SIGNIFICANCE OF SHAFTING LENGTH: A SUEZMAX TANKER DESIGN PROBLEM*

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ABSTRACT

In this paper the influence of shafting length on torsional vibration behavior of suezmax tanker propulsion system is investigated and discussed. Besides, some countermeasures to harmful vibration stresses are also addressed. Finally, an example of suezmax tanker shafting design is given. The initial propulsion plant design, characterized with excessive vibration stresses is improved to satisfy the classification society stress limits and to save relatively short engine room length of 23 meters. For the sake of comparison, the two competitor's designs require the engine room lengths of more than 25 and 27 meters, respectively.

INTRODUCTION

The main propulsion shafting belongs to a narrow group of the most vital systems on the ship. It influences on reliability of the propulsion plant as well as the overall design of the vessel. Therefore, it is of great importance to design reliable and cost-effective shafting.

From the shipowner's point of view, the payable space of the ship has to be enlarged as much as possible, to improve the transportation efficiency and to increase their revenues. That means that the engine room, as a *non-payable* space, has to be reduced, i.e. shortened, as much as possible. On the other hand, shipbuilders tend to satisfy the prospective shipowner's needs in order to be competitive on the turbulent world market scene. Therefore, all these factors stress the importance of the shafting length, as one of the main factors influencing the engine room overall length. Namely, shorter shafting opens the possibility of engine room length reduction. Unfortunately, short propulsion shafting usually generates excessive torsional vibration stresses in the shafting and therefore it could not be reduced endlessly.

This paper stresses the importance of a proper torsional vibration analysis of a shafting system in an early stage of the project development cycle. Namely, it is widely accepted that the torsional vibration behavior of the shafting system has the major impact on the dimensions chosen, e.g., Long, C.L., 1980; Hakkinen, P., 1987.

SCANTLINGS OF INTERMEDIATE AND PROPELLER SHAFTS

According to International Association of Classification Societies; Anon., 2000a; the minimum diameter of an intermediate shaft is not to be less than the calculated from the following formula:

$$d = F \cdot k \cdot \sqrt[3]{\frac{P}{n} \cdot \frac{1}{1-\left(d_i/d_o\right)^4} \cdot \frac{560}{\sigma_B + 160}} \qquad (1)$$

where:

- d – minimum diameter, mm
- d_i – actual diameter of shaft bore, mm
- d_o – actual outside diameter of shaft, mm
- F – factor for type of propulsion installation
- k – factor for different shaft design features
- P – rated power of the main engine, kW
- n – rated speed of intermediate shaft, min^{-1}
- σ_B – tensile strength of the shaft material taken for calculation, N/mm^2.

Similarly; Anon., 2000b; the minimum diameter of the propeller shaft is not to be less than that calculated from the following formula:

$$d_p = F \cdot k \cdot \sqrt[3]{\frac{P}{n} \cdot \frac{560}{\sigma_B + 160}} \qquad (2)$$

where d_p is a propeller shaft minimum diameter and n is a rated speed of propeller shaft. Other symbols are defined in Eq. (1).

These two equations neglect bending loads, as well as alternating, mainly torsional, loads. Moreover, they neglect the influence of stress risers, too. However, despite their simplicity, these equations still provide a sound basis for preliminary design of intermediate and propeller shafts. However, it should be clearly understood that the final shafting diameters may be established after a thorough torsional vibration analysis of the propulsion system only.

SHAFTING LENGTH

The length of shafting is a function of main engine location. In general, the main machinery is located as far aft as practicable; Long, C.L., 1980. The propeller shaft length is mainly determined by the shape of the hull afterbody and therefore, the only variable is the intermediate shaft length. To save an engine room space a short intermediate shafts are favorable, but it should be clearly realized that it could not be reduced endlessly. Namely, two factors are especially significant:

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- When developing a shafting arrangement the provisions for removing propeller shaft from the stern tube must be considered,
- Short propulsion shafting usually generates excessive torsional vibration stresses in the propulsion system.

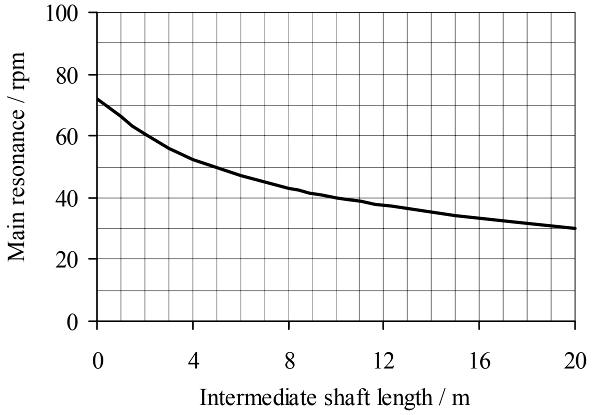


Figure 1: The main resonance as a function of intermediate shaft length (corresponds to I-node natural frequency)

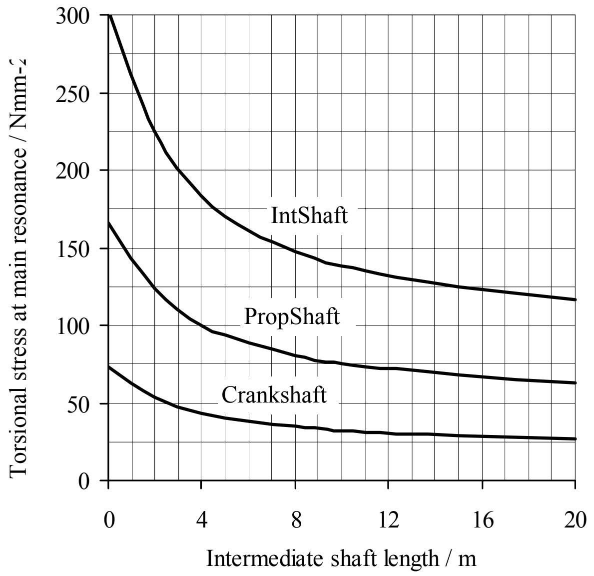


Figure 2: Peak torsional stress as a function of intermediate shaft length

The general influence of the intermediate shaft length on the torsional vibration behavior of the propulsion system is shown in Figs. 1 and 2. Fig. 1 shows the main critical speed variation as a function of the intermediate shaft length. On the other hand, Fig. 2 shows the variation of the peak torsional vibration stress at the main critical speed given in Fig. 1. The results presented in Figs. 1 and 2 correspond to suezmax tanker propulsion system preliminary design, described in more detail in the next section. In the case of changed propulsion system, the actual values will be different, but the nature of characteristics shown will remain unchanged.

SUEZMAX TANKER DESIGN PROBLEM

Brodosplit Shipyard, Croatia, has accomplished a *SUPERCARGO* project in which the new generation of full hull-form cargo vessels is developed. The design of a 167,000 dwt suezmax tanker, Figs. 3 and 4; Čudina, P., 2001; is developed under the following two performance requirements:

- Deadweight to be as large as possible;
- Speed, daily fuel oil consumption and service range to be in accordance with modern suezmax tanker designs.

The main particulars of the designed tanker are given in Fig. 3. The tanker is capable to sail through the Suez Canal on the draught of 17.06 meters carrying 166,500 deadweight tons. The main engine fuel oil consumption amounts to 56.7 tons per day and the trial speed at *CSR* and design draught amounts to 15.8 knots. On the other hand, the service speed at *CSR*, design draught and a 15% sea margin amounts to 15.2 knots. Finally, the cruising range achieved amounts to approximately 23,000 nautical miles.

Designed tanker is comparable to recent suezmax tanker designs developed by some famous Far East shipbuilders (Daewoo, Halla, Hyundai and Samsung). Moreover, according to Čudina, P., 2001; it can carry through the Suez Canal from 6,500 to 18,300 deadweight tons more than the competitors; the cargo tanks volume is close to the volume of the largest design, and the daily fuel oil consumption is the best.

The chosen main engine is *Brodosplit MAN B&W 6S70MC-C* with the selected maximum continuous rating (*SMCR*) of 16,780 kW at 82 rpm and a continuous service rating (*CSR*) of 14,270 kW at 77.7 rpm.

The propulsion shafting designers were faced with the following problems:

- Propulsion system preliminary torsional vibration has shown that the peak vibration stresses were much higher than the generally allowed stress limits; Anon., 2000c; Fig. 5.
- The main engine tentative location gives little opportunity for improving dynamic characteristics of a shafting system.

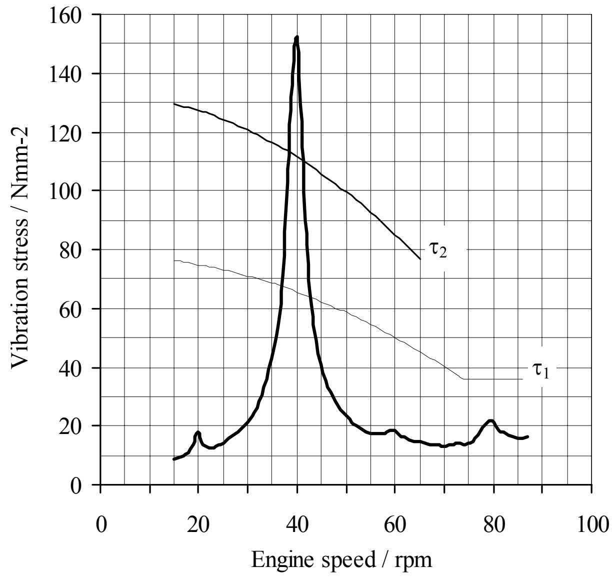


Figure 5: Torsional vibration stress in the intermediate shaft of the propulsion shafting initial design (stress limits for continuous, τ_1, and transient, τ_2, running are also included)

Before the design can progress further, satisfactory solution to torsional vibration problems must have been found. Therefore, available solutions were suggested and evaluated. In general, two design approaches were possible:

- The first approach incorporates the mounting of a torsional vibration damper. The torsional vibration damper is a device that has to be mounted on the front end of the engine crankshaft. Unfortunately, this approach is characterized with high investment costs of nearly US$ 100,000 a peace. Therefore, it was adopted as a *last chance* solution.
- The second one, so called the flexible shafting system concept;

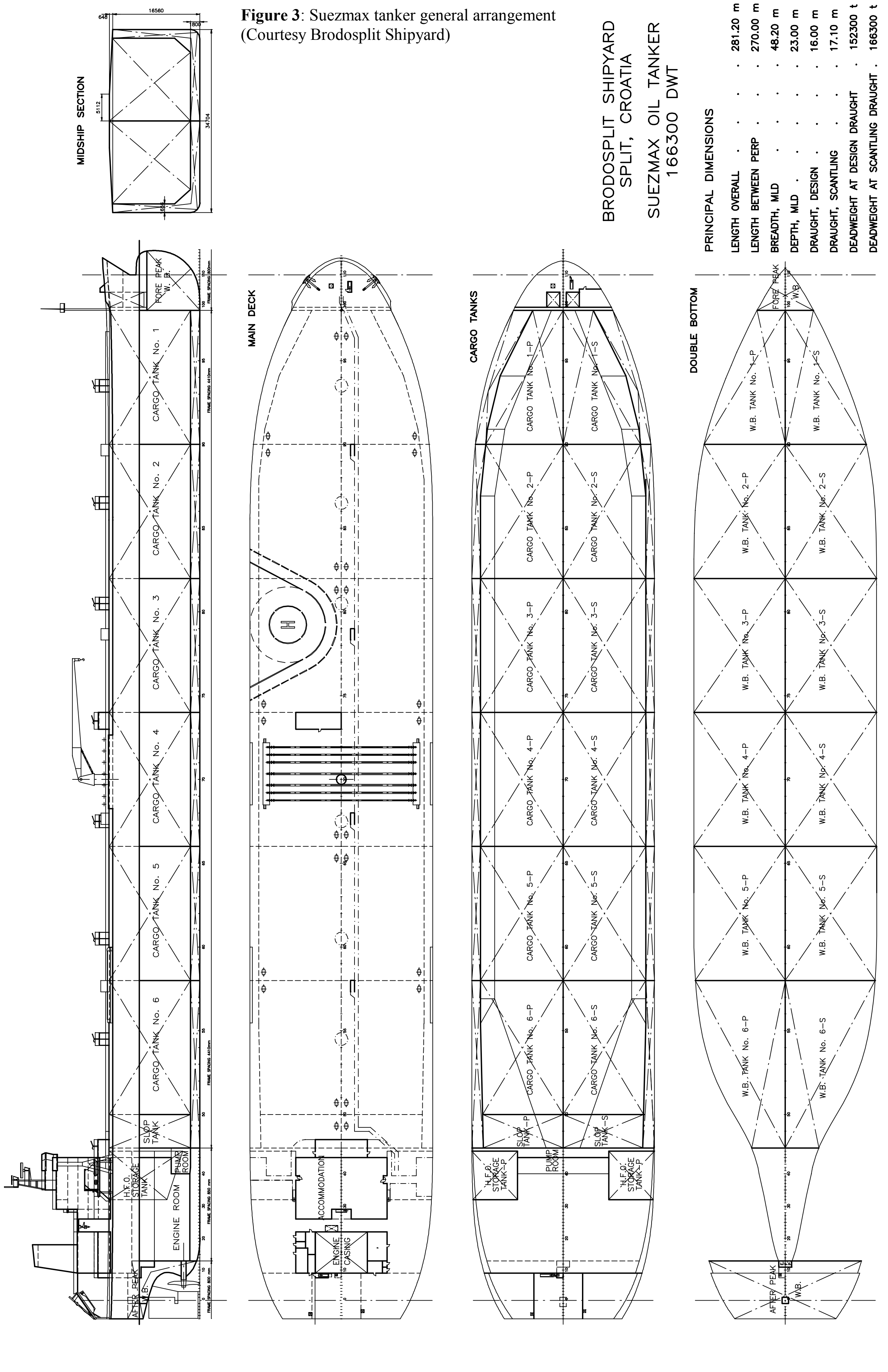


Figure 3: Suezmax tanker general arrangement (Courtesy Brodosplit Shipyard)

Anon., 1988; Magazinović, G., 2000a; in which shafting of high-strength steel is employed in conjunction with the enlarged turning wheel and/or tuning wheel. Additional supportive measure is locating the main engine in the foremost position allowed by the engine room arrangement.

The second approach is iterative in its nature and it may be stated and solved as an optimization problem; Magazinović, G., 1995. Although it may be solved numerically, on this occasion it is solved manually, by simple trial and error. Each design iteration contained a new shafting arrangement development and a complete torsional vibration analysis performed by the *TorViC* computer system; Magazinović, G., 2000b. To obtain propulsion system successful design, more than 12 iterations were needed.

CALCULATION PROCEDURE

The dynamic behavior of the torsional system may be described with a system of the linear differential equations, presented in matrix form; Magazinović, G., 2000b:

$$\mathbf{J} \cdot \varphi'' + \mathbf{C}_T \cdot \varphi' + \mathbf{G}_T \cdot \varphi' + \mathbf{K}_T \cdot \varphi = \mathbf{f}_T \qquad (3)$$

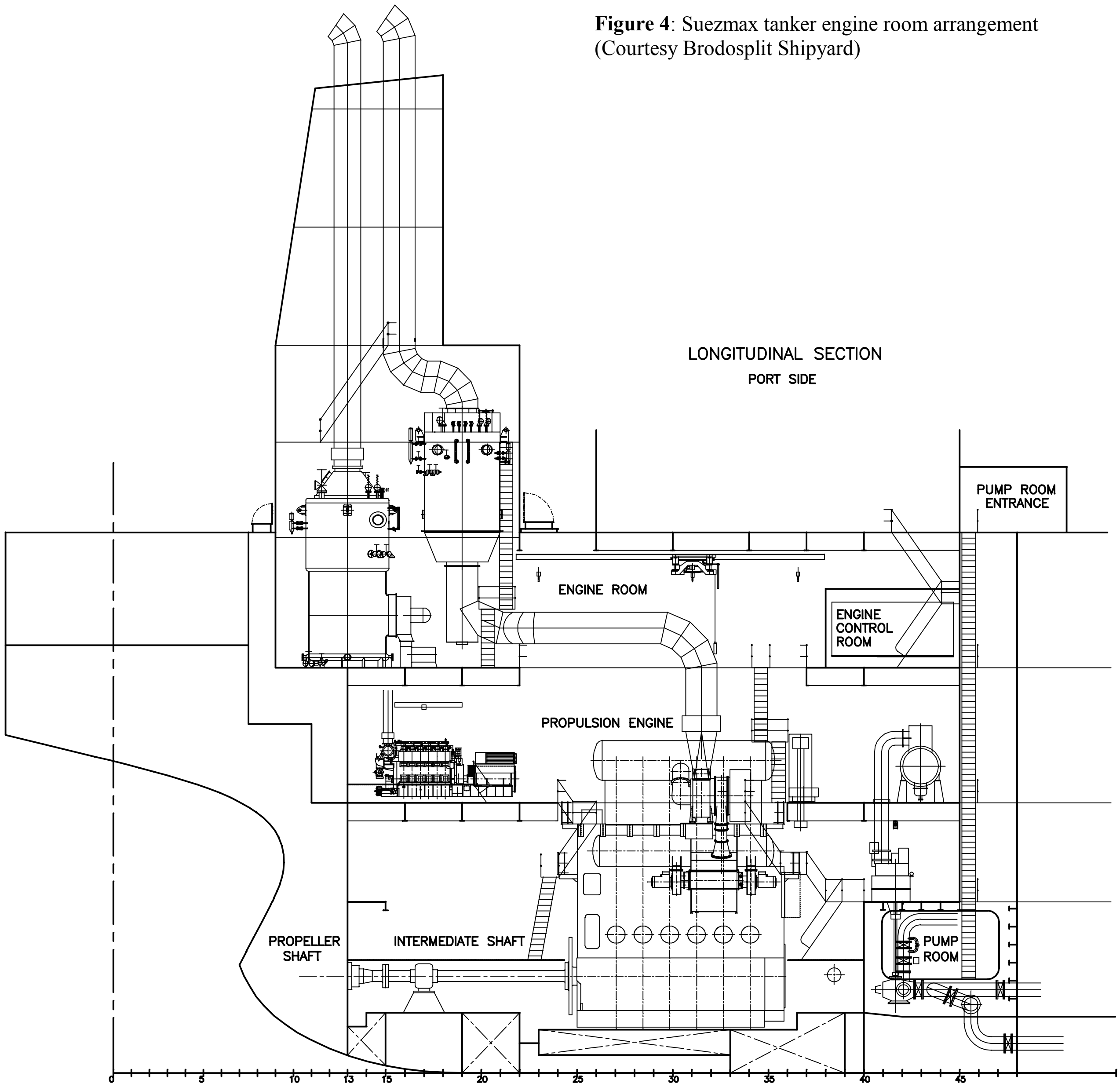
where:

- $\mathbf{J}$ – mass moment of inertia matrix (diagonal matrix),
- $\mathbf{C}_T$ – 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).

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 4: Suezmax tanker engine room arrangement (Courtesy Brodosplit Shipyard)



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:

$$f_n = T_n \cdot \sin\left(n\omega t + \Theta_n\right) \tag{4}$$

where:

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

For linear systems, the n 'th order excitation, Eq. (4), produces the n 'th order response:

$$\varphi_n = \Phi_n \cdot \sin\left(n\omega t + \Psi_n\right) \tag{5}$$

where:

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

Taking in account Eqs. (4) and (5), by multiple solving of Eq. (3) for each excitation order, determination of the unknown angular displacements φ is enabled:

$$\varphi = \sum_{1}^{N} \Phi_n \cdot \sin\left(n\omega t + \Psi_n\right) \tag{6}$$

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

RESULTS

The main particulars of the propulsion shafting final design are summarized in Table I.

Table I. Propulsion Shafting Final Design

Turning wheel inertia	13,150 $\text{kg} \cdot \text{m}^2$ (standard wheel)
Tuning wheel inertia	45,000 $\text{kg} \cdot \text{m}^2$
Intermediate shaft material	Forged steel, $\sigma_B = 600$ N/mm^2
Intermediate shaft length	8,000 mm
Intermediate shaft diameter	585 mm
Propeller shaft material	Forged steel, $\sigma_B = 600$ N/mm^2
Propeller shaft length	8,430 mm
Propeller shaft diameter	650 mm
Flange fillet radius	26 mm / 113 mm
Engine room length	22.95 m

Torsional vibration behavior of the shafting final design is thoroughly given in Magazinović, G., 2001. As an excerpt, Figs. 6 and 7 show torsional vibration stress variation in the intermediate shaft.

The barred speed range between 40 engine rpm and 49 engine rpm is imposed due to the I-node, 6th order torsional critical, when vibration stresses are higher than allowed for continuous running. However, the stress limits for transient running; Anon., 2000c; still remain satisfied.

In the event of one cylinder misfiring the maximum engine speed is not to exceed 62 rpm due to the I-node, 4th order torsional critical.

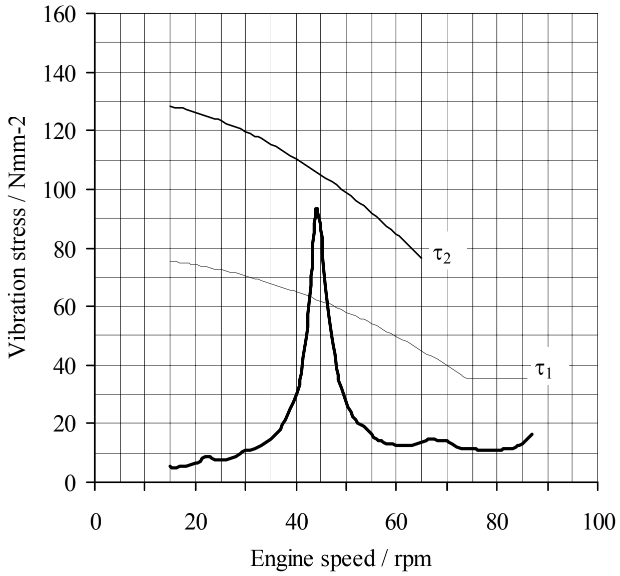


Figure 6: Torsional vibration stress in the intermediate shaft of the propulsion shafting final design – engine normal operation case. Stress limits for continuous, τ_1 , and transient, τ_2 , running are also included.

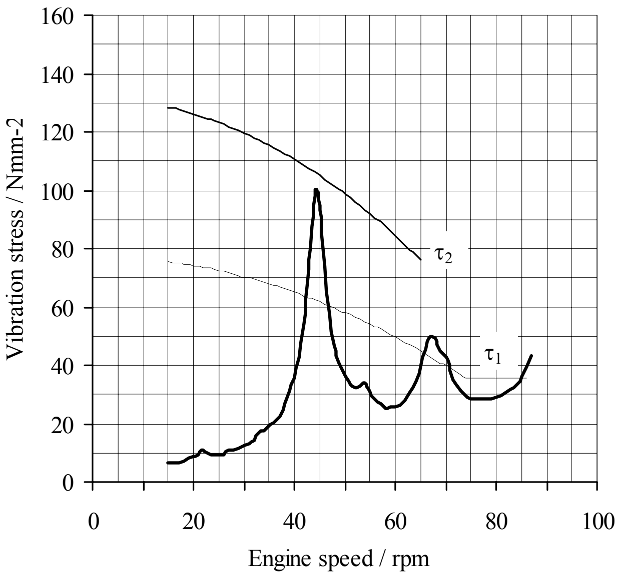


Figure 7: Torsional vibration stress in the intermediate shaft of the propulsion shafting final design – one cylinder misfiring operation case. Stress limits for continuous, τ_1 , and transient, τ_2 , running are also included.

Since all design requirements imposed by the classification society rules were satisfied, Bureau Veritas, as an actual classification society in charge, has approved the design.

Regarding the achieved engine room length, it is interesting to compare it with the available data for engine room lengths accomplished by two of the four competing shipbuilders stated in previous sections; on this occasion let's call them the *Competitor A* and *Competitor B*; Table II. Engine room length stated in Table II corresponds to distance between the aft bulkhead and the pump room station aft bulkhead.

Table II. Engine Room Length Comparison

Design	DWT	Engine	Length
Brodosplit	167 000	6S70MC-C	22.95 m
Competitor A	158 300	6S70MC	25.5 m (abt.)
Competitor B	153 000	6S70MC	27.2 m (abt.)

For the sake of honesty it should be stated that the 6S70MC engine is 0.5 to 0.7 meters longer than the MC-C engine type counterpart. However, it should be also noted that the achieved engine room length reduction is significantly higher than the engine length difference.

CONCLUSIONS

Propulsion shafting length becomes a significant factor in achieving efficient, cost-effective and reliable tanker designs. Shorter shafting enables engine room length reduction, but generally enlarges torsional vibration stresses too. Therefore, it could not be reduced endlessly.

The presented suezmax tanker design has numerous advantages compared with the designs of Far Eastern competitors; for details refer to Čudina, P., 2001. When propulsion shafting is concerned, it may be noted that presented design is characterized with relatively short engine room of slightly less than 23 meters. On the other hand, the two competitor's designs require engine room lengths of more than 25 and 27 meters, respectively. Such achievement was possible due to extensive torsional vibration analyses performed, during which more than 12 different shafting layouts were analyzed. Therefore, the key action in obtaining successful propulsion shafting system design was a proper torsional vibration analysis in the early stage of design evolution process.

Acknowledgments

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REFERENCES

Anon., *Vibration Characteristics of Two-Stroke Low Speed Diesel Engines*, 2. edition, revised from P.8703-165, MAN B&W Diesel A/S, Copenhagen, 1988.

Anon., *Scantlings of Intermediate Shafts*, Requirements Concerning Machinery Installations, Document M33, International Association of Classification Societies, London, 2000a.

Anon., *Scantlings of Propeller Shafts*, Requirements Concerning Machinery Installations, Document M37, International Association of Classification Societies, London, 2000b.

Anon., *Permissible Limits of Stresses due to Torsional Vibrations for Intermediate, Thrust and Propeller Shafts*, Requirements Concerning Machinery Installations, Document M48, International Association of Classification Societies, London, 2000c.

Čudina P., *Research & Development Supercargo Project (II), Brodogradnja*, 49 (2001) pp. 55-60.

Hakkinen P., *Are Torsional Vibration Problems All in the Past?*, The Motor Ship 9th International Marine Propulsion Conference, London, March 12-13 (1987) pp. 43-52.

Long C.L., *Propellers, Shafting, and Shafting System Vibration Analysis*, in: Harrington R.L. (Ed.), *Marine Engineering*, The Society of Naval Architects and Marine Engineers, New York, 1980.

Magazinović G., *Optimization as a Marine Propulsion Shafting System Design Tool*, Proceedings of the 10th International Conference on Engineering Design – ICED 95, Vol. 1, Prague, August 22-24 (1995) pp. 199-200.

Magazinović G., *Utility of High-Strength Steels for Main Propulsion Shafting Design*, Proceedings of the IX Congress IMAM 2000, Vol. I, Session B, Ischia, April 2-6 (2000a) pp. 1-8.

Magazinović G., *TorViC v1.1 – Program for Propulsion System Torsional Vibration Analysis: User's Guide*, CADEA, Split, 2000b. (in Croatian)

Magazinović G., *Torsional Vibration Analysis of the Propulsion System, Brodosplit Yards 433/434, Report TVC 116-1.00,* CADEA, Split, 2001.