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Evaluation of the increased stiffness for the elastic coupling under the dynamic loading conditions in a ship

HyungSuk Han *, KyoungHyun Lee, SungHo Park

Naval System Research Team, Busan Center, Defense Agency for Technology and Quality, 525-2,Gwangan 1 dong, Busan, Korea

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ABSTRACT

Torsional vibration is restricted during design because it can cause fatigue fractures of the shaft due to repetitive vibratory torque and should be verified by analytical calculations as well as experiments. Torsional vibration is a characteristic of propulsion systems and is strongly related to their natural frequency; differences between measured and analytically predicted torsional vibration could be caused by variation of parameters related to natural frequency such as stiffness, damping, and components' inertial masses. In this investigation, extreme torsional vibration caused by increased coupling stiffness is expected to be the main root cause of fracture. The variation of coupling stiffness is investigated in a laboratory excitation test in order to determine the root cause of increased stiffness. Torsional vibration analysis error is identified as the main cause, where calculated values are quite different from measured results. It is shown that this difference is caused by torsional stiffness differences between the cases of the analytic model and the real ship.

1. Introduction

The torsional vibration of the propulsion shaft of a ship is very important because it can give rise to shear fatigue fracture. Therefore, torsional vibration analysis should be conducted during the design of a ship and verified by measurement in sea trail testing. In order to diminish the significant vibration transferred from the engine to the shaft system, elastic coupling is used and rubber material is widely applied for this elastic coupling. When a rubber coupling is applied, complex nonlinear characteristics should be considered because the stiffness and damping of the rubber coupling can vary according to the temperature and the amplitude and frequency of the vibratory torque.

Snowdon [1] described the nonlinear characteristics of rubber materials such as natural rubber, neoprene rubber, SBR rubber, and butyl rubber according to the frequency of the applied load, showing that the stiffness and damping of rubber materials increased with frequency. Dickens [2] has researched rubber vibration isolators for maritime machinery applications and investigated the stiffness and damping of rubber isolators using an electric shaker and a temperature conditioning unit, considering the effects of temperature and excitation frequency. Han et al. [3] have researched the variation of the dynamic stiffness and damping for the rubber mount installed in the shipboard equipment according to the frequency of excited load.

Related to rubber coupling, Lee et al. [4] performed torsional vibration analysis using the stiffness of the rubber coupling in considering non-linear properties of the rubber material, which they evaluated by excitation testing in the laboratory. In the coupling manufacturer's catalog [5–7], it is stated that the coupling stiffness can vary according to the amplitude of the vibratory torque and the temperature. Therefore, it is recommended that the coupling stiffness should be defined considering variation of the stiffness under various machine operating conditions.

* Corresponding author at: Defense Agency of Technology and Quality, 525-2, Gwangan 1-dong, Suyeong-gu, Busan 613-808, Korea.
E-mail address: daerihan@hanmail.net (H. Han).

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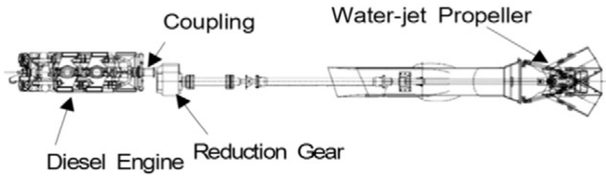

Fig. 1. Propulsion shaft system of the water jet for the naval vessel described in this research.

Sasaki and Kamata [8] conducted full-scale torsional vibration measurement for two ships with rubber couplings and suggested a practical calculation method for torsional vibration of the propulsion shafts considering dynamic factors of the rubber which varied according to its non-linear properties. Zhao et al. [9] have investigated torsional stiffness of the disc coupling used in compressor, gas turbines and aerospace application. He investigated the factors influencing the torsional stiffness of the disc coupling such as the amplitude of the transmitted torque and torque fluctuation.

When a propulsion shaft with a rubber coupling is designed, the stiffness and damping of the rubber coupling should be defined considering ship operating conditions. The variation range of the stiffness and damping is suggested by the coupling manufacturer; therefore, it is commonly adopted by ship builders. However, careful investigation should be conducted to define the stiffness and damping of the rubber coupling since the stated value is not always representative and cannot cover all applications of a ship.

In this research, a typical ship is described where torsional fatigue fracture of the shaft occurred because of incorrect definition of the coupling dynamic stiffness. Since the stiffness of the coupling in the analysis report was quite different from that in the real ship, a torsional vibration problem occurred and was followed, consequently, by shaft fracture. During the investigation of the root cause of this fracture, the incorrect definition of the coupling stiffness can be identified as the main root cause of this fracture. In this research, a comparison is made between the stiffness of the coupling evaluated from the full-scale torsional vibration measurement in a ship and that measured directly with an exciter in the laboratory, and the root cause of the fracture is verified.

2. Shaft structure

Fig. 1 shows a schematic diagram of the shaft described in this research, where the main propulsion system is a diesel engine and a water jet. In Fig. 1, the fracture occurred at the reduction gear shaft between the coupling and the reduction gear. Fig. 2 shows the fracture of the reduction gear input shaft, where the crack developed at 45°, starting from the end of the keyway. Considering the fracture pattern in Fig. 2, fractures can be expected to occur owing to the extreme torsional vibration caused by the high stress concentration factor at the end of the keyway. Therefore, the root cause of the fracture is investigated as related to the torsional vibration of the propulsion shaft in the next chapter.

3. Analysis

Fig. 3 shows the analytic model for the propulsion shaft in Fig. 1, and the position where the fracture occurs is the shaft between J14 and J15. Based on torsional vibration modeling (shown in Fig. 3), the equation of motion of the torsional vibration can be represented as shown in Eq. (1):

$$[J]\left[\ddot{\theta}\right] + [C]\left[\dot{\theta}\right] + [K][\theta] = [T]. \tag{1}$$

Here, [J] is the matrix of the mass moment of inertia, [C] is the damping matrix, [K] is the matrix of stiffness,$[\ddot{\theta}]$ is the angular acceleration, $[\dot{\theta}]$ is the angular velocity, $[\theta]$ is the angular displacement, and [T] is the exciting torque of the engine.

Fig. 2. Torsional fatigue failure of the shaft.