Design of Marine Diesel Propulsion System Including Highly-Elastic Rubber Couplings*

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In the marine propulsion system with geared diesel engine, highly-elastic elements such as rubber couplings are installed between main engine and reduction gear mainly to prevent the chattering of gear wheel.

The failures of rubber couplings are, in many cases, found in the shafting system with large engine, and sometimes occur in a short period of time. Most of the failures are caused by the torque fluctuations due to torsional vibration of shafting system. Continuous running of main engine at peak of torsional vibration particularly increases the possibility of failures because of the heat generation within rubber elements.

This paper shows the investigated results on service period of rubber couplings and the causes of failure. In terms of the torsional vibration calculations, the outline of calculating method using modal analysis technique is described. And, as the guidelines for shafting design, allowable vibratory torques, allowable power loss and fatigue curves for rubber couplings are described.

1. Introduction

Highly elastic rubber couplings have been used for ship's propulsion shafting system with geared diesel engine. Among those the couplings of large size were used with a high concentration during the period from 1976 to 1984 according to statistical data of NK-class ships[1]. This is because the large 4-stroke cycle engines with gear are preferably equipped for such ships as oil tankers, pure car carriers and bulk carriers in those days.

In the shafting system with geared diesel engine, the torque fluctuations generated by the engine and the chattering of gear wheel are serious problem. To solve this problem, rubber couplings are installed between main engine and reduction gear. Rubber's high elastisity makes it possible to adjust a major critical speed of torsional vibration far from the normal speed range, and rubber's high damping properties diminish the amplitude of torsional vibration.

By the way, unlike other power transmission components, the rubber couplings used for ship's propulsion system are not always capable of serving semi-permanently. High polymers such as rubber naturally deteriorate due to the effect of ozone in the environment, and the internal heat of rubber elements generated by torque fluctuations affects rubber's strength. Actually, several failures involving rubber's melting have been reported[1)2)].

Considering the above, first, this paper shows the investigated results on the situation of failures and maintenance of rubber couplings. From a statistical point of view, we describe the probability of failure and the durability limit of rubber couplings in those application to ship's propulsion system. The causes of failure are explained with those distinctive features.

For preventing the failures of rubber couplings, the correct evaluation of both torque fluctuations and energy loss is indispensable. In 2nd section, the outline of theoretical calculations authors carried out is described. This calculating method which was verified in detail based on the data of full-scale measurements makes it possible to perform a reliable evaluation of rubber's strength.

Lastly, in view of the strength of rubber couplings, we present the guidelines for shafting design with the causes of failure. Furthermore, as a new attempt to predict the service period of rubber couplings, the method based on fatigue curves is discussed.

2. Failures and maintenance of rubber couplings

2.1 Failures and maintenance

According to the former investigation on failures of rubber couplings[1], the failures are frequently found in the couplings of large size. Based on the results, our investigations of this time were made into the couplings equipped for engines of larger power output than 2,000 kW.

Fig. 1 shows the service periods of couplings

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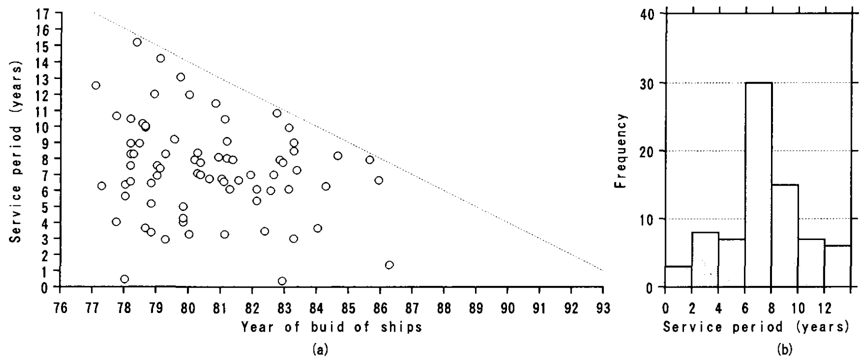

Fig. 1 Service period of couplings renewed due to failure

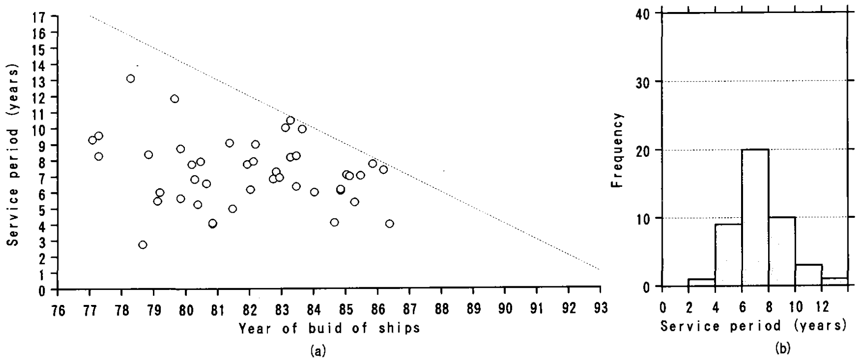

Fig. 2 Service period of couplings renewed as shipowner's maintenance

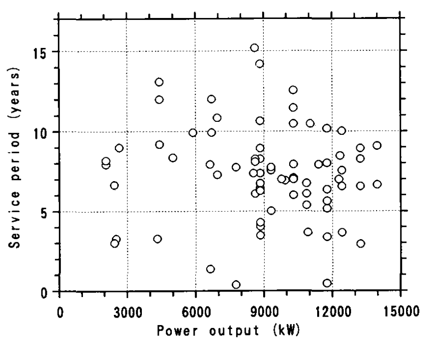

Fig. 3 Effect of power output of engine on failure of couplings

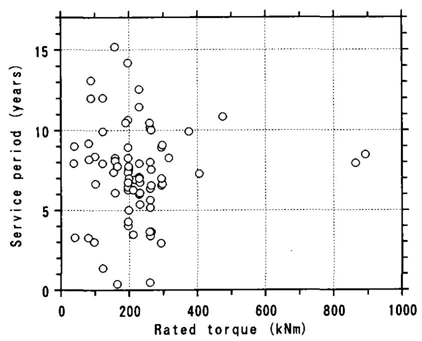

Fig. 4 Effect of rated torque of engine on failure of couplings

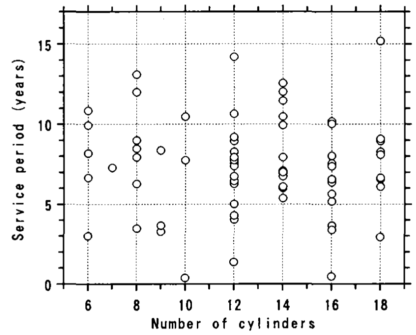

Fig. 5 Effect of number of cylinders on failure of couplings

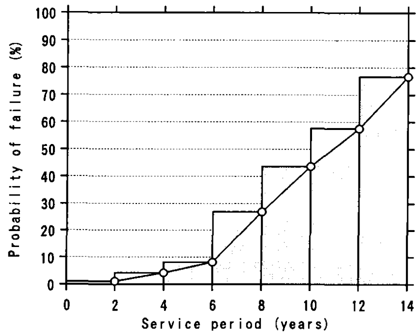

Fig. 6 Probability of failure in rubber couplings

renewed due to failure, and Fig. 2 shows those of couplings renewed as a shipowner's maintenance. In both figures, (a) is the scatter-plot of service periods, and (b) is the histogram of the same data. Year of build of ships in (a) is for seeing the aspect of the alteration of shafting design. In histograms, class interval is 2 years except the last class covering the range of 12 years and above. For example, when the renewals due to failure were carried out in 1985 and 1992 in the ship built in 1980, two data of (80,5), (80,7) are plotted in Fig. 1(a) and classified into each corresponding class of Fig. 1(b).

As shown in (a) of both figures, the dispersion of service periods of couplings is large, and these changes with the time of shafting design are not seen. No data after 1987 are associated with the decreasing of 4-stroke cycle engines installed in NK-class ships; therefore, this situation does not necessarily mean the decreasing of failures or maintenance work of couplings.

On the other hand, it is evident from two histograms that comparatively many couplings were renewed in the period of 6-8 years after those installation in either case of failures and maintenance. These histograms are not correct in view of statistics because of the difference of each ship's age, but the results agree with the information from many shipowners and are worthy of notice.

In addition to the above, we investigated the cause of the dispersion seen in service periods using different approaches. As a few examples, the effects of rated power output, rated torque and number of cylinders of main engine on failures of couplings are shown in Fig. 3, 4 and 5 respectively.

In Fig. 3 and Fig. 4, the concentration of data within the range of 8,500-12,000 *kW* and 190-270 *kNm* indicates that M.C.R. of many engines is located in these ranges and as a result many failures are also found there. As for the dispersion of data in the vertical direction of each figure, distinctive features are not seen in any figures. Investigated couplings are of the similar type in torque transmission. The results, therefore, suggest that the dispersion of service periods seen in failed couplings is not related to engine outputs or the structure of shafting system, but to other causes such as the operating conditions of engine, engine troubles and so on.

The probability of failure and durability limit in rubber couplings are described below. We investigated all couplings equipped for the ships built in 1977-92 among NK-class ships currently in operation. Data of each coupling were classified by the similar method to the foregoing histogram, and from the results the probability of failure was roughly estimated.

For example, when a rubber coupling has the next history:

- registered date of ship : '80,12.
- date of failure/renewal : '85,12.
- date of maintenance/renewal : '90,12.
- date of investigation : '93,12.,

these data were classified into 2 classes of service period and counted as follows:

failure counter:	renewal counter:
$FC[2,4] = 0$	$RC[2,4] = 1$
$FC[4,6] = 1$	$RC[4,6] = 2$

where $[j, j+2]$ denotes the period from j to $j+2$ years after the installation of couplings. The date of investigation was counted as that of maintenance/renewal. All history data of couplings were gathered in this manner and stored as an universal set of service periods of couplings.

If the rate of failure $p\,[j,\ j+2]$ is given by the following expression:

$$p\,[j,\ j+2] = \frac{FC[j, j+2]}{\sum_{i=j}^{N} RC[i, i+2]}$$

and the probability of failure is defined as a cumulus of $p[j,\ j+2]$, it will give a clue to the service life of rubber couplings used in ship's shafting system.

Fig. 6 shows the probability of failure in rubber couplings obtained by above procedure. This figure has the following features:

- The probability of failure within 6 years is comparatively low.
- The probability of failure increases sharply after the service for 6 years and reaches approximately 80% in 14 years.

That is, as many as 80% of rubber couplings will not endure up to 14 years in service in those application to ship's propulsion system.

2.2 Causes of failures

In reference (1), authors discussed the causes of failure in rubber couplings and classified them into four (4) typical causes except their structural or material defect. These are illustrated in Fig. 7 using the example of torsional vibration curves of 12-cylinder vee-type engine system and are summarized as follows.

(a) Impulsive large torque

Under next operating conditions large torque acts impulsively on the rubber coupling, and cracks are sometimes generated on the rubber surface.

- starting/emergency stop of engine
- crash astern
- on/off of clutch device

The cracks are generated when the quality of rubber is in deteriorated condition to some extent. In that sense, these operations are considered to be the direct cause of failure rather than the principal cause.

Among others crash astern operation generates the largest torque. The peak torque generated by crash astern operation is as large as astern steady torque despite quick passing through the critical speed of the engine[3]. This is actually the vibratory torque due to 1-node torsional vibration, but acts on rubber as an impulsive large torque.

Emergency stop of engine and crash astern operation feature the direction of cracks. That is, by the large reverse torque, cracks are generated in the direction of strain caused during ahead running of the engine[1].

(b) Torque fluctuation at the resonant peak of torsional vibration

When an engine operates properly under the steady condition, the maximal torque fluctuation is observed at the resonant peak of torsional vibration. The vibration damping of rubber is caused by the energy loss and this energy component is converted into heat in rubber elements. Therefore, the continuous running of engine at resonant speed increases the internal temperature of rubber elements and sometimes causes the failures involving rubber's melting.

The failure appears as the perfect breakdown of rubber elements. The fracture surface shows a pattern of serration with an inclination of 45 degrees in the circumferential direction, which is peculiar to torsional vibration phenomena.

(c) Reduced-cylinder operation

Engines are not always in a proper condition, but are sometimes forced to drive in reduced-cylinder operation when engine troubles such as misfiring occured. Reduced-cylinder operation increases the exciting torque of 0.5th and 1.0st order; consequently, the corresponding responses reach so high level that the engine can not be continuously used at the resonant point as shown in Fig. 7.

The reduced-cylinder operation, in most cases, causes a rapid decrease in the service period of rubber couplings. The shape of failure is, similarly to the case (b), characterized by the breakdown of rubber elements with heat generation. However, when torque fluctuations are excessively large in amplitudes, the breakdown occurs momentarily without the accumulation of heat.

(d) Fatigue

Fatigue fractures of rubber are generally said to take place when the bond of filler materials is destroyed by mechanical strain, heat generation and associated oxidation caused in the rubber.

In this paper, however, we use the term "fatigue" to explain the failure caused by the torque fluctuations at the flank of torsional vibration resonance curves. The shafting system for propulsion is usually so designed that N.O.R. (or M.C.R.) of engine will be located in the flank region of torsional vibration. Therefore, the failure which occured in the speed range near N.O.R. is classified as fatigue if other causes are not found. The failure occurs without excessive heat generation and appears as cracks on rubber surface or increased permanent strain of rubber body.

The allowable limit associated with fatigue is given as the "allowable vibratory torque" by the manufacturer of couplings and classification societies. According to our investigations, however, not a few failures of couplings occur in a shorter period than the foregoing service life even if the vibratory torque acting on rubber is lower than the allowable limit. This problem is taken up in the last section.

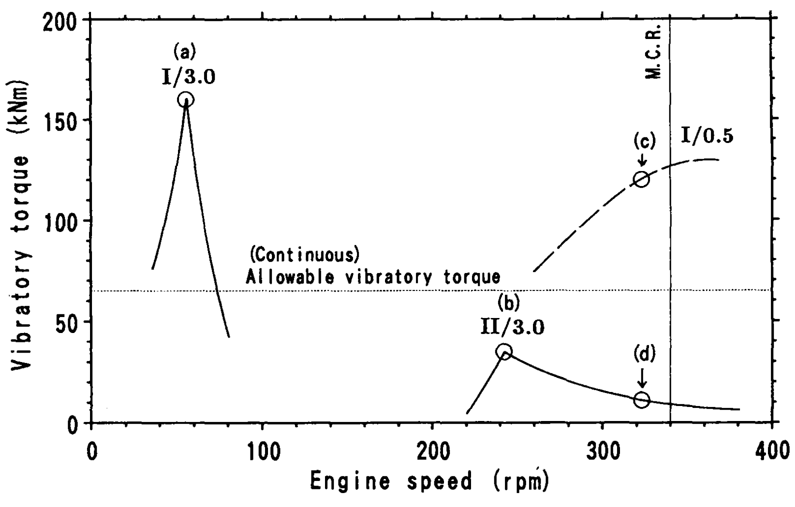

Fig. 7 Causes of failure in rubber couplings

3. Theoretical calculation for evaluating the strength of rubber couplings

Authors developed the torsional vibration calculating system for shafting system with rubber couplings using modal analysis technique[4]. Using this calculating system the torque fluctuations which act on rubber and the corresponding energy loss (power loss) are accurately obtained. The outline is described below.

3.1 Torque fluctuations

In torsional vibration system with the damping proportional to vibration speed, an equation of motion can be generally written as following matrix form:

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=\{f\} \tag{1}$$

where $[M]$, $[C]$ and $[K]$ are the matrices denoting inertia moment of mass, damping and torsional stiffness respectively, and $\{x\}$ and $\{f\}$ are the vectors denoting angular displacement and exciting torque respectively.

There are several methods to solve this equation; however, to the shafting system with highly-damping elements such as rubber, the assumption of a general viscous damping is to be applied. From the assumption, the amplitude and phase of vibratory torque are obtained with high accuracy.

In the case of free vibration ($\{f\}=\{0\}$), Eq. (1) is rewritten to the following last form as a standardized eigenvalue problem:

$$-[D]^{-1}[E]\{Y\}=\lambda\{Y\} \tag{2}$$

where

$$[D]=\begin{bmatrix}[C] & [M]\\ [M] & [0]\end{bmatrix},\ [E]=\begin{bmatrix}[K] & [0]\\ [0] & -[M]\end{bmatrix},$$

$$\{y\}=\begin{Bmatrix}x\\ \dot{x}\end{Bmatrix}=\{Y\}e^{\lambda t} \tag{3}$$

From Eq. (2) eigenvalues λ_r and eigenvectors $\{\Psi_r\}$ (both are complex numbers) can be obtained. λ_r consist of a conjugate pair $(\lambda_r, \bar{\lambda}_r)$ except for a rigid mode and over-damping mode. Real parts and imaginary parts of λ_r indicate the degree of damping and damped angular natural frequencies respectively.

Eigenvectors also have a conjugate pair similarly to eigenvalues. Let $[\Psi]$ be the matrix in which eigenvectors $\{\Psi_r\}$ are arranged in the direction of column, then $[\Psi]$ can be divided into four matrices:

$$[\Psi]=\begin{bmatrix}[\phi] & [\bar{\phi}]\\ [\phi][\Lambda'] & [\bar{\phi}][\bar{\Lambda}']\end{bmatrix} \tag{4}$$

where $[\Lambda']$, $[\bar{\Lambda}']$ are the matrices denoting the diagonal arrangement of eigenvalues λ_r, $\bar{\lambda}_r$ respectively, and $[\phi]$ is the eigenvector matrix of Eq. (1), that is the vibration modes of shafting.

Vibration responses are obtained using above eigenvalues and eigenvectors. By $[\Psi]$, matrices $[D]$, $[E]$ are diagonalized. We express the diagonalized matrices as $\lceil d \rfloor$, $\lceil e \rfloor$ and these elements as d_r, e_r respectively. If input and output are given by $\{f\}=\{F\}e^{\lambda t}$, $\{x\}=\{X\}e^{\lambda t}$ respectively, the steady vibration response is expressed as follows:

$$\{X\}=([\phi]\lceil u \rfloor[\phi]^T+[\bar{\phi}]\lceil \bar{u} \rfloor[\bar{\phi}]^T)\{F\} \tag{5}$$

where $\lceil u \rfloor$, $\lceil \bar{u} \rfloor$ are the diagonal matrices, and these have

$$1/(\lambda d_r+e_r),\ 1/(\lambda \bar{d}_r+\bar{e}_r)$$

respectively as the component.

Therefore, by substituting the engine exciting torques into $\{F\}$ of Eq. (5), the angular displacement responses of all mass position are calculated. And, the corresponding responses of torque can be easily obtained from the angular displacement responses.

The calculating accuracy was verified in detail using the results of full-scale measurements. Fig. 8 shows an example of calculating model. All of damping coefficients in this calculation are treated as viscous damping: the damping of torsional damper and rubber couplings which is proportional to relative speed of adjacent two masses and the damping of engine and propeller which is próportional to these vibration speed.

Fig. 9 shows the calculated and measured vibratory torques of 2-node 3rd order in the shafting system of Fig. 8. Calculated torques agree with those measured not only at resonant peaks but also at the flank of resonant curves.

Fig. 10 shows the example of response amplitudes of torque in reduce-cylinder operation, and the comparison is made at the position of output shaft of couplings. While measured results are limited to narrow speed range, good correspondence between calculated and measured results is observed in amplitudes of each order of engine speed.

3.2 Energy loss (Power Loss)

It is well known that if torque periodically acts on high polymers such as rubber, the hysteresis shown in Fig. 11 is created by the time lag between torque (T) and torsional angle (θ). The vibration damping effect of rubber is caused by the energy loss expressed by the inside area A_v of a closed loop, and this energy component is converted into heat within the rubber element. Generally, the damping effect is expressed as the relative damping (or damping rate) ψ denoting the ratio of A_v to the corresponding strain energy A_p:

$$\psi=\frac{A_v}{A_p} \tag{6}$$

The relative damping can be adjusted by the changing the vulcanizing condition employed in the manufacturing process for rubber or with filler materials. Rubber couplings used for the propulsion system of ships normally have the relative damping of

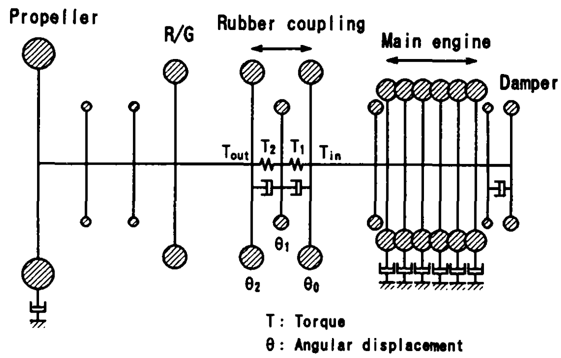

Fig. 8 Calculating model

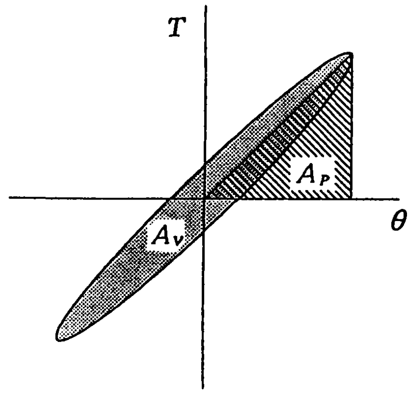

Fig. 11 Hysteresis curve of rubber

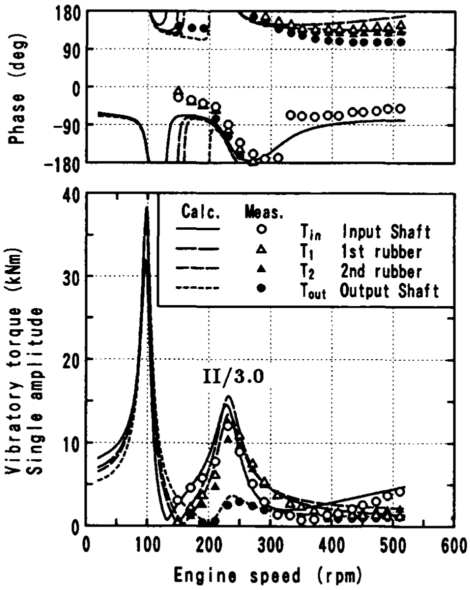

Fig. 9 Response of torque (2-node 3rd order)

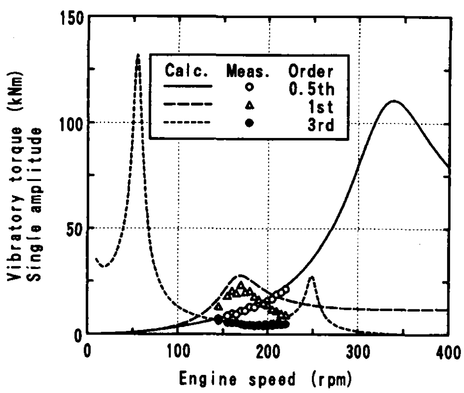

Fig. 10 Response of torque in reduced-cylinder operation

0.5 to 1.5. Within the service range of marine applications, the changes of relative damping are considered to be insignificant.

Using the relative damping, energy loss E_i in one vibration cycle can be expressed as follows:

$$E_i = \frac{\pi\psi}{\sqrt{4\pi^2+\psi^2}} \cdot \frac{T_i^2}{C_{dyn}} \tag{7}$$

where i, T_i are the order of engine speed and corresponding vibratory torque respectively, and C_{dyn} is dynamic spring constant. By further approximation, Eq. (7) becomes

$$E_i = \frac{\psi}{2} \cdot \frac{T_i{}^2}{C_{dyn}} \tag{8}$$

From Eq. (8), power loss P_v at engine speed n (rpm) can be given by

$$P_v = \sum_i E_i \cdot \frac{n}{60} \cdot i \tag{9}$$

The allowable power loss P_w is so established that the maximum temperature in rubber elements will be lower than the vulcanizing temperature (150～160 ℃). It is the normal practice at present to set the allowable power loss at 1～4 kW per row[5]. In addition, P_w is corrected for the ambient temperature in practical applications.

4. Guidelines for shafting design

As the allowable limits for rubber couplings, following vibratory torques are prescribed in NK's guidance note[6]

$$T_1 = 2.5\times10^3\times H/N \quad (0.8\sim1.05N)$$
$$T_2 = 8\times T_1 \quad (\text{below } 0.8N)$$

where H (kW) is maximum continuous output of engine and N (rpm) is engine speed at M.C.R. These torques correspond to manufacturer's allowable vibratory torque and allowable maximum torque respectively.

Table 1 Allowable limits for rubber couplings

causes of failures	allowable limits
• impulsive large torque	$T_v < T_2$
• reduced-cylinder operation	$T_v < T_1$ and $P_v < P_w$
• torsional vibration (resonance)	$T_v < T_1$ and $P_v < P_w$

Allowable power loss P_w is not mentioned in NK's rule.

As already described in section 2, failures of rubber couplings sometimes occur in a short period of time. To avoid the premature failures, the shafting is to be so designed that the vibratory torque T_v and power loss P_v due to torsional vibration will be lower than the corresponding allowable limits. Table 1 shows the allowable limits classified by failures of rubber couplings. If T_v (or P_v) exceeds the limit T_1 (or P_w), the barred speed range for avoiding continuous operation is to be provided.

In case of reduced-cylinder operation including accidental one-cylinder misfiring, attention should be payed particularly to the operation range of engine. Then, the torque fluctuations of 0.5th order are relatively large even in low speed range corresponding to the flank region of vibrations, and the resonance of 1.0st order also appears within this speed range. Accordingly, the operation range of engine to be selected during reduced-cylinder operation becomes considerably narrow in view of the strength of rubber couplings. In that meaning, it is effective to fit the caution plate which explicitly indicates the safe operation range on the control console.

For long-term use of rubber couplings, vibratory torques at N.O.R. of engine are to be diminished to as low level as possible. As for the small coupling in size, the existence of its fatigue limit is confirmed by the manufacturer's test; however, the test results leave many uncertainties in connection with the size of coupling, internal temperature of rubber elements and so on.

To clarify above points, we made the fatigue curves for rubber couplings in a similar way to S-N curves of metals. Selected rubber couplings are of the same type (radial type[2)]) in torque transmission. Cycles of fatigue data are based on the service period of failed couplings. The results are shown in Fig. 12. Vibratory torques were measured or calculated at N.O.R. of engine, and were plotted as a ratio to the nominal torque of rubber coupling. And, the power loss of rubber elements as a parameter was calculated by Eq. (9).

It is seen from this figure that there exist the curves regarded as fatigue and the curves change depending on power loss. Power loss dependent on vibratory torque may not be appropriate for a parameter. However, power loss generally has the same effects as both coupling size and rubber's internal temperature have. That is, in general, high power loss is generated in large couplings in which large vibratory torque is permitted and leads to high temperature in the rubber element. Therefore, power loss is considered to be one of the parameters that can effectively express the difference of fatigue curves.

By the cycles (N_c) obtainted from fatigue curves, the service period of rubber couplings in continuous running of $n(rpm)$ is estimated as follows:

$$S.P. = \frac{N_c}{5.126 \times 10^5 \cdot n \cdot i} \, (years) \tag{10}$$

This method is based on a few assumptions, but can be regarded as one of the effective methods for investigat-

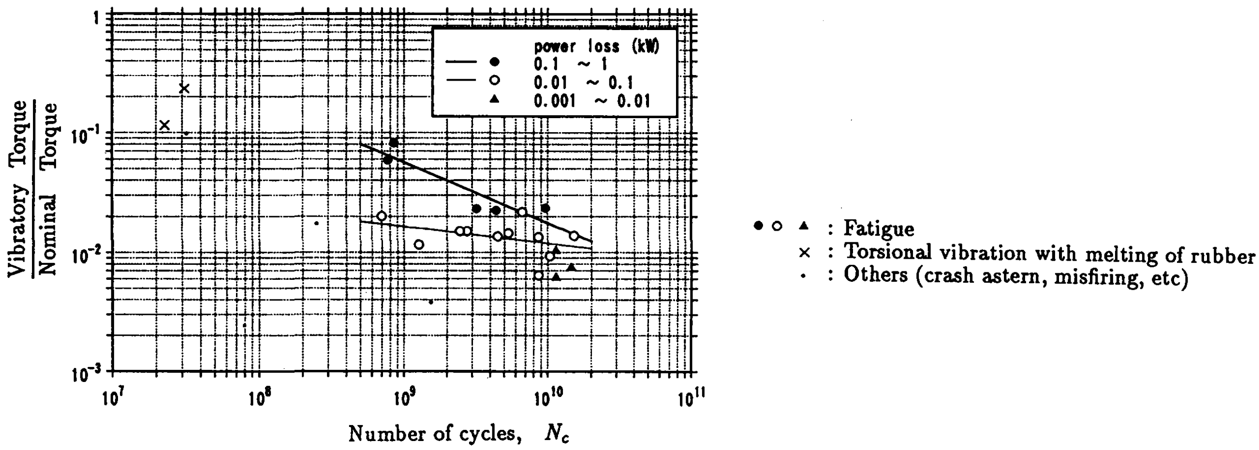

Fig. 12 Fatigue curves for rubber couplings

ing the durability of rubber couplings. If many fatigue data are accumulated and fatigue curves are exactly drawed from those, this method will be the most important means to predict the service period of rubber couplings in the design stage of shafting system.

5. Conclusion

Regarding the failure of rubber couplings, the results of our investigations are summarized as follws:

- A large number of rubber couplings was renewed in the period of 6-8 years after those installation.
- The dispersion seen in service periods of failed couplings is related to the operating conditions of engine, engine troubles such as misfiring and so on.
- As many as 80% of rubber couplings will not endure up to 14 years in service.

In terms of the strength of rubber couplings, we presented the guideline for shafting design including the method for predicting the service period of the coupling. This method is based on fatigue curves of rubber, and such trials are in progress.

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