Coupling analysis of torsional vibration and engine rotational speed control system of marine propulsion shafting

Shuwen YU[1]; Yan LIU[1]; Xiao HAN[1]; Meilong CHEN[1]; Wanyou LI[1]

[1] College of Power and Energy Engineering, Harbin Engineering University, China

ABSTRACT

For general simulation of engine rotational speed control system, the shafting is always assumed to be rigid. The torsional vibration characteristics of the shafting are neglected. The possible relationship between the rotational speed oscillation phenomenon and coupling of the torsional vibration and the engine rotational speed control system will not then be revealed in such simulation. In this paper, by assuming the shafting to be elastic instead of rigid, a coupling simulation model is established. The influences of different PID control parameters and variation of the propeller damping to the speed-control performance are analysed to illustrate the coupling characteristics of the shafting torsional vibration.

Keywords: Torsional vibration, Rotational speed control system, Coupling

I-INCE Classification of Subjects Number(s): 44

1. INTRODUCTION

According to provisions of classification societies in “Rules for Building and Classing Steel Vessels” [1-2], torsional vibration of marine shafting is classified as strength problems while engine rotational speed control system as steady-state or transient droop problems. In general simulation model of complete engine rotational speed control process, the shafting is always assumed to be rigid.

With continuous development of marine technology, wide usage of propulsion multi-load system and combined power plants, marine propulsion shafting is getting more and more complex. During the operation of an engine driving marine propulsion system with single propeller, rotational speed oscillation phenomenon occurred; the controller rod also vibrated intensely at the same time. However, all corresponding performance analyses of the marine are up to standard: including both vibration of the shafting system and simulation of the rotational speed control system. To find the reason for rotational speed oscillation of the shafting, these two systems should be analysed in a couple way.

In this paper, a coupling model of the marine propulsion system is established by refining the rigid shafting to be an elastic one. The influences of different PID control parameters and variation of the propeller damping to the speed-control performance are analysed.

2. COUPLING MODEL

2.1 Shafting model

1) According to the principle of reserving characteristics of vibration [3], the lumped parameter model is established by simplifying the shafting into an elastic model with 7 inertias and 6 shaft sections, as shown in Figure 1. The descriptions of the inertias are shown in Table 1.

[1] 253352615@qq.com

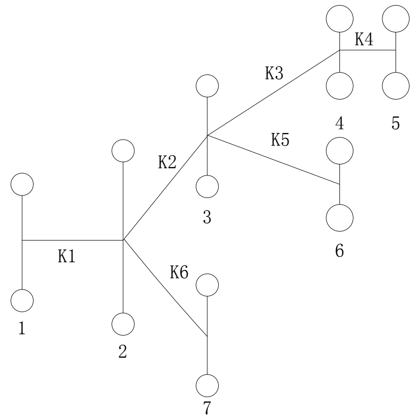


Figure 1 – Torsional vibration model of shafting

Table 1 – Scheme of the inertias

NO.	Inertia
1	Engine
2	Gears and elastic Plate, shaft sections
3	Gears and elastic Plate, shaft sections
4	Tail shaft
5	Propeller
6	Elastic plates, flanges and shaft sections
7	Elastic plates, flanges and shaft sections

The equation of free torsional vibration can be written as [4]:

$$\mathbf{J}\ddot{\boldsymbol{\theta}}+\mathbf{K}\boldsymbol{\theta}=\mathbf{0} \tag{1}$$

The form of solution of the equation can be set as:

$$\{\boldsymbol{\theta}\}=\{\mathbf{A}\}\cos\omega\mathbf{t} \tag{2}$$

By substituting Eq. (2) to Eq. (1), the matrix equation can then be obtained:

$$[\mathbf{K}]\{\mathbf{A}\}=\lambda[\mathbf{J}]\{\mathbf{A}\} \tag{3}$$

Where, $\mathbf{K}$ is the torsional stiffness matrix, $\mathbf{J}$ is the inertia matrix, λ represents eigenvalues of shafting system. The natural frequencies are calculated by $\omega=\sqrt{\lambda}$.

Program for calculating the natural frequencies of torsional vibration is written with MATLAB software. The first three-order natural frequencies are listed in the Table 2.

Table 2 – Natural frequencies of torsional vibration

NO.	Natural frequencies, Hz
1	2.44
2	4.27
3	18.79

2) The simulation model of the shafting which is a linear multi-input, multi-output system is presented based on state-space method [5]. Forced torsional vibration differential equation is shown as follow:

$$\mathbf{J}\ddot{\boldsymbol{\theta}}+\mathbf{C}\dot{\boldsymbol{\theta}}+\mathbf{K}\boldsymbol{\theta}=\mathbf{M_n} \tag{4}$$

According to theories of state-space method, it can be expressed in the form of state equation:

$$\begin{aligned}\dot{\mathbf{x}}&=\mathbf{A}\mathbf{x}+\mathbf{B}\mathbf{u}\\ \mathbf{y}&=\mathbf{C}\mathbf{x}+\mathbf{D}\mathbf{u}\end{aligned} \tag{5}$$

Where, $\mathbf{x}$ is the state variable; $\mathbf{u}$ is the input variable; y is the output variable;

$\mathbf{A}=\begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{J}^{-1}\mathbf{K} & -\mathbf{J}^{-1}\mathbf{C} \end{bmatrix}_{14\times14}$ is the system matrix reflecting characteristics of the shafting system; $\mathbf{B}=\begin{bmatrix} \mathbf{0} \\ \mathbf{J}^{-1} \end{bmatrix}_{14\times7}$ is the input matrix; $\mathbf{C}=\begin{bmatrix} \mathbf{I}_{7\times7} & \mathbf{0}_{7\times7} \end{bmatrix}$ is the output matrix; $\mathbf{D}=\begin{bmatrix} \mathbf{0}_{7\times7} \end{bmatrix}_{7\times7}$ is called the direct-effect matrix of the system; **J** is the inertia matrix; **K** is the stiffness matrix; **C** is the damping matrix [6].

Depending on the modern control theories, the stability of the state-space model can be distinguished by eigenvalues of matrix **A**: if all the eigenvalues have negative real parts, the system is stable; otherwise the system is unstable. So the plot (Figure 2) of the shafting matrix eigenvalues shows that the shafting model is stable.

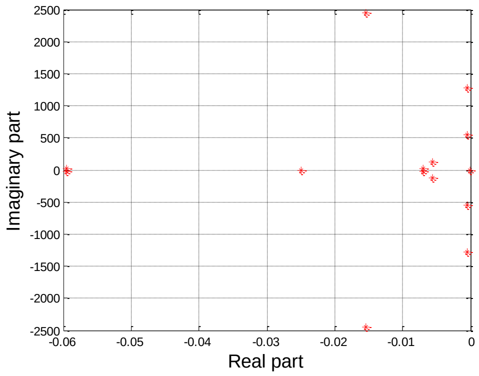
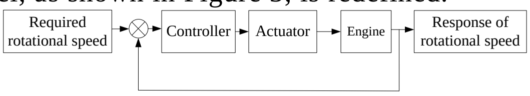


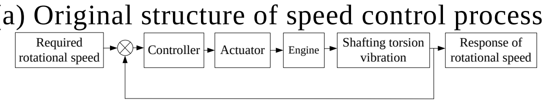
Figure 2 – Eigenvalues of the shafting matrix

2.2 Coupling model of the propulsion system

To establish the coupling model, the elastic shafting model is substituted for the original rigid rotor model. Therefore, torsional vibration is coupled with the rotational speed control process. After that, the closed-loop control model, as shown in Figure 3, is redefined.



(a) Original structure of speed control process



(b) Coupling structure of speed control process

Figure 3 – Structure of rotational speed control process

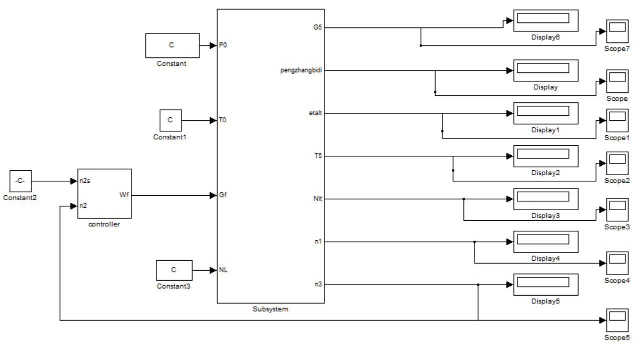


Figure 4 – Simulink simulation model

The premise of the coupling process is that torsional vibration and rotational speed control form a closed-loop control system. That is, the signals collected by the sensors and passed to the speed controller are actually the derivatives of the angle responses of the elastic shafting torsional vibration. After receiving the rotational speed signal, speed controller's regulation performance is reflected as an exciting torque applied upon the shafting. In turn, the torque affects the response of the shafting torsional vibration. The closed-loop coupling process is shown in Figure 5.

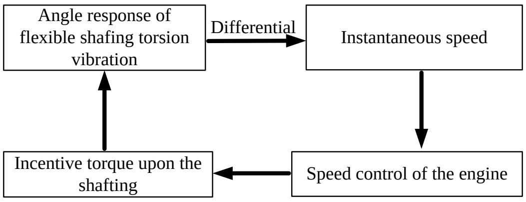


Figure 5 – Closed-loop control process of the coupling model

3. COUPLING ANALYSIS

1) The rotational speed of the propulsion system and the output power of the engine can remain stable with the PID control parameters shown in Figure 6. The output power of the engine determines the exciting torque applied upon the shafting.

Some simulation results are presented. Taking Inertia 2 as an example, the relative rotational speed curve is shown in Figure 7. And relative output power curve is shown in Figure 8. With these two figures, the effectiveness of the PID control parameters and correctness of the coupling simulation model are both verified.

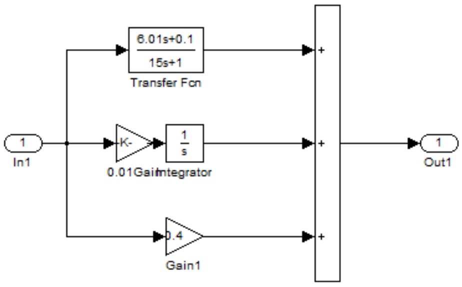


Figure 6 – PID control parameters

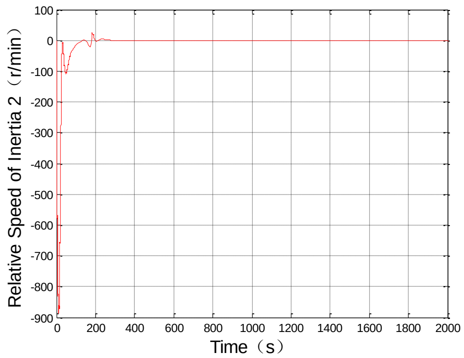


Figure 7 – Rotational speed curve of Inertia 2

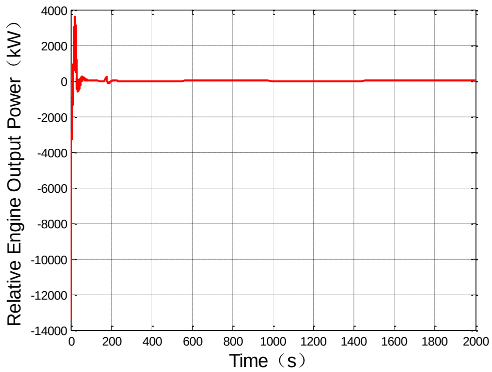


Figure 8 – Output power curve of engine

2) Under the adjustment of the PID control with parameters as shown in Figure 9, the rotational speed tends to fluctuate smoothly within the permissible range around 4r/ min (Figure 10). The reason of the acceptable fluctuation is due to torsional vibration of the elastic shafting.

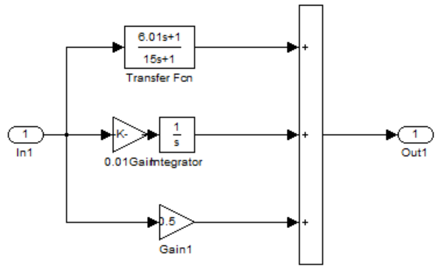


Figure 9 – PID control parameters

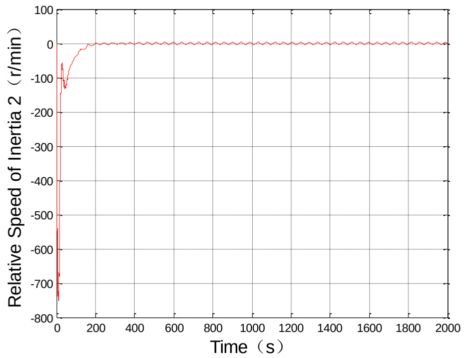
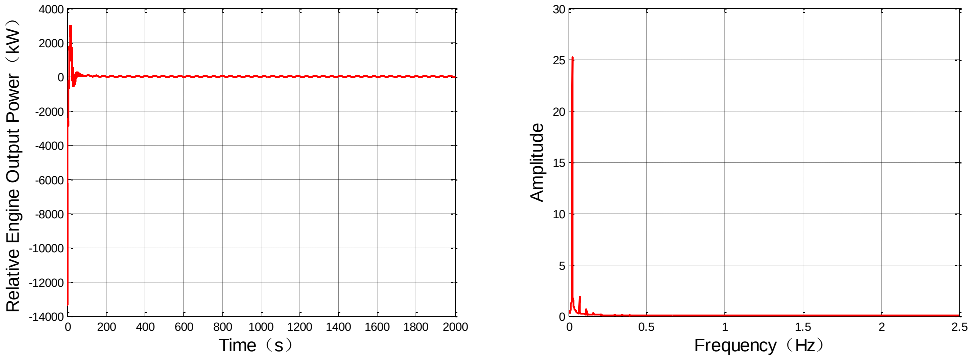


Figure 10 – Rotational speed curve of Inertia 2



(a) Time domain (b) Frequency domain

Figure 11 – Output power curve of engine

It is obvious that there are no components of torsional vibration around natural frequencies in the signals of the closed loop according to the spectrum of Figure 11 (b).

3) Under the adjustment of the PID control with parameters as shown in Figure 12, the rotational speed tends to diverge and become unstable. Due to the closed loop, the output power of the engine also shows tendency of divergence, as shown in Figure 13, and Figure 14 (a).

As Figure 14 (b) show, two distinct peak frequencies (2.397Hz, 4.267Hz) in the spectrum of the output power are equal to the first two-order natural frequencies of the shafting torsional vibration (see Table 2) , indicating that there are components of natural frequencies of torsional vibration in the signals of the closed loop.

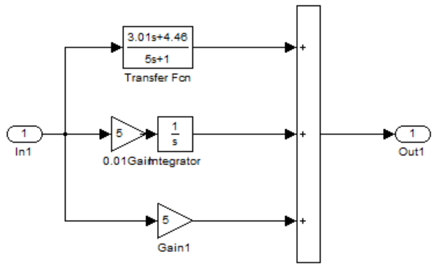


Figure 12 – PID control parameters

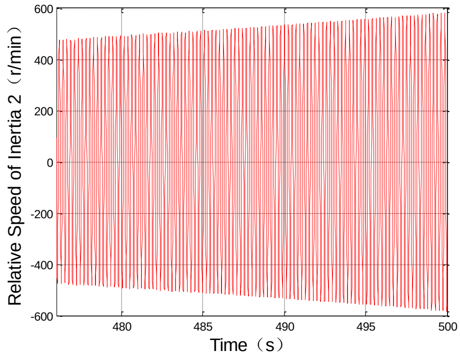
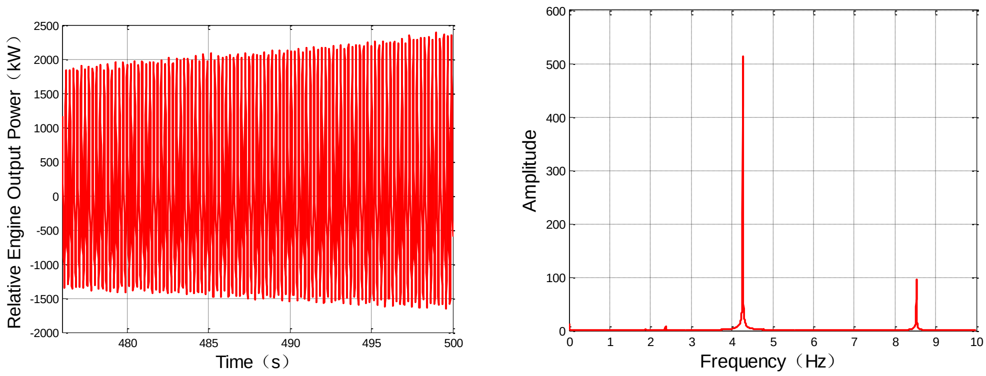


Figure 13 – Rotational speed curve of Inertia 2



(a) Time domain (b) Frequency domain

Figure 14 – Output power curve of engine

Accordingly, the divergence is due to resonance when the torque applied upon the shafting determined by the PID control parameters contains the components of natural frequencies of torsional vibration. It is essential to take torsional vibration into account in the design stage of the rotational speed control system.

4) In the variable working conditions, propeller damping may change in a certain range while the PID control parameters stay constant. After the change of the working condition, there is a fault situation that the original PID control parameters do not fit the shafting system so that the rotational speed tends to diverge.

For instance, under the control of the certain parameters as shown in Figure 15, the rotational speed (Figure 16) tends to be stable with acceptable fluctuations with the propeller damping of 10000Nms/rad. However, when the propeller damping is changed, the performance of the PID control parameters shows large differences. The rotational speeds are unstable obviously with the propeller damping of 1000Nms/rad and 0Nms/rad.

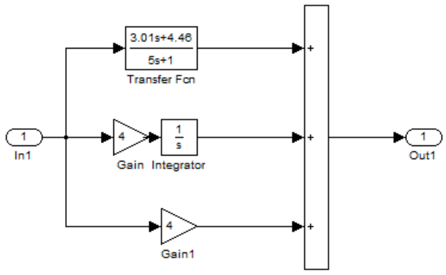


Figure 15 – PID control parameters

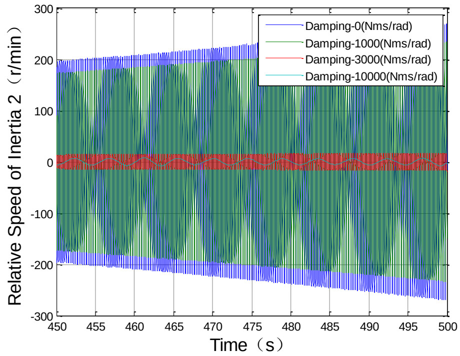


Figure 16 – Rotational speed curve of Inertia 2

The spectrums of the rotational speed curves with damping of 10000Nms/rad and 0Nms/rad are displayed respectively in Figure 17 and Figure 18. There are no peak frequencies near the natural frequencies so that the rotational speed is stable in Figure 17. While in Figure 18, there is exactly one peak frequency equal to second-order natural frequency so that the rotational speed diverges.

It follows that variation of the propeller damping would affect the stability of the rotational speed control.

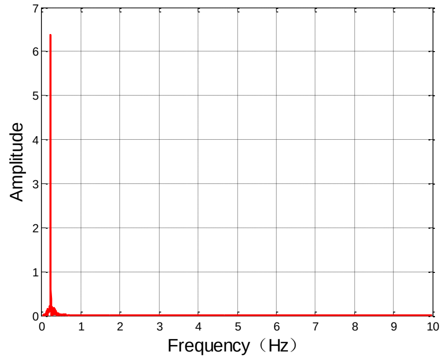


Figure 17 – Rotational speed spectrum of propeller damping 10000Nms/rad

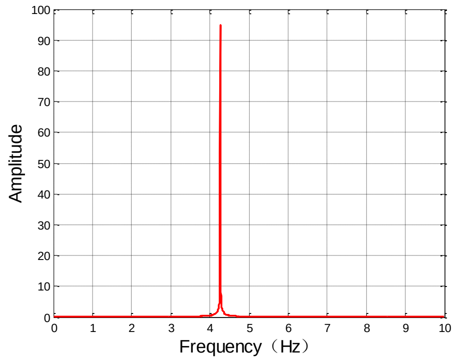


Figure 18 – Rotational speed spectrum of propeller damping 0Nms/rad

The PID control parameters are reset as shown in Figure 19. Under the adjustment of this group of parameters, simulation results in Figure 20 show that the rotational speeds with different damping values are all stable.

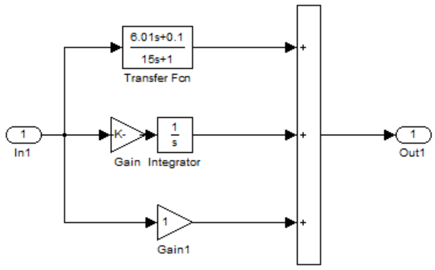


Figure 19 – PID control parameters

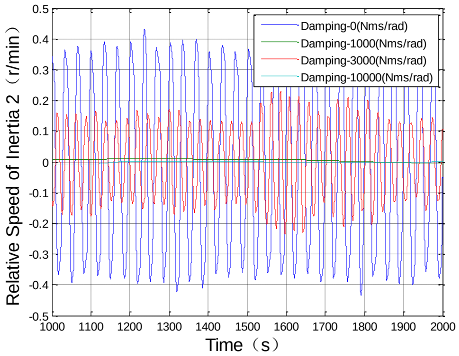


Figure 20 – Rotational speed curve of Inertia 2

4. CONCLUSION

For analysing the shafting rotational speed oscillation phenomenon in real engineering, a coupling rotational speed control system model considering torsional vibration of shafting is established in this paper.

By analysing speed-control simulation results of the coupling model under different PID control parameters, the direct reason of the rotational speed oscillation is due to resonance of the shafting at low-order natural frequencies.

Due to alteration of working conditions, propeller damping would vary in a certain range and the simulation results showed that the variation of damping would affect the speed-control performance with certain PID control parameters. Therefore, in practical engineering applications, it is essential to choose appropriate control parameters to stabilize the rotational speed due to possible variation of the propeller damping.

In conclusion, the coupling between torsional vibration of shafting system and the speed control system should be taken into account during the design of control system of a marine propulsion system. How to choose appropriate control parameters is of vital importance for making sure that the system would operate in safe and stable condition.

REFERENCES

1. CCS. Rules for Building and Classing Steel Vessels.2005, 3(12):127-129.
2. ABS. Rules for Building and Classing Steel Vessels.2008, Part 4.
3. CHEN Zhi-yan. Vibration of Ship Propulsion Shafting.Shanghai: Press of Shanghai Jiaotong University, 2000.p.60-63.
4. WANG Qi. Torsional Vibration of Engine Crankshaft. Dalian: Dalian University of Technology Press, 1991.p.48-53.
5. LI Ying, ZHU Bo-li, ZHANG Wei. Simulink-Dynamic System Model and Simulation. Xian: Press of Xian University of Electronica Science and Technology, 2004.7
6. ZENG De-hui. Simulation of Mechanical Vibration Systems with Multiple Degrees of Freedom Based on Matlab/Simulink. Journal of Hubei Institute for Nationalities, 2008, 26(1):57-60.