SHAFTLINE DESIGN CONSIDERATIONS OF FIVE-CYLINDER LOW-SPEED PROPULSION PLANTS*

Gojko Magazinović, M.Sc.

CADEA d.o.o.	University of Split, FESB
Trg M. Pavlinovića 6	R. Boškovića b.b.
21000 Split, Croatia	21000 Split, Croatia

Torsional vibration behavior of the recent five-cylinder two-stroke low-speed marine diesel engine propulsion plants is more complex than ever. Therefore, this paper stresses the importance of a proper torsional vibration analysis of a shafting system in an early stage of the project development cycle. Three basic design approaches, that is the flexible shafting system, the rigid shafting system and the shafting system with a torsional vibration damper are reviewed and discussed. One of them, the flexible shafting system approach is emphasized and described in more detail. Besides, some practical design and manufacturing recommendations are given. Finally, an example five-cylinder diesel engine shafting system design is presented. The sea trial vibration measurements have proved the concept as well as the performed calculations.

PROJEKTIRANJE OSOVINSKIH VODOVA PETCILINDARSKIH SPOROHODNIH PORIVNIH SUSTAVA

Gojko Magazinović, mr.sc.

CADEA d.o.o.	Sveučilište u Splitu, FESB
Trg M. Pavlinovića 6	R. Boškovića b.b.
21000 Split	21000 Split

Obilježja torzijskih vibracija suvremenih petcilindarskih, dugostapajnih, sporohodnih dizel motornih postrojenja su složenija nego ikada. Zbog toga ovaj rad naglašava važnost temeljite analize torzijskih vibracija porivnoga sustava već u ranoj fazi razvoja projekta. U radu su prikazana i raspravljena tri načelna pristupa projektu osovinskoga voda. To su: podatljivi osovinski vod, kruti osovinski vod i osovinski vod s ugrađenim prigušivačem torzijskih vibracija. Posebno je naglašen i obrađen prvi pristup kod kojega se u postrojenje ugrađuje izrazito podatljivi osovinski vod. Pored toga, dane su preporuke za konstrukcijska i proizvodna poboljšanja projekta. Konačno, u radu je dan primjer projekta jednoga stvarnoga petcilindarskoga dizel motornog porivnoga sustava. Mjerenja vibracija na pokusnoj plovidbi potvrdila su valjanost pristupa, kao i izvršenih proračuna.

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INTRODUCTION

In the last few decades we have been faced to the dramatic changes in the shipping and shipbuilding practice. Worldwide utilization of low-speed, two-stroke marine diesel engines with a low number of cylinders has taken place. Besides, shaftline length tends to be minimized, while stroke to bore ratios, as well as mean indicated pressures, permanently rise. All of these factors have resulted in an obvious fact that the torsional vibration behavior of the recent marine diesel engine propulsion plants became more complex than ever [1].

In the light of the modern trends in shipbuilding practice, five-cylinder two-stroke low-speed marine diesel engine propulsion plants need special attention and consideration.

PROBLEM FORMULATION

Due to the economic pressure to reduce production, as well as operation costs, the five-cylinder two-stroke low-speed diesel engine propulsion plants emerge as a natural answer to the growing needs. Unfortunately, such propulsion plants are faced to the significant torsional vibrations.

If the diameter of the shafting is chosen according to the Classification Society Rules, the resulting torsional vibration stresses are significantly higher than the permissible limits, Fig. 1. Figure 1 represents the typical situation when the shafting design follows the Classification Society Rules. The lower stress limit τ_c is applied to continuous engine running, and it may be exceeded for a short time only, requiring a barred speed range. On the other hand, the upper stress limit τ_t may not be exceeded at all, and is applicable for the transient running only. Therefore, from the torsional vibration viewpoint, Figure 1 depicts a completely unacceptable shafting design solution.

In order to improve the situation and to obtain acceptable torsional vibration behavior of the system, the design engineer has to choose among the three possible design approaches. The first of them, so called the flexible shafting system, is a very common solution [2]. Its counterpart, so called the rigid shafting system is also a common design solution [2]. Finally, the third design approach incorporates the mounting of a torsional vibration damper. According to major engine licenser [2], this design solution is not very common.

REVIEW OF POSSIBLE SOLUTIONS

A - The flexible shafting system

This design approach incorporates the following measures:

- the intermediate shaft, as well as the propeller shaft, has to be manufactured from the high quality steel with the highest ultimate tensile strength,

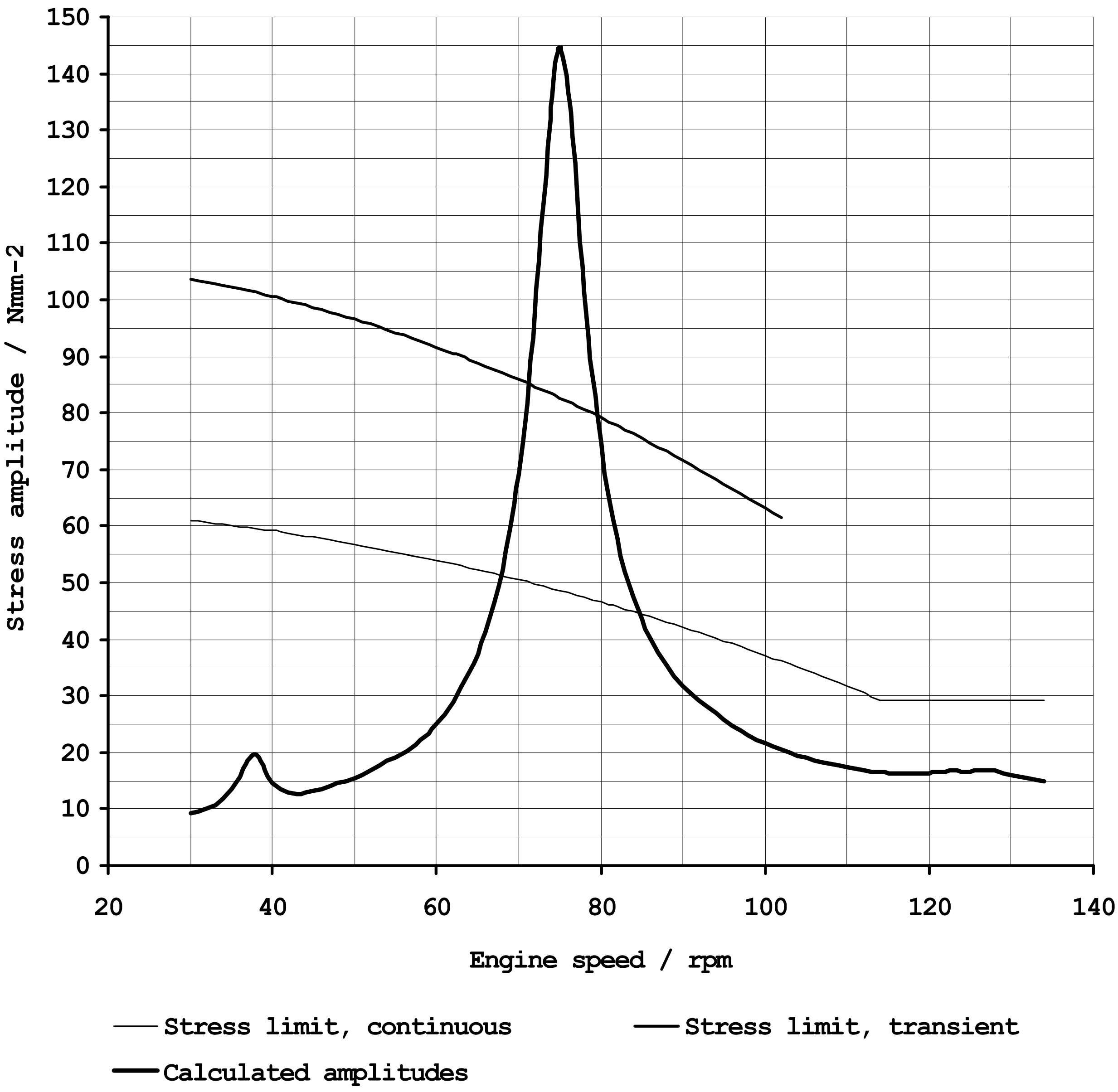


Figure 1. Vibratory stress amplitudes in the intermediate shaft of a sample shaftline (diameters chosen according to the Classification Society Rules)

- shafting diameters have to be reduced bellow the class diameters,
- tuning wheel of the appropriate inertia has to be mounted on the front end of the engine crankshaft.

These measures result in the following achievements:

- permissible stress limits rise, as they depend on the material quality and the shaft diameters,
- the main critical resonance, as well as the barred speed range, moves to the lower engine speed,
- engine excitation falls,
- torsional vibration stresses reduce due to the lower excitation and due to better dynamic behavior of the whole system.

B - The rigid shafting system

This design approach is very simple in its nature. The only measure needed is to increase the intermediate and propeller shaft diameters to far above the class diameters. Due to changed dynamic behavior of the whole system, this measure results in the following achievements:

- the main critical resonance moves to far above the nominal engine speed,
- torsional stresses are lower than permissible stress limits for continuous running in the whole engine speed range.

C - The shafting system with a torsional vibration damper

According to this design approach, the problem of the unacceptably high vibration stresses is solved by means of a torsional vibration damper. The torsional vibration damper is a device which has to be mounted on the front end of the engine crankshaft. Due to the improved dynamic behavior of the whole system, this measure caused the following achievements:

- torsional stresses are lower than the permissible stress limits for continuous engine running in the whole engine speed range,
- barred speed range may be completely removed,
- in some cases the barred speed range may be also required, but the torsional stresses are still bellow the permissible stress limits for transient conditions (τ_t).

The more complete list of advantages and disadvantages of the possible design solutions is given in Table I.

CALCULATION PROCEDURE

The dynamic behavior of the torsional system [3-5] may be described with a system of the linear differential equations, presented in matrix form [6]:

$$\mathbf{J}\,\varphi'' + \mathbf{C}_T\,\varphi' + \mathbf{G}_T\,\varphi' + \mathbf{K}_T\,\varphi = \mathbf{f}_T \ , \qquad (1)$$

where:

- $\mathbf{J}$ – mass moment of inertia matrix (diagonal matrix with main diagonal elements $j_{ii} = J_i$),
- $\mathbf{C}_T$ – torsional absolute damping matrix (diagonal matrix),
- $\mathbf{G}_T$ – torsional relative damping matrix (includes the internal damping and other miscellaneous damping),
- $\mathbf{K}_T$ – stiffness matrix,
- φ – angular displacement vector,
- $\mathbf{f}_T$ – excitation moment vector (includes the engine excitation, as well as the propeller excitation).

Table I - Advantages and disadvantages of the proposed design approaches

Solution	Advantages	Disadvantages
A	• very common solution [2] • the cheapest solution [2] • torsional vibration problem is solved in a "natural" way, by utilizing dynamic characteristics of the system • lighter, as well as cheaper, shaftline • smaller and cheaper shafting bearings • smaller and cheaper propeller hub • lower torsional vibration induced propeller thrust • simple shaft alignment • hull's afterbody design is not influenced by the shafting diameters	• tuning wheel mounting is necessary • high-grade steel for shafting is needed • high quality manufacturing is essential • the plant will have a barred speed range • very accurate torsional vibration analysis is needed • shafting is subjected to high torsional stresses
B	• common solution [2] • torsional vibration problem is solved in a "natural" way, by utilizing dynamic characteristics of the system • the plant will have not a barred speed range • low-grade steel for shafting is needed • standard quality manufacturing is appropriate • shafting is subjected to low torsional stresses	• high torsional vibration induced propeller thrust is very possible [2] with a consequence of the ship's super-structure excessive vibrations • large shafting bearings have to be applied • high bearing reactions are usual • propeller hub is greater, as well as the propeller cost • shaft alignment is harder • hull's afterbody design is influenced by the shafting high diameters
C	• the barred speed range is not necessary • shafting is subjected to moderate torsional stresses • medium-grade steel for shafting is needed • standard quality manufacturing is appropriate • lower torsional vibration induced propeller thrust • simple shaft alignment • hull's afterbody design is not influenced by the shafting diameters	• not very common solution [2] • torsional vibration problem is solved by the device which may be subjected to failure • very accurate torsional vibration analysis is needed • mounted damper significantly increases the overall cost of the plant (order of magnitude is $100.000)

Legend:

A - Flexible shafting system; B - Rigid shafting system; C - Shafting system with a torsional vibration damper

All the matrices, that is the inertia matrix, the absolute damping matrix, the relative damping matrix, as well as the stiffness matrix, are the characteristics of the system. On the other hand, the excitation moment vector represents the forces imposed to the system.

The system main excitation origins from the variable gas pressure produced in the cylinder. Besides, this excitation is superimposed with the excitation produced by the inertial forces of the crankshaft/connecting rod mechanism, Figure 2.

Excitation moment is periodic in its nature. Therefore, using the Fourier analysis, it can be easily broken down into a number of appropriate harmonic excitation moments. The n'th order excitation moment may be represented with [2]:

$$f_n = T_n \sin(n\varpi t + \Theta_n) \ , \tag{2}$$

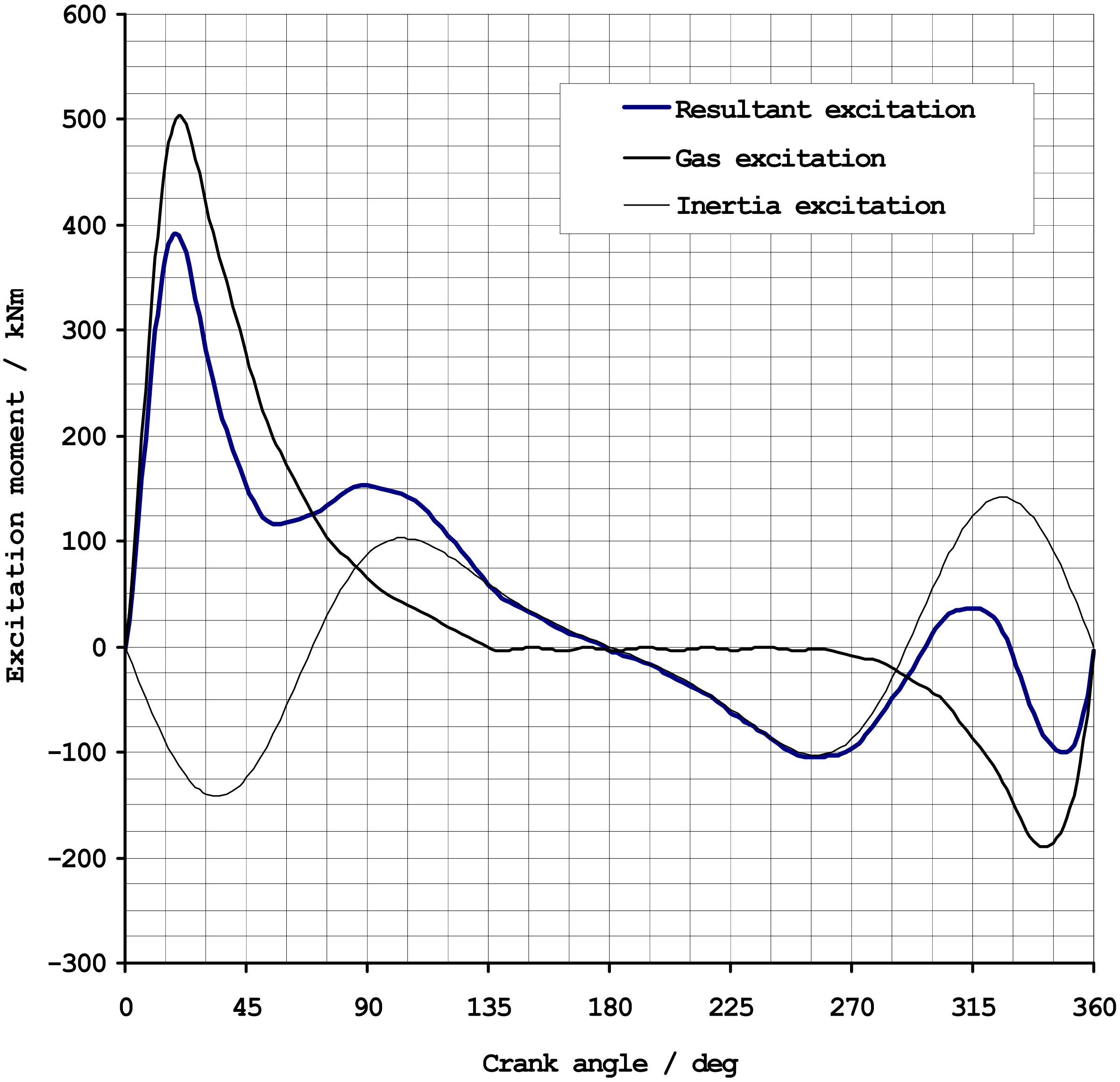


Figure 2. An example of engine excitation produced in one cylinder

where:

T_n – n'th order excitation moment amplitude, Nm,
n – order of excitation,
ϖ – engine angular velocity, rad/s,
t – time, s,
Θ_n – n'th order excitation phase angle, rad.

For linear systems [2], the n'th order excitation (2) produces the n'th order response:

$$\varphi_n = \Phi_n \, sin(n\varpi t + \Psi_n) \; , \tag{3}$$

where:

Φ_n – n'th order response amplitude, rad,
Ψ_n – n'th order response phase angle, rad.

Taking in account equations (2) and (3), multiple solving of system (1), for each excitation order, enable determination of the unknown angular displacements φ:

$$\varphi = \sum_{1}^{N} \Phi_n \, sin(n\varpi t + \Psi_n) \; , \tag{4}$$

where N is a number of applied excitation orders. The knowledge of the vibratory angular displacements of all masses in the system enables determination of the vibratory torques. After that, the vibratory torsional stress calculation is straightforward.

The author has developed the *VIBRAP* computer program to solve for the vibratory responses in torsional vibration. Based on the prepared input data the program provides:

- undamped natural vibration analysis that includes natural modes and frequencies, vector summations and critical speeds,
- forced damped vibration analysis that includes angular displacements, torque and vibration stresses, as well as the response synthesis.

The *VIBRAP* computer program is very efficient, designed to solve the most complicated multi-branched shafting systems.

SAMPLE ANALYSIS

In order to present a shaftline design of five-cylinder low-speed propulsion plant, a specific real-life problem was considered. The sample vessel was a 45.000 dwt bulk carrier, Table II.

According to plant specification, using the Classification Society Rules as a guide, Design Department proposed the following shaftline initial design, Table III.

Torsional vibration analysis of the proposed plant has shown that the main resonance vibratory stresses, at the engine speed of 75.2 rpm, were excessively high and beyond of all permissible limits, as shown in Figure 1.

Table II. Machinery data

Engine	**MAN B&W 5S50MC**
MCR output	**7150 kW**
MCR speed	**127 rpm**
Number of cylinders	**5**
Cylinder diameter	**500 mm**
Stroke	**1910 mm**
Connecting rod length	**2190 mm**
Crankpin diameter	**560 mm**
Reciprocating mass per cylinder	**3229 kg**
Firing order	**1-4-3-2-5**
Mean indicated pressure	**19 bar**
Maximum pressure	**140 bar**
Flywheel inertia	**1625 kgm^2**
Shafting length (overall)	**13619 mm**
Propeller inertia (in water)	**17600 kgm^2**

Table III. Shaftline initial design

Intermediate shaft material	**Forged steel, 440 N/mm^2 UTS**
Intermediate shaft diameter	**380 mm**
Intermediate shaft length	**6680 mm**
Propeller shaft material	**Forged steel, 440 N/mm^2 UTS**
Propeller shaft diameter	**460 mm**
Propeller shaft length	**6939 mm**
Flange fillet radius	**40 mm**

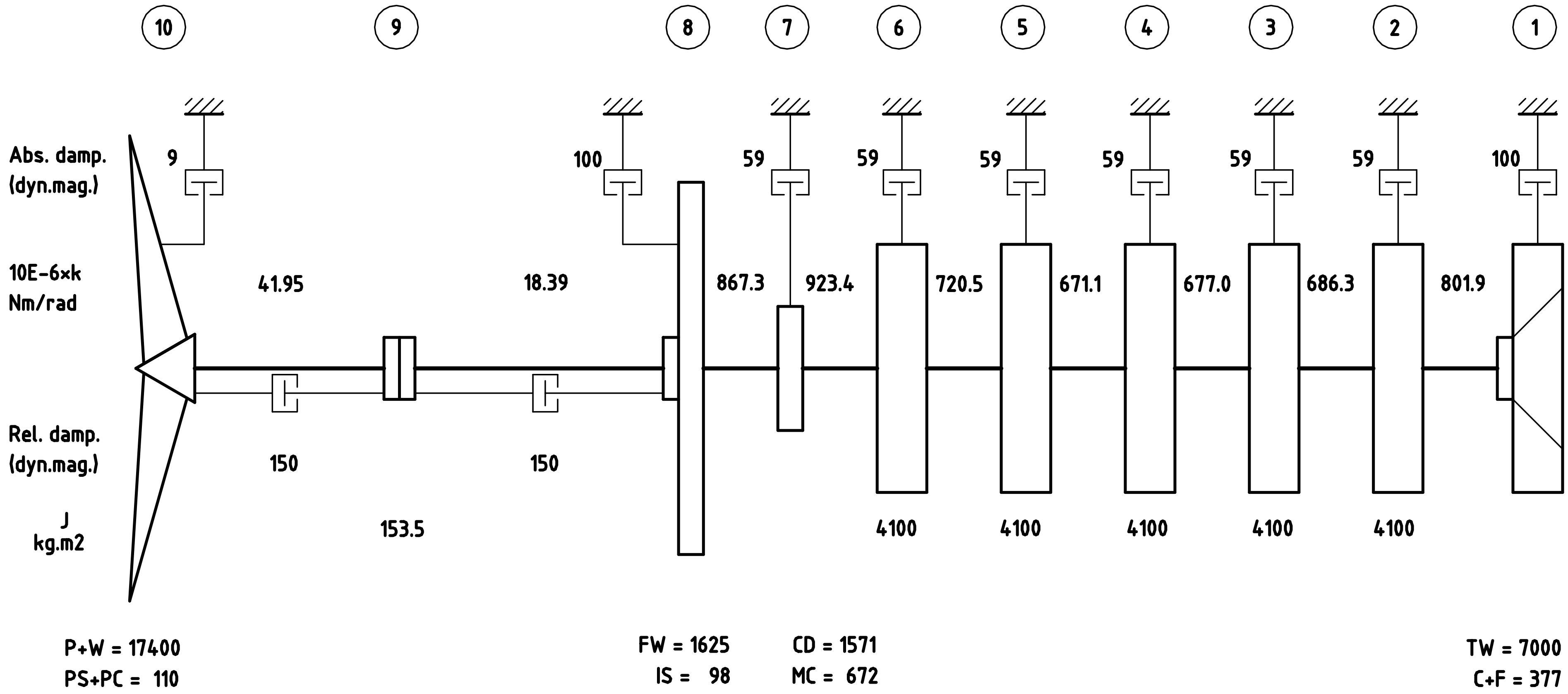
Due to the initial design inability to conform to the Classification Society Rules, the alternative designs were considered. The results of the consideration may be summarized in the following:

- mounting of the torsional vibration damper is a costly solution and therefore it was abandoned,
- the rigid shafting design solution may provoke the ship super-structure excessive vibrations due to high torsional vibration induced propeller thrust, and, therefore it was also abandoned,
- the flexible shafting design solution may fulfil the imposed requirements and therefore it was marked as the most reasonable design solution.

After many trials and errors, the final design of the shaftline has been chosen, Table IV:

Table IV. Shaftline final design

Tuning wheel inertia	**7000 kgm^2**
Intermediate shaft material	**Forged steel, 800 N/mm^2 UTS**
Intermediate shaft diameter	**350 mm**
Intermediate shaft length	**6680 mm**
Propeller shaft material	**Forged steel, 600 N/mm^2 UTS**
Propeller shaft diameter	**423 mm**
Propeller shaft length	**6869 mm**
Flange fillet radius	**113 mm**



LEGEND:

P+W - Propeller + added water
PS - Propeller shaft
PC - Propeller cap
FW - Turning wheel
CD - Camshaft drive + thrust cam

IS - Intermediate shaft
MC - Moment compensator
TW - Tuning wheel
C+F - Chain drive + flange

1 - Tuning wheel
2 - Cylinder No. 1
3 - Cylinder No. 2
4 - Cylinder No. 3
5 - Cylinder No. 4

6 - Cylinder No. 5
7 - Camshaft drive
8 - Turning wheel
9 - Flange
10 - Propeller

Figure 3. Torsional scheme of the propulsion plant

Dynamic scheme of the final system is given in Figure 3 and the *VIBRAP* computer program was used to obtain the vibratory response. These results for the intermediate shaft are given in Figure 4. As required, all the Classification Society Rules regarding the torsional vibrations were met.

The actual response of the system was measured during the sea-trials. The strain gauge method torsional vibration measurements were performed by the specialists of the Marine Research & Special Technologies, Zagreb. The Full Wheatstone Bridge, mounted at the midpoint of the intermediate shaft, was composed of the *HBM Type 6/350 XY 21* strain gauges while the wireless signal transmission was enabled using the *ETF ODP-1, ETF APMO-2* and *ETF IPMO-2*. Other equipment used were the *HP 3310A* function generator, the *B&K 7007* tape recorder and the *B&K 2033* FFT analyzer.

The intermediate shaft vibratory stress amplitudes of the 5-th order, recorded during the sea trials, are summarized in Figure 4. As expected, the comparison of the measured and the calculated amplitudes confirms the agreement very well.

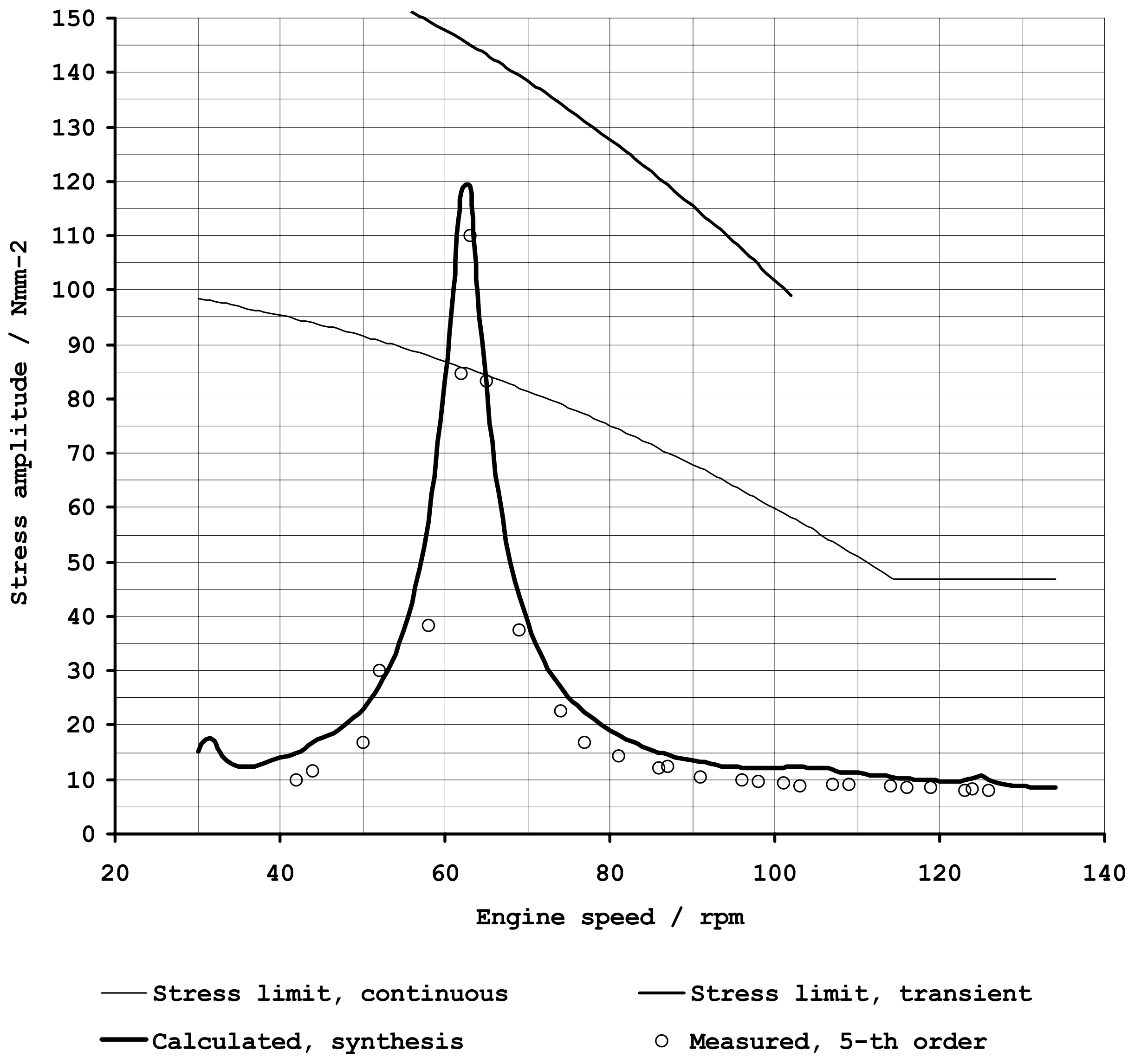


Figure 4. Vibratory stress amplitudes in the intermediate shaft of a final shaftline design

CONCLUSIONS

The problem of torsional vibration behavior of the five-cylinder two-stroke low-speed marine diesel engine propulsion plants may be satisfactory solved if the special attention, in an early stage of the project development cycle, is provided. Modern computer technology means exist to provide the reliable prediction of the shaftline behavior covering various driving conditions. A continuous research into and the development of the improved alternative designs is essential if the move forward in the reliability of the five-cylinder two-stroke low-speed marine diesel engine propulsion plants is needed.

Finally, two additional points have to be emphasized:

- Special care is needed during the shaftline detailed design. Flange fillets, as well as the other miscellaneous stress risers, have to be considered with maximum care. The mild fillets with the proper surface machining are essential for bounding the stress concentration factors.

- Shaft forging heat treatment has to be done in strict conjunction with the specified procedures. The proper heat treatment is a must for the reliable operation of the shafting made of high strength steels.

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