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MODAL CHARACTERISTICS ANALYSIS OF BLOCK AND CRANKSHAFT OF 18V32/40 DIESEL ENGINE FOR OFF-SHORE PLATFORM POWER STATION

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With the intensification of the energy crisis, the offshore oil exploitation has aroused more and more attention. As one of the important power source of the offshore platform, the excessive vibration and noise level of the medium and high speed diesel engine will have a significant influence on its secure operation and comfort. In this paper, taking the 18V32/40 offshore platform diesel engine as the analysis model, the three-dimensional solid model for the engine block and crankshaft is built and meshed. The free and constraint modal analysis are performed for the engine block and crankshaft structure using the software of Virtual. Lab. Numerical simulation is conducted for the comparison of modal characteristics of engine crankshaft, crankshaft with flywheel, and crankshaft with flywheel and balance weight. The results show that the natural frequencies of the engine block will be increased with the addition of restraints, and the vibrational displacement of the upper part of engine block will be reduced for the higher mode, while the vibration level for the internal main bearing wall is still significant. The natural frequency will be decreased for the crankshaft with the flywheel and balance weight added, and vibrational displacement will be increased. The vibration can be improved for some low order mode of engine block at the location where the isolator is applied, and the introduction of balance weight can attenuate the crankshaft vibration to some extent.

Keywords: offshore platform, diesel engine generator, block and crankshaft, modal analysis

1. Introduction

The diesel engine is widely used in land vehicles, ships and many other engineering fields, because of its high economy, wide power range and convenient maintenance. Marine diesel engine is an important power element, and it is also a significant source of vibration and noise. The vibration is transmitted to the hull structure through the base to form the external radiation noise [1]. At the same time, too much vibration can cause damage to the components, such as the connecting rod, piston and shafting of diesel engine, but also affect the crankshaft and connecting rod big end and small end bearing lubrication and wear, detrimental to the normal operation of the diesel engine [2]. Marine diesel engine vibration and noise problem has been widely concerned, because of the adverse effect on safe operation of power plant and comfort of ship. A variety of methods are used to optimize the vibration and noise of diesel engine.

In the 1950s, the vibration and noise of diesel engine begun to enter the field of research. In 1973, Lalor from University of Southampton published a paper on the low noise design of diesel engine. The effects of the engine's overall mode and local plate mode on the engine noise were systematically expounded [3]. Simply supported shafting is taken as the research object for Khadem and Hossein to study the free vibration response of the extensible and inextensible shafting under the action of vertical and horizontal coupling [4]. They believe that the longitudinal inertia can be ignored and they also studied the active control response and further simplified the dynamic response equation [5]. Dynamic software ADAMS and Virtual. Lab et al are widely used in engine industry, such as the German Volkswagen Company using ADAMS software to calculate the engine vibration caused by valve train obtain good results [6]. Li used ADAMS software to calculate the main load of the diesel engine under the two rigid and flexible models, motivated by it, the vibration of the whole machine is calculated by finite element method [7]. Novotny and Pistek proposed a method combined finite element analysis and dynamic analysis in order to tremendous cost of vibration numerical computation [8].

With the intensification of the energy crisis, the offshore oil industry further increases. High power diesel engine suitable for offshore platforms is more and more widely used. The operating environment of the engine increases the requirements for the reliability of diesel engine. There are strict regulations on the vibration and noise of the high speed running diesel engine in the actual specification of offshore platform. On the one hand, the resonance between the platform and equipment should be prevented, on the other hand, severe vibration may cause damage to the equipment of the generating set [9].

Based on the literature research, it can be found that the research on vibration of the offshore platform diesel engine is relatively rare, and the study is mainly confined to model and simulation of offshore drilling platform. As the main bearing part of the engine, the engine block is subjected to complex excitation, and its vibration has an important influence on the vibration and noise radiation of the whole machine. The crankshaft is the most complex part of the diesel engine, and the vibration is complicated.

In this paper, we combine the modal analysis with the finite element method to mesh and model the engine block and crankshaft of 18V32/40 diesel engine. As for engine block without cylinder cover, Virtual. Lab software will be used to calculate two cases of the free mode and the constraint mode. Then the free mode of crankshaft under three conditions the crankshaft, the crankshaft with flywheel and the crankshaft with flywheel and balance weight will be analysed. Finally, the vibration characteristic of two parts are obtained and analysed.

2. Simulation Model

2.1 Finite Element Modal Analysis Foundation

The research object in this paper is large and complex, it is difficult to obtain satisfactory results by certain single analytical theory. To this end, we use the finite element method for analysis. The stiffness matrix and mass matrix of each element are set up according to the serial number of node freedom, the overall stiffness matrix and total mass matrix are obtained. In this section, the diesel engine block is taken as an example. In the usual physical coordinate system, the dynamic equation can be expressed as:

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

in which, $\{x\}$, $\{\dot{x}\}$, $\{\ddot{x}\}$, $\{f\}$ are displacement, velocity, acceleration and excitation force column vector. For the analysis of vibration, they are harmonic change with time. $[M]$ is the mass matrix, $[C]$ is the damping matrix, and $[K]$ is the stiffness matrix, all of which are symmetric fixed length real matrices, and is positive definite, $[C]$ and $[K]$ is definite or semidefinite, $[C]$ is under damping, the conditional damping is also satisfied:

$$[C]=\alpha[M]+\beta[K] \tag{2}$$

The $\{f\}$ is a zero matrix, and the two order linear constant coefficient homogeneous equation is obtained:

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=\{0\} \tag{3}$$

Suppose the solution is:

$$\{x\}=\{X\}e^{\lambda t} \tag{4}$$

Substitute the equation (4) into the equation (3), one will get:

$$\left(\lambda^2[M]+\lambda[C]+[K]\right)\{X\}=\{0\} \tag{5}$$

In order to obtain the nonzero solution, the coefficient determinant of the formula (5) is zero:

$$\det\left(\lambda^2[M]+\lambda[C]+[K]\right)=0 \tag{6}$$

This is $2n$ order linear algebraic equation of λ , which is called the characteristic equation of the system. M is a positive definite or positive definite symmetric matrix. In the case of damping, the complex eigenvalue of n in the form of conjugate pairs can be obtained by solving this equation.

The eigenvalues λ_i and the corresponding eigen-vectors are called the eigen-pairs of the system. Its number is equal to the number of degrees of freedom of system. The n eigen vectors are arranged in order of size, and the $n\times n$ matrix is called the modal matrix of the system.

When the multi degree freedom system of engine block or crankshaft is subjected to arbitrary excitation, the differential equations of motion are shown in Eq. (1). The system is assumed to be in equilibrium position without initial velocity. By using the Laplasse transform, the formula (1) is transformed into the complex plane of the $s=-\alpha+j\omega$ of the Laplace variable:

$$\left(s^2[M]+s[C]+[K]\right)\{X(s)\}=\{F(s)\} \tag{7}$$

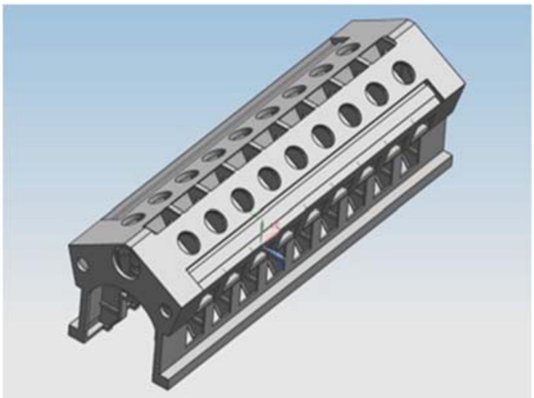
In the formula, $\{X(s)\}$ and $\{F(s)\}$ are respectively the dynamic response of the system and the Laplace transform of the excitation force:

$$\{X(s)\}=[H(s)]\{F(s)\} \tag{8}$$

$\{X(s)\}=[H(s)]\{F(s)\}$ is the transfer function matrix of the system, which reflects the dynamic characteristic of the system.

2.2 Solid Modelling and Meshing

The UG software is used to establish the three-dimensional geometric model of the engine block and crankshaft as shown in Figure 1 (a) and (b). Because of the complex structure, there are various kinds of stiffener, lug, bearing hole and oil hole. When the three-dimensional model is established, the small bolt hole and the oil hole with an internal diameter less than 5mm and the lug which cannot play a major role are neglected, but stiffener is retained.

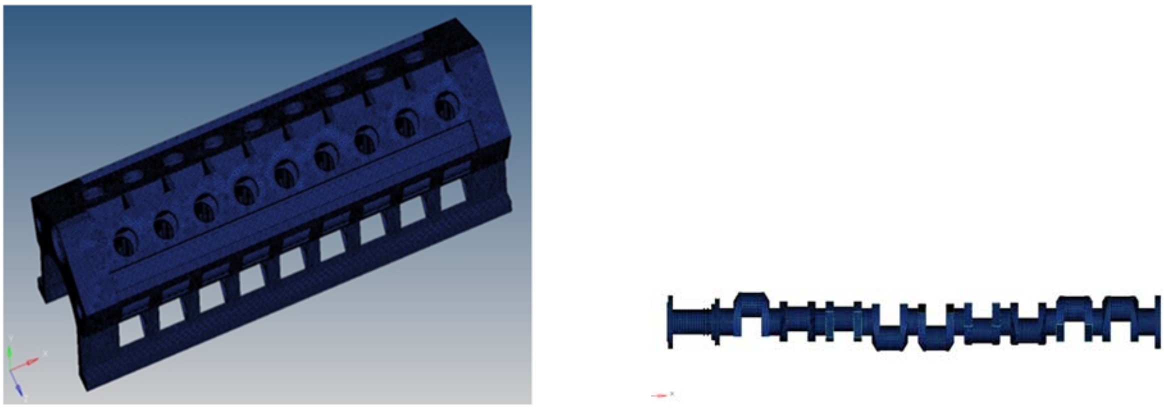


(a) Model 1, engine block model

(b) Model 2, crankshaft model

Figure 1: The simplified geometrical model of diesel engine block and crankshaft model obtained in NX software

Hypermesh software is used to mesh the model of 18V32/40 diesel engine block and crankshaft, and the finite element analysis model is established. The finite element mesh model is shown in Figure 2 (a) and (b). By using tetrahedron element, the model of diesel engine block is obtained with 258470 nodes and 1357229 cells, the crankshaft model consists of 69996 nodes and 378910 cells.



(a) Model 1, engine block model (b) Model 2, crankshaft model

Figure 2: Finite element model of 18V32/40 diesel engine block and crankshaft.

3. Numerical Results and Discussion

In this section, the Virtual.Lab software is used to carry out the finite element analysis of the crankshaft and the block. Select the density DENS=7800Kg/m, elastic modulus EX=209000GPa, Poisson ratio σ =0.25, solid 185 grid element.

3.1 Modal Analysis of Diesel Engine Block

Firstly, modal analysis of the diesel engine block without cylinder cover is conducted. Through the comparison of the two modes of the free mode and the constrained mode, the influence of the constrained on the vibration of the block is studied. When the structure resonance occurs, usually lower order resonance is more dangerous. In this paper, the first order modes of the free mode of the block are calculated (expected for the first six constrained modes). Table 1 and Table 2 describe the parameters and position coordinates of the vibration isolator, and Figure 3 shows the position of the isolator.

Table 1: Device parameter of diesel engine vibration isolation.

Isolator types	X static stiffness (kN/mm)	Y static stiffness (kN/mm)	Z static stiffness (kN/mm)
AV/C2L 65	12.8	12.8	9.6
	X damping coefficient	Y damping coefficient	Z damping coefficient
	5%	5%	5%
	X dynamic stiffness coefficient	Y dynamic stiffness coefficient	Z dynamic stiffness coefficient
	1.6	1.6	1.6
Isolator types	X static stiffness (kN/mm)	Y static stiffness (kN/mm)	Z static stiffness (kN/mm)
AV/C2L 55	9.8	9.8	6.3
	X damping coefficient	Y damping coefficient	Z damping coefficient
	5%	5%	5%
	X dynamic stiffness coefficient	Y dynamic stiffness coefficient	Z dynamic stiffness coefficient
	1.4	1.4	1.4

To calculate the constrained mode, the constraints are added to the position of the isolator. Figure 4 (a) and (b) give the first order natural frequency and vibration mode of the block in the free and constrained mode. Table3 and Table 4 are the results of the natural frequencies of free and constrained mode.

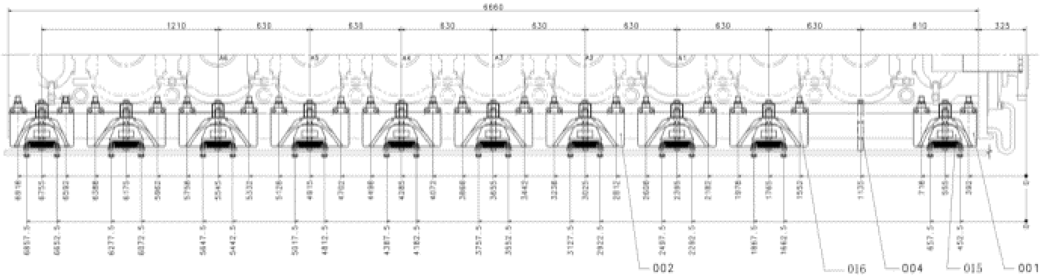


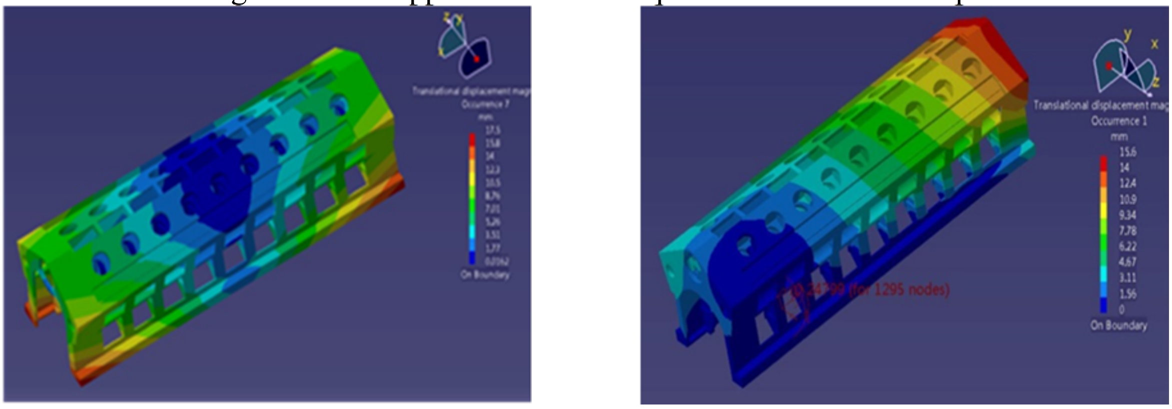
Figure 3: Arrangement of diesel engine vibration isolator.
(The free end is on the left, the flywheel side is on the right.)

Table 2: Parameter of vibration isolation arrangement coordinate (Coordinate of 20 vibration isolation, number 1 to 12 close to the flywheel side, number 13 to 20 close to the free end).

Isolator number	Name	X(mm)	Y(mm)	Z(mm)
1	AV/C2L 65	555	1025	-1410
2	AV/C2L 65	555	-1025	-1410
3	AV/C2L 65	1765	1025	-1410
4	AV/C2L 65	1765	-1025	-1410
5	AV/C2L 65	2395	1025	-1410
6	AV/C2L 65	2395	-1025	-1410
7	AV/C2L 65	3025	1025	-1410
8	AV/C2L 65	3025	-1025	-1410
9	AV/C2L 65	3655	1025	-1410
10	AV/C2L 65	3655	-1025	-1410
11	AV/C2L 65	4285	1025	-1410
12	AV/C2L 65	4285	-1025	-1410
13	AV/C2L 55	4915	1025	-1410
14	AV/C2L 55	4915	-1025	-1410
15	AV/C2L 55	5545	1025	-1410
16	AV/C2L 55	5545	-1025	-1410
17	AV/C2L 55	6175	1025	-1410
18	AV/C2L 55	6175	-1025	-1410
19	AV/C2L 55	6755	1025	-1410
20	AV/C2L 55	6755	-1025	-1410

The rated speed of 18V32/40 diesel engine is 750r/min, the rated operating frequency is 12.5Hz. When the constraint is added, the frequency is higher than that of the free mode. Restrict reduces the block's freedom, but the overall stiffness becomes larger.

From first to sixth order, the block can be seen as a whole in bending and torsional vibration, we called global mode. In the seventh and eighth order, in addition to the local area of the bottom bending vibration in the XOY plane, other parts no longer appear large vibration. This reflects that the stiffness of the bottom place is weaker than other parts. The ninth order mode is the plane bending vibration of the whole block. The tenth order, the middle and bottom parts of the block are in different forms of vibration. In the middle of the block bending vibration occurs, and in the bottom of the block reverse bending vibration happens on the XOY plane with different amplitudes.



(a) Model 1, without constraint (b) Model 2, in constraint

Figure 4: The first mode of diesel engine block without constraint and in constraint

After constraints are added, the first order natural frequency is 99.55Hz, which is higher than that of the free mode 39.359Hz. Under the constraint conditions, the overall block degree of freedom is reduced, the whole stiffness of the block is increased, so the frequency is greater than free mode. The second order, upper part is displaced in the positive direction of the X axis, and is approximately symmetrical with the first displacement on the XOZ plane. From third to tenth order vibration diagram, although there is on obvious vibration in the whole block, the local area of the main bearing wall is displaced and vibration in the X axis. This reflects that the stiffness of these places is weaker than other parts.

Table 3: Natural frequency calculated results of diesel engine block free mode.

Order	Natural Frequency (Hz)	Order	Natural Frequency (Hz)
1	39.359	6	136.222
2	72.18	7	140.694
3	97.464	8	145.893
4	118.877	9	152.583
5	135.449	10	160.337

Table 4: Natural frequency calculated results of diesel engine block constraint mode.

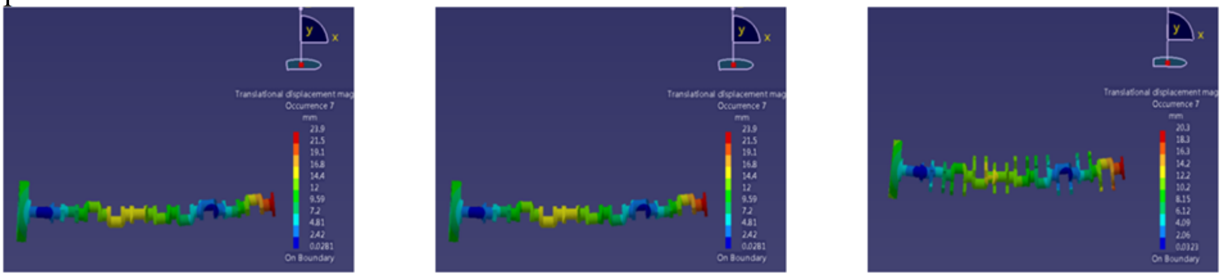
Order	Natural Frequency (Hz)	Order	Natural Frequency (Hz)
1	99.55	6	227.541
2	132.759	7	228.65
3	159.275	8	228.742
4	220.588	9	228.88
5	223.31	10	229.069

3.2 Modal Analysis of Diesel Engine Crankshaft

Crankshaft is the heart of diesel engine, belong to long shaft parts. Its characteristics are large long neck ratio, disconnected axis, irregular section shape and complex structure. The crankshaft is subjected to various forms of loads and constraints during operation. Its performance is very important to the normal operation of the diesel engine.

Figure 5 (a), (b) and (c) show the first order natural frequency and vibration mode of the crankshaft, the crankshaft with flywheel and the crankshaft with flywheel and balance weight.

The whole crankshaft is in the first and second order respectively for one order bending vibration on XOY plane and XOZ plane, with maximum amplitude reached at both ends. In the third order, the whole crankshaft two order bending vibration is on the XOZ plane, the maximum amplitude is at the two ends of the crankshaft. The fourth order, crankshaft stretching and torsional vibration in the direction of Z axis. The maximum amplitude is on the two end of the crankshaft, and the amplitude is the smallest in the middle and the two ends of the spindle neck. Two order bending and torsional vibration of the crankshaft in the XOY plane of the order fifth. The maximum amplitude of sixth to eighth order crankshaft bending torsional coupling vibration occurs on the two ends of free end face and output end face. Two order of couple the bending and torsional vibration in the Z axis direction of the ninth order mode. The maximum amplitude occurs on the two ends. The tenth order mode of the crankshaft is at Z axis stretching and bending vibration coupling. The maximum amplitude occurs at both ends.



(a) crankshaft model (b) crankshaft with flywheel model (c) crankshaft with flywheel and balance weight model

Figure 4: The first mode of diesel engine crankshaft, crankshaft with flywheel and crankshaft with flywheel and balance weight without constraint.

Table 5: Calculation results of natural frequencies of crankshaft free mode

Order	Natural frequencies of crankshaft (Hz)	Natural frequencies of crankshaft with flywheel (Hz)	Natural frequencies of crankshaft with flywheel and balance weight (Hz)
1	16.398	11.677	9.97
2	16.881	11.731	10.018
3	43.2	32.686	27.653
4	46.22	33.483	28.194
5	80.171	56.353	34.757
6	88.795	62.472	50.932
7	109.132	66.147	54.712
8	129.633	85.704	74.109
9	134.611	101.38	81.538
10	145.547	114.315	88.823

First and second order modes of the crankshaft with flywheel respectively for one order bending vibration of XOZ plane and XOY plane, the maximum amplitude is located at free end of the crankshaft. The third, fourth and fifth order modes are two order bending vibration on XOZ plane, fourth and fifth order occur axial torsional vibration also. Sixth and seventh order modes are bending torsional coupled vibration. The maximum amplitude of the sixth order is at the free end, and of the seventh on the seven crank throw. Eighth and ninth order modes are two order bending and torsional vibration of XOY plane.

The first and second order modes of crankshaft with flywheel and balance weight are one order bending vibration on XOZ plane and XOY plane. Third, fourth and fifth order modes are two order bending vibration on XOZ plane. The forth order also occurs torsional vibration in Z axis direction.

The mode of sixth order is two order bending vibration of XOY plane. The maximum amplitude is at the free end of the crankshaft. The seventh mode is the bending torsional coupling vibration, the maximum amplitude is located on the edge of the flywheel. The eighth and ninth order modes are XOY plane and XOZ plane of two order bending and torsional vibration. The tenth order mode is the coupling of bending and torsional vibration, the maximum amplitudes located at the edge of each balance weight.

4. Conclusion

This paper analyses the vibration characteristics of 18V32/40 offshore platform power station diesel engine block and crankshaft. When the block at isolator position is restrained, the overall degree of freedom reduced and stiffness increase, so the frequency is larger than the free condition. Vibration of upper parts of the block are not obvious from the fourth order mode, the vibration of main bearing wall is obvious. In the actual installation, the block subject to various constraints and forces from the outside, the vibration will be complicated.

Free modes under three conditions of crankshaft, crankshaft with flywheel and crankshaft with flywheel and balance weight are simulated. The low order vibration mode is dominated by plane bending vibration. The higher order mode is dominated by bending torsional coupled vibration. As the order increase, the frequency increases. For each order, the maximum displacement is concentrated at the free end. The free mode frequency of crankshaft is higher than that of the crankshaft with flywheel and the frequency increases rapidly. The free mode frequency of crankshaft with flywheel is greater than that of the crankshaft with flywheel and balance weight and the frequency is faster. When a similar mode of vibration occurs, compared with crankshaft the displacement of crankshaft with flywheel and balance weight is larger and the shape change is more obvious.

It is shown that the constraint can only improve the lower order vibration of the engine block, adding flywheel and balance weight just improve vibration in a certain extent. To solve the vibration problem of diesel engine block and crankshaft radically, other measures need to be taken.

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