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PREDICTION OF THE ONSET OF FATIGUE FAILURE OF SHIP PROPELLER SHAFTS

Statement of the problem in general form

Most ship structural components operate under cyclic loading, which creates favorable conditions for the development of fatigue cracks. Restoration, repair, and replacement of these components, particularly the underwater section, are performed only after the vessel is decommissioned.

With the increase in cargo transportation volumes, ship sizes, particularly tanker fleets, significantly increase. This leads to increased power of ship propulsion systems and, consequently, increased loads on ship propeller shafts. It should be noted that failures of ship propeller shafts cause damages that far exceed the cost of the shaft. They can entail vessel downtime costs and loss of operating profits; vessel repair (dock) costs; replacement of the shafts themselves; and, in some cases, lost propellers.

Furthermore, failures of this kind can affect the ship's controllability and, under certain conditions, lead to the loss of the vessel. Therefore, ensuring the strength of propeller shafts is essential for reducing accidents at sea.

As practice shows, contact of the propeller shaft with seawater is unavoidable. To protect against corrosion and reduce the coefficient of friction, propeller shaft liners (jackets) are used. These are metal cylinders tightly fitted to the shaft journals.

When exposed to seawater, the propeller shaft surface is subject to severe corrosion. Identified causes of propeller shaft damage indicate that the majority of damage (over 60% of cases) are due to fatigue and corrosion fatigue of the propeller shafts.

Statement of the problem

The objective of this study is to predict the initial stage of fatigue failure of a propeller shaft, which allows us to determine the moment when a fatigue crack enters the macrocrack development stage.

1. Analysis of the latest researches and publications

The problem of damaged and broken propeller shafts has been around for a long time; the first statistical data on this issue were published by foreign classification societies back in the 1930s.

Figure 1 shows a comparative analysis of the economic costs to a shipowner for a scheduled propeller shaft replacement and the damage caused by its failure, using the example of a propeller shaft of a 3,500-ton deadweight tanker built in Vietnam with a diameter of 290 mm and a length of 7,850 mm [1]. This tanker is transporting crude oil from the Bach Ho (White Tiger) offshore field to the Dung Quat refinery in central Vietnam (a distance of approximately 500 km). The shipowner's earnings in this case are approximately \$242,000.

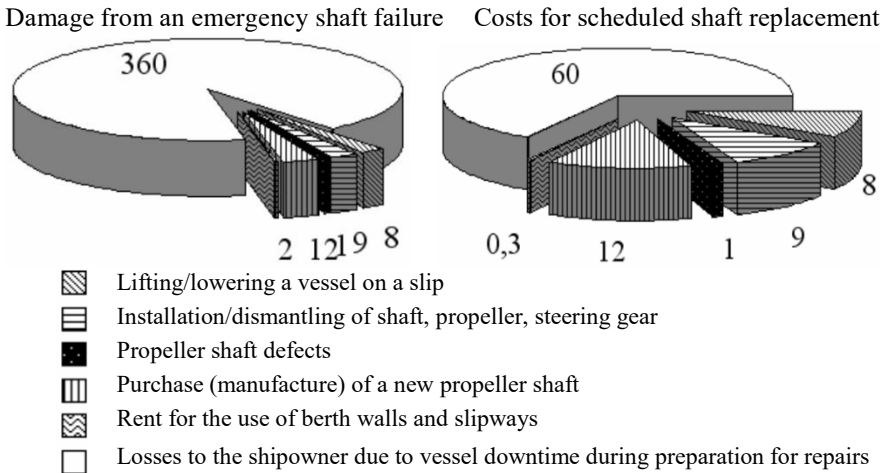


Fig. 1. Analysis of economic costs for emergency failure and scheduled replacement of a propeller shaft (in thousands of US dollars)

Figure 1 shows that the main difference, in terms of economic costs, is the amount of damage to the shipowner due to vessel downtime. In the case of a shaft failure, this figure is six times higher than the damage due to downtime due to a scheduled propeller shaft replacement. Summarizing statistical data on propeller shaft damage is a rather complex task, as the data from various sources relate to different time periods and vessel types.

Furthermore, most of this data is incomplete, often failing to indicate the causes and locations of cracks.

Contribution of the main material of the investigation.

The main places of origin of cracks in propeller shafts

An analysis of literature sources [2-13] showed that cracks in propeller shafts can be categorized by location into the following main types (Fig. 2):

1. directly in the shaft body;
2. in or near the keyway extending into the shaft body, including near the keyway mounting holes;
3. at the shaft transition from a cone to a cylinder at the aft end;
4. directly in the liners;
5. at the edges of the liners (at the bow and aft ends of the liners);
6. in the welds and weld-affected zones of the liners.

The formation of cracks in the area of the keyway (Fig. 2 a, b) and the keyway mounting holes is quite typical for propeller shafts and is caused by the keyway and holes acting as stress concentrators within the propeller shaft. The presence of cracks in the large diameter of the propeller shaft seating cone is caused by the greatest loads in this section of the shaft.

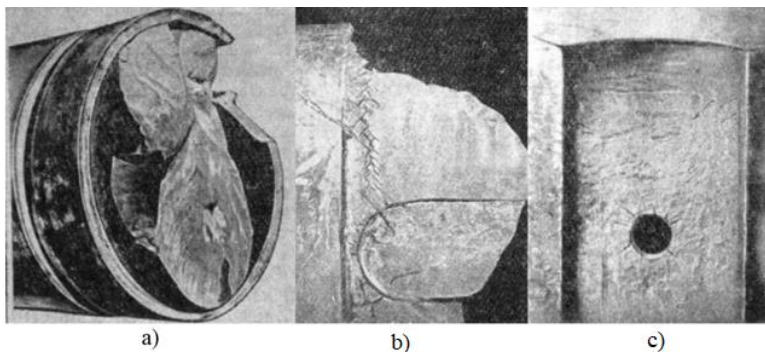


Fig. 2. Cracks and damage in the area of the keyway.

The large diameter of the propeller shaft cone essentially serves as the sealing point for the cantilever portion of the propeller shaft within the ship's hull, via the sterntube support. The primary cause of cracks at the edges of the liners is electrochemical corrosion due to the formation of a galvanic couple (in the case of liners made of non-ferrous metals). This results in large corrosion pits (10-12 mm deep [2]) forming around the shaft circumference near the ends of the liners, reducing the fatigue limit

due to stress concentration. Cracks in the welds and weld-affected zones of the liners are a consequence of changes in the mechanical properties of the material due to thermal effects during the welding process. Cracks directly in the shaft body or liner may be the result of poor machining or the presence of metallurgical defects in the material. Studies have shown [1] that, when cracks were found directly in the shafts, 67.5% of shafts were replaced and 32.5% were repaired; due to cracks in the linings, 34.5% of shafts were replaced and 65.5% were repaired.

Analysis of these data (Table 1) allowed us to plot graphs of changes in the specific crack detection rates (in shafts and linings) and propeller shaft replacements for different diameters (Fig. 3).

Table 1. Shaft damage statistics for different shaft diameters.

Shaft diameter, mm	Number of vessels	Number of dockings	Cracks in the shaft body	Cracks in the cladding	Shaft replacements
to 400	449	3113	63	79	39
400 - 500	88	730	19	13	20
500 - 600	75	590	15	20	10
over 600	44	262	3	6	1

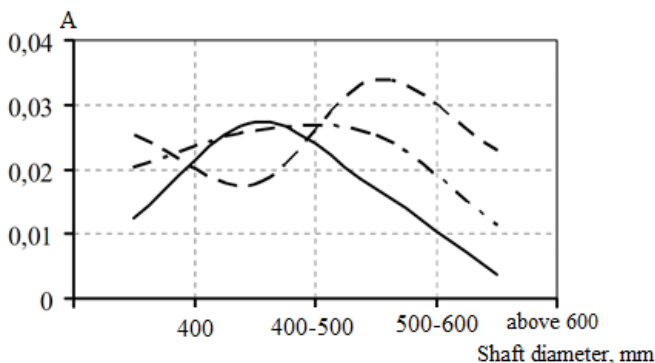


Fig. 3 Specific value of crack detection in shafts and linings and shaft replacements per one docking

The diagram (Fig. 3) shows that the majority of damage occurs to shafts with diameters of 400–600 mm, which is consistent with statistics from Veritas France [14]. However, as shaft diameter increases, damage rates decrease. Furthermore, it should be noted [15, 16] that in recent

years, instances of propeller shaft failures on ships have increased sharply. During this period, there were 73 cases of propeller shaft damage, accounting for 15% of the total number of damages to ships and their components across all supervised vessels.

This is due to the high degree of wear on mixed (river-sea) navigation vessels, excess operating hours of propeller shafts, violations of repair procedures and poor quality inspection, fatigue and corrosion-fatigue processes, and changes in the navigation conditions of the vessels. Thus, it can be concluded that the problem of damage and breakage of propeller shafts, according to data [18], remains relevant today, especially for mixed-navigation vessels, which may be due to the design features, operating conditions, and high degree of wear of the tanker fleet (river-sea type vessels). The main sites of crack initiation in propeller shafts [17, 19] are the shaft seat cone and the keyway area. Fatigue and corrosion fatigue are the main causes of damage to propeller shafts.

2. The fatigue failure process, existing energy criteria for fatigue failure

2.1 Stages of the fatigue failure process.

The entire process of fatigue failure development cannot be described by a single criteria equation. Most existing criteria are based on the relationships of linear fracture mechanics theory, which cannot be applied to the description of small crack growth kinetics [20 - 22].

Crack growth is a staged process, but the division of this process into stages varies. One such option is the following [17]:

- Stage one is the number of loading cycles to which a structural element is subjected until a small crack capable of propagating develops;
- Stage two is the number of loading cycles during which the small cracks develop until a macrocrack appears;
- Stage three is the number of loading cycles during which macrocracks develop until the structural element fails.

Thus, the entire process of fatigue failure of carbon steels can be divided into several stages:

$$N_p = N_j + N_{jj} + N_{jjj} + N_{jjj},$$

Where N_p is the overall life of the structural element; N_j is the number of cycles corresponding to the initiation and growth of microcracks within a single microstructural grain; N_{jj} is the number of cycles corresponding to crack propagation from a size corresponding to the grain diameter to the appearance of a macrocrack; N_{jjj} is the number of cycles corresponding to

the development of a macrocrack; N_{jy} is the number of cycles corresponding to the fracture process.

The first stage of fatigue failure is associated with the initiation and growth of a microcrack within a single microstructural grain of the material. In this case, we are talking about a microcrack of the order of one to two tens of micrometers, and for each specific material, this will be determined by the average grain diameter of the microstructure.

The second stage is the stage of small crack development. Experience shows [18, 19] that the second stage of fatigue failure accounts for approximately 40-60% of the total service life, and for some materials, this figure can reach 80-90% [20]. During this stage, abrupt changes in crack velocity are observed, which is due to the influence of microstructural barriers on crack growth. Crack sizes in the second stage reach approximately 0.5-1 mm, which corresponds to the possibility of detection using dye or magnetic flaw detection.

The third stage, which, as a rule, is also quite lengthy, is associated with the development of macrocracks, i.e., cracks whose growth is not affected by microstructural barriers, and is described by the equations of linear fracture mechanics. Cracks at this stage are generally visible to the naked eye and can have various sizes, depending on the dimensions and geometric parameters of the structural element being examined. The fourth stage is associated with the fatigue failure of the structural element; this stage can occur very quickly, almost instantaneously. The fourth stage may be absent for some materials or structural elements with certain geometric parameters and operating load conditions, and is typically characterized by the absence of a second vertical segment on the crack growth rate curve.

Thus, it can be noted that for a number of critical components and structural elements (such as propeller shafts) of seagoing vessels, the operation of which is prohibited if cracks are present, the most pressing issue is determining the duration of the first and second stages, rather than predicting the overall service life of the element. Therefore, to ensure the safe operation of the propulsion system of seagoing vessels, studying the initiation and growth of small cracks is particularly important. The presence of a macrocrack in a propeller shaft or liner, the growth of which is described by linear fracture mechanics, suggests the need for urgent replacement of the shaft.

2.2 Analysis of existing energy criteria for fatigue failure

Energy criteria are most suitable for the mathematical description of the initial stage of fatigue failure. Energy criteria for fatigue failure are based on the assumption that the ultimate limit state is determined by the critical value of irreversibly dissipated energy. In this case, the energy per loading cycle, the total number of cycles to failure, or other parameters derived from these are considered.

The parameter of specific irreversibly dissipated energy per cycle is calculated using the formula:

$$D = K_f \sigma_a \Delta \varepsilon_n,$$

Where K_f is the hysteresis loop shape factor; σ_a is the stress amplitude of the cycle; $\Delta \varepsilon_n$ is the inelastic strain of the material.

Figure 4 shows a series of dependences of the specific dissipated energy per cycle on the number of cycles to failure. According to these data, the number of cycles to failure increases with decreasing specific dissipated energy per cycle. It should be noted that, for the same number of cycles to failure, the absolute energy values change significantly, even for alloys with the same base material.

The parameter of total dissipated energy is also of interest in developing energy criteria for fatigue failure. Figure 4 shows the dependences of total dissipated energy on the number of cycles to failure in bending [21].

The data in Fig. 5 show that with an increase in the number of cycles till failure, the total dissipated energy increases significantly for most of the alloys studied.

Various relationships have been proposed to describe these relationships. Thus, in addition to the assumption that all dissipated energy goes toward fatigue failure:

$$\sum_1^{N_p} D = const,$$

In the works of Goltsev [22], Hanstock [15], and other authors, it is assumed that only the difference between the total energy and the energy dissipated at stresses equal to the fatigue limit is used for the development of fatigue damage.

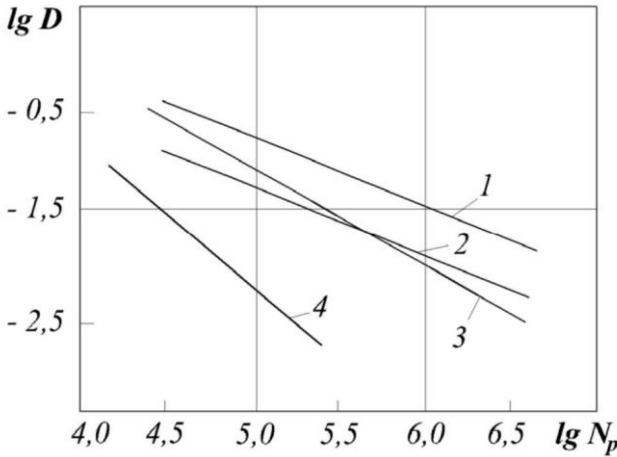


Fig. 4. D - N_p dependence in logarithmic coordinates: 1 - EI726 steel; 2 - 20Kh steel; 3 - 25 steel; 4 - 45 steel.

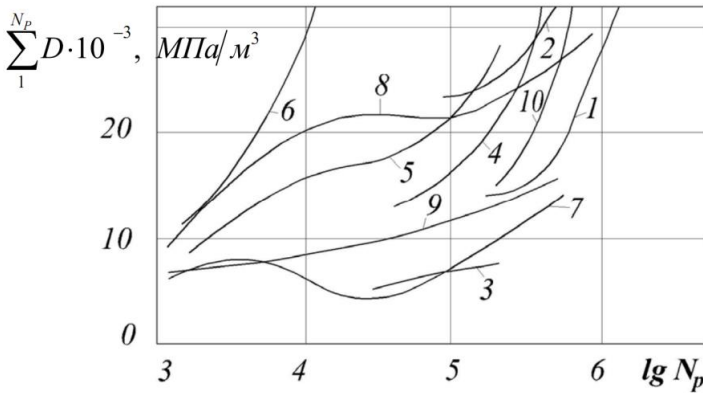


Fig. 5. Change in energy depending on the number of cycles before failure in bending: 1 - steel 20X; 2 - steel 25; 3 - steel 45(II); 4 - steel EI726; 5 - steel 12X13; 6 - steel 45; 7 - steel 40X; 8 - steel 45 (hardening and high tempering); 9 - steel 45 (hardening and low tempering); 10 - copper.

The remaining energy per cycle remains constant at stresses above the fatigue limit.

$$\sum_1^{N_p} (D - D_r) = const,$$

where D_r is the specific energy dissipated per cycle, when stresses equal to the fatigue limit.

Troshchenko [14] suggests that the portion of the energy that does not contribute to the development of fatigue damage is not constant, but increases with increasing dissipated energy per cycle:

$$\sum_1^{N_p} \left[D - D_r \left(\frac{D}{D_r} \right)^\alpha \right] = const,$$

Where α is a parameter characterizing the rate of increase of that portion of the energy that does not contribute to the development of fatigue damage.

The parameter α varies between 0 and 1. At $\alpha = 0$, Equation 5 becomes Equation 4; at $\alpha = 1$, all dissipated energy does not contribute to the development of fatigue damage in the material. It was also noted that the energy values determined according to Equation 5 are close to the energy equivalent to the latent heat of fusion of metals.

Feltner and Morrow [5] suggested that fatigue failure is associated with the parameter of critical irreversibly dissipated energy, which is equal to the ultimate work of deformation under static loading. It was assumed that the mechanical hysteresis loop has constant dimensions during cyclic loading.

$$D_c = 2N \int_0^{\Delta \varepsilon_i} \sigma d\varepsilon,$$

where D_c is the ultimate energy value.

Morrow subsequently proposed [6] that the total dissipated energy is inversely proportional to the fourth power of the ratio of the true tensile strength to the stress amplitude:

$$D_c = S_u \left(\frac{\sigma_p}{\sigma_a} \right)^4, \quad S_u = \frac{1}{1 + m_0} \sigma_p e_p,$$

where S_u is the fracture energy with a monotonic increase in load; m_0 is the strain hardening coefficient.

This equation is also valid for torsion. Morrow's controversial points include the dependence of dissipated energy on stress, which is essentially constant for all materials and stress ranges.

Stall, in his work [7], proposed describing the stress-strain diagram for both static and cyclic loading using the following formula:

$$e = \frac{\sigma_u}{E} + e_0 sh \frac{\sigma_u}{\sigma_0},$$

where e_0 and σ_0 are constants, and σ_u is the true voltage.

The hysteresis loop area adopted by Stoll for a pulsating cycle is written as follows:

$$D = \sigma_e - 2 \int_0^{\sigma_B} ed\sigma_u,$$

Considering the variable nature of the hysteresis loop, that in an axisymmetric cycle only the half-cycle of stretching is taken into account, we obtain:

$$D_{cp} = \frac{1}{2} \sigma_0 e_0 \left[\frac{\sigma_u^2}{\sigma_0^2} sh - \left(2ch \frac{\sigma_u}{\sigma_0} - 1 \right) \right],$$

Thus, the failure condition for a pulsating loading cycle will have the form:

$$N_p D_{cp} = S_u,$$

Chang in his work proposes the following condition for fatigue failure:

$$N_p (D - D_r) = D_c,$$

Martin, Muratov, and others consider the total hardening energy as a fatigue failure criterion. Thus, in Martin's work:

$$N_p E_T^{II} \Lambda \varepsilon_u = D_c,$$

$$E_{Th}^{II} = tg\beta, E = tg\alpha.$$

Muratov's energy dissipated during the cycle, associated with hardening:

$$D = (\sigma_a - \sigma_{III}^{II})^2 \frac{E - E_T^{II}}{EE_T^{II}}, D_c = DN_p,$$

In Ivanova's work, the energy expended in destruction is compared with the parameter of latent heat of fusion:

$$D_{cp} (N_p - N_i) = L_{pl},$$

where D_{cp} is the average energy value per loading cycle; N_i is the number of cycles at which crack initiation begins; N_p is the number of cycles until failure; L_{pl} is the latent heat of fusion of the metal.

In the works of Troshchenko [17, 18], the following energy criterion is presented:

$$D_c = S_u \left(\frac{\sigma_p}{\sigma_0} \right),$$

where b is a parameter characterizing the intensity of the increase in the total dissipated energy compared to the value S_u with a decrease in the stresses σ_a .

Conclusions

1 The problem of damage and breakage of propeller shafts remains relevant nowadays. Propeller shafts of river-sea vessels, with shaft diameters of approximately 200 mm, are particularly susceptible to damage.

2 The main causes of damage to ship propeller shafts are fatigue and corrosion fatigue, with the primary damage sites being the propeller seat cone and the keyway area, which is consistent with calculation data.

3 A calculation model for the initial stage of fatigue failure, developed based on the elastic deformation energy parameter of the material, allows us to determine the duration of the initial stage of fatigue failure of a propeller shaft and determine the need for scheduled replacement, as current regulations prohibit the operation of cracked propeller shafts on seagoing vessels.

4 The data obtained during the experiment allow us to conclude that the specific elastic deformation energy accumulated per unit length of the resulting crack is a constant value, independent of the level of applied stresses.

5 Timely replacement of the propeller shaft allows the shipowner to avoid costs associated with lost operating profits due to vessel downtime, which can occur in the event of a propeller shaft failure. These costs amount to approximately \$300,000.

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