200	0,145600	0,165000	0,191200	0,219100	0,250500	0,282600
7,0	0,061400	0,074200	0,087000	0,099800	0,112600	0,126900	0,142500	0,161600	0,181500
8,0	0,043100	0,048800	0,054500	0,060100	0,065800	0,069600	0,075100	0,084300	0,094400
9,0	0,029900	0,031400	0,032800	0,034300	0,035800	0,034600	0,035600	0,040600	0,046300
10,0	0,020700	0,020600	0,020500	0,020400	0,020300	0,016300	0,015000	0,017200	0,020200
11,0	0,014300	0,011900	0,009500	0,007100	0,004600	-0,001600	-0,004800	-0,004600	-0,003900
12,0	0,009700	0,006700	0,003600	0,000600	-0,002500	-0,008600	-0,011500	-0,011800	-0,011600
13,0	0,006600	0,003500	0,000500	-0,002600	-0,005700	-0,011900	-0,015000	-0,015900	-0,016300
14,0	0,004600	0,001000	-0,002600	-0,006100	-0,009700	-0,015500	-0,018400	-0,019300	-0,019800
15,0	0,003000	-0,000200	-0,003400	-0,006600	-0,009800	-0,014700	-0,017000	-0,017700	-0,018100
16,0	0,002000	-0,001000	-0,003900	-0,006900	-0,009900	-0,014400	-0,016400	-0,017000	-0,017400

BMIP ☑ Extend table Harmonic components units: MPa OK Cancel

Fig 6.1 Gas pressure harmonic coefficients table for MAN diesel engine

Tangential harmonic components

Order BMIP Recalculate... Clear... Save... Load...

Sin | Cos | Resultant | Phase

Order \ BMIP [Bar]	0,0000	22,9800
0,5	0,103102	0,817975
1,0	0,148655	1,104496
1,5	0,171889	0,922966
2,0	0,149264	0,795700
2,5	0,151487	0,636062
3,0	0,128331	0,523912
3,5	0,113124	0,431907
4,0	0,094313	0,341360
4,5	0,074918	0,266831
5,0	0,062511	0,215434
5,5	0,052353	0,176113
6,0	0,044827	0,139918
6,5	0,038384	0,107445
7,0	0,032431	0,086614
7,5	0,026958	0,072370
8,0	0,022281	0,058739
8,5	0,018334	0,045135
9,0	0,015074	0,037018
9,5	0,012723	0,032054

BMIP ☑ Extend table Harmonic components units: MPa OK Cancel

Fig 6.2 Gas pressure harmonic coefficients table for Caterpillar diesel engine

These data for engine normal operation as well as for misfiring conditions are the engine manufacturer proprietary. Where the engine manufacturer provides TVA of the propulsion system, the gas excitation harmonic coefficients, as a rule, are not included in calculation report. It should be pointed out that such policy unfortunately became a standard practice in present days.

From our point of view it is not a good practice, because it leads to monopolization of technical data. As a consequence ship owner has no possibility to fulfil the calculation in the future which may be required in the case of ship renovating. In such cases he is forced to order calculation from engine manufacturer again, if he still exists of course.
Less preferable is a way of gas harmonic coefficients calculation using theoretical cylinder pressure curve. But in this case some specific values must be received from engine manufacturer too.
When the gas pulsation data are not available from engine manufacturer the general source data should be used. There are a lot of the gas excitation coefficients tables and graphs that exist since 1925 (Lewis [8]), Porter [9]). Among general sources Lloyd Register Excitation pack should be mentioned too [???]. Unfortunately general source data are not updated in connection with diesel engines of newest design and using it we do not make the torsional vibration calculation more accurate.

6.2.2 Propeller excitation

Fluctuating torque component of hydrodynamic propeller loads is the result of non uniform wake flow. Wake flow usually is presented by axial and tangential components of velocity. The common way to have a wake flow data is towing tank test of the ship model. Axial and tangential components are measured in the disc where the propeller will be installed, Fig 6.3.

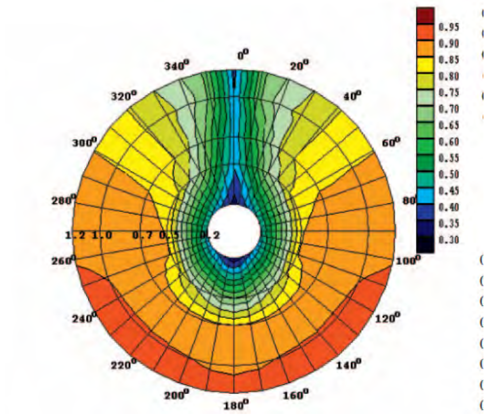

Fig 6.3 Axial component of one screw ship wake field [10]

Measured wake flow data a is used to calculate mean values and fluctuating component of the six integral hydrodynamic loads including torque fluctuating component, Fig 6.4.

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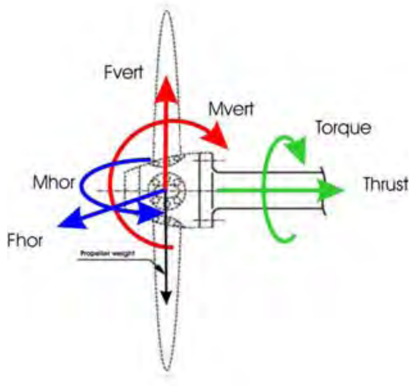

Fig 6.4 Propeller hydrodynamic loads

Main components of the fluctuating torque are of the first and second propeller blade order. Dynamic torque amplitudes may be calculated as a percent of the mean torque. Det Norske Veritas recommends the following table for dynamic torque amplitudes depending on propeller blade number:

DNV recommendations for propeller excitation amplitudes
Table 6.1

Propeller excitation amplitudes in % of actual mean torque		
Number of blades	1-st blade frequency	2-nd blade frequency
3	8,0	2,0
4	6,0	2,0
5	4,0	1,5
6	4,0	1,5

As can be seen from the table, increasing of propeller blade number reduces excitation torque.
DNV recommendations were checked using ShaftDesigner software (Fig.6.5) based on unsteady lifting surface theory. In the most cases of the ten calculated projects the results did not exceed the recommended values. In the studied projects the hull form and propeller characteristics were optimized by model tests to have low vibration level.

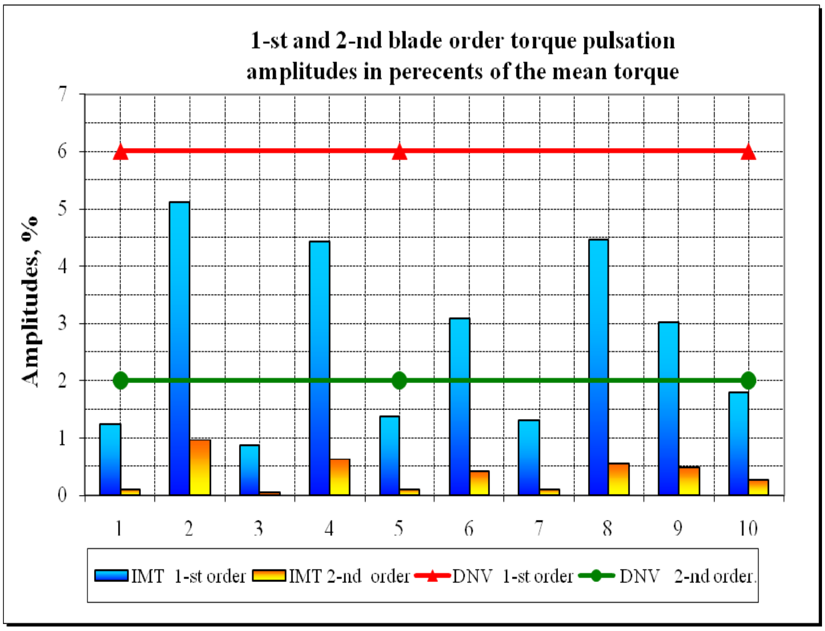

Fig 6.5 Dynamic torque amplitudes calculated with ShaftDesigner software

To set the harmonic excitation thoroughly the excitation phase angles must be determined. One may expect that the propeller phase angle relatively to the first engine crank will change the torsional vibration parameters but really the influence is negligibly small. Therefore all excitation torque phase angles can be set to zero.
The main problem is the wake field characteristics. If they are obtained from the model test the influence of propeller presence is not taken into account. In addition scale distortions are possible too.
As to the modern CFD applications there no results regarding excitation torque determination are known to the author.

6.2.3 Other excitations

Propeller/ice interaction, water jet impeller, cardan (universal) joints, gear meshing are the other excitation sources. Ice impact vibration will be discussed in the next section.

6.3 Damping data

6.3.1 Representation of damping in vibration analysis

The most problematic for the shaft designers is a damping setting. Nobody can be definitely sure in damping data used in calculation until measurement of a torsional vibration stresses is fulfilled.
Damping is any effect that tends to reduce the vibratory amplitude of any oscillated system. Energy dissipation always accompanies damping. Damping effect in vibration equation is usually represented by the damping moment $\mathbf{D}(t)$:

$$\mathbf{M}\ddot{\mathbf{X}}(t)+\mathbf{D}(t)+\mathbf{K}\mathbf{X}(t)=\mathbf{F}(t).$$

Energy dissipation manner depends on the details of a mechanical system. Three main types of damping are important for calculation of propulsion system vibrations. They are:

- viscous damping;
- fluid damping;
- internal damping;
- structural damping.

Viscous damping is caused by such energy losses as occur in liquid lubrication between moving parts. The viscous-damping force is directly proportional to the relative velocity between the moving parts of the vibrating system: $D(t)=C\dot{X}(t)$. The viscous damping is referred as an absolute damping if one of the parts belongs to the vibrating system environment. The viscous damping is referred as a relative damping if both moving parts belong to the vibrating system.

Only the linear viscous damping admits simplified mathematical analysis. Therefore in propulsion system torsional vibration calculation all types of damping are replaced by an equivalent viscous damping. Equivalent linear viscous damping is a property of the computer model and is not a property of a real propulsion system.
Fluid damping is caused by dynamic interactions of a propeller and sea water. In torsional vibration calculation propeller damping is presented as absolute viscous damping.

Internal (material) damping results from mechanical energy dissipation within the shaft material, flexible coupling material and torsional vibration dampers. In torsional vibration calculation material damping is presented as relative viscous damping.

Structural damping is caused by relative friction motions between propulsion system elements that have contact points. In torsional vibration calculation structural damping is presented as relative viscous damping.
The most reliable method to estimate damping is by measurements in field tests of propulsion trains. There are no commonly used methods for damping measurement in industry. As a consequence several types of parameters currently are used for a damping representation. Conversion formulas for various damping presentation are shown in Tab. 4.2.

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Damping conversion

Table 6.2

	c	κ	ψ	*Q*	ε	*M*
c	1	$\frac{\kappa K}{\omega}$	$\frac{\psi K}{2\pi\omega}$	$\frac{K}{\omega}\sqrt{\frac{1}{Q^2-1}}$	$\frac{2\varepsilon K}{\omega}$	$\frac{K}{M\omega}$
κ	$\frac{c\omega}{K}$	1	$\frac{\psi}{2\pi}$	$\sqrt{\frac{1}{Q^2-1}}$	$2\ \varepsilon$	$\frac{1}{M}$
ψ	$\frac{2\pi c\omega}{K}$	$2\pi\kappa$	1	$\frac{2\pi}{\sqrt{Q^2-1}}$	$4\pi\varepsilon$	$\frac{2\pi}{M}$
Q	$\frac{\sqrt{K^2+c^2\omega^2}}{c\omega}$	$\frac{\sqrt{1+\kappa^2}}{\kappa}$	$\frac{\sqrt{4\pi^2+\psi^2}}{\psi}$	1	$\frac{\sqrt{1+4\varepsilon^2}}{2\varepsilon}$	$M\sqrt{1+\frac{1}{M}}$
ε	$\frac{c\ \omega}{2K}$	$\frac{\kappa}{2}$	$\frac{\psi}{4\pi}$	$\frac{1}{2}\sqrt{\frac{1}{Q^2-1}}$	1	$\frac{1}{2M}$
M	$\frac{K}{c\ \omega}$	$\frac{1}{\kappa}$	$\frac{2\pi}{\psi}$	$\sqrt{Q^2-1}$	$\frac{1}{2\ \varepsilon}$	1

where:

c – linear viscous damping [Nms/rad],

κ – undimensioned damping factor,

ψ – ratio of damping energy,

Q – vibration magnifier,

ε – percent of critical damping [%],

M – dynamic magnifier,

ω – phase velocity of vibration [rad/s],

K – stiffness [Nm/rad].

It is no problem to measure damping parameters in the case of the systems with one- or two-degree-of-freedom. For multi-degree of-freedom structural systems the measurement will involve the response of more modes. The test and the analysis method required to predict the damping ratios ζ_i are more complex.

Introducing of new energy dissipation equipment to the propulsion system the structural engineers are forced to treat the energy dissipation in a more exact manner and use full solution and time domain methods to predict torsional vibration parameters. These methods require the knowledge of damping coefficients for each node and element of mass-elastic system separately.

6.3.2 Engine damping

The most reliable damping values are supplied by the engine manufacturer who measures it for each produced engine. When damping data for certain engine are not known and cannot be obtained from the engine manufacturer the following default values may be used in torsional vibration calculation:

Engine damping defaults **Table 6.3**

Engine part	ε **[%] (MAN & BW)**	***M*** **(DNV)**
Cylinder	0.85	70
Turning wheel	0.50	n/a
Tuning wheel	0.50	n/a
Crank	1.00	50

6.3.3 Propeller damping

When propeller vibrates the water around the blades is involved in vibration motion too. Part of the shaft energy dissipates in the water and amplitudes of propulsion system torsional vibration decrease. Fluid damping of the propeller is one of most significant.
There are several formulas for propeller damping estimation known from publications. Choosing the appropriate formula is a shaft designer problem.
One of the oldest is the following Ker Wilson formula for propeller damping coefficient estimation [11]:

$$c_p = 196.2 D^4 \frac{A_e}{A_0}$$

where:

c_p – propeller damping coefficient [Nms/rad];

D – propeller diameter;

A_e / A_0 – expanded area ratio (EAR).

Currently for propeller damping determination very often Archer's formula is used:

$$c_p = A\frac{T_p}{n_p} \quad \text{or} \quad c_p = A\frac{30}{\pi}\frac{P_p}{n_p^2},$$

A – Archer's factor (25÷35);

T_p – propeller torque;

n_p – propeller speed;

P_p – propeller power.

Where load characteristic according to a propeller law is used the following formula can be used:

$$c_p = A \frac{30}{\pi} \frac{P_{p0}}{n_{p0}^3} n_p ,$$

where:

n_{p0} – nominal propeller speed;

P_{p0} – propeller power at nominal propeller speed.

Archer's formula is an approximation based on open water characteristics of the Wageningen B-srew series [12].
H. Dien and H. Schwanecke formula [13] is used too:

$$c_p = \frac{\rho \omega}{\pi} D^5 \left(\frac{P}{D} \right)^2 \frac{A_e}{A_0} 0.0231 ,$$

where:

ρ – specific mass of the sea water;

ω – propeller angular speed;

P/D – propeller pitch ratio.

This formula derived from the calculations based on unsteady propeller theory.

MAN & BW recommends set propeller damping as $\varepsilon = 5.0\%$.

Modern system for calculation of structure vibrations must have the options to set the damping in any possible form. Sample of the dialog windows for damping setting is shown on Fig.2.1.

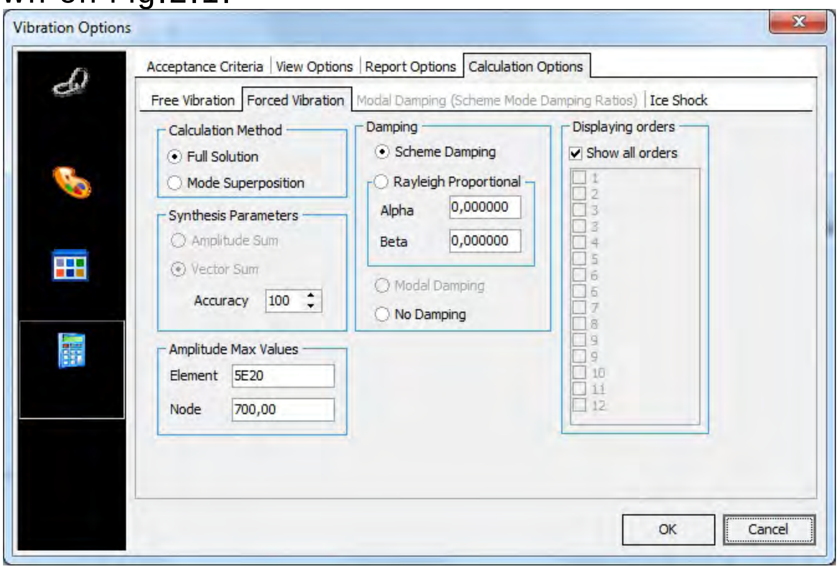

Fig.6.6 Options for damping type setting

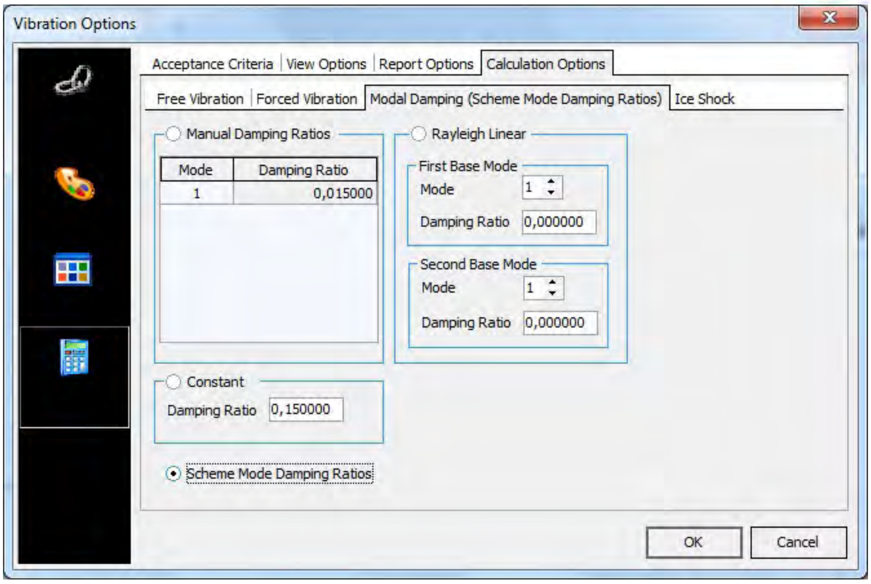

Fig.6.7 Options for modal damping ratios setting

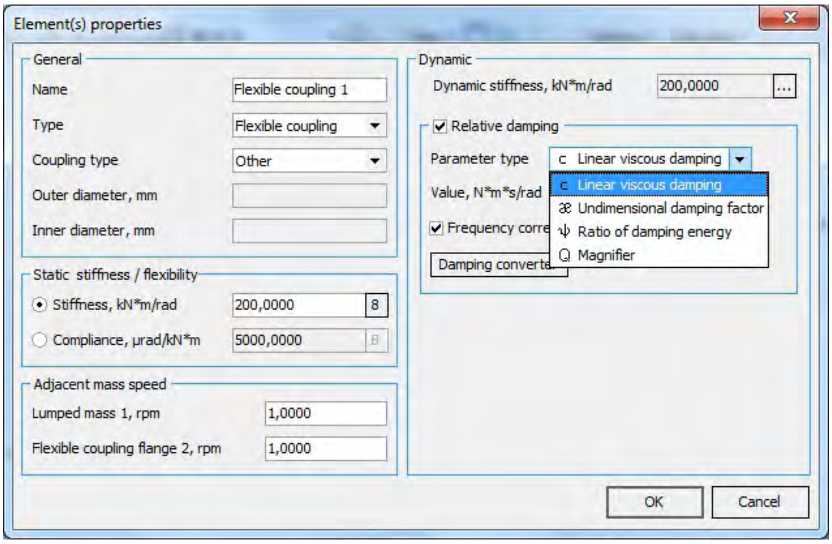

Fig.6.8 Options for relative damping setting

The variety of formulas and recommendations for propeller damping coefficient determination indicates that problem of propeller damping has no ultimate solution yet. Quite probably there is no simple universal analytical solution at all, because of complexity of modern propeller form and operating condition variety. Unfortunately most recent investigations intended for propeller damping characteristics estimation, if judge by John Carlton book [12] references, are 25 years old. Quite possible the Computational Fluid Dynamics applications (CFD) will solve the problem of propeller damping determination in the nearest future.

7. Vibration excited by ice torque

IACS members changed their Classification Rules last summer. Main changes coming into force 1 January 2011. New regulations shall be applied to propulsion machinery of the ice strengthened ships and polar class ships. The regulations cover open- and ducted-type propellers with controllable pitch or fixed pitch design.

Propeller shaft line components of above mentioned ships are to be designed to withstand the design torque Q_r. Design torque Q_r is a maximum response torque along the propeller shaft line, taking into account the dynamic behaviour of the shaft line for ice excitation (torsional vibration) and hydrodynamic mean torque on propeller. Design torque Q_r is to be calculated in transient torsional vibration analysis using ice-induced torque resulting from propeller/ice interaction on one propeller blade, including hydrodynamic loads on that blade.

For the ice strengthened ships transient torsional vibration analysis of ice torque response shall be performed and submitted for approval if a first blade order torsional resonance is within operational speed range ± 20% ($0.8n_{\min} \div 1.2n_{\max}$). For the polar class ships (ships for arctic and icebreaking service) transient torsional vibration analysis of ice torque response shall be performed and submitted for approval unconditionally.

The propeller ice torque excitation for shaft line transient torsional vibration analysis is described by a sequence of blade impacts (repetitive ice shocks). The sequence of blade impacts is referred hereafter as ice milling sequence.

For the estimation of design ice torque on propeller during milling sequence, a maximum size of ice block entering the propeller is determined. It is a rectangular ice block with the dimensions $H_{\text{ice}} \times 2H_{\text{ice}} \times 3H_{\text{ice}}$, Fig.7.1.

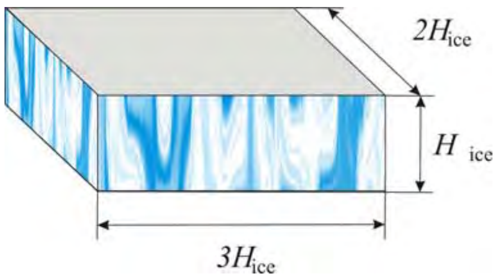

Fig.7.1 Maximum ice block size

The thickness of the ice block H_{ice} is dependent on ship ice class.

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Three different cases of milling sequences in propeller/ice interaction are to be examined, Tab.7.1, where C_q and α_i – are torque excitation parameters (C_q is impact effectiveness, α_i – duration of propeller blade/ice interaction expressed in propeller rotation angle).

Torque excitation parameters **Table 7.1**

Torque excitation	**Propeller/ice interaction**	C_q	α_i
Case 1	Single ice block	0,75	90
Case 2	Single ice block	1	135
Case 3	Two ice blocks (phase shift 45 deg.)	0,5	45

The torque resulting from a single blade ice impact as a function of propeller rotation angle φ for each of three cases is

$$Q(\varphi) = Q_{\text{peak}} \cdot \sin\left(\frac{180}{\alpha_i}\varphi\right), \qquad \text{when } \varphi = 0...\alpha_i,$$

$$Q(\varphi) = 0, \qquad \text{when } \varphi = \alpha_i...360,$$

where:

$$Q_{\text{peak}} = C_q \cdot Q_{\max}$$

$Q_{\max}$ – maximum design ice torque on propeller resulting from propeller/ice interaction depends on propeller design and ship ice class.

Sample of a single blade ice impact torque for Case 1 is shown in the Fig.7.1.

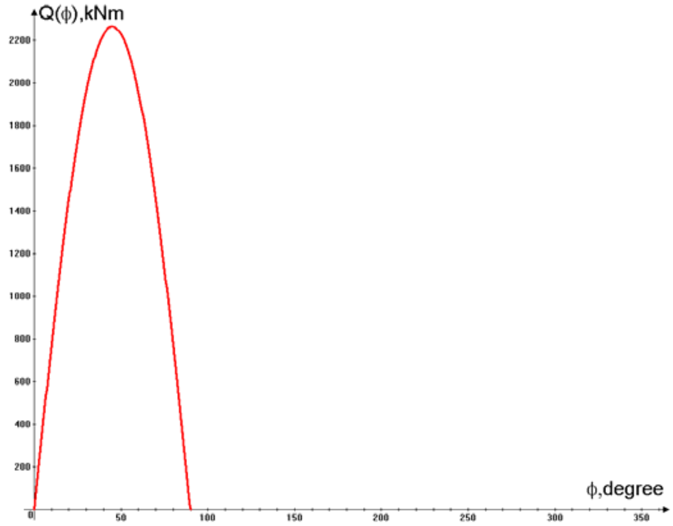


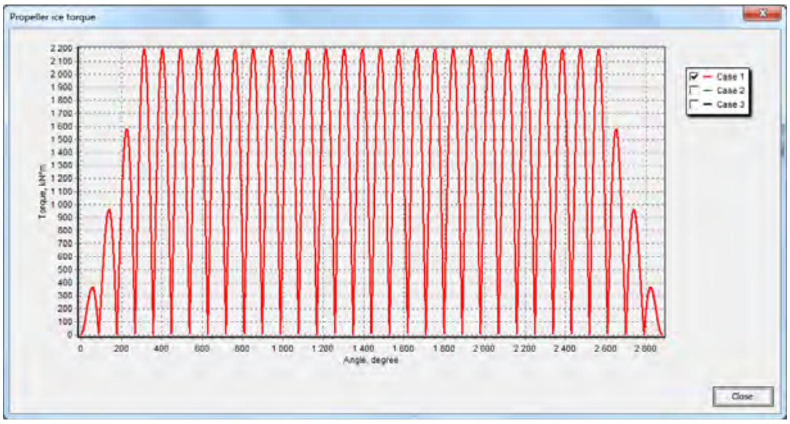
Fig.7.2 Single blade ice impact torque for Case 1

The total ice torque is obtained by summing the torque of single blades, taking into account the phase shift $\frac{360^\circ}{Z}$, where Z is a blade number.

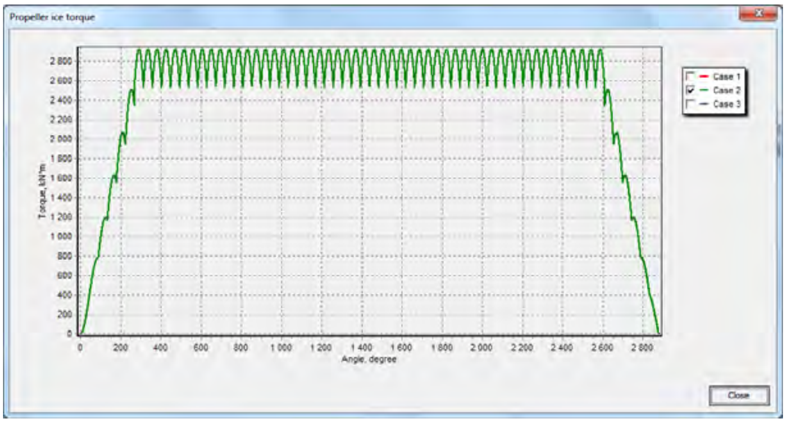
In addition, at the beginning and at the end of the milling sequence a linear ramp functions for 270 degrees of rotation angle shall be used.

The total ice torques graphs for 4-blade propeller of DNV Polar class PC-1 ship ($H_{\text{ice}} = 4.0$ m) are shown in the Fig.7.3.

Case 1



Case 2



Case 3

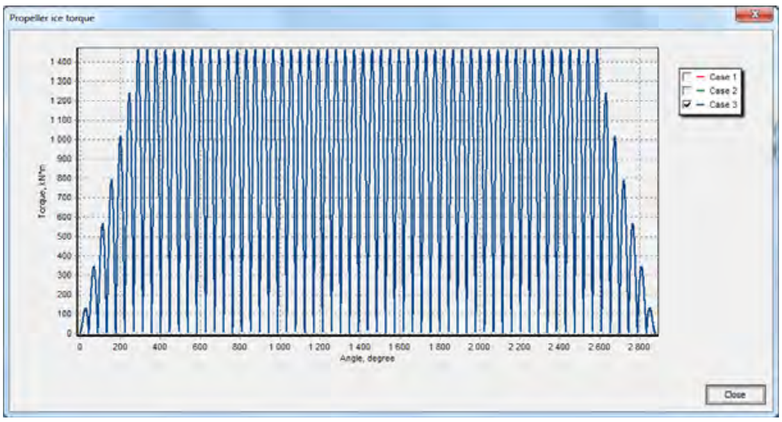


Fig.7.3 Three cases of total ice impact torque for 4-blade propeller of PC-1 class

The peak torque Q_{peak} is used for calculating of propeller mounting characteristics such as contact pressure for shrink fit and keyed connections, for calculation of flange connections parameters.
The number of propeller revolutions during a milling sequence is
$N_Q = 2 * H_{\text{ice}}$.

The number of impacts is $Z \cdot N_Q$ is the base for calculation of N_{ice} – total number of ice loads on propeller blade during ship's service life used in fatigue calculations.

Milling sequence time of the propeller rotating with the speed n (rpm) can be calculated as:

$$T_{impact} = N_Q \frac{60}{n}, \text{ s}$$

Milling sequence time is an important parameter for calculation of transient torsional vibration in time domain by direct numerical integration methods.

The maximum design ice torque on propeller $Q_{\max}$ resulting from propeller/ice interaction is as follows

$$Q_{\max} = K_Q \cdot \left[1 - \frac{d}{D}\right] * \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot (nD)^{0.17} \cdot D^3, \qquad [kNm]$$

when $D \le D_{\text{limit}}$

$$Q_{\max} = K_Q \cdot \left[1 - \frac{d}{D}\right] * \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot (nD)^{0.17} \cdot D^{1.9} \cdot H_{\text{ice}}^{\;1.1}, \qquad [kNm]$$

when $D > D_{\text{limit}}$

where: $D_{\text{limit}} \le 1.8 H_{ice}$ [m]

$P_{0.7}$, m – propeller pitch at $0.7R$ radius. For CP propellers, $P_{0.7}$ shall correspond to MCR in bollard condition. If not known, $P_{0.7}$ is to be taken as $0.7P_{0.7n}$ where $P_{0.7n}$ is the propeller pitch at MCR in free running condition;

D, m – propeller diameter;

d, m – external diameter of propeller hub;

K_Q	**Open propeller**	**Ducted propeller**
$D \le D_{\text{limit}}$	10.9	7.7
$D > D_{\text{limit}}$	20.7	14.6

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n, rpm – rotational speed in bollard condition. If not known n is to be taken as from the following table

Propeller type	Rotational speed n
CP propellers	n_n *)
FP propellers driven by turbine or electric motor	n_n
FP propellers driven by diesel engine	$0.85\,n_n$
*) n_n refers to MCR free running condition	

$P_{0.7n}$, m propeller pitch at 0.7R radius at MCR in free running condition;

When response torques are calculated some of characteristics must be determined for each element: peak torque T_{peak}, average torque T_{aver} and double torque amplitude T_{double}, Fig.7.4.

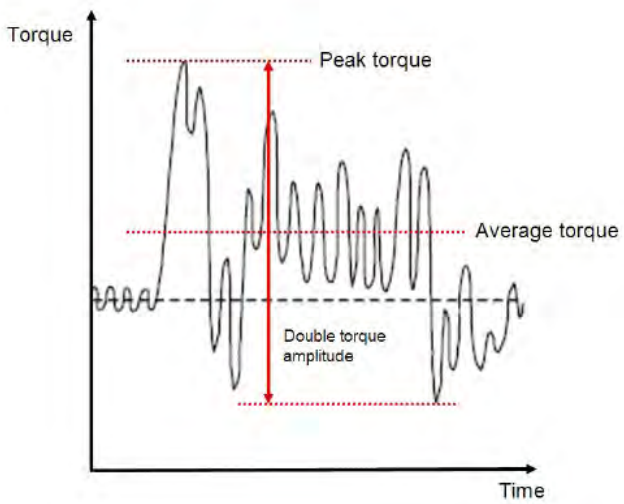

Fig.7.4 Response torque parameters

T_{peak} is the highest peak torque between the various lumped masses in the system. The highest torque amplitude during a sequence of impacts is to be determined as half of the range from max to min torque and is referred to as $T_{A\max}$:

$$T_{A\max} = \frac{1}{2} T_{\text{double}}.$$

Taking into account that milling sequence can start at the arbitrary moment the worst phase $T0$ between the milling sequence and any high torsional vibrations caused by engine excitations (e.g. 4th order engine excitation in direct coupled 2-stroke plants with 7-cyl. engine) should be considered in the analysis.

Some samples of ice excited torsional vibration calculations for directly driven installation are presented below. The project has first blade order resonance at 103 rpm, Fig. 7.5.

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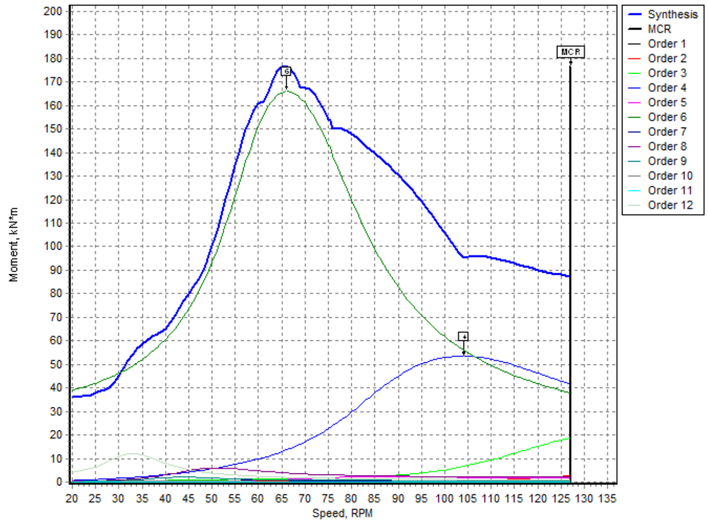

Fig.7.5 Steady response of the propeller shaft

The moment when the milling sequence shall be applied defines the initial conditions for time domain transient vibration calculation: twisting angles $X(T0)$ and twisting velocities $\dot{X}(T0)$ at the mass-elastic scheme nodes. Graphs of twisting angles and twisting velocities for propeller shaft are shown in Fig. 7.6.

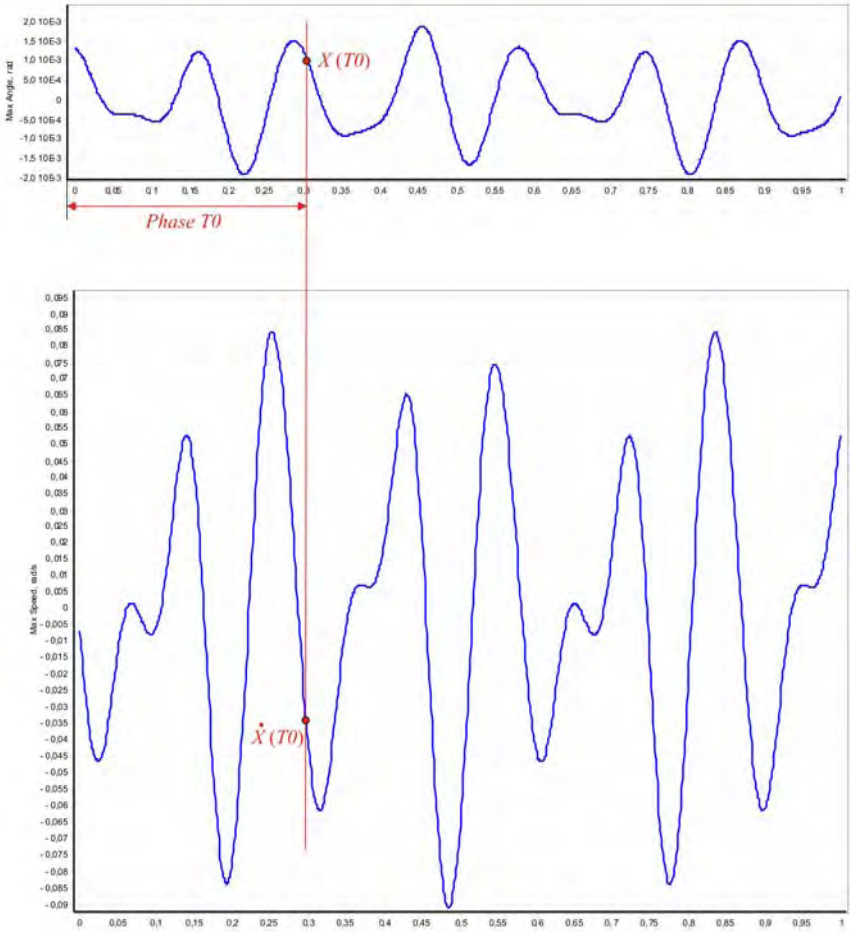

Fig.7.6 Twisting angles and twisting velocities in steady response at 103 rpm

Calculation at 103 rpm with zero initial conditions gave the maximum response torques Fig.7.7. As can be expected the maximum torque is excited in the propeller shaft. Propeller shaft torque was calculated without and with engine excitation, Fig.7.8, 7.9. Engine throw 2 torque is shown in Fig.7.10.

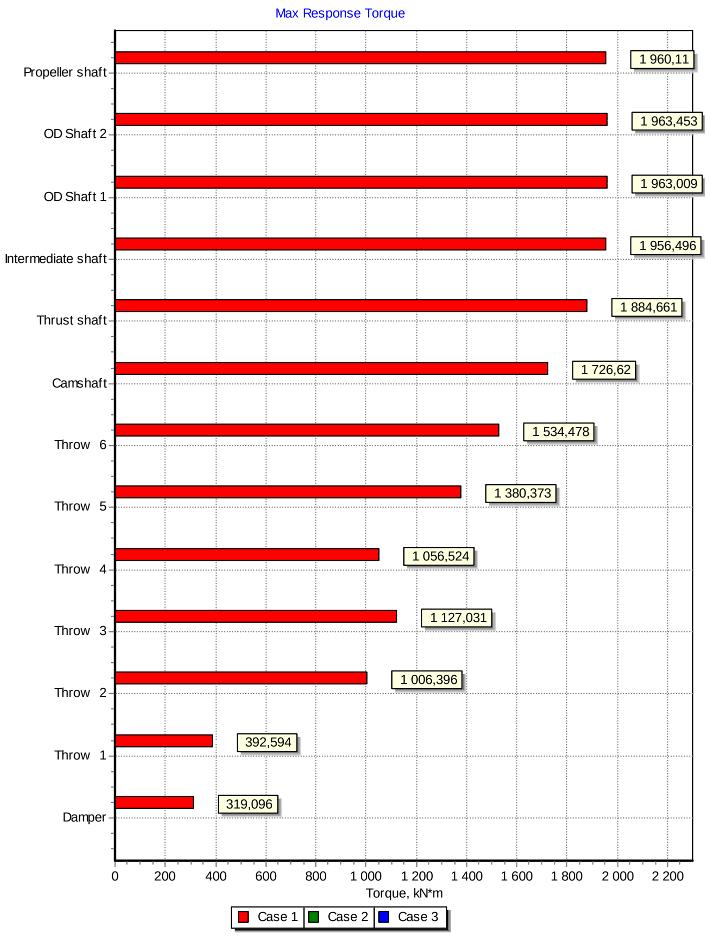

Fig.7.7 Maximum response torques due to ice impacts at 103 rpm

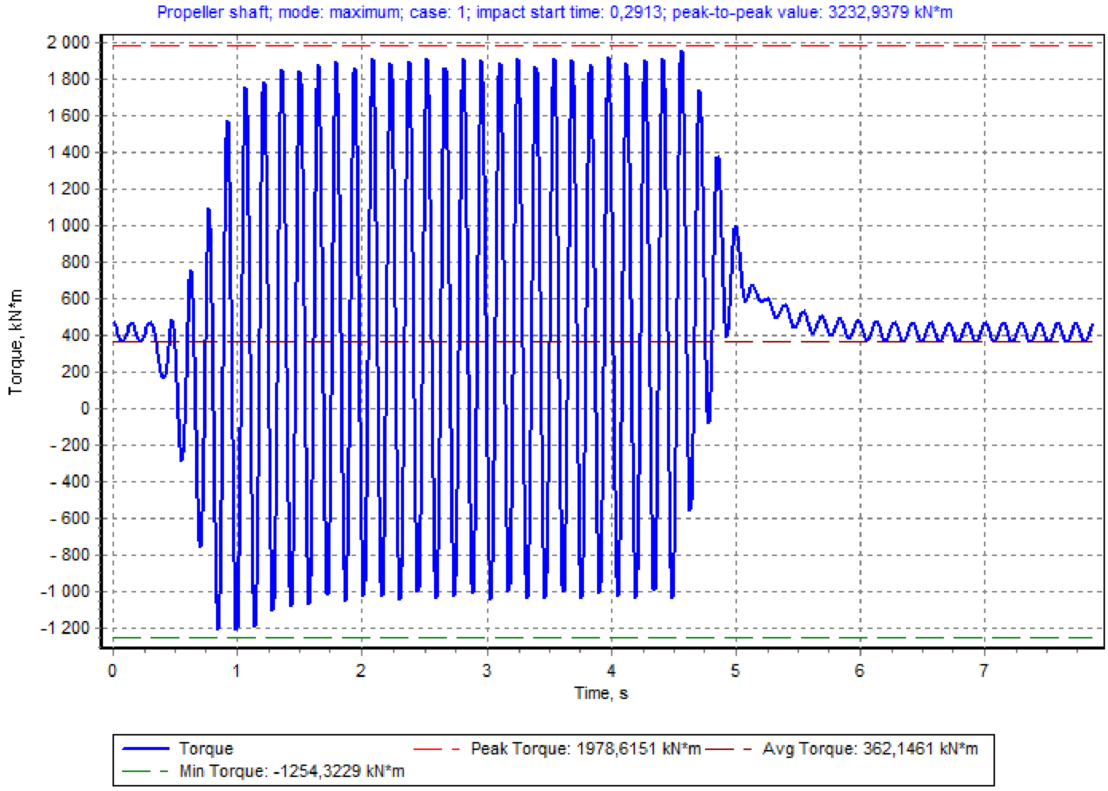

Fig.7.8 Propeller shaft torque (no engine excitation) at 103 rpm

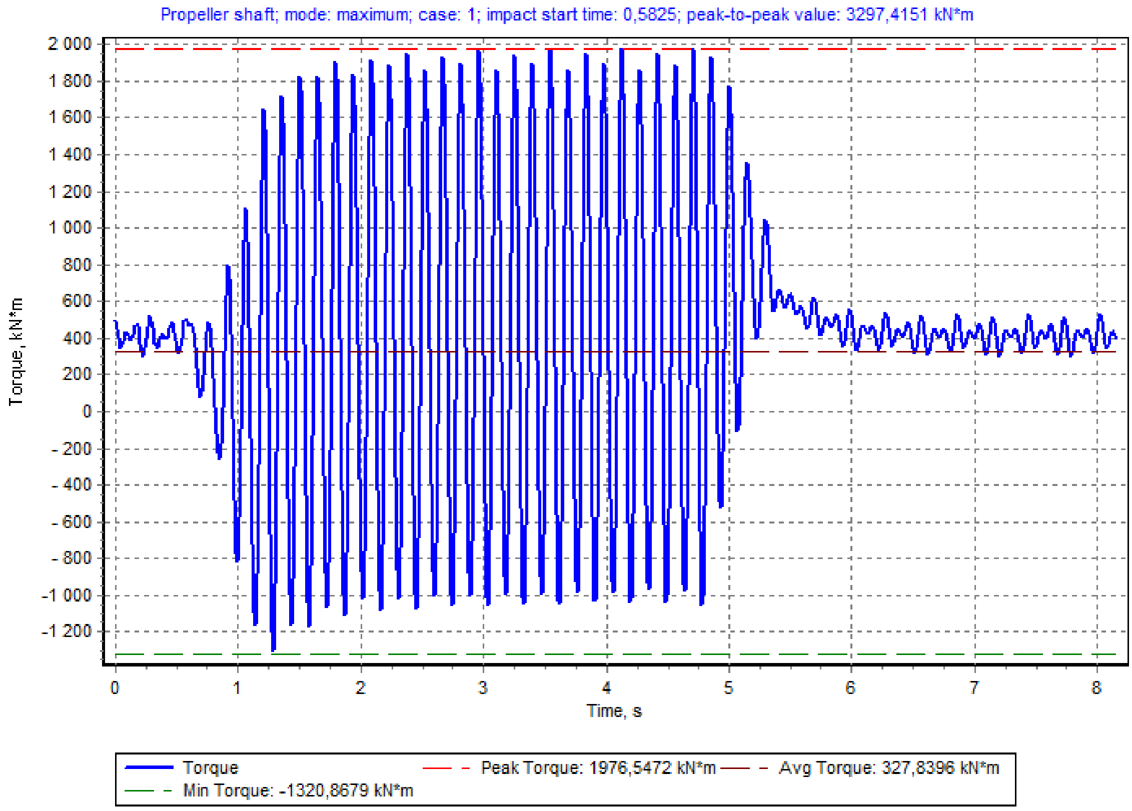

Fig.7.9 Propeller shaft torque (with engine excitation)

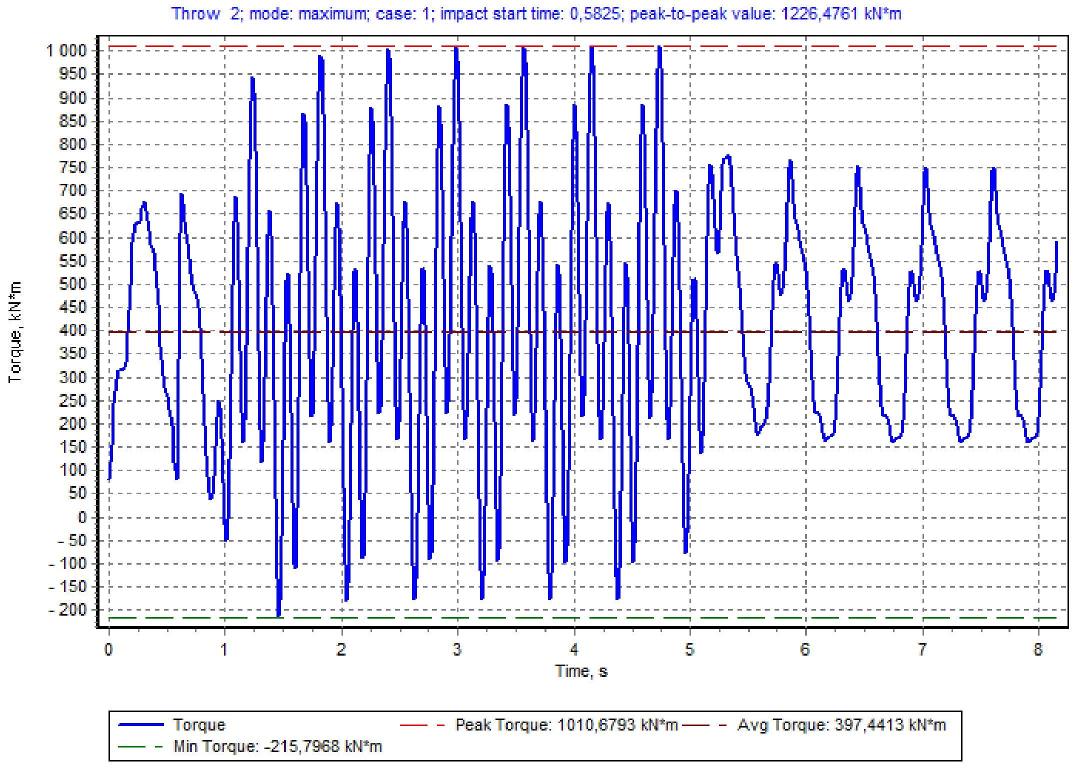

Fig.7.10 Engine throw 2 torque

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