A new approach for marine propulsion shafting design

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In this paper, a new design approach for propulsion shafting system is presented. The aim of this approach is to improve the dynamic response of the shafting system concerning torsional vibrations. The approach results in raising the permissible stress limits set by the Rules of the Classification Societies, and reduce the shafting response due to engine excitation without any barred speed range. A computer program to calculate the Vibration Response Of Torsional Stresses (VIBRTS) using the proposed method has been developed. A numerical example of a 2-stroke, 6 cylinder marine diesel engine is investigated and the results are compared with those obtained from the basic design approaches based on the flexible shafting system, and the rigid shafting system.

يستعرض هذا البحث طريقة جديدة لتصميم عمود الدفع تهدف إلى تحسين السلوك الديناميكى للنظام تجاه اهتزازات اللى. بتطبيق هذه الطريقة يمكن الحصول على تصميم تزيد فيه حدود الإجهادات المسموح بها من قبل هيئات الإشراف، وتقل استجابته لإثارة الماكينة بدون وجود حدود محظورة للتشغيل.ولقد تمت برمجة الطريقة المقترحة وتطبيقها على محرك ثنائي الاشواط ذى ست اسطوانات لدراسة سماتِ إهتزازَ النظام ودراسة إجهادات اللي في مكونات عمود الدفع مع المقارنة بنتائج الطرق التقليدية المعروفة لتصميم عمود الدفع.

Keywords: Torsional vibration, Propulsion shafting, Flexible shafting system

1. Introduction

Propulsion shafting deserves special consideration during design, manufacturing and later, operation. 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]. The dynamic behavior of the propulsion shafting is the major factor influencing its overall reliability and efficiency. Three kinds of shafting vibrations are identified; namely, torsional, axial and lateral vibrations, each with specific sources, characteristics and consequences. In general, axial and/or lateral vibrations are less severe than torsional vibrations that are the most important factor influencing shafting design [2]. Torsional vibrations may lead to fatigue failure in the shafting system and can adversely affect the efficiency of operating machinery and auxiliaries. They are due to the periodical vibrations of the masses on the shaft in the plane of rotation, i.e., the periodical changes of the tangential forces acting on the crankshaft which result from torsional resistance of the shaft. Vibration characteristics have to be satisfactory with respect to vibration limits set by the requirements of the International Association of Classification Societies (IACS) [3] where it is also imperative to check the propulsion shafting of a main engine for torsional vibration. However the given formulae take into account static loads only and, thus, may not be sufficient for proper shafting design. Many studies had been carried out to improve the situation in order to obtain acceptable torsional vibration stresses of the system [4-7]. The most common possible approaches are the flexible shafting system [6], and the rigid shafting system [5]. These approaches are reviewed in this paper and a new approach which combines the benefits of the two preceding methods is then presented. The basic core of this approach is increasing the shafting diameter designed by the classification societies recommended equation by a proposed correction factor. This factor depends upon the main engine operating cycles, mean effective pressure, and the number of cylinders.

2. Analysis of torsional response

The equations of motion of torsional vibration systems may be written as

$$J\ddot{\theta} + C\dot{\theta} + K\theta = T(t), \quad (1)$$

where K is the stiffness matrix, J the inertia matrix, C the matrix of external and internal damping, θ the angular displacement vector and $T(t)$ the external excitation. The first step in determining the torsional response is to calculate the torsional natural frequency of the system conforming to the homogeneous form of eq. (1). This may be carried out by means of the Transfer Matrix method (Holzer) [1], that requires the stiffness and mass inertia of the shaft components being analyzed (referred to as mass-elastic data) [10]. The system is then modeled as a multi-rotor system. An example of such a system is shown in fig. 1[2]. The model is essential to define the dynamic magnifier of the engine and propeller required to determine the torsional vibratory stresses within the system.

2.1. Engine torsional excitation

The torque excitation in marine diesel engines is due to two sources of variation, namely, the varying piston gas pressure and the varying inertia loads due to the cylinder reciprocating masses. A tangential–effort diagram can be developed from a cylinder gas pressure indicator card where the cylinder pressure is related to the piston crankpin angle [1]. This curve is then modified by harmonic components resulting from the effect of inertia of the moving parts.

A tangential effort caused by the cylinder gas pressure, T_{Eg} can be represented by a Fourier series consisting of a constant term and a series of harmonically varying terms. The constant term is the mean tangential effort, T_m and does not excite torsional vibration. The harmonically varying terms are the principal sources of torsional vibration and do not contribute to the useful work output of the engine. A Fourier series representing the gas–pressure tangential

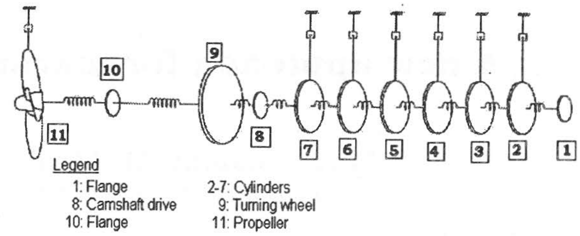

Fig.1. Torsional model of a propulsion shafting system.

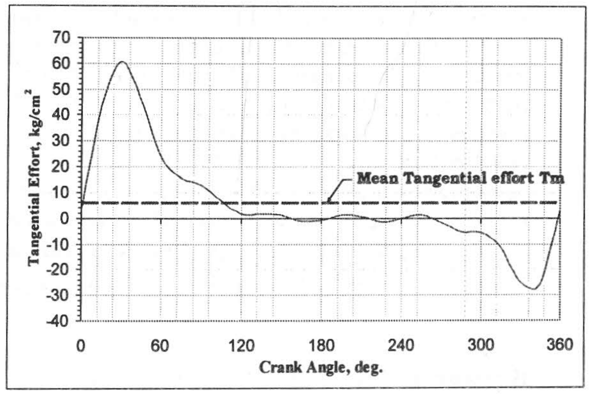

Fig. 2. Tangential-effort diagram with harmonic components [1].

efforts diagram illustrated by fig. 2 can be written as [1, 2]:

$$\begin{aligned} T_{Eg} &= T_m + \sum (A_n \sin n\varphi + B_n \cos n\varphi) \\ &= T_m + \sum T_n \sin(n\varphi + \beta_n). \end{aligned} \quad (2)$$

where

$$T_m = \frac{p_m}{\pi i}, \quad \tan\beta_n = \frac{B_n}{A_n}, \quad T_n = \sqrt{A_n^2 + B_n^2}.$$

In general, in elementary calculations of shaft torsional vibrations, the inertia force harmonics are completely ignored [8].

2.2. Propeller torsional excitation

The pattern of fluid flow through the propeller can be reduced to a series of harmonic components. The frequency of excitation applied to the propeller and the shaftline can be calculated as follows [9]:

$$f = n_P . N_P . \quad (3)$$

The propeller order of excitation in single-screw installation is not the same as in twin-screw installation. It depends on the existence of odd or even number of propeller blades Z_P.

- For twin-screw installation with odd or even number of blades $n_P = Z_P$
- For single-screw installation with an even number of blades $n_P = Z_P$
- For single-screw installation with an odd number of blades $n_P = 2\,Z_P$.

As a broad guideline, the number of propeller blades should be selected to keep the propeller excitation frequency and excitation order away from combination with engine excitation frequency and order [10].

3. Shaft design

According to the Classification Societies [3], the minimum shaft diameters are determined according to the shaft's intended service, transmitted power, shaft speed and applied material. The diameter of the intermediate or propeller shaft is not to be less than determined by the following formula:

$$d = FK\sqrt[3]{\frac{P}{MCR}\left(\frac{560}{\sigma_u + 160}\right)}. \tag{4}$$

The IACS requirement concerning the shaftline diameters is based on the transmitted power originated from mean torque, i.e. mean indicated effective pressure. The Rules diameter neglect alternating loads and vibration behavior of the shafting system. The series of harmonically varying orders of exciting torques that are superimposed with the mean torque are neglected by the Rules formula given by eq. (4). This situation may be overcome by adding a proposed correction factor "α" for the effect of the altering loads and exciting torque according to each propulsion engine characteristics, namely, the mean indicated pressure, operating cycles, and the number of engine cylinders. This factor can be estimated as

$$a = \sqrt[3]{1 + \frac{T_{ng}}{T_m}}. \tag{5}$$

where T_{ng} is the resultant value of the harmonic component of gas pressure tangential effort curve for n^{th} order of excitation for the marine internal combustion engine and can be calculated from [9]:

$$T_{ng} = \begin{cases} \left[\dfrac{140.62}{(50/\sqrt{n}) + n^3}\right] + \left[\dfrac{10.194\,p_m}{20 + n^3}\right] & \text{for 2-stroke engines} \\[2ex] \left[\dfrac{70.31}{(50/\sqrt{n}) + n^3}\right] + \left[\dfrac{5.097\,p_m}{18 + n^3}\right] & \text{for 4-stroke engines.} \end{cases} \tag{6}$$

It is considered that, the major orders of excitation are those which are integer multipliers of the number of cylinders; in the case of two-stroke cycle engines these are Z, 2Z, 3Z etc., and in the case of four-stroke cycle engines they are ½Z, Z, 1½Z, etc. Figs. 3 and 4 introduce the value of the correction factor (α) based on the mean indicated pressure p_m and number of engine cylinders Z for two and four-stroke diesel engines.

Thus, the minimum shaft diameter according to the Rules can be modified considering the effect of the exciting torque as:

$$d = \alpha FK\sqrt[3]{\frac{P}{MCR}\left(\frac{650}{\sigma_u + 160}\right)}, \tag{7}$$

where α is the proposed correction factor.

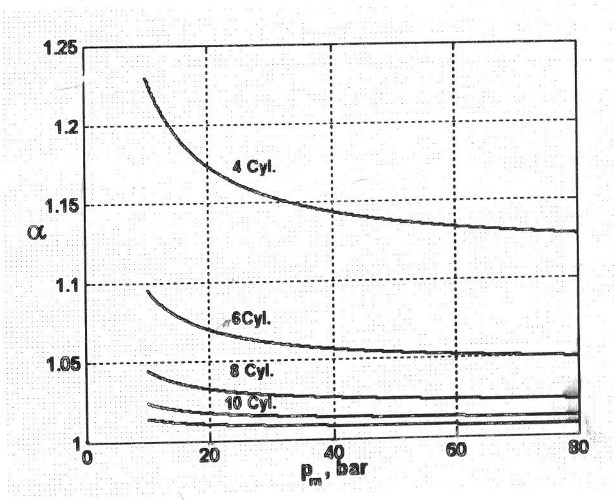

Fig. 3. Correction factor (α) of 2-stroke engines.

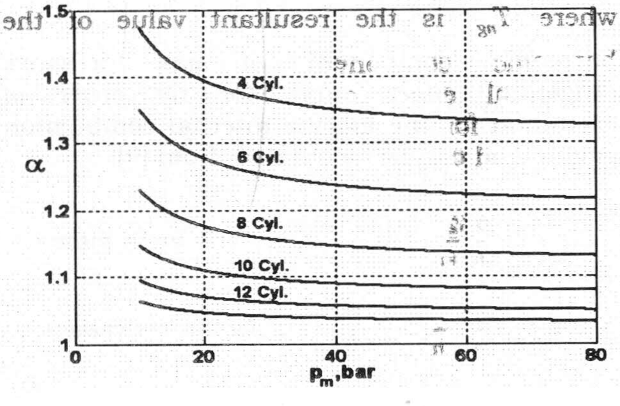

Fig. 4. Correction factor (α) of 4-stroke engines.

4. Stress consideration

Classification societies prescribe the amount of allowable torsional vibration stresses for engine crankshaft, intermediate shaft and propeller shaft. These stress limits are determined by the purpose, shape, material selected, dimensions and intended operation of shafting. Moreover, the stress limits are not constant; instead they are a function of engine speed.

According to the worldwide accepted requirements [3, 6] the torsional vibration stress limits are defined as follows:

"In no part of the propulsion system may the altering torsional vibration stresses exceed the value of τ_c for continuous operation and τ_t for transient running".

For continuous operation the permissible stresses due to altering torsional vibration are not to exceed the values given by the following formula

$$\tau_c = \begin{cases} \pm \dfrac{\sigma_u + 160}{18} c_k c_D \left(3 - 2\lambda^2\right) & \text{for} \quad \lambda < 0.9 \\ \pm \dfrac{\sigma_u + 160}{18} c_k c_D 1.38 & \text{for} \quad 0.9 \le \lambda \le 1.05. \end{cases} \tag{8}$$

$C_D = 0.35 + 0.93 d^{-0.2}$

For transient running, the permissible stresses due to altering torsional vibration are not in any case to exceed the value given by

$$\tau_t = \pm 1.7 \tau_c / \sqrt{c_k}, \qquad \text{for} \quad \lambda < 0.8. \tag{9}$$

Fig. 5 shows the influence of the selected shaft diameters and the ultimate tensile stress (UTS) on the permissible stress due to torsional vibrations of the shaft. It shows that at certain speed ratios, an increase in the UTS of the selected shafting material leads to an increase in the permissible stress for continuous operation at constant diameter. This design approach is called the flexible shafting system [6]. However, it introduces some unfavorable side effects if, for instance, the shafting diameters may be reduced below the Rules diameters resulting in high torsional stresses approaching the permissible limit[5]; moreover, the need for high quality manufacturing is essential. It is noticed that the increase in shafting diameter allows higher permissible stress for the same UTS of the shafting material. This is the rigid shafting design approach. It depends on increasing rationally the shaft diameter. This will change the dynamic behavior of the whole system which results in decreasing the torsional stresses; these will be less than the permissible stress limits for continuous running in the whole engine speed range [5]. The difficulty of this approach is the time consumed in the design iterations to obtain the suitable shaft diameter of the system.

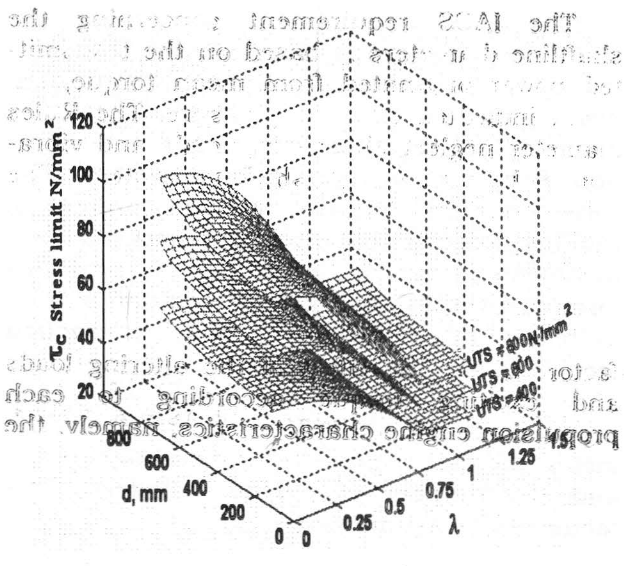

Fig. 5. Influence of shafting material on permissible stress limit.

5. The proposed design approach

Starting from the aforementioned approaches, good benefits can be obtained by combining the advantages of the flexible and rigid design approach. The shafting can be designed from high quality steel with a large ultimate tensile strength, considering the proposed correction for the altering loads in the Rules diameter equation.

The proposed approach design is illustrated by a shaft line design of a slow speed propulsion plant with the following particulars:

2-stroke, 6-cylinders marine diesel engine

Cylinder bore	600 mm
Length of stroke	2400 mm
Connecting rod length	2460 mm
Max. continuous power output	13530 kW
Max. Continuous speed	105 rpm
Mean indicated effective pressure/MCR	20 bars
Firing order	1-5-3-4-2-6
Oscillating mass/cylinder	5003 kg

The mass-elastic data required for the torsional model adopted fig. 1 has been computed and is given in table1 [10]. The original shafting design using the Rules as a guide is summarized in table 2.

Fig. 6 shows the torsional vibration analysis of the original design. The resonance vibratory stress occurs at the engine speed of 60 rpm. It is excessively high and beyond all permissible limits.

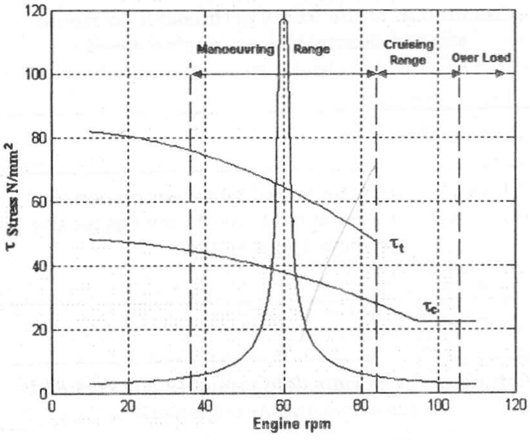

Fig. 6. Torsional Stress in intermediate shaft for initial design (table 2)

The flexible shafting approach using high tensile steel with UTS 600 N/mm² is illustrated in table 3. The corresponding torsional vibration analysis of this design is shown in fig. 7. It shows that the excessive torsional vibration stress has been decreased with decreasing of resonance speed to 50 rpm.

To apply the proposed design approach, the factor α is simply taken from fig. 3. The shafting design is summarized in table 4. The result of the torsional vibration analysis of this design is illustrated in fig. 8. This analysis has been carried out by the computer program VIBRTS "Vibration Response of Torsional Stresses" [10]. The flow chart of this program is given in fig. 9.

Table 1
Mass-elastic data of the model

No	Element	Mass inertia N.m.s²	Stiffness 10^6 N.m/rad	Diameter cm
1	Flange	212	1720	72
2	Cylinder 1	11160	1370	72
3	Cylinder 2	11160	1390	72
4	Cylinder 3	11160	1350	72
5	Cylinder 4	11160	1380	72
6	Cylinder 5	11160	1440	72
7	Cylinder 6	11160	1880	72
8	Camshaft drive	4802	2740	72
9	Turning wheel	4982	80	51
10	Flange	612	162	61.6
11	Propeller	64800		

Table 2
The Rules shaft line design

Intermediate shaft material	UTS = 400 MPa
Intermediate shaft diameter	510 mm
Propeller shaft material	UTS = 400 MPa
Propeller shaft diameter	616 mm

Table 3
Design by flexible sytem approach

Intermediate shaft material	UTS = 600 MPa
Intermediate shaft diameter	460 mm
Propeller shaft material	UTS = 600 MPa

Table 4
Proposed design

Intermediate shaft material	UTS = 600 MPa
Intermediate shaft diameter	500 mm
Propeller shaft material	UTS = 600 MPa
Propeller shaft diameter	560 mm
Correction " α"	1.07

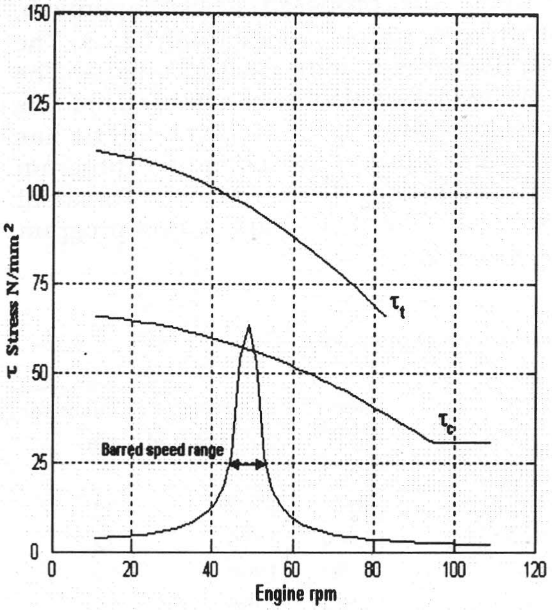

Fig. 7. Torsional Stress in intermediate shaft flexible shafting approach (table 3).

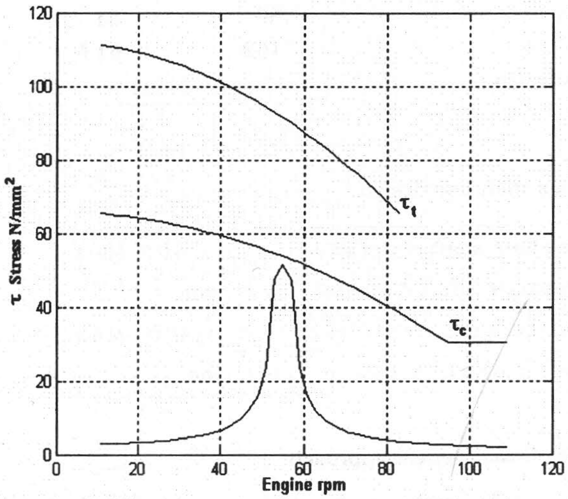

Fig. 8 Torsional Stress in intermediate shaft for proposed approach (table 4).

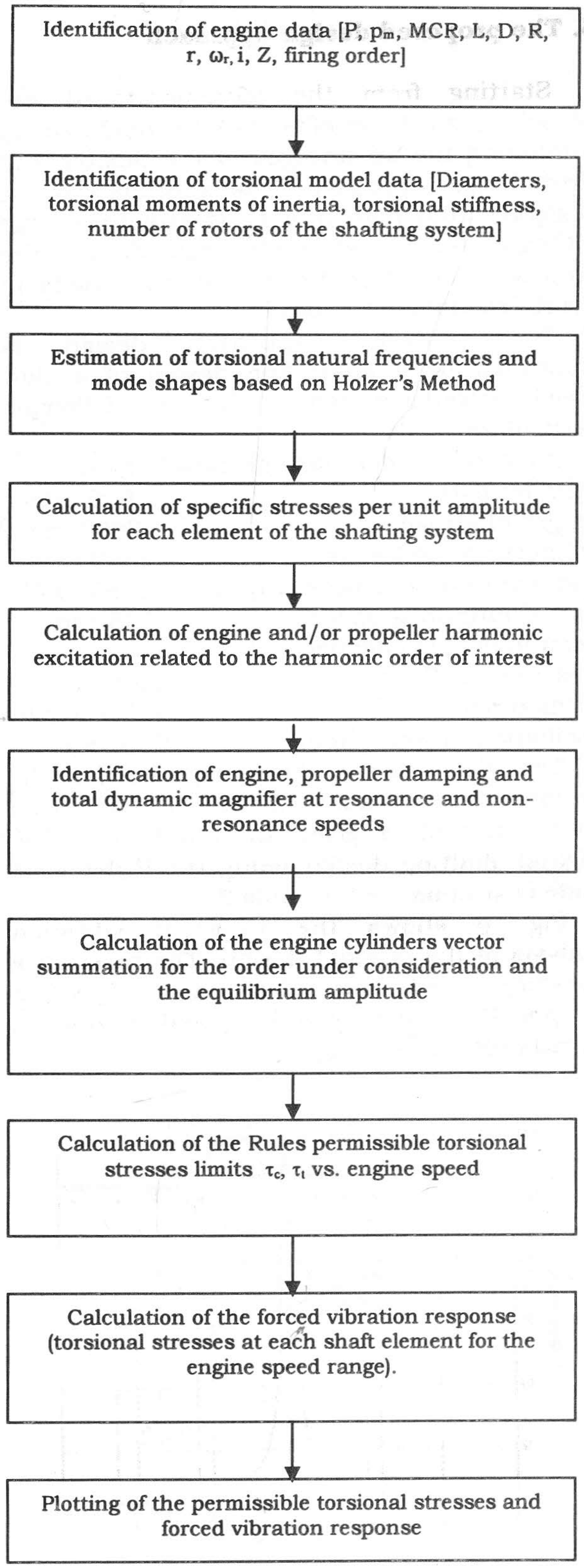

Fig. 9. Flow chart of VIBRTS program.

The output fig. 8 shows that the proposed approach gives a shafting design that conforms to the permissible stress level set by the IACS requirement [3] without any barred speed range. The permissible stress limits raise, the resonance speed moves far from the nominal engine speed consequently, the shafting response due to engine excitation falls, and the torsional vibration stresses reduce to lower level due to better dynamic behavior.

6. Conclusions

- The problem of excessive torsional vibration stresses had not been completely solved with the flexible shafting approach that results in a barred speed range for engine operation, and the rigid approach that does not specify an upper limit to the shafting diameter.
- A shafting design approach that overcomes these defects is proposed. This has been achieved by the use of high tensile steel and a proposed correction factor to increase the Rules shafting diameter.
- The correction factor accounts for the inertia force harmonics that have been ignored in elementary calculations of shaft torsional vibrations. The factor depends directly on the characteristics of the main engine, such as the mean indicated effective pressure, the number of engine cylinders, and operating cycle.
- The proposed approach raises the permissible torsional stress limits and reduces the shafting response due to engine excitation without any barred speed range.
- The application of this approach to a propulsion shafting of a 2-stroke, 6-cylinder marine diesel engine has resulted in a reduction of about 8% in the torsional stresses compared with the flexible approach.

Nomenclature

D	is the cylinder bore, mm
F	is the factor describing shaft service,
K	is the factor describing shaft design,
L	is the stroke length, mm
MCR	is the Maximum continuous rating, rpm
N_p	is the propeller rpm,
R	is the crank radius, mm
P	is the transmitted power at MCR, kW,
T_{Eg}	is the tangential effort, bar
T_m	is the constant mean tangential effort, bar
T_{ng}	is the nth order resultant harmonic component due to gas pressure, bar
Z	is the number of engine cylinders,
c_k	is the factor for different shaft design,
c_D	is the size factor,
d	is the shaft diameter, mm
i	is the 1 for 2-stroke, 2 for 4-stroke engines,
r	is the connecting rod/crank radius ratio,
w_r	is the reciprocating mass per cylinder, kg
n	is the order n° of engine excitation,
n_p	is the order n° of propeller excitation,
p_m	is the indicated mean effective pressure, bar
α	is the proposed correction factor,
β_n	is the phase angle,
φ	is the crank angle,
λ	is the speed ratio,
σ_u	is the ultimate tensile strength of shaft material, MPa
τ_c	is the permissible stress due to torsional vibration for continuous operation, MPa and
τ_t	is the permissible stress due to torsional vibration for transient running. MPa

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