UTILITY OF HIGH-STRENGTH STEELS FOR MAIN PROPULSION SHAFTING DESIGN*

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Abstract

The main propulsion shafting belongs to a narrow group of the most vital systems on the ship. It influences reliability of the propulsion plant as well as the overall design of the vessel. As high-strength steels possess a favorable property to allow reduced shafting diameters, which results in enlarged torsional flexibility of the shaftline, in this paper, the flexible shafting system approach is presented. In general, this concept offers better torsional vibration behavior of the propulsion system, reduces overall investment costs, enables smoother shaftline alignment and finally, permits shaftline, as well as the engine room, length reduction. Unfortunately, this approach introduces some unfavorable side effects. Therefore, some practical design and manufacturing notes are also given. Finally, two real-life main propulsion shafting design examples are presented. The sea trial vibration measurements proved the concept as well as the performed analyses.

Introduction

Shafting designers have a premiere task to design reliable and cost-effective shaftlines. On the other hand, shafting design is governed by the Rules set by the given classification society. Unfortunately, the Rule diameters neglect alternating loads and therefore, it should be used as a guide only. Namely, it is widely accepted that the torsional vibration behavior of the shafting system has the major impact on the dimensions chosen. 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 [1−3]. Besides, it investigates the influence of the applied material on the torsional vibration behavior of the shafting system.

Although the principles stated in this paper apply to all types of propulsion shafting systems, the discussion has been primarily directed towards a conventional arrangement of a low-speed diesel propulsion plant with a fixed-pitch propeller.

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Problem formulation

The design of a shafting system is, by necessity, an iterative process because the various system design parameters are, to some extent, mutually dependent [1]. Shafting designs have to fulfill at least the following requirements:

- Diameters chosen have to meet the minimum standards set by the classification society Rules,
- Vibration characteristics (torsional, axial and lateral) of the shafting system have to be satisfactory with respect to vibration limits set by the classification society Rules,
- Dimensions chosen have to provide shaftline smooth alignment,
- Dimensions chosen have to provide shaftline smooth mounting and removal,
- Shaftline length, as well as the engine room overall length, has to be minimized in order to maximize the ship's payload efficiency.

In general, mild steel is traditional choice for building shafts. Sometimes, however, high-strength steels are also used. Reduced shafting diameters result in enlarged torsional flexibility of the shaftline and offer better torsional vibration behavior of the propulsion system. Moreover, it reduces overall investment costs, enable smoother shaftline alignment and finally, permit shaftline, as well as the engine room, length reduction. To obtain reduced shafting diameters it is necessary to apply the steel with the enlarged specified minimum tensile strength. However, on the other hand, this approach introduces some unfavorable side effects.

Shaft diameters

Classification Society, which classes the vessel, defines shafting minimum diameters and these values cannot be reduced by any reason. Rule diameters are determined with the shaft's intended service, transmitted power, shaft speed and applied material.

According to the major Classification Society [4] the shaft minimum diameters are defined as (excerpts from the Rules):

Diameter of the intermediate or propeller shaft is to be not less than determined by the following formula:

$$d = F \cdot k \sqrt[3]{\frac{P}{R}\left(\frac{560}{\sigma_u + 160}\right)} \tag{1}$$

where:

d – shaft diameter, mm
F – factor describing shaft service
k – factor describing shaft design
P – transmitted power at MCR, kW
R – shaft speed at MCR, rpm
σ_u – specified minimum tensile strength (UTS) of the shaft material, N/mm^2

The specified minimum tensile strength of forging for shafts is to be selected within the following general limit [4]:

- Carbon and carbon-manganese steel – 400 to 600 N/mm^2
- Alloy steel – not exceeding 800 N/mm^2

When applied to propeller shaft design the specified minimum tensile strength is not to be taken as greater than 600 N/mm^2 [4].

The following figure depicts the influence of the selected ultimate tensile strength on the Rule diameters of the intermediate and the propeller shaft. Other parameters assumed are:

- rated power is 8240 kW
- rated speed is 122 rpm
- forged intermediate shaft with integral coupling flanges
- forged propeller shaft carrying a keyless propeller

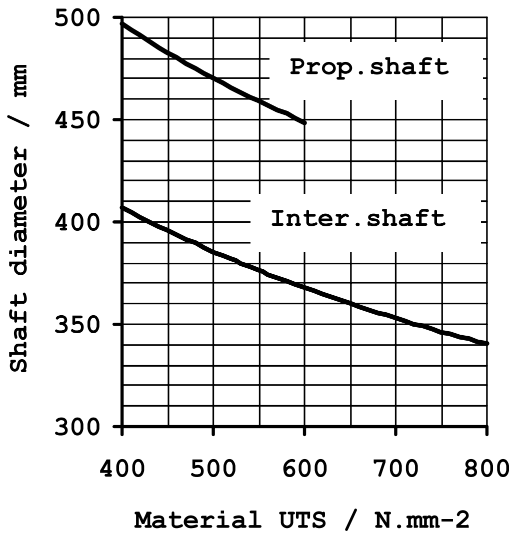

Figure 1. Material influence on the shaft minimum diameters

When compared to the basic ultimate tensile strength of 400 N/mm^2, the higher strength steel of 600 N/mm^2 allows the intermediate shaft as well as the propeller shaft diameter reduction for almost 10 %. Moreover, the high-strength steel of 800 N/mm^2 allows the intermediate shaft diameter reduction for over 16 %.

The Rule diameters neglect alternating loads and vibration behavior of the shafting system. Very often, due to vibration considerations, it is necessary to modify an otherwise satisfactory shafting system design. Namely, modern practice shows that the torsional vibration behavior of the shafting system has the predominant impact on the dimensions chosen. Therefore, the Rule diameters should be viewed as a guide only, not as a suggested one.

Torsional vibration stress limits

According to the world wide accepted Requirements [5] the torsional vibration stress limits are defined as follows (excerpts from the Document):

In no part of the propulsion system may the alternating torsional vibration stresses exceed the values of τ_1 for continuous operation and τ_2 for transient running.

For continuous operation the permissible stresses due to alternating torsional vibrations are not to exceed the values given by the following formula:

$$\tau_1 = \pm \frac{\sigma_\mathrm{u} + 160}{18} \cdot c_k \cdot c_D \cdot (3 - 2\lambda^2) \qquad \text{for } \lambda < 0{,}9 \qquad (2)$$

$$\tau_2 = \pm \frac{\sigma_\mathrm{u} + 160}{18} \cdot c_k \cdot c_D \cdot 1{,}38 \qquad \text{for } 0{,}9 \le \lambda \le 1{,}05 \qquad (3)$$

where:

τ_1 – permissible stress due to torsional vibrations for continuous operation, N/mm^2

c_k – factor for different shaft design

c_D – size factor

$$c_D = 0{,}35 + 0{,}93 \cdot d^{0{,}2} \qquad (4)$$

d – shaft diameter, mm

λ – speed ratio

$$\lambda = n/n_0 \qquad (5)$$

n – speed in question, rpm

n_0 – rated speed, rpm

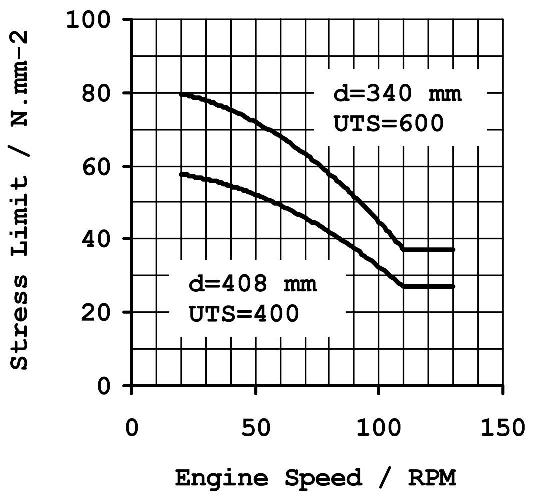

Figure 2. Permissible stress limits for the intermediate shaft

For transient running the permissible stresses due to the alternating torsional vibrations are not, in any case, to exceed the values given by the following formula:

$$\tau_2 = \pm 1{,}7 \cdot \tau_1 / \sqrt{c_k} \qquad \text{for} \quad \lambda \le 0{,}8 \qquad (6)$$

where:

τ_2 – permissible stress due to torsional vibrations for transient running

Figure 2. shows the influence of the selected shaft diameter on the permissible stress due to torsional vibrations for the intermediate shaft. The same relation for the propeller shaft is given in Figure 3. Other assumed parameters are the same as in Figure 1.

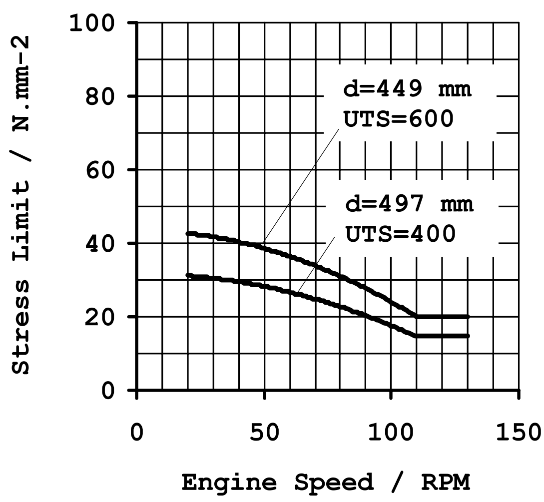

Figure 3. Permissible stress limits for the propeller shaft

Flexible shafting system concept

Previous discussion clearly shows that:

- high-stress steels allow shafting diameter reduction
- shafting diameter reduction enables greater torsional vibration stress limits

From torsional vibration viewpoint [6, 7], shafting diameter reduction means:

- shafting system torsional natural frequency is reduced which place the main torsional resonance in the region of enlarged torsional stress limits
- propeller damping rise which results in reduced alternating torsional vibration torque
- shafting cross section moments of inertia reduce which result in increased torsional stresses

Advantages of the proposed approach may be summarized in:

- lighter and less expensive shaftline
- smaller and less expensive shafting bearings
- smaller and less expensive propeller hub
- lower torsional vibration induced propeller thrust
- smoother shaftline alignment
- hull's afterbody design is less influenced by the shafting diameters
- shaftline length, as well as the engine room, may be reduced

Disadvantages of the proposed approach are:

- shafting is subjected to high torsional stresses
- high quality manufacturing is needed
- very precise torsional vibration analysis is needed

To alleviate these unfavorable side effects, two measures are essential [3]:

- 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 forgings heat treatment have 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.

Example 1

The flexible shafting system approach is applied to a shaftline design of a six-cylinder low-speed propulsion plant. The sample vessel was a 45.000 dwt tanker for oil and chemicals.

Machinery particulars are given in Table I. and the shafting initial design is summarized in Table II. Torsional vibration analysis of the proposed plant has shown the unaccepted stress level, Figure 4.

Table I. Machinery data

Engine	MAN B&W 6S50MC
MCR output	8240 kW
MCR speed	122 rpm
Number of cylinders	6
Cylinder diameter	500 mm
Stroke	1910 mm
Firing order	1-5-3-4-2-6
Mean ind. pressure	19 bar
Maximum pressure	140 bar
Flywheel inertia	1600 $kg.m^2$

Table II. Shaftline initial design

Interm. shaft material	440	N/mm^2
Interm. shaft diameter	440	mm
Propeller shaft material	440	N/mm^2
Propeller shaft diameter	535	mm
Flange filler radius	50	mm

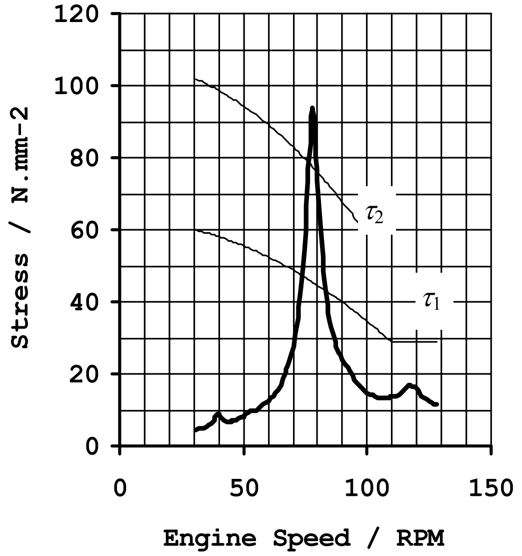

Figure 4. Torsional stress in intermediate shaft for initial design

After few design iterations, applying the flexible shafting system concept, the final shafting design has been defined, Table III.

Table III. Shaftline final design

Interm. shaft material	600	N/mm^2
Interm. shaft diameter	420	mm
Propeller shaft material	600	N/mm^2
Propeller shaft diameter	465	mm
Flange filler radius	26/113	mm

As shown in Figure 5., the final shafting design conforms to the permissible stress level set by the IACS Requirements [5]. Therefore, the Classification Society has approved the design.

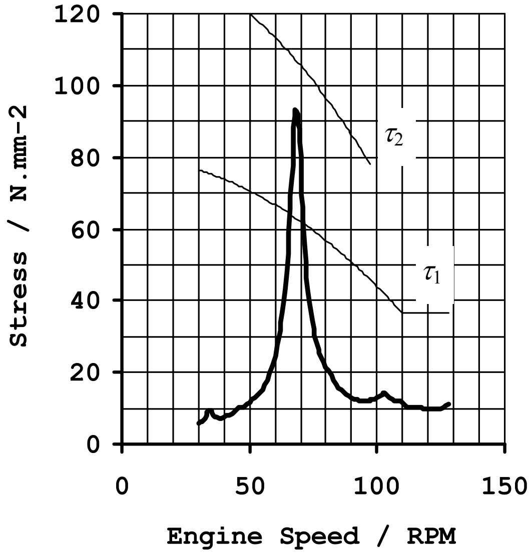

Figure 5. Torsional stress in intermediate shaft for final design

Example 2

The sample vessel was a 45.000 dwt bulk carrier, propelled by the five-cylinder low speed diesel engine [5], Table IV. Initial shafting design is summarized in Table V.

Table IV. 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
Firing order	1-4-3-2-5
Mean ind. pressure	19 bar
Maximum pressure	140 bar
Flywheel inertia	1625 kg.m^2

Table V. Shaftline initial design

Interm. shaft material	440	N/mm^2
Interm. shaft diameter	380	mm
Propeller shaft material	440	N/mm^2
Propeller shaft diameter	460	mm
Flange filler radius	40	mm

Table VI. Shaftline final design

Interm. shaft material	800	N/mm^2
Interm. shaft diameter	350	mm
Propeller shaft material	600	N/mm^2
Propeller shaft diameter	423	mm
Flange filler radius	26/113	mm

Preliminary torsional vibration analysis of the proposed plant has shown that the main resonance vibratory stresses were excessively high and beyond all permissible limits, as shown in Figure 6.

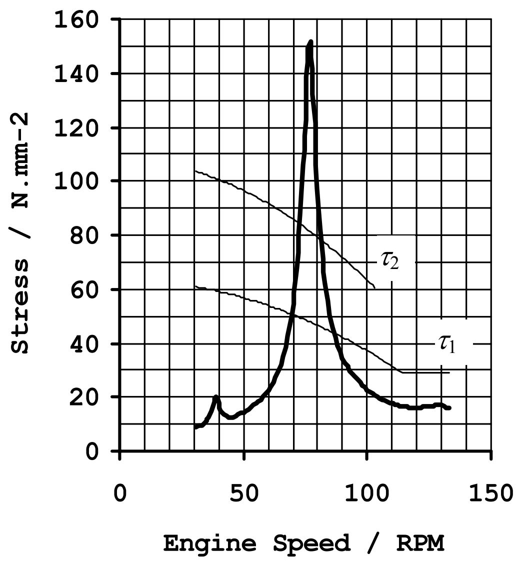

Figure 6. Torsional stress in intermediate shaft for initial design

Therefore, the alternative shafting designs and layouts were analyzed. After many design iterations, the final design of the shaftline has been defined, Table VI. Two countermeasures were needed:

- for the intermediate shaft a high-stress steel with 800 N/mm^2 UTS is adopted
- a tuning wheel of 7000 $kg.m^2$ inertia is mounted on the front end of the engine.

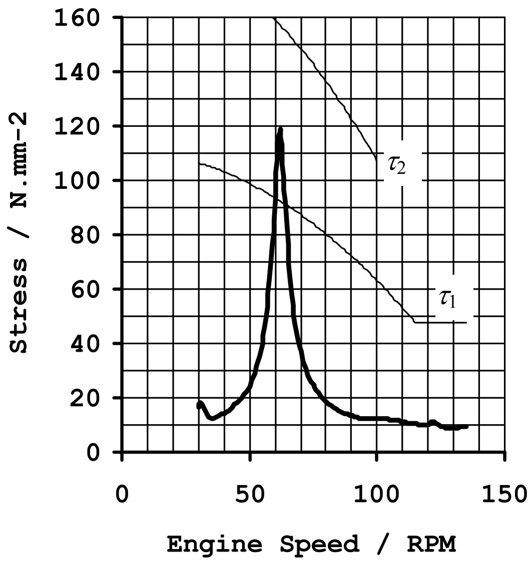

Figure 7. Torsional stress in intermediate shaft for final design

Vibration measurements, performed during the sea trials, proved the concept as well as the performed analyses.

Conclusion

In the light of the modern trends in shipbuilding practice, the utility of high strength steel for main propulsion shafting design needs special attention and consideration. By applying the high strength steel it is possible to satisfactory resolve otherwise hard to solve design problems. Moreover, using the high strength steel it is possible to achieve a reliable and cost effective main propulsion shafting design.

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