Research on the Theory and Simulation of Torsional Vibration for CODAD Systems

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Abstract—Two machines driving one propeller, especially CODAD (Combined Diesel and Diesel) has been widely used in marine propulsion shafting for power performance and fuel economy, but there were few researches about the device for shaft vibration. In this paper, based on the independent design multi-functional test bench of the marine power plant, the torsional vibration calculation method of the test bench was established and the calculation results were gotten by the theory analysis. Then the corresponding system simulation model was set up on the basis of constructing discrete dynamics model of the two diesel engines with different cylinders, and response curves of the shaft under the different rotational speed were obtained. The resonant frequency of the shaft system was estimated through Fast Fourier Transform Algorithm (FFT), and the simulation result has better consistency with the theoretical value, which proved that the natural frequency in complex shafting could be obtained by the simulation method.

Keywords-propulsion shafting; CODAD; torsional vibration; simulation; discrete model

I. INTRODUCTION

The traditional power plant in the marine with single propeller driven by one machine (engine) have been unable to meet the need of some large ships for the dynamic reason, so the power plant that two machines get together and drive one propeller increase day by day. Currently, high power ships, engineering ships, energy-saving ships and Military vessels abroad can meet the application of the corresponding devices [1-3]. Low carbon economy putting forward raised higher request to the ship power plant performance: for the diesel engine as the main engine for power plant, i.e., make the output power of diesel engine meet the requirements of dynamic performance, and hope that diesel engine could works in the most economic conditions at the same time, and hope the whole device function maximization and purchasing cost minimization. CODAD (Combined Diesel and Diesel) systems had been gotten more and more extensive application in ship power device [4].

With the application widely of this system, the shafting torsional vibration problems are also beginning to cause the attention of many scholars. Marine shafting vibration influences the ship navigation performance and security directly, while accurate analysis of this kind of shafting torsional vibration characteristics can provide reliable scientific and theoretical basis in ship design, manufacture, the reduction of vibration noise and accident [5-6], so the research to shafting vibration has the very important meaning.

II. PRACTICAL MODEL

The multi-functional test bench of the marine power plant was established, which could be seen in Fig .1, in order to research shafting coupled vibration better for CODAD device [7-8].. The core device is 2GWH100 gearbox, in which there were four sets of clutches, was produced by Hangzhou Advance Gearbox Group CO.LTD, the ratio is 1, the four were interfaces connected with a three cylinder diesel engine, a four cylinder diesel engine, a frequency conversion motor and a dynamometer respectively. According to the frequency conversion motor working status and clutch closed or release, the gearbox can achieve a variety of transmission ways, so as to realize single diesel engine driving dynamometer, CODAD driving dynamometer, electric propulsion, single diesel engine driving dynamometer and PTO (power take-off) and so on In the following analysis, only the CODAD working conditions was illustrated.

Figure 1. the multi-functional test bench

III. CALCULATION

A. Model

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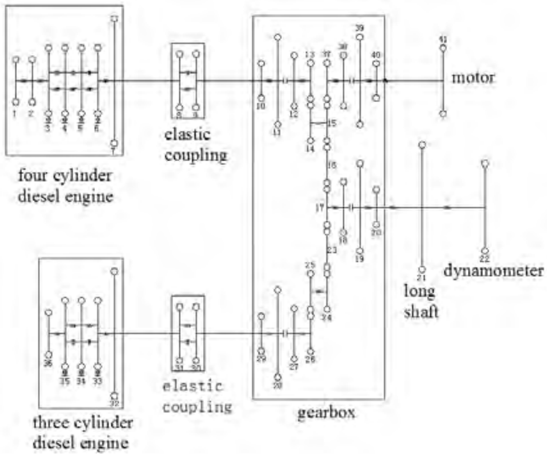


Figure 2. The equivalent diagram of the multi-functional test bench

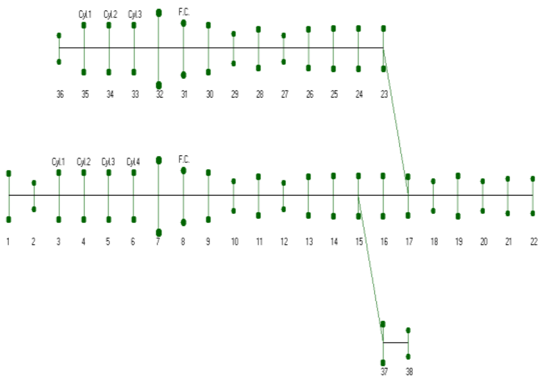


Figure 3. The calculation equivalent diagram

B. Analysis and Calculation

From Fig .2 the torsional free vibration equation is defined as below

$$[J]\{\ddot{\theta}\}+[k]\{\theta\}=\{0\} \qquad (1)$$

Where $[J]$, $[k]$, $\{\ddot{\theta}\}$, $\{\theta\}$ represent the total mass matrix, total stiffness matrix, angular acceleration vector and angular displacement respectively.

According to Ref. [10], in the transmission system, each shaft runs in different speed, so transmission system with different speed is often transformed into an equivalent system with the same rotation speed, and in engine propulsion shafting the diesel engine speed is often treated as the standard speed in the torsional vibration calculation. Meanwhile, the meshing gears are processed as one component, or the stiffness between the gears is infinite Equivalent inertia and equivalent stiffness calculation formula is given in (2).

$$J_l = i^2 J_f\,,\; k_l = i^2 k_f \qquad (2)$$

Where subscript f and l represent the physical quantity after and before equivalent calculation respectively. Transmission ratio $i = n_f / n_l$ But in the gear transmission's time, the gear tooth profile in addition to rotate to exactly the same, namely the ratio of 1.

Therefore, the total mass matrix could be written as:

$$[J]=diag(J_1,J_2,\Lambda\,,J_{38})$$

And the total stiffness matrix could be defined as:

$$[k]=\begin{bmatrix}a_1 & a_2 & \Lambda & a_{37} & b_{38}\end{bmatrix}^T$$

Where

$$a_1=[k_{12},-k_{12},0,\Lambda\,]_{38}$$
$$a_2=[-k_{12},k_{12}+k_{23},-k_{23},0,\Lambda\,]_{38}$$
$$a_3=[0,-k_{22},k_{23}+k_{34},-k_{34},0,\Lambda\,]_{38}$$
$$\Lambda$$
$$a_{15}=[0,\Lambda\,,0,-k_{14-15},k_{14-15}+k_{15-16}+k_{16-37},-k_{15-16},0,\Lambda\,,0,-k_{16-37},0]_{38}$$
$$a_{16}=[\underbrace{0,\Lambda,0}_{15},-k_{15-16},k_{15-16}+k_{16-17},-k_{16-17},0,\Lambda\,]_{38}$$
$$a_{17}=[\underbrace{0,\Lambda,0}_{16},-k_{16-17},k_{16-17}+k_{17-18}+k_{17-23},-k_{17-18},0,0,0,0,0,-k_{17-23},0,\Lambda\,]_{38}$$
$$\Lambda$$
$$a_{22}=[\underbrace{0,\Lambda,0}_{21},-k_{21-22},k_{21-22},0,\Lambda\,]_{38}$$
$$a_{23}=[\underbrace{0,\Lambda,0}_{13},-k_{17-23},\underbrace{0,\Lambda,0}_{8},k_{17-23}+k_{23-24},-k_{23-24}0,\Lambda\,]_{38}$$
$$\Lambda$$
$$a_{36}=[\underbrace{0,\Lambda,0}_{35},-k_{35-36},k_{35-36},0,\Lambda\,]_{38}$$
$$a_{37}=[\underbrace{0,\Lambda,0}_{15},-k_{16-37},0,\Lambda\;0,k_{16-37}+k_{37-38},-k_{37-38}]_{38}$$
$$a_{38}=[0,\Lambda\;0,-k_{37-38},k_{37-38}]_{38}$$

The free torsional vibration is calculated by self-made software, and tab. I gives first ten orders natural frequency in this system.

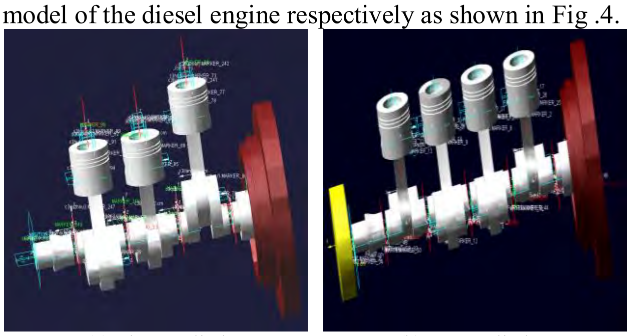
TABLE I. FIRST FIVE ORDERS OF NATURAL FREQUENCY (R/MIN)

Order	*1*	*2*	*3*	*4*	*5*
Frequency	*835.02*	*1769.98*	*5070.28*	*10478.57*	*15703.97*

IV. THE SIMULATION ANALYSIS

A. Diesel engine crankshaft simulation

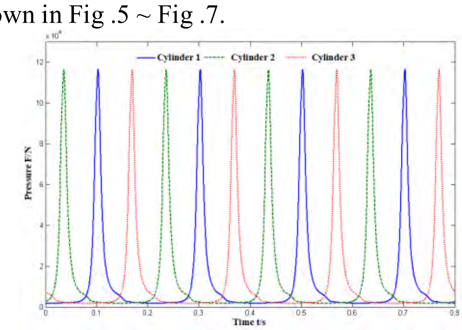
The vibration components in the diesel engines are mainly composed of the crankshaft, piston and connecting rod, and there is a silicone oil shock absorber in the crankshaft free end of the four cylinders diesel. In order to add torsional rigidity and torsional damper between the crankshaft main journals in simulation model easily, the actual crankshaft model have been established in discrete model. After inputting the model in into ADAMS, add the appropriate motion pairs, constraint and drive among the components deputy. The crankshaft crank would fluctuate in rotation speed for the torsional spring, so the cylinder explosion pressure was no longer a function of crank angle, but the function of time, and the function should be written as "AKISPL (SPLINE_i time * n, 0, 0)", where n is the rotation speed, the same as the crankshaft speed set by the free end, "SPLINE_i" for the corresponding to the pressure of the cylinder pressure data, the dynamic simulation model of the diesel engine respectively as shown in Fig .4.



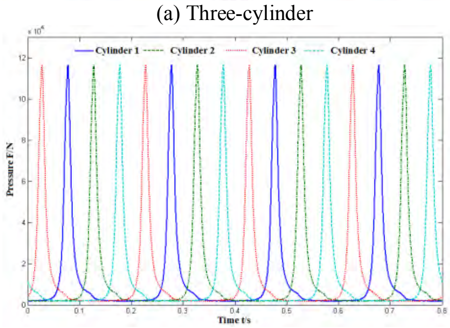
(a) Three-cylinder (b) Four-cylinder

Figure 4. Diesel engine dynamic simulation models

When the rotating speed is 600 r/min, that the input speed is "3600d * time", the diesel engine cylinder explosion pressure, piston acceleration and the angular velocity curve of the flywheel can be obtained respectively, as shown in Fig .5 ~ Fig .7.



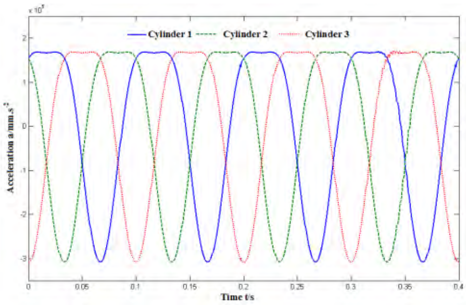
(a) Three-cylinder



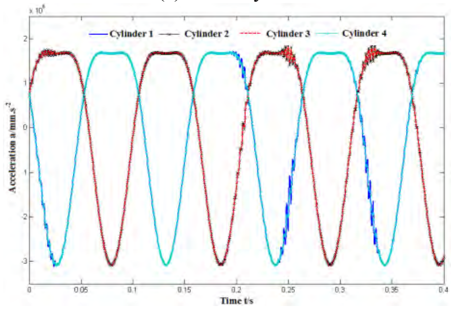
(b) Four-cylinder

Figure 5. diesel engine cylinder explosion pressure (600 r/min)

Due to the outbreak of diesel engine cylinder pressure is a function of time, so the Fig .5 curve according to the change of firing order in diesel engine; Fig .6, the overall trend of acceleration of the piston is completely consistent, however, there will be a slight fluctuations in change process considering the influence of crankshaft torsional vibration; Fig .7 reacts the relationship between the diesel engine flywheel angular velocity changes over time, the cycle is 0.2s, the peak value of each period in the three cylinder diesel engine with three, and four cylinder diesel engine in each period of peak has four, correspond to the main order of diesel engine.

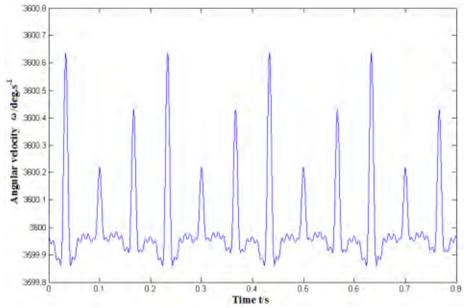


(a) Three-cylinder

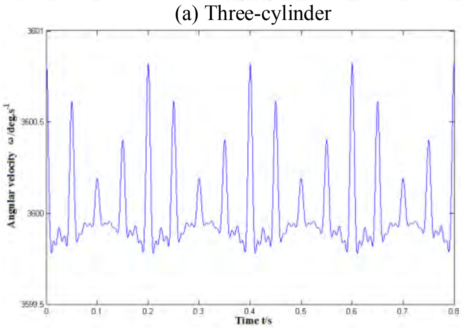


(b) Four-cylinder

Figure 6. diesel engine cylinder piston acceleration (600 r/min)



(a) Three-cylinder



(b) Four-cylinder

Figure 7. diesel engine flywheel angular velocity curve (600 r/min)

B. The shaft system simulation

According to the actual operation of the system status, add appropriate constraints and drive in the shafting parts , as shown in Fig .8, add "contact" contact between gear pair, added torsional spring between the shaft and the gearbox output flanges, the stiffness is determined by shafting parameters. This model is mainly used for the influence that the simulation of gear system's played on the system response.

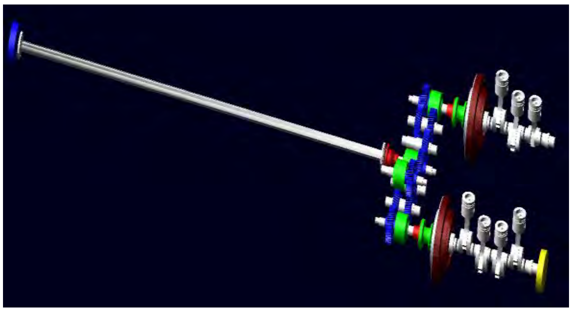


Figure 8. ADAMS dynamics simulation model of the system

Speed parameters change, the "5400d * time" at "300 d" intervals until "9000d * time," which means speed at 50 r/min intervals from 900 r/min to 1500 r/min. The long axis angular velocity (the ADAMS software can't directly get angular displacement) curve in 900r/min and 1050r/min as shown in Fig .9 (initial impact is very big, only show the steady state was 0.5 ~ 1.5s, hereinafter the same).

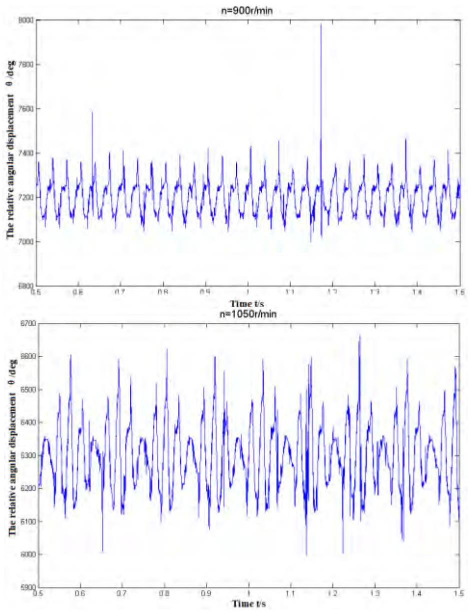


Figure 9. Shaft angular velocity under different rotation speed change curve

FFT was used for all data in different speed, and the spectrum was drawn in one figure, which was shown in Fig .10. It can be seen from the Fig .10, two of the biggest amplitude appeared between 40.14 Hz and 43.77 Hz. According to the basic principle analysis of vibration test, the system between the two horizontal there will be a resonance point, namely the resonance speed should appear in the 2408.4 r/min, and 2626.2 r/min, and the working condition of three or four in table 1 order resonance speed were 5070.28 r/min and 10478.571 r/min, just four times in the simulation results of double frequency and frequency, and between the theoretical calculation results and simulation results are well corresponding. In fact, the accurate resonance point, not a range, would be obtained by enough simulation times.

V. CONCLUSION

1. According to the established CODAD test bench, the first five order natural frequency of the model was calculated based on the shafting multi-branch torsional vibration calculation method

2. The discrete diesel engine dynamic simulation model the test bench was set up, and the simulation curves showed the results meet to the practical test data.

3. The experimental system of rigid dynamics model was established, and the change of the intermediate shaft's angular was analyzed in different rotational speed, so the spectrum diagram was gotten by FFT and the resonance of the system could be estimated, which matched the theoretical value. So the natural frequency in complex shafting could be obtained by the s imulation method with changing rotational speed and FFT.

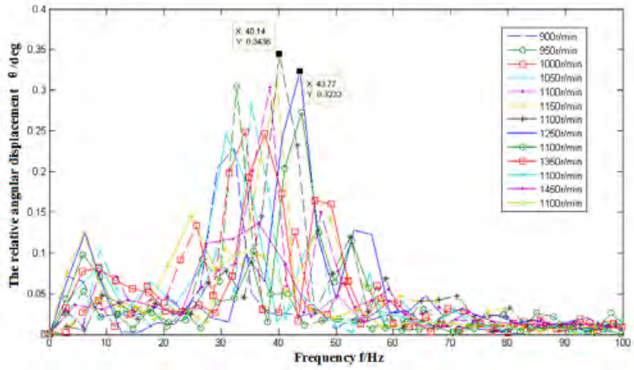


Figure 10. Rigid model of the rotational speed of frequency domain

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