A new method for strength evaluation of propulsion shafting coupling bolts based on the actual failure mode

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ABSTRACT

Damages to coupling bolts connecting propeller shaft, intermediate shaft and crankshaft of the main engine while infrequent also occur repeatedly. This paper identifies the exact causes of such damage through an in-depth examination of several bolt failure cases. It is found that contrary to their design criterion to avoid the shear failure at the face of the couplings, the bolts break off within the coupling flanges in bending. Examinations of the fracture surfaces reveal that the cracks initiate on the bolt surface of the reamer portion due to fretting fatigue that occurs in between bolt and flange hole surfaces. After initiation, the cracks are driven by torque-induced bending stress. The longitudinal location of maximum bending stress is determined by both experimental and theoretical analysis. Experimental and theoretical analysis results show that the maximum bending stress is mainly controlled by the bolt pretension force, the diametral interference between the reamer and the flange holes, and the contact length of the bolt reamer portion, in addition to the transmitted torque itself. Adequate bolt pretension forces reduce the bending stress in bolts by transmitting more torque through the friction between the flange coupling faces. The extent of shrinkage between reamer and hole is believed to change the supporting stiffness from flanges to bolts; the stiffer the supports, the smaller the bending stress. In addition, a shorter contact portion between reamer and hole shows a favorable effect on the bolt bending. As for the amplitude of the transmitted torque, the damage experienced cases are all estimated to be subjected to relatively large torsional vibratory torque. As the result of clearly identifying the affecting factors to the bolt failures, the Society’s rule requirement for determining coupling bolt diameter has been changed to take into account the effect of vibratory torque. In addition to the change in rule requirement, the authors recommend preventive measures addressing the effects of the reamer shrinkage extent and the length of the reamer contact portion as well as the bolt pretension.

1. INTRODUCTION

Shafting coupling bolts are important components for transmitting power from the prime mover to the propeller. However, while coupling bolts joining the propeller shaft, intermediate shaft, and crankshaft fail infrequently, they often fail repeatedly, which in most cases resulted in complete loss of propulsion power. The failures occurred at an average rate of approximately one incident per year among ClassNK classed vessels. Similar failures were also reported among other vessels classed with other Societies1-3. Fig. 1 shows a latest example of such failures aboard a tanker. In this case, all eight bolts connecting propeller shaft and intermediate shaft broke off at once while going astern. Due to the stern momentum of the propeller, the flange of the propeller shaft impacted the stern tube seal, damaging it.

Fig. 1 Example of coupling bolt failure

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Thus far, partially due to their relatively low occurrence, there were few in-depth analyses of the causes of such failures. In many cases, the loosening of bolts has often been determined as the cause. In the view of the authors, although a number of factors of design, manufacturing, installation or maintenance may have contributed to these failures, if the main cause can be identified and controlled through thorough failure analysis, the failure frequency can then be further reduced.

In this paper, the main actual failure mode of coupling bolts is examined. Then the root causes of such failure and several influential factors are determined through experiments and numerical analyses. Finally a short explanation about the latest amendment to ClassNK’s rule requirement based on this research results is given.

2. Main failure mode of coupling bolts
2.1 Presumed failure mode

Bolted flanges in ship propulsion shafting are mainly loaded by transmitting torque T as shown in Fig. 2(a). Therefore bolts for coupling these flanges are assumed to be subjected to pure shear produced by tangential force, S which can be simply calculated from transmitted torque, T. Accordingly, the presumed failure mode of coupling bolts is the shear failure at the face of the coupling

In addition, the current required diameter of coupling bolts in most major classification society’s rules, including ClassNK, is based on the unified requirement of the International Association of Classification Societies (IACS). The unified requirement concerning the diameter of coupling bolts is based on the presumed failure mode using nominal rated torque as external load

2.2 Actual failure mode

However, actual coupling bolt failures were differed completely from the predicted failure mode. The bolts did not fail due to excessive shear stresses at the coupling face but rather broke off within the flanges, obviously caused by repeated bending as shown in Fig. 3(a).

As can be seen on the fracture surfaces in Fig. 3 (b), the cracks propagated in the direction of tangential force before final separation. Judging from the direction of the crack propagation, it was determined that the coupling bolts fractured in fatigue caused by repeated bending due to fluctuating tangential force coming from vibratory torsional torque. While the zones on the fracture surfaces created by crack propagation in the same direction with the tangential forces in ahead condition accounted for a large percentage of the zones, relatively small cracking zones in the opposite direction were also confirmed. These smaller cracking zones in the opposite side may have been caused by the torsional vibratory torque for a
small number of cycles during astern condition. Furthermore, it can be said from the features of the fracture surfaces that recent failures reflect different external forces from those seen in similar failures in the 1970s. Most of coupling bolt failures in the 1970s were caused by large bending moment acting on the flange resulting from excessive wear down of stern tube bearings made of lignum vitæ.

3. Crack initiation

A close-up of one of the fracture surfaces is shown in Fig. 4. It is noteworthy that tongue-shaped protrusions can be seen at both crack initiation sites on the bolt head side fracture surface. These protrusions strongly indicate that fretting had occurred between the bolt and hole surfaces. Fretting is generally caused by small scale oscillatory motions between two tightly fitting parts. Relative small motions and contact pressure create friction forces causing surface shear. In fretting, crack initially develops at an angle to the surface of between 35° to 55° driven by the stress state dominated by the surface shear. The fatigue strength of a material can be reduced to 50 to 70% of its original fatigue strength when fretting occurs.

Fig. 4 Tongue-shaped protrusions observed on the fracture surface of bolt head side

In this case, the small scale oscillatory motions between bolt and hole surfaces arose from the axial displacement on the bolt surfaces, under repeated bending, while the contact pressure developed from diametral interference fitting the reamer bolts into flange holes as well as the supporting pressure from the flange to the bolt under bending, as illustrated in Fig. 5. After initiation near the contact surface, whether these cracks will further propagate depends on the bulk stress field. In this case, the cracks advanced in a direction almost perpendicular to the bolt axis driven by the bending stresses.

Other noteworthy facts are that most of the fractures occurred at around the middle of the reamer section but somewhat closer to the flange mating surfaces and that the fractures always occurred in the longer sections when lengths of reamer sections in both flanges were different. These facts are considered to be related to the longitudinal distribution of the bending moment and will be discussed below in details.

Fig. 5 Illustration showing mechanism of fretting fatigue of coupling bolts

As mentioned above, bending was the direct cause of the bolt fractures. Therefore, bending in bolts is analyzed below both experimentally and numerically.

4. Experiment

4.1 Test specimen and set-up

Details of the specimen bolts and test setup are shown in Fig. 6.

Fig. 6 (a) Test specimen bolt; (b) Test set up
The test specimen bolts are of the same size and steel grade as that of the failure example shown in Fig. 3. Several specimen reamer bolts were prepared to measure the bending stress distributions along the longitudinal length under different bolt pretension and bolt-hole diametral interferences conditions. The nominal diameter of the bolts at the reamer section where the bolt is interference fitted in the coupling bolt hole in the flange is 48 mm, and the other principal dimensions are as shown in Fig. 6. Details of the specimen bolts and test setup are shown in Fig. 7.

![Specimen bolt](image)

![Strain gauges details](image)

![Bolt subjected to shear forces](image)

Fig. 7 (a) Specimen bolt; (b) Strain gauges details; (c) Bolt subjected to shear forces

The forged steel has a tensile strength of between 590 and 710 MPa and a minimum yield stress of 295 MPa. Two shallow grooves opposite to each other were machined on the bolt surface in the axial direction to accommodate the strain gauges and cables. Bending stresses in six different cross sections were measured. Bolt pretension level in tensile strain was also monitored during the test by measuring it at two different sections. Three different diametral shrink interferences of 0.007 mm, 0.013 mm and -0.016 mm were used. The minus interference has no special technical meaning, expect to ensure that there was a clearance between the bolts and holes, i.e. it is the same as zero interference. The bolts were pre-tensioned to five different levels corresponding to axial strains of 0, 2 x 10^-4, 4 x 10^-4, 6 x 10^-4 and 8 x 10^-4 respectively. The maximum pretension of 8 x 10^-4 tensile strain corresponds to about 55% of the yield stress of the material. The experiment was conducted by applying shear force to the bolts up to 200 kN by a constant increment of 50 kN and then by unloading it by the same decrement.

4.2 Test results and discussion

The measured bending stress under 200 kN shear loading and three different diametral interference conditions are shown in Fig. 8. The results under other loading conditions show the same tendency.
Discussions about the experiment results are made as follows.

(1) Longitudinal distribution of bending stress measured during the experiment
The first thing noteworthy in Fig. 8 is that the maximum bending stress was in the flange with the longer reamer contact portion, and occurred at those locations somewhat away from the centre of the reamer portion towards the flange mating face, regardless of the diametral interference. These locations are very close to those locations where the bolts actually broke off. As why bolt reamer portion with longer contact section shows higher bending stress level will be elaborated later in this paper, it can be seen from the result that the maximum bending stress will reduce by about 40% for the 0.007 mm diametral interference, as the contact portion length changed from 58 mm to 44 mm.

(2) Effect of pre-tension level of bolts on bending stress
As can be seen in Fig. 8, bending stress can be reduced to some extent by increasing the pre-tension forces, as increased friction force between flange faces will carry more transmitted torque, but the effectiveness was not as significant as anticipated. It is noteworthy that the bending stress did not drop below that seen in non-pre-tension condition at all, until the pre-tension was increased to 8 x 10⁻⁴. The reason why a peak value appeared when the pre-tension level was 2 x 10⁻⁴ is not clear. However, one possible explanation for this is that pre-tension will change the diametral interference between bolt and hole. In this case, a 20% increase in bending stress was observed, indicating the danger of bolt loosening.

When the diametral interference from a minus value (diametral clearance) increased to 0.007 mm and 0.013 mm, the maximum bending stress decreased by about 40%. It is though that this result occurs because greater diametral interference increases the stiffness of flanges supporting the bolts and, in turn, reduces the bending stress as expressed later by analytical model.

As already briefly explained earlier, the measured bending stress shown in Fig. 8, indicated that the head side bolts with a reamer portion length of 58 mm exhibit about 140% to 200% greater maximum bending stress compared to that of the nut side bolts with a reamer portion length of 44 mm, depending on the diametral interference. Although the reason for this is still not well known, the quantitative relation between the reamer portion length and the maximum bending stress is examined later in this paper by numerical analysis.

Fig. 10 Effect of diametral interference on maximum bending stress

5. Theoretical Analysis
5.1 Analysis model
The bolt was modeled as a continuous beam with elastic supports representing the flange as shown in Fig. 11. The stiffness of these supports is obviously related to the diametral interference. The greater the diametral interference is, the stiffer the supports are. Using the 0.007 mm diametral interference as an example, the stiffness of each elastic support was taken as 1560 kN/mm by making the calculated maximum bending stress the almost same as the value obtained from experiment.

Fig. 11 Calculation model
5.2 Longitudinal distribution of bending stress obtained from calculation

The result calculated using the model is shown in Fig. 12. The result shows that the maximum bending stress occurs in a location a little away from the middle point of the reamer portion towards the flange faces. This coincides with above mentioned experimental results.

![Fig. 12 Calculated bending stress](image)

5.3 Effect of Diametral interference on bending stress

As can be seen from the experimental results shown in Fig. 10, bending stress decreases as diametral interference increases. This result can be explained by thinking that the greater diametral interference will increase the stiffness of flanges supporting the bolts and, in turn, reduce the bending stress. The effect of the stiffness on bending stress was numerically simulated and is shown in Fig. 13.

Although estimating the relationship between the diametral interference and the stiffness needs further examination, a tighter fit between the bolts and holes will certainly make the supports stiffer, and therefore it will be beneficial to reducing the bending stress as far as practical.

Since the relationship between the flange stiffness and the maximum bending stress was inferred from data shown in Fig. 13 as nearly linear, within a practical range, to the same as the relationship between diametral interference and the maximum bending stress shown in Fig. 10, it is reasonable to believe that there is also a linear relationship between the flange stiffness and diametral interference.

![Fig. 13 Effect of supporting stiffness on bending stress](image)

5.4 Effect of reamer portion length on bending stress

As indicated in the test result, the maximum bending stress will increase as the reamer portion length increases. Numerical analysis was performed to examine this phenomenon and the result is shown in Fig. 14. As can be seen from the result, when the reamer portion length is reduced from 50 mm to 20 mm, the maximum bending stress is almost halved.

![Fig. 14 Effect of reamer portion length on bending stress](image)

6. Margin of bolt size and Dynamic loads

6.1 Margin of bolt size

Fig. 15 shows the comparison of the margin of bolt size between the bolts connecting propeller shaft and intermediate shaft (Aft), and the bolt connecting intermediate shaft and the crank shaft (Fore). It is obvious that fore bolts usually have a higher size margin despite having almost the same load condition. This may be the reason behind the fact that the failure incidence of fore bolts is only about half of that of aft bolts.

![Fig. 15 Comparison of the diametral margins between fore and aft coupling bolts](image)
6.2 Dynamic loads

Ratio of amplitude of torsional torque at resonant point (Q) to the nominal rated torque (Q_m) for about 100 ships are shown Fig. 16. The horizontal axis represents the serial number of sample ships. It can be seen from this figure that failures tend to occur on ships that have greater vibratory torque.

Furthermore, if we divided the torque ratio into three ranges of 0.5-1.5, 1.5-2.5 and 2.5-3.5, the failure rate at each range can be calculated by dividing the number of failures at the range by the number of all ships at the range estimated from the distribution determined by the above data as shown in Fig. 17.

It can be seen from Fig. 17, when torque ratio exceeds 2.5, the failure rate goes up sharply.

\[ \alpha = 0.951 \frac{Q}{Q_m} \]  

From Eq. (1) and (2), it is obvious that when Q/Q_m is less than 1.166, the required diameter of bolt will remain unchanged and when Q/Q_m is greater than 1.166 the required diameter of bolt will increase, depending on the vibratory torque ratio.

As the result of this amendment, the diameter margins of almost all fractured bolts have dropped below 1.0, or in other words, they need to be increased. The average actual diameter will increase about 16%, supposing that the margins of bolt diameter remain unchanged.

8. CONCLUSIONS

Rule requirement for the diameter of coupling bolts has been changed to take into account the effect of ship-specific vibratory torque. This is a conceptual change in design. As a result of this change, the coupling bolts are to be designed against their actual failure mode, namely fatigue failure under repeated bending, rather than only considering the static nominal rated torque.

As the required diameter of the bolts will have to be increased for the majority of the cases where failure occurred, failures should be significantly reduced.

The safety of coupling bolts can be further improved, if the following factors are properly addressed in addition to the change in rule requirements.

(1) To make the length of reamer section as short as practical in order to reduce the bending moment acting on the bolts. A shorter reamer section length will also facilitate the pulling out of bolts when disconnecting flanges.

(2) To make the diametral interference as great as practical in order to reduce the bending moment acting on the bolts.

(3) To ensure that all the bolts are adequately pre-tightened in order to reduce the bending moment acting on the bolts.
REFERENCES

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