To cite this article: Hu Z C, He L, Xu W, et al. Optimization design of resonance changer for marine propulsion shafting in longitudinal vibration[J/OL]. Chinese Journal of Ship Research, 2019, 14(1). http://www.ship-research.com/EN/Y2019/V14/I1/107.

DOI: 10.19693/j.issn.1673-3185. 01077

Optimization design of resonance changer for marine propulsion shafting in longitudinal vibration

Hu Zechao[1,2], *He Lin*[1,2], *Xu Wei*[*1,2], *Li Zhengmin*[1,2], *Zhao Xingqian*[1,2]

1 Institute of Noise & Vibration, Naval University of Engineering, Wuhan 430033, China

2 National Key Laboratory on Ship Vibration & Noise, Wuhan 430033, China

Abstract: [**Objectives**] The transmission path of the longitudinal vibration of propeller shafting can be changed and the base response attenuated via a Resonance Changer (RC) integrated in the thrust bearing, enabling the natural frequency of the shafting to be avoided by the propeller's blade frequency and multiplier frequency excitation force. In this way, the purposes of vibration reduction and frequency adjustment can be achieved. [**Methods**] In this paper, the mechanical model of longitudinal vibration of propeller shafting is established and the vibration model of the propeller shaft system is calculated on the basis of the transfer matrix method. The influence of the main parameters of the RC on the vibration isolation effect of propeller shafting is analyzed by taking the force transmission rate as the index. The methods of the minimization of maximum value and parameter correction of the curve area are used to optimize the main parameters of the RC. [**Results**] The results show that the vibration isolation effect of propeller shafting is significantly improved when the RC is installed, and the vibration reduction effect of the RC can be improved by using the design method of the parameter correction of the curve area. [**Conclusions**] The rational design of the RC's parameters can produce a vibration isolating system with excellent vibration isolation effects.

Key words: Resonance Changer (RC); propulsion shafting; transfer matrix method; longitudinal vibration

CLC number: U664.2

0 Introduction

Under the uneven wake field, the pulse excitation force generated by the periodic operation of the propeller is the main noise source when the ship is sailing at a medium or high speed. The longitudinal excitation force is transmitted to the hull by the thrust bearing, which causes the vibration of the shafting and the hull, affects the operation safety of the ship, and reduces the acoustic performance of the hull. In order to reduce the transmission of the longitudinal excitation force to the hull, a shock absorber can be installed on the shafting. Considering that the transmission of thrust bearing has characteristics of large thrust and small displacement, it is necessary to design a vibration isolating device with low stiffness and high resistance. The hydraulic damping device adjusts the stiffness and damping of the system based on the compressibility of fluid. Through rational design, the dynamic characteristics of propulsion shafting can meet requirements. The installation of Resonance Changer (RC) at the thrust bearing not only can adjust the longitudinal natural frequency of propulsion shafting to deviate from the pulse excitation frequency of the propeller, but also can reduce the longitudinal vibration response of the stern of the hull to achieve vibration isolation. Goodwin [1] believed that RC can be equivalent to a

Received: 2017 - 10 - 02

Supported by: Fund Project of National Key Laboratory on Ship Vibration & Noise (9140C280301)

Author(s): Hu Zechao, male, born in 1991, Ph.D. candidate. Research interest: vibration noise control of ship power plant. E-mail: huzechao68@qq.com

He Lin, male, born in 1957, professor, doctoral supervisor, academician of Chinese Academy of Engineering. Research interest: ship stealth technology. E-mail: Helin202@vip.sina.com

Xu Wei, male, born in 1980, Ph.D., associate professor. Research interest: vibration noise control of ship power plant. E-mail: xuwei216@126.com

***Corresponding author:** Xu Wei

mass-spring-damper unit, and designed a hydraulic damping device which can reduce the longitudinal vibration of shafting in a specific frequency band. Dylejko et al. [2] and Li et al. [3] used the transfer matrix method to establish a mathematical model of the propeller-shaft-hull system, and analyzed the influence of the main parameters of RC on the force transmission rate of propeller shafting. Li et al. [4] and Wang et al. [5] optimized the RC design using the dynamic harmonic vibration elimination theory, and obtained the optimal natural frequency ratio and damping ratio of RC. However, the analysis model is simple, and the application range is limited.

In this study, the mechanical model of longitudinal vibration of propeller shafting is proposed, and the vibration response of propeller excitation force transmitted to shell is calculated by the transfer matrix method. With the force transmission rate as the index, the influences of the piston cylinder diameter d_0, connecting pipe length l_1, connecting pipe diameter d_1 and tank volume V_1 on the vibration isolation effects of the propeller shafting are analyzed. Moreover, the main parameters of RC are optimized using the method of the minimization of maximum value of force transfer rate and the correction method of minimum area enclosed by the force transfer rate and coordinate axis respectively.

1 Dynamic model of RC

RC consists of a tank filled with oil, an external pipe system and a piston cylinder. The working fluid in the device can change the longitudinal stiffness and damping of the shafting [6]. Fig. 1 shows the principle model of RC, in which P indicates the pressure difference between the two sides of the piston, and x_0 and x_1 indicate the displacements at both ends of the piston cylinder.

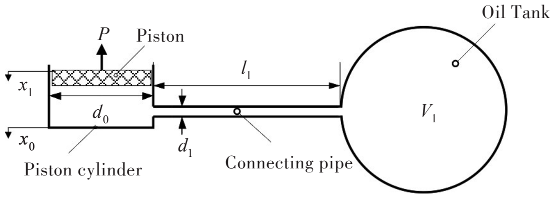


Fig.1 Structure diagram of RC

In order to facilitate the derivation of the dynamic equation of RC, the following assumptions should be made [7]:

1) The tank wall is rigid and all the compression of the fluid occurs in the tank.

2) The fluid in the connecting pipe is in a laminar flow state.

3) The fluid in the pipe can be regarded as the lumped mass.

4) The hydraulic effective length of the fluid is equal to the actual length of the connecting pipe.

5) Compression effect in the pipeline is not considered.

According to these assumptions and the D'Alembert principle, it can be known that the pressure acting on the connecting pipe in the piston cylinder is equal to the sum of the inertia force of the oil in the pipe, the damping force in the pipe and the force required to compress the oil in the tank. Then the dynamic equation of RC can be described as follows:

$$A_1P=\rho_1A_1l_1\frac{(\ddot{x}_1-\ddot{x}_0)A_0}{A_1}+8\pi\mu_1l_1\frac{(\dot{x}_1-\dot{x}_0)A_0}{A_1}+A_1B_1\frac{(x_1-x_0)A_0}{V_1} \quad (1)$$

Where $A_0=\pi(\frac{d_0}{2})^2$ and $A_1=\pi(\frac{d_1}{2})^2$ respectively indicate the sectional areas of piston cylinder and connecting pipe; B_1 indicates the bulk modulus of the oil; μ_1 and ρ_1 respectively indicate the viscosity and density of the oil; $\dot{x}_0$, $\ddot{x}_0$ and $\dot{x}_1$, $\ddot{x}_1$ respectively indicate the speeds and accelerations at both ends of the piston cylinder. The Eq. (2) can be obtained by multiplying the two ends of Eq. (1) by A_0/A_1.

$$A_0P=\frac{\rho_1A_0^2l_1}{A_1}(\ddot{x}_1-\ddot{x}_0)+\frac{8\pi\mu_1l_1A_0^2}{A_1^2}(\dot{x}_1-\dot{x}_0)+\frac{A_0^2B_1}{V_1}(x_1-x_0) \quad (2)$$

Assuming that

$$M_h=\frac{\rho_1A_0^2l_1}{A_1}$$

$$K_h=\frac{8\pi\mu_1l_1A_0^2}{A_1^2}$$

$$C_h=\frac{A_0^2B_1}{V_1}$$

$$F_0=A_0P$$

Where M_h, K_h and C_h represent the mass, stiffness and damping of RC, respectively, Eq. (2) can be converted into a mathematical model of mass-spring-damping:

$$F_0=M_h(\ddot{x}_1-\ddot{x}_0)+C_h(\dot{x}_1-\dot{x}_0)+K_h(x_1-x_0) \quad (3)$$

Where F_0 refers to the external force acting on the piston.

2 Mathematical model of longitudinal vibration of propeller shafting

The mechanical model of longitudinal vibration of propeller shafting is shown in Fig. 2. The model can be decomposed into five subsystems, and each subsystem can use the transfer matrix to represent the transfer relation of longitudinal vibration at the left and right ends of the element. In Fig. 2, the subscripts p, t, c, b, h respectively indicate the propeller, thrust collar, coupling, base and RC; M_p , M_t , M_b , M_c respectively indicate the mass of the propeller, thrust collar, base and coupling; K_0 and K_b respectively indicate the stiffness of oil film and base; C_0 indicates the damping of oil film; L_s and L_{se} respectively indicate the actual length and the effective length of stern shaft; and L indicates the length of intermediate shaft. Units 1–5 are respectively located from the propeller to the coupling. $\boldsymbol{T}_i$ (i = 1, 2, 3, 4, 5) represents its corresponding unit transfer matrix; U_j^{L} and U_j^{R} are respectively the displacement response of the left and right end faces of unit j; F_j^{L} and F_j^{R} are respectively the force response of the left and right end faces of unit j; and subscript j can be replaced by b, h, c, t, p.

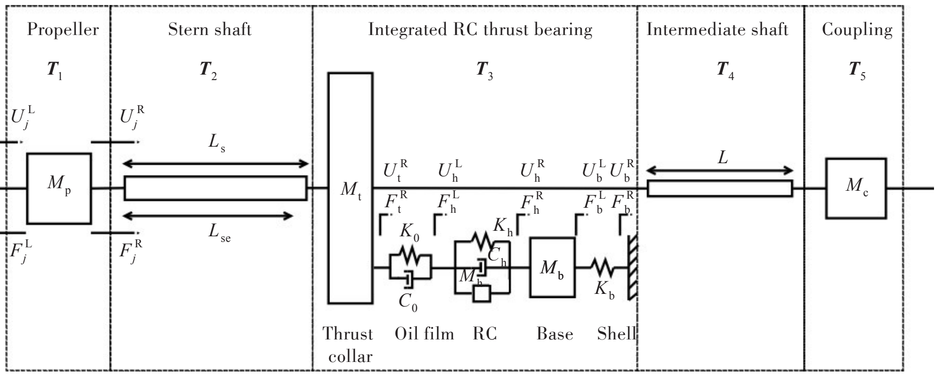


Fig.2 Mechanical model of longitudinal vibration of propeller shafting

2.1 Transfer matrix of longitudinal vibration points of shafting

Considering the effect of the propeller pulse excitation force F_p, the longitudinal transfer matrix of propeller shafting should be rewritten to the form of $\boldsymbol{T}_{3\times3}$, and Eqs. (4)–(7) represent the transfer matrix of each subsystem of propeller shafting:

$$\boldsymbol{T}_1=\begin{bmatrix}1 & 0 & 0\\ -M_p\omega^2 & 1 & F_p\\ 0 & 0 & 1\end{bmatrix} \tag{4}$$

$$\boldsymbol{T}_2=$$

$$\begin{bmatrix}\mathrm{j}\omega\cos k(L_s-L_{se}) & \mathrm{j}\omega\dfrac{\cos k(L_s-L_{se})-\cos kL_s\cos kL_{se}}{EAk\sin kL_s} & 0\\ -\dfrac{EAk\sin kL_s}{\cos k(L_s-L_{se})} & \cos kL_{se} & 0\\ 0 & 0 & 1\end{bmatrix} \tag{5}$$

$$\boldsymbol{T}_4=\begin{bmatrix}\cos kL & \dfrac{\sin kL}{EAk} & 0\\ -EAk\sin kL & \cos kL & 0\\ 0 & 0 & 1\end{bmatrix} \tag{6}$$

$$\boldsymbol{T}_5=\begin{bmatrix}1 & 0 & 0\\ -M_c\omega^2 & 1 & 0\\ 0 & 0 & 1\end{bmatrix} \tag{7}$$

Where $k=\omega/c$ refers to the longitudinal wave number of the shafts (the material properties and sectional areas of stern shaft and intermediate shaft are the same), in which ω refers to the angular frequency and $c=\sqrt{E/\rho}$ (ρ is the density of shafts) refers to the longitudinal wave velocity of the shafts. Besides, E and A refer to the elastic modulus and sectional area of stern shaft and intermediate shaft, respectively. Propeller and coupling can be considered as the lumped mass blocks. Due to the long stern shaft, the effective length should be generally taken into account during calculation, and the corresponding transfer matrix is $\boldsymbol{T}_2$.

The integrated RC thrust bearing can be further decomposed into three units: thrust collar, oil film and RC. The transfer matrix equation from the right end face of thrust collar to the hull can be expressed as Eq. (8):

$$\begin{bmatrix}U_b^R\\ F_b^R\\ 1\end{bmatrix}=\begin{bmatrix}1 & \dfrac{1}{K_b} & 0\\ 0 & 1 & 0\\ 0 & 0 & 1\end{bmatrix}\begin{bmatrix}1 & 0 & 0\\ -M_b\omega^2 & 1 & 0\\ 0 & 0 & 1\end{bmatrix}$$

$$\begin{bmatrix}1 & \dfrac{1}{-M_h\omega^2+K_h+\mathrm{j}\omega C_h} & 0\\ 0 & 1 & 0\\ 0 & 0 & 1\end{bmatrix}\begin{bmatrix}1 & \dfrac{1}{K_0+\mathrm{j}\omega C_0} & 0\\ 0 & 1 & 0\\ 0 & 0 & 1\end{bmatrix}\begin{bmatrix}U_t^R\\ F_t^R\\ 1\end{bmatrix} \tag{8}$$

The shell connected with the base has a large stiffness, which can be regarded as the stiffness boundary condition of propeller shafting (namely $U_b^R=0$) [8]. Then the relationship between F_t^R and U_t^R can be deduced by substituting U_b^R into Eq. (8):

$$\begin{cases} F_{t}^{R} = K_{e} U_{t}^{R} \\ K_{e} = \dfrac{(K_{b} - M_{b}\omega^{2})(K_{0} + j\omega C_{0})(-M_{h}\omega^{2} + K_{h} + j\omega C_{h})}{(K_{b} - M_{b}\omega^{2} + K_{0} + j\omega C_{0})(-M_{h}\omega^{2} + K_{h} + j\omega C_{h}) + (K_{b} - M_{b}\omega^{2})(K_{0} + j\omega C_{0})} \end{cases} \tag{9}$$

Where K_e refers to the equivalent stiffness between thrust collar and shell.

According to Eq. (9), the thrust bearing-base-shell model can be simplified to the equivalent mechanical model shown in Fig. 3, and the transfer matrix $\boldsymbol{T}_3$ corresponding to the integrated RC thrust bearing can be simplified to Eq. (10):

$$\boldsymbol{T}_3 = \begin{bmatrix} 1 & 0 & 0 \\ -M_t\omega^2 + K_e & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{10}$$

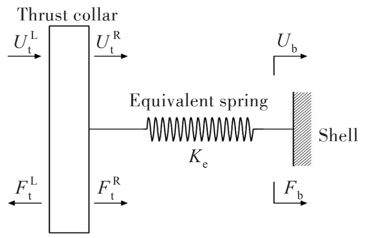


Fig.3 Equivalent mechanical model of thrust bearing and base

2.2 Calculation of longitudinal vibration response of shafting

After the transfer matrices of $\boldsymbol{T}_1$ - $\boldsymbol{T}_5$ are obtained, the vibration response of propeller shafting can be solved according to the boundary conditions. Eq. (11) refers to the transfer matrix from the input end of the propeller to the output end of the coupling. In the equation, $T_{r,q}$ represents the element of $\boldsymbol{T}$ in the r-th row and the q-th column ($1 \leqslant r, q \leqslant 3$). When pulse excitation force F_p acts on the propeller, the left end face of propeller and the right end face of coupling can be regarded as the free boundary conditions (namely, $F_1^L = F_5^R = 0$), and Eq. (12) can be obtained by substituting the constraint conditions into Eq. (11):

$$\boldsymbol{T} = \boldsymbol{T}_5\boldsymbol{T}_4\boldsymbol{T}_3\boldsymbol{T}_2\boldsymbol{T}_1 \tag{11}$$

$$U_1^L = -\frac{T_{2,3}}{T_{2,1}} F_p \tag{12}$$

Eq. (13) and Eq. (14) indicate the transfer matrix $\boldsymbol{T}'$ of the input end of propeller from base to shell, where $T'_{r,q}$ represents the element of $\boldsymbol{T}'$ in the r-th row and q-th column ($1 \leqslant r, q \leqslant 3$). After Eq. (14) is substituted into Eq. (9), the response force F_b of the excitation force transmitted to the shell by the propeller shafting is obtained, from which the force transmission rate $T_f = F_b / F_p$ of propeller shafting can be calculated.

$$\boldsymbol{T}' = \boldsymbol{T}_3\boldsymbol{T}_2\boldsymbol{T}_1 \tag{13}$$

$$U_t^R = T'_{1,1}U_1^L + T'_{1,3}F_p \tag{14}$$

$$F_b = F_t^R = K_e U_t^R \tag{15}$$

3 Optimization design method of RC structure

In order to make the propeller shafting have a good vibration isolation effect, it is necessary to design the parameters of RC. The rational design of structure parameters enables the RC device to absorb most of the vibration energy of propeller shafting, which not only limits the force transmission rate to a certain range and makes the resonance peak become smaller, but also adjusts the natural frequency of propeller shafting, so as to avoid the propeller's blade frequency excitation and achieve the purpose of vibration reduction and frequency adjustment. By taking the parameters of l_1 , d_0 , d_1, V_1 of RC as the design objectives, we optimize the RC structure using the method of the minimization of maximum value of force transmission rate [9] and the correction method of the minimum curve area of force transmission rate [10].

3.1 Optimization method of minimization of maximum value

The optimization of RC structural parameters can be expressed as the design problem of minimizing the maximum value of peak value of the force transmission rate curve, namely that through some algorithm, a set of design variables (l_1, d_0, d_1, V_1) are searched to minimize the multiple peak values of $T_f^a(l_1, d_0, d_1, V_1)$ of force transmission rate of propeller shafting within the analysis frequency range. This optimization method can be described as Eq. (16):

$$\text{obj1} = \min\{\max_{1 \leqslant a \leqslant m} T_f^a(l_1, d_0, d_1, V_1)\} \tag{16}$$

Where a represents the order of longitudinal mode; m represents the number of peaks of the force transmission rate in the analyzed frequency band.

The optimization method of minimization of maximum value focuses more on the variation of the peaks of force transmission rate, and is suitable for the optimization of constraint conditions with strict limits on the magnitude of forces transmitted to the shell or on the strength of shafting. Therefore, it is essentially a local optimization method.

3.2 Minimum area correction method

For each set of design variables (l_1, d_0, d_1, V_1) the area enclosed by the force transmission rate curve $T_f(l_1, d_0, d_1, V_1)$ and the coordinate axis f is denoted as $A_f^a(l_1, d_0, d_1, V_1)$ and the method can be described as Eq. (17).

$$\text{obj2} = \min_{1\leqslant a\leqslant m} A_f^a(l_1, d_0, d_1, V_1) \tag{17}$$

In other words, for the minimum value of $A_f^a(l_1, d_0, d_1, V_1)$ in the whole variable range, the corresponding (l_1, d_0, d_1, V_1) is searched. This method optimizes the force transmission rate curve in the entire analyzed frequency band. Although the local peak values of the curve may be increased, the overall vibration isolation effect of propeller shafting can be improved in the analyzed frequency band, so it is a global optimization method.

4 Case analyses

Taking the propeller shafting of a certain type of ship as an example, the calculated parameters are as follows: $M_p = 7\,000$ kg, $M_c = 1\,000$ kg, $M_b = 4\,000$ kg, $M_t = 500$ kg, $\rho_1 = 860$ kg/m³, $\rho = 7\,850$ kg/m³, $E = 200$ GPa, $A = 0.02$ m², $L_s = 14.6$ m, $L_{se} = 14$ m, $L = 2$ m, $K_b = 5 \times 10^9$ N/m, $B_1 = 1.38$ GPa, $\mu_1 = 0.23$ Pa·s, and number m of propeller blades being 7.

According to the experimental data, the oil film stiffness K_0 and the damping C_0 are related to the rotating speed n and the load F acting on the propeller. When $F = 200$ kN, the variation curve of K_0 and C_0 with the rotating speed is shown in Fig. 4. Assuming that $n = 220$ r/min (the corresponding blade frequency is 25.7 Hz), the corresponding $K_0 = 1.4 \times 10^{10}$ N/m, and $C_0 = 6.5 \times 10^8$ N·s/m. The preliminarily designed RC parameters are $l_1 = 1$ m, $d_0 = 0.06$ m, $d_1 = 0.01$ m and $V_1 = 1.6$ L.

The influence of RC parameters (l_1, d_0, d_1, V_1)

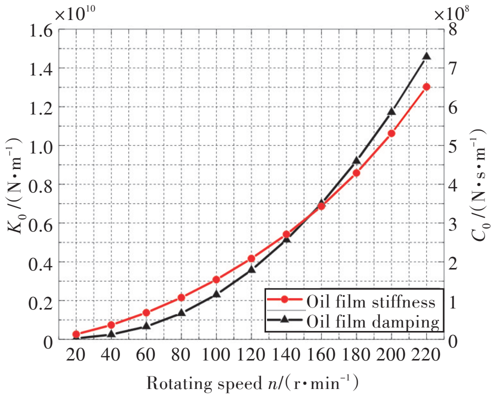
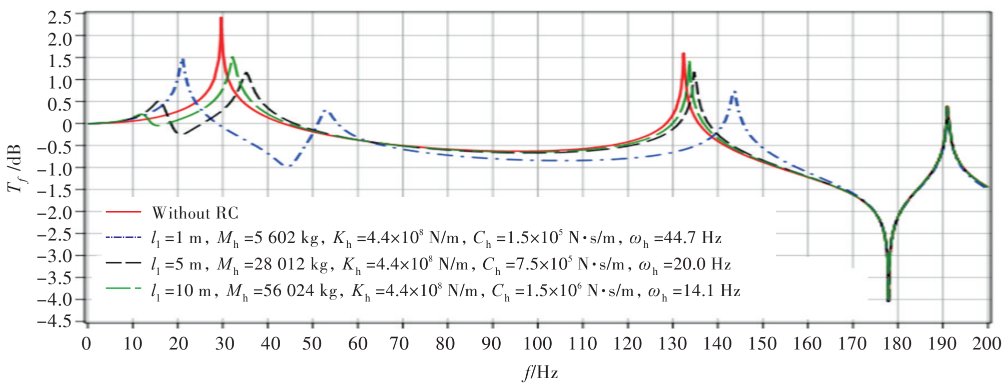


Fig.4 Variation curve of oil film stiffness and damping with rotating speed

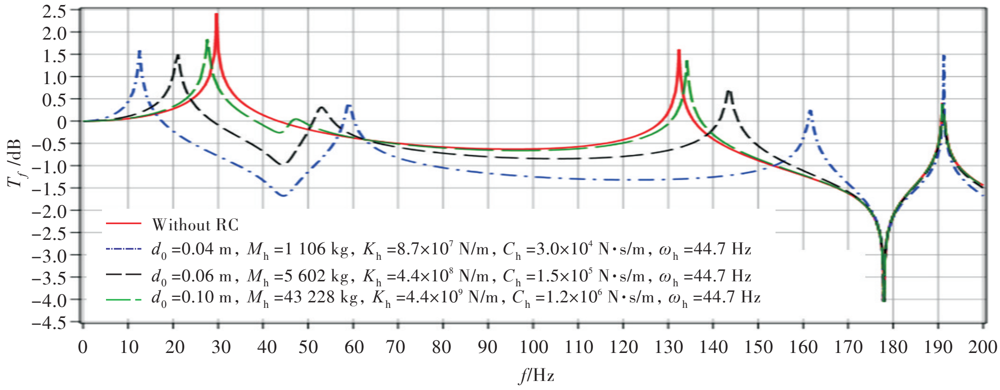
on vibration isolation effect of propeller shafting is shown in Fig. 5. This figure shows that when RC is not installed, the resonance peaks of first two orders of propeller shafting appear near the 30 and 132 Hz, and the first-order resonance peak is large. Compared with the force transmission rate curve in the case without installing the RC system, the first-order mode peaks of propeller shafting after the installment of RC are converted into the low order modes with smaller second-order peaks, which are respectively located at both sides of the first-order mode frequency when the RC system is not installed. Obviously, in the analyzed frequency band of 0–200 Hz, RC has a great influence on the modes of the first two orders in propeller shafting.

According to the variation trends of the M_h, K_h, C_h, ω_h frequencies of RC with the l_1, d_0, d_1, V_1, the following conclusions can be drawn:

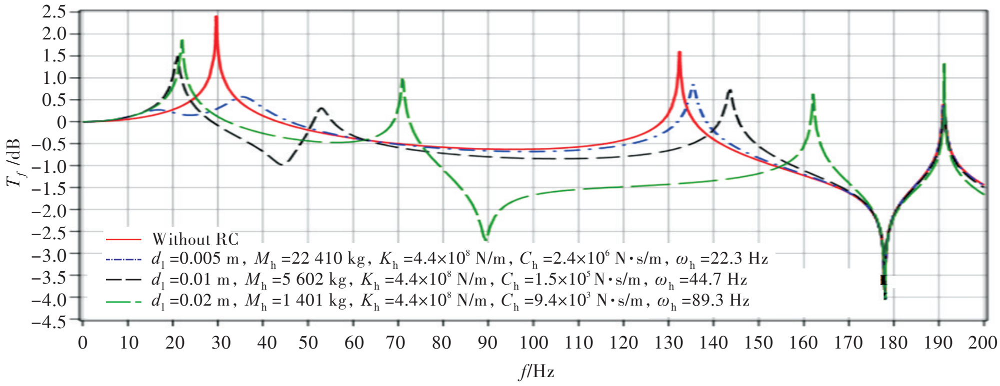
1) As l_1 increases, the RC equivalent damping C_h increases, and the peak of the force transmission rate of propeller shafting decreases accordingly. If l_1 is too long, the mass of RC will increase. If l_1 is too short, the peak of the first-order mode of propeller shafting is large. Therefore, the value range of l_1 must be constrained, and it can be constrained as $l_1 \in [0.5\text{ m}, 10\text{ m}]$.



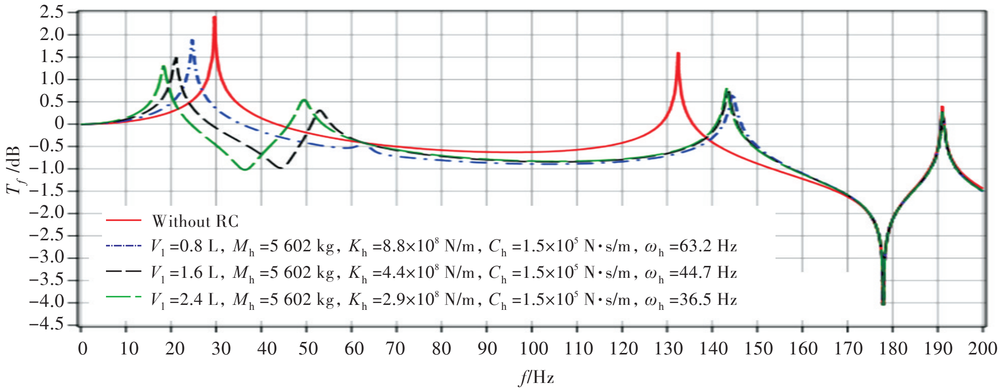
(a) Connecting pipe length l_1



(b) Piston cylinder diameter d_0



(c) Connecting pipe diameter d_1



(d) Oil tank capacity V_1

Fig.5 Influence of RC's main parameters on vibration isolation effect of propeller shafting system

2) When ω_h is constant, as d_0 increases, K_h and M_h increase at the same proportion. If $d_0 \to \infty$ RC can be regarded as a rigid body, and the force transmission rate of RC will be close to the case without RC. Therefore, the value of d_0 should not be too large when the RC is designed. To achieve good vibration isolation effect, we can set $d_0 \in [d_1, 0.1\ \mathrm{m}]$.

3) When K_h is constant, the equivalent mass M_h of RC increases as d_1 decreases. When RC is designed, its equivalent mass should not be designed as too large, and we can set $d_1 \in \left[0.005\ \mathrm{m}, d_0\right]$.

4) When M_h is constant, the increase of V_1 is equivalent to the thickening of oil layer in the oil tank, the stiffness K_h of RC will decrease accordingly, and the vibration isolation effect will be enhanced. However, the excessive V_1 will cause the displacement response of the coupling end to exceed its allowable range and thus affect the normal operation of the motor, while the too small V_1 will make the force transmission rate curve close to the curve without RC, resulting in RC failure. In order to avoid the above two phenomena, we can set $V_1 \in [0.8\ \mathrm{L}, 2.4\ \mathrm{L}]$.

After the constraint conditions of RC parameters are determined, the objective functions obj1 and obj2 are used to optimize RC parameters, and the optimization results are shown in Fig. 6. The followings can be seen from the figure.

1) The optimized force transmission rate of RC is significantly reduced in the analyzed frequency band

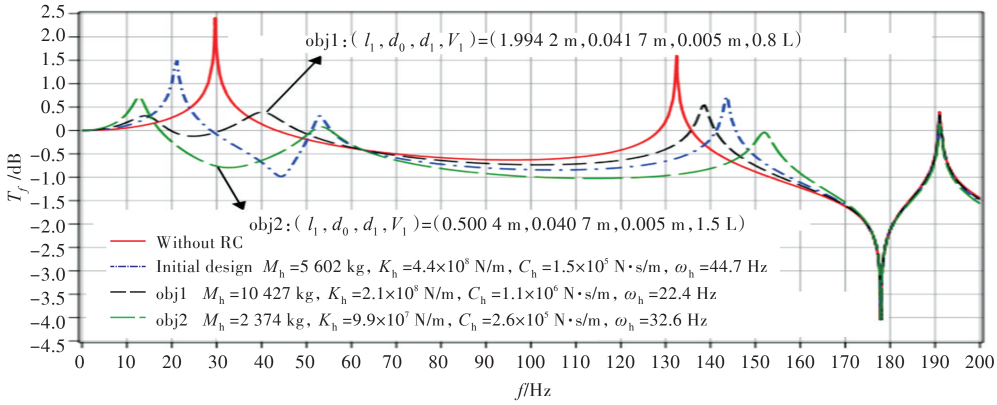


Fig.6 Optimal results for RC parameters

compared with that of the initially designed RC, and the peaks in the modes of the first three orders are reduced.

2) The method of the minimization of maximum value can minimize the peaks of the force transmission rate, but cannot guarantee that RC has excellent vibration isolation performance in the entire analyzed frequency band.

3) The minimum area correction method can make RC have the best vibration isolation performance in the analyzed frequency band, but it does not rule out that there are large peaks at some frequencies.

For this example, according to the optimization results, the objective function obj2 should be used to optimize the RC parameters. Although the peaks of the force transmission rate curve is slightly increased relative to obj1, the vibration isolation effect of the optimized RC in the frequency range of 0 – 200 Hz is obviously improved. Compared with the initial design, the optimization method of obj2 reduces the peaks at the first-order mode by about 5.3 times, and avoids the blade frequency excitation force of 25.7 Hz, which obtains an evident effect of vibration reduction and frequency adjustment, and satisfies the use requirements.

5 Conclusions

In this paper, the mathematical model of longitudinal vibration of propeller shafting is established, and the influence of RC parameters on the vibration isolation performance of propeller shafting is analyzed using the transfer matrix method with the force transmission rate as the index and based on the actual ship data. The constraint conditions of design parameters are given by theoretical analysis, and the RC structure is optimized based on the method of the minimization of maximum value and the minimum curve area correction method. The following conclusions are obtained:

1) The introduction of RC eliminates the first-order mode of propeller shafting, but brings the low order modes with smaller second-order peaks, which are respectively located on both sides of the first-order mode frequency in the case without RC. The second-order mode frequency moves in the high-frequency direction, but the peak does not change much and has no effect on the third-order mode.

2) After the installation of RC, the vibration isolation effect of the shafting has been significantly improved.

3) RC parameters have great influence on the vibration isolation performance of propeller shafting, and the rational design of RC structure can make the system obtain a good vibration isolation effect.

4) Compared with the initial design scheme of RC, both the optimization methods can improve the vibration isolation performance of propeller shafting and keep its natural frequency away from the propeller's blade frequency and multiplier frequency excitation force. The optimization method of the minimization of maximum value can minimize the peaks of the force response transmitted to the shell, and the design method of the minimization of curve area can make the vibration reduction effect of the RC device better in the whole analyzed frequency band.

References

[1] Goodwin A J H. The design of a resonance changer to overcome excessive axial vibration of propeller shafting [J]. Transactions of the Institute of Marine Engineers, 1960, 72: 37-63.

[2] Dylejko P G, Kessissoglou N J, Yan T, et al. Optimization of a resonance changer to minimize the vibration transmission in marine vessels [J]. Journal of Sound and Vibration, 2007, 300(1/2): 101-116.

[3] Li Z M, He L, Cui H G, et al. Study of the influence of the resonance changer on the longitudinal vibration of marine propulsion shafting system [C]//International Conference on Vibroengineering. Nanjing, China:

JVE International Ltd., 2015: 65-70.

[4] Li L W, Zhao Y, Lu P, et al. Optimization of dynamic absorber parameters for reducing axial vibration of ship shafting[J]. Shipbuilding of China, 2010, 51(2): 139-148 (in Chinese).

[5] Wang J, Jiang Y K, Wang Z Y, et al. Design and optimization of resonance changer for longitudinal vibration of ship shafting[J]. Computer Aided Engineering, 2015, 24(6): 52-57 (in Chinese).

[6] Li L W, Zhao Y. Research on longitudinal nonlinear vibration characteristic of marine shafting resonance changer[J]. Shipbuilding of China, 2011, 52(4): 74-82 (in Chinese).

[7] Chu W, Zhao Y, Zhang G B. Dynamic anti-resonance vibration isolation theory of resonance changer and application[J]. Journal of Ship Mechanics, 2016, 20(1/2): 222-230 (in Chinese).

[8] Zhang G B, Zhao Y, Li T Y, et al. Propeller excitation of longitudinal vibration characteristics of marine propulsion shafting system[J]. Shock and Vibration, 2014, 3(1): 413592.

[9] Hou X L, Li H Y, Liu J. Optimal algorithm for minimization of maximum value problems and application of dynamic absorb or with multi-DOF[J]. Journal of Vibration and Shock, 2008, 27(1): 100-103 (in Chinese).

[10] Wang W, Zhao Q H, Zhang H Y. Analysis of optimal parameters of dynamic reducer[J]. Journal of Vibration and Shock, 2006, 25(5): 180-182 (in Chinese).

船舶推进轴系纵向振动共振转换器的优化设计

胡泽超[1,2], 何琳[1,2], 徐伟[*1,2], 李正民[1,2], 赵兴乾[1,2]
1 海军工程大学 振动噪声研究所,湖北 武汉 430033
2 船舶振动噪声重点实验室,湖北 武汉 430033

摘　要:[**目的**]在推力轴承上集成共振转换器(RC)可以改变轴系纵向振动的传递路径,衰减传递到基座的响应使轴系的固有频率避开螺旋桨叶频及其倍叶频激励力,从而实现减振、调频的目的。[**方法**]为此,建立推进轴系纵向振动的力学模型,基于传递矩阵法计算桨轴系统的振动响应,以力传递率为指标,分析RC的主要参数对推进轴系隔振效果的影响,分别采用最大值最小化方法和曲线面积最小的参数修正方法,对RC的主要参数进行优化设计。[**结果**]研究结果表明:加装RC后,轴系的隔振效果得到了明显的改善,采用曲线面积最小修正的优化设计方法可使RC的减振调频效果更佳。[**结论**]通过对RC结构参数的合理设计能使减振系统获得优良的隔振效果。

关键词:共振转换器;推进轴系;传递矩阵法;纵向振动