International Journal of Smart Engineering, Volume 2, Issue 2, 2018

Vibration Characteristics and Isolation of a Diesel Engine and Electric Generator System

Qingkai Han[1]*, Shuo Han[2], Pingping Ma[1], Qingyu Zhu[1], Siqun Ma[2], Tianmin Guan[2]

1 School of Mechanical Engineering, Dalian University of Technology, Dalian, PR China

2 School of Mechanical Engineering, Dalian Jiaotong University, Dalian, PR China

Abstract: In this paper, a diesel engine and electric-generator set worked on locomotive or a diesel-generator system is investigated for its vibration problems. Its system dynamics is analyzed, obtaining the influence of stiffness and position change of machine foot vibration isolator on vibration response of generating unit. In the beginning, the dynamics of the diesel-generator set is simplified, and a multi-body dynamic simulation model of the complete system is established. The simulation results show that with the reduction of the stiffness of the machine foot, the vibration amount of the shaft end and machine feet increases, and the spectrum characteristics change more significantly than the axis does. The field vibration test is locomotiveried out at the speed of 1800r/min of the diesel-generator set. By comparing and analyzing the vibration amount of different positions, it is found that the vibration of the machine feet is 10 times higher than that of the base plate. And the frequency components in the spectrum are all related to the frequency conversion, and there are no other frequency components. It shows that the vibration isolator has a good vibration isolation effect, and the stiffness of the base plate meets the requirements. Finally, a comparative analysis of the vibration of the locomotive with excessive vibration and the normal locomotive shows that the amplitude of the transverse vibration of the locomotive is the largest.

Key words: diesel-generator set, dynamic simulation, vibration isolation analysis

* Han Qingkai. Professor, hanqingkai@dlut.edu.cn

 ISSN 2572-4975 (Print), 2572-4991 (Online)

1 Introduction

The traction power unit of the freight locomotive is a large medium-speed diesel engine. The diesel engine is connected to the main generator through a coupling. The electric energy generated by the main generator is the power source for the dispersed drive motor of each locomotiveriage. In engineering, the problem of excessive vibration of the diesel engine and the main generator often take occur, which causes the locomotive to be unable to leave the factory or to run normally.

The main factors that cause vibration in diesel engine-generator units include: crank connecting rod mass imbalance, crankshaft bending, shaft misalignment, coupling loosening, oil film oscillation, structural resonance, loose bearing housing, insufficient cover tightness and improper magnetic force, etc. Geng Zunmin et al.[1] analyzed and classified the impact excitations inside the engine, and put forward an analytic model of the non-stationary engine vibration considering its time-varying transfer properties, and then discussed in details its time domain and time-frequency domain characteristics, designing an applicable signature extracting and diagnosing approach to extract engine vibration. Simone Delvecchi et al.[2]. used several signal processing tools for monitoring and diagnosis of connecting rod with incorrectly tightened screws assembly faults in diesel engines through the cold test technology. And a method based on the image correlation of symmetrized dot patterns is proposed in order to quickly detect the faulty engines in cold tests Based on vibration theory, Wang Rui et al.[3] developed the software for the optimization of single layer vibration isolation system parameters of diesel engines, and used the method of diesel engine vibration isolation parameters comparison optimization to study vibration isolation effects of different isolators, ensuring the best optimization of vibration isolation, and achieving quick calculation and high efficiency. Gao Wenzhi et al.[4] presented a theoretical study and proposed an active vibration control scheme for controlling torsional vibration of a rotor shaft due to electromagnetic disturbances or unsteady flow in large steam turbine generator sets. Based on the theory of random vibration and system identification, MA Ren-le et al. [5] performed ambient vibration tests of a certain type of wind turbine towers, and put forward a method of coupling overall modeling of blade, hub, nacelle and tower.The numerical stimulation and tests results showed that the wind turbine towers can effectively avoid resonance, and the overall modeling showed excellent consistency with the test results. To solve the non-linear torsional vibration problem of engine and generator shaft system, Zhang Wei et al. [6] established mathematical model of the non-linear torsional vibration through lumped parameter model method, and got the non-linear differential equations of the system torsional vibration, and then multiple scales method was used to solve the equations, getting the critical speed of the shafting torsional vibration.Xiao Nengqi et al. [7] introduced the lumped parameter method into a marine diesel-electric hybrid propulsion system, calculating and analyzing the nonlinear vibration characteristics of the hybrid propulsion shaft system via the incremental harmonic balance method.Yang Zhe [8]made researches on basic support design of diesel generator set and main pipe support design, through the implementation of improved solutions, the vibration level and mechanical failure rate of the unit were effectively reduced.Kambrath et al.[9] simplified the diesel engine-generator system as a 7 inertia-6 stiffness system, and developed an electrical control solution to alleviate torsional stresses, restraining the torsional oscillation and hence improving the reliability of the system under engine transients.Sun Yuhua et al. [10]proposed a dynamic method to optimize the stiffness of the isolator for the double-layer vibration isolation system of diesel engine that must consider flexibility effect of the intermediate mass, and the effectiveness of the method was verified by experiments.

The anti-vibration measures of diesel-generator sets generally isolate the vibrations generated during the operation, usually using vibration isolators. The vibration isolator has three functions: first, isolate the vibration generated by the generator set from the outside to prevent its transmission; second, isolate the external vibration from the generator set to prevent its disturbance; third, change the frequency of the installed units and reduce them. Therefore, it is required that the vibration isolator must have a low natural frequency to isolate the high-frequency vibration, because the high-frequency vibration will cause structural damage due to fatigue, and the isolator should also have the ability to bend in an adjustable and repeatable range to absorb the impact load transmitted to the separated devices. The performance of the isolator is a very important factor in reducing the vibration of the diesel -generator set. Meanwhile, the matching relationship between multiple isolators is also very critical.

In addition to the performance of the isolator, there are several factors that affect the vibration of the diesel-generator set. If the vibration of diesel-generator set can be analyzed and predicted in the design stage, and the sensitive structural parameters and assembly process parameters that affect the vibration of the diesel-generator set can be analyzed through simulation and test tests, it is of great significance to better control the vibration of the diesel-generator set, and improve the robustness of the vibration of the generator set.

2 Dynamics system description of diesel-generator unit

2.1 Structure description

Figure 1 shows the structure of a certain type of locomotive diesel-generator sets manufactured by a locomotive factory. The diesel engine is fixed on the baseplate of the frame through six symmetrically installed vibration isolator. The generator is fixed on the frame base plate by two isolators mounted symmetrically. In order to ensure the neutrality, the output shaft of the diesel engine is connected with the generator shaft through the plate coupling and the accuracy of the coaxiality is ensured. The connection between the diesel engine housing and the generator housing is through the spigot assembly.

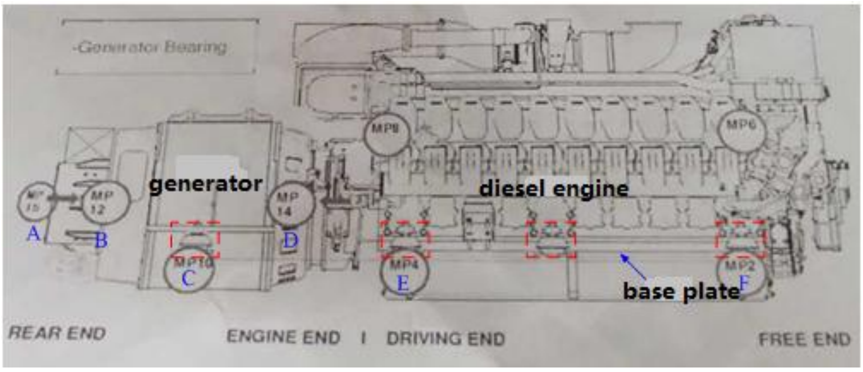


Fig.1. The structure of diesel engine and generator system

2.2 Description of rotor system of diesel engine and generator

The locomotive diesel engine is a 20-cylinder and 4-stroke engine. The working cycle consists of 4 piston strokes, namely, intake stroke, compression stroke, power stroke and exhaust stroke, and the symmetrical distribution of the piston drives the crankshaft to rotate.

The main generator is AC synchronous generator, and the generator rotor mainly includes excitation winding and magnetic pole core. One end of the generator rotor is directly connected to the output shaft of the diesel engine through the disc coupling, and the other end is supported by a double row roller bearing. The bearing housing is connected to the outer casing of the generator through the web plates.

The rotor parts of the diesel engine-generator constitutes a two-rotor, three-fulcrum system.

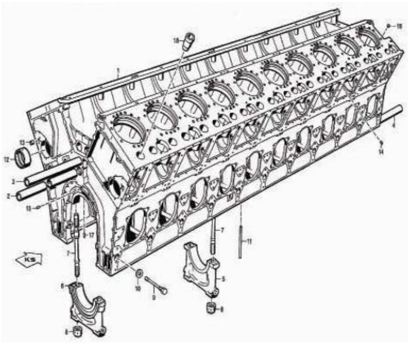


Fig.2. 4 stroke and 20 cylinder diesel engine

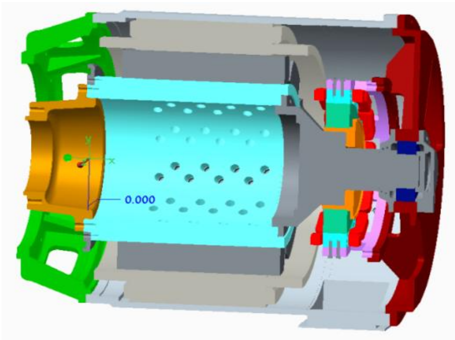


Fig.3. Internal structure of main generator

3 Vibration analysis of diesel-generator unit based on multi-body dynamics simulation

3.1 Simplification of system dynamics and establishment of simulation model

The physical model of the diesel engine shown in Fig. 1 is simplified to form a design model as shown in Fig. 4. The crankshaft structure of the diesel engine is simulated as an equivalent eccentric mass disc. The excitation winding and the pole core of the generator are simulated as two mass plates, respectively, and generator shell is simplified into cylindrical structure. In order to reveal the vibration mechanism of the diesel-generator set, and to study the influence of the parameters on the diesel-generator set, model equivalents are made according to the actual structural parameters and characteristics of the diesel- generator set. The results can reflect the mechanism and the law, and it is different from the actual vibration.

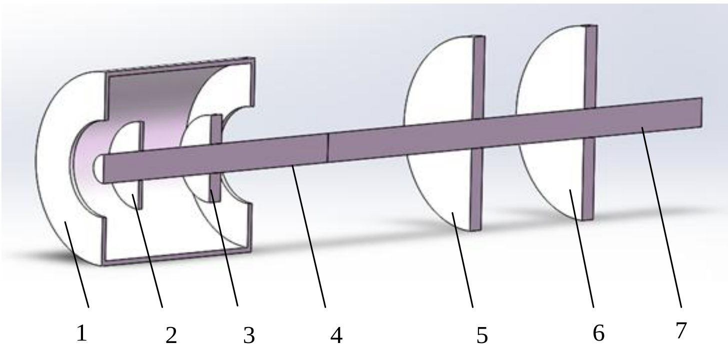


Fig.4. System structure semi profile diagram of diesel-generator set
1—generator shell 2—excitation winding equivalent disk 3—magnetic core equivalent disk 4—generator shaft 5、6—crankshaft and stroke equivalent disc 7—diesel engine shaft

Table.1. Parameters of the rotor system

No.	Name	Value	No.	Name	Value
1	Diesel engine shaft diameter	60mm	11	Casing outer diameter	400mm
2	Main generator shaft diameter	60mm	12	Casing thickness	10mm
3	Diesel engine shaft length	1000mm	13	Casing aperture	200mm
4	Shaft length of main generator	600mm	14	Imbalances	98g
5	Simulated disk diameter of magnetizing coil	200mm	15	Eccentricity of unbalanced quantity	160mm
6	Simulated disk thickness of magnetizing coil	10mm	16	Support stiffness K1 of the crankshaft No. 1	1e8
7	Analog disk diameter of excitation winding	200mm	17	Support stiffness K2 of the crankshaft No. 2	1e8
8	Simulated disk thickness of excitation winding	30mm	18	Main generator bearing stiffness K3	1e8
9	Crankshaft simulative disc diameter	400mm	19	Foot support stiffness K4 of main generator	1e7
10	The thickness of the simulated disk of the crankshaft	30mm			

ISSN 2572-4975 (Print), 2572-4991 (Online)

3.2 Setting of multi-body dynamics simulation model

The simplified system dynamics diagram is shown in Figure 5, where the X is the vertically downward direction, Y is the transverse direction, and Z is the axial direction. The vibration isolator and the supporting device are simulated as the spring units, and the wheel disc and the unbalanced mass are used to simulate the crankshaft, and mpulse excitation is simulated by adding impact force on the crankshaft. The ADAMS simulation model is set up as shown in Figure 6. In simulation, the rotating speed of the shaft is 1800r/min, and the rotating speed of the disc 1 and 2 is 900r/min.

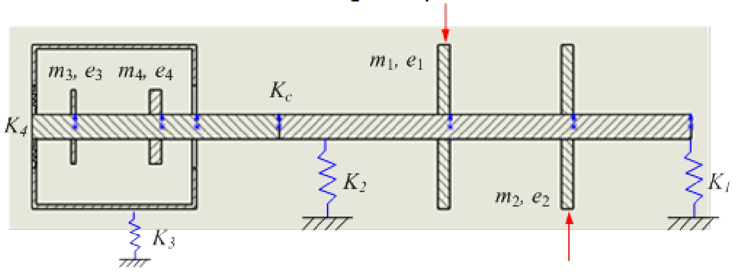
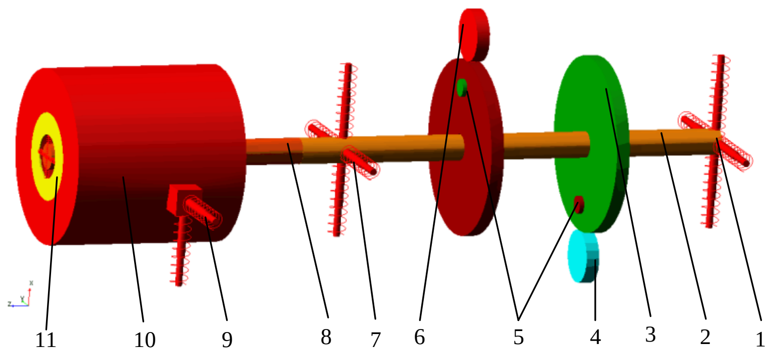


Fig.5. Schematic diagram of simulation mechanics



1-K1 2-Shaft1 3- Mass plate 1 4- Rotating wheel 1 5-Unbalance mass M1,M2 6- Rotating wheel 2
7 -K2 8-Shaft 2 9-K3 10- Generator casing 11- Plate

Fig.6. Adams simulation model diagram

3.3 Analysis of simulation results and influence of parameters

3.3.1 The influence of the vibration isolator stiffness change

The position of the generator feet are set at the center of gravity, and the rigidity of the isolator is set to 1e7N/m, 1e6N/m, and 1e5N/m respectively, and the horizontal and vertical vibration speeds the bearing end and the foot position of the generator.

(1) Stiffness of machine foot K3=1e7N/m

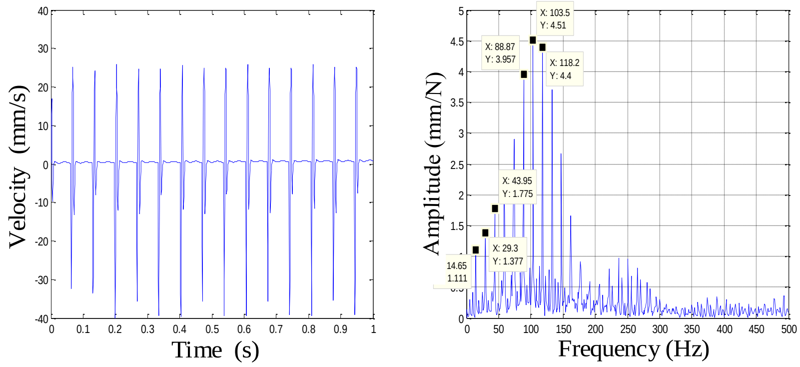


Fig.7. Time domain and spectrum diagram of vertical velocity at shaft end

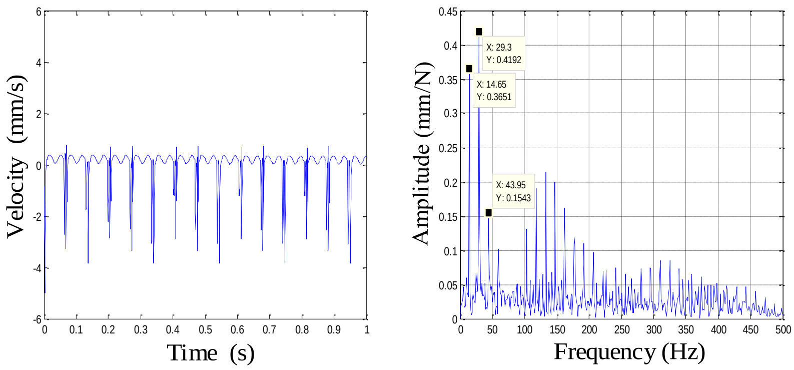


Fig.8. Time domain and spectrum diagram of transverse velocity at shaft end

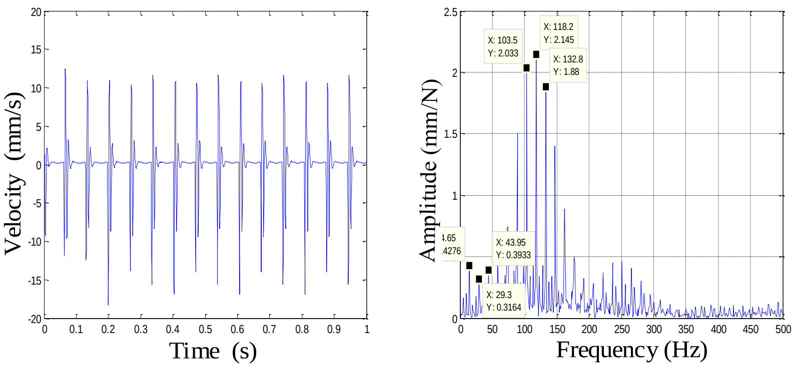


Fig.9. The time domain and spectrum diagram of the vertical velocity of the generator's foot

 ISSN 2572-4975 (Print), 2572-4991 (Online)

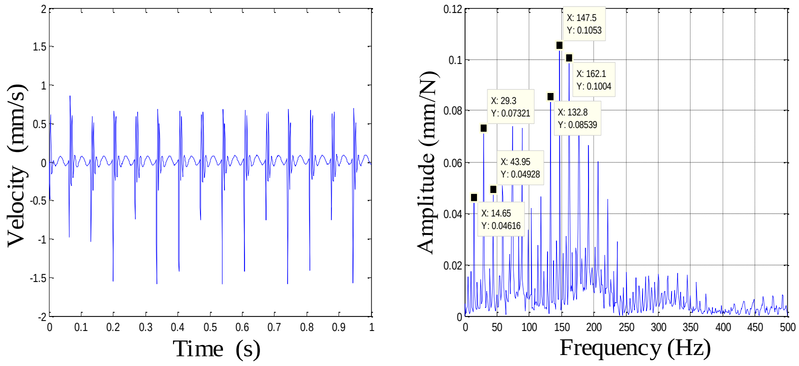


Fig.10. The time domain and spectrum diagram of the transverse velocity of the generator's foot

(2) Stiffness of machine foot K3=1e6N/m

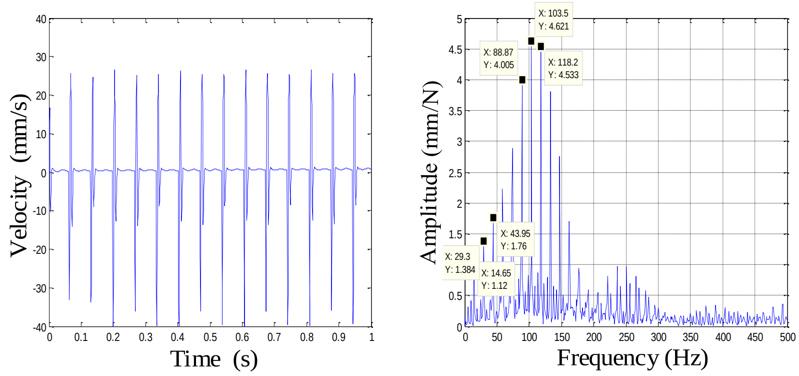


Fig.11. Time domain and spectrum diagram of vertical velocity at shaft end

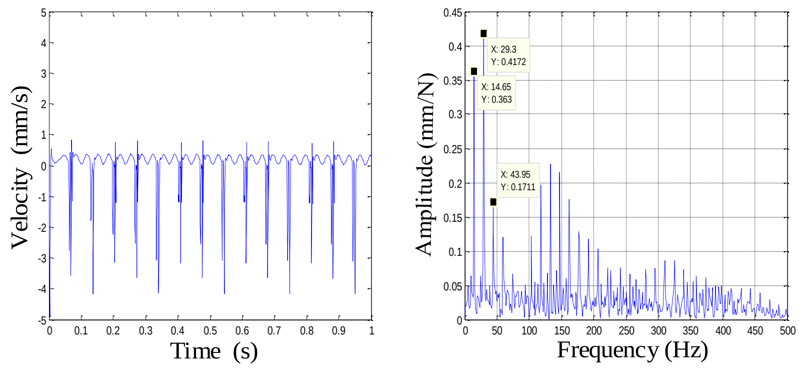


Fig.12. Time domain and spectrum diagram of transverse velocity at shaft end

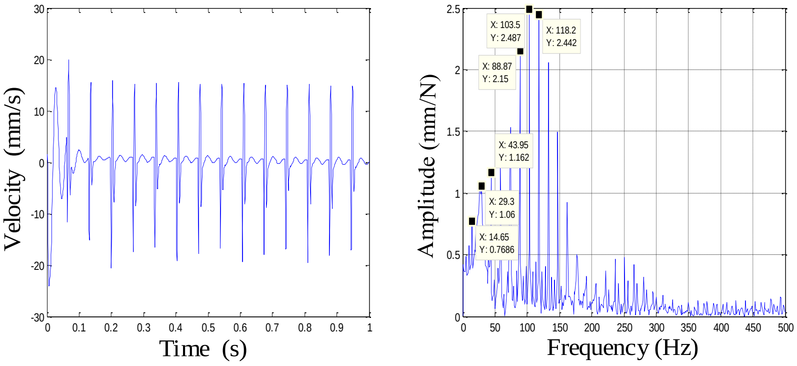


Fig.13. The time domain and spectrum diagram of the vertical velocity of the generator's foot

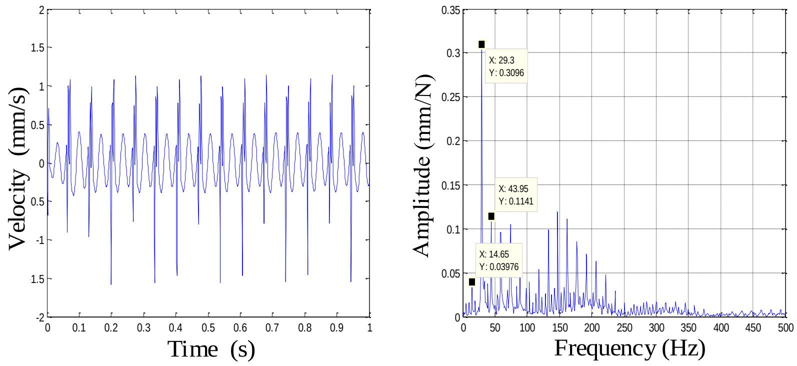


Fig.14. The time domain and spectrum diagram of the transverse velocity of the generator's foot

(3) Stiffness of machine foot K3=1e5N/m

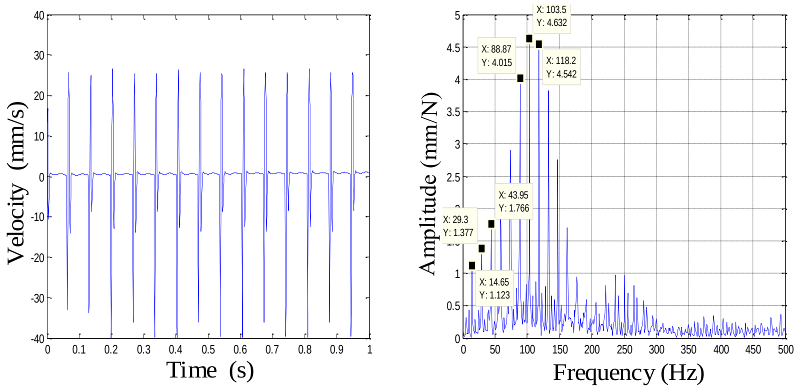


Fig.15. Time domain and spectrum diagram of vertical velocity at shaft end

ISSN 2572-4975 (Print), 2572-4991 (Online)

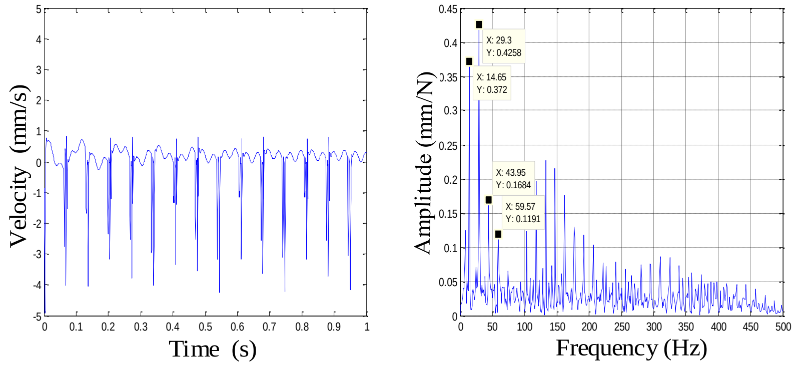


Fig.16. Time domain and spectrum diagram of transverse velocity at shaft end

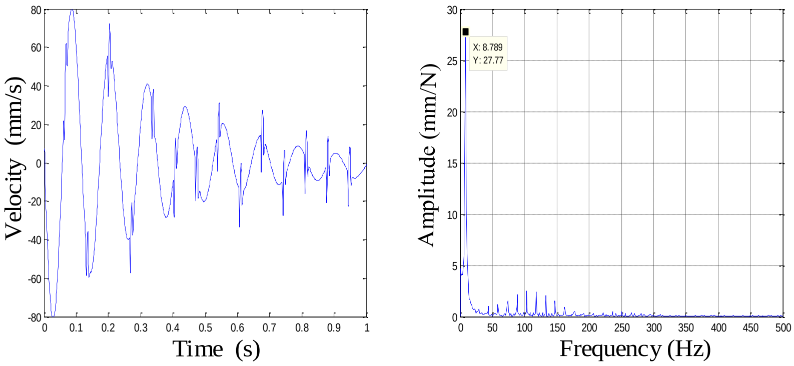


Fig.17. The time domain and spectrum diagram of the vertical velocity of the generator's foot

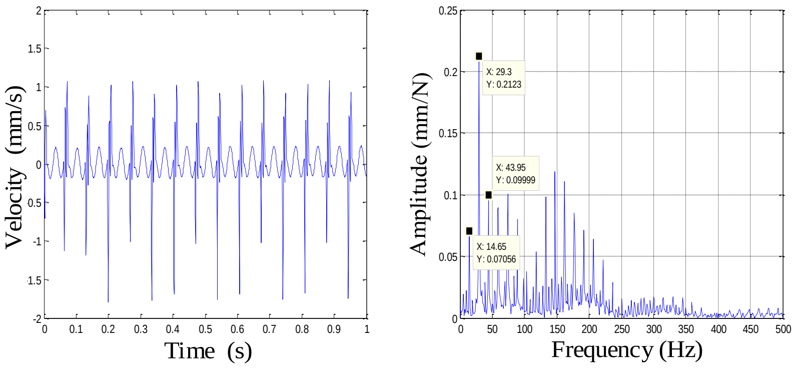


Fig.18. The time domain and spectrum diagram of the transverse velocity of the generator's foot

3.3.2 The influence of the machine foot position change

Given the supporting stiffness K3=1e7N/m of the main generator, and change the position of the feet support of the generator. Take the plane where the centroid of the generator is located as the original plane, and the offset to the generator end is positive, and the offset to the diesel engine end is negative. Given z1 = 20mm and z2 = -20mm, respectively, as shown in Figure 19. Calculate the lateral and vertical vibration speeds of the generator shaft end and the feet as shown below.

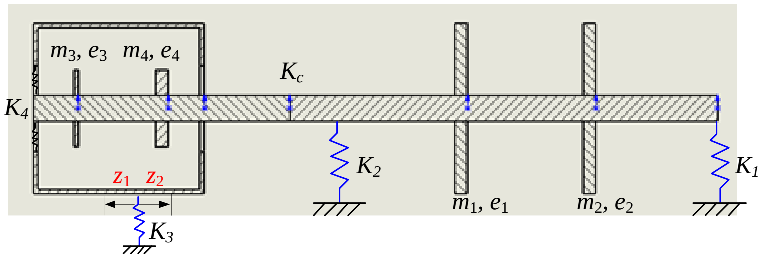


Fig.19. Schematic diagram of the machine foot position change

(1) Deviation z_1=20mm

Speed and spectrum of the shaft end and the machine foot.

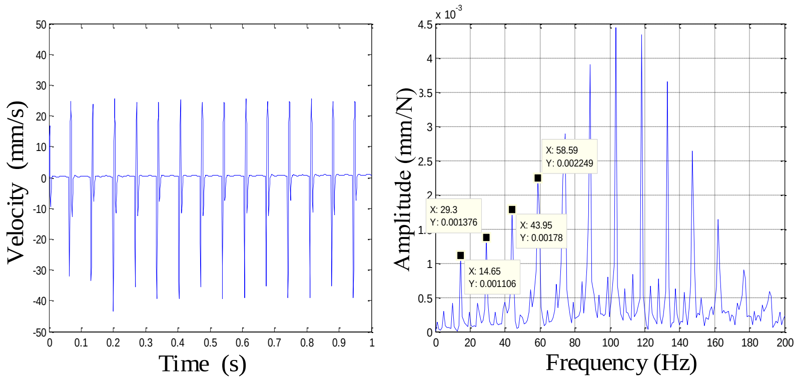


Fig.20. Time domain and spectrum diagram of vertical velocity at shaft end

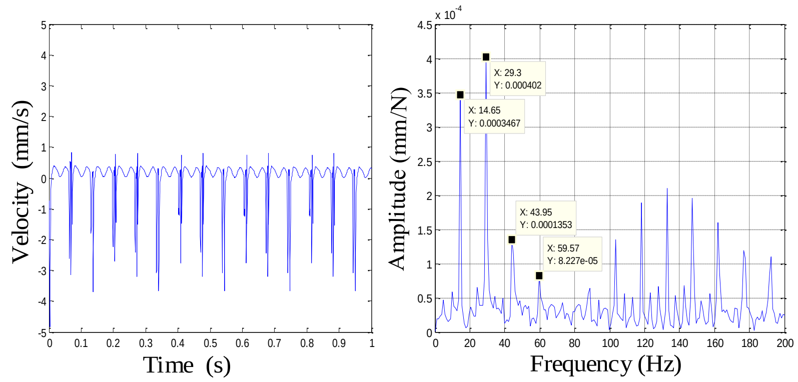


Fig.21. Time domain and spectrum diagram of transverse velocity at shaft end

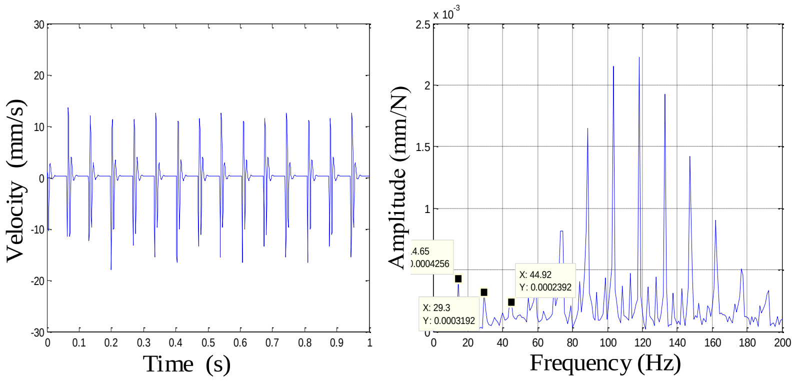


Fig.22. The time domain and spectrum diagram of the vertical velocity of the generator's foot

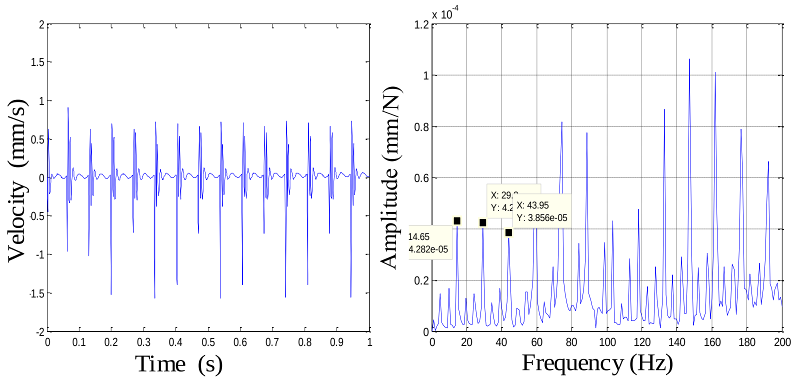


Fig.23. The time domain and spectrum diagram of the transverse velocity of the generator's foot

(2) Deviation z_2=-20mm

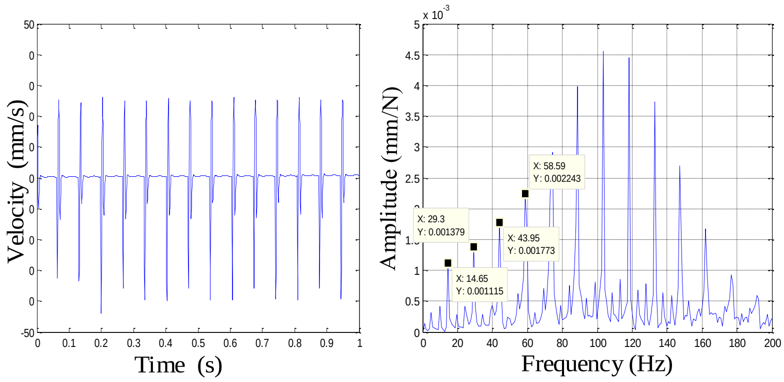


Fig.24. Time domain and spectrum diagram of vertical velocity at shaft end

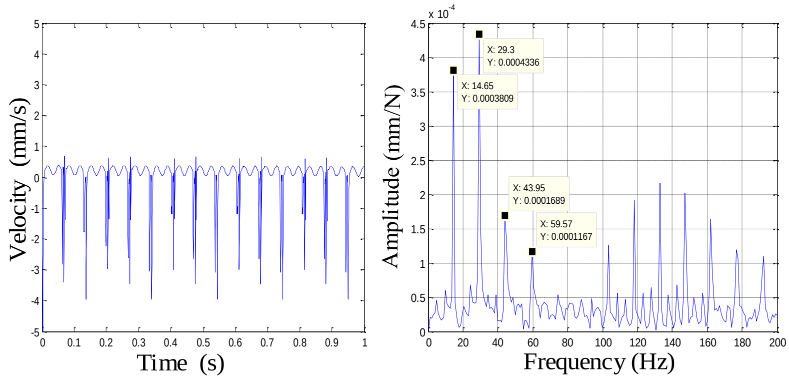


Fig.25. Time domain and spectrum diagram of transverse velocity at shaft end

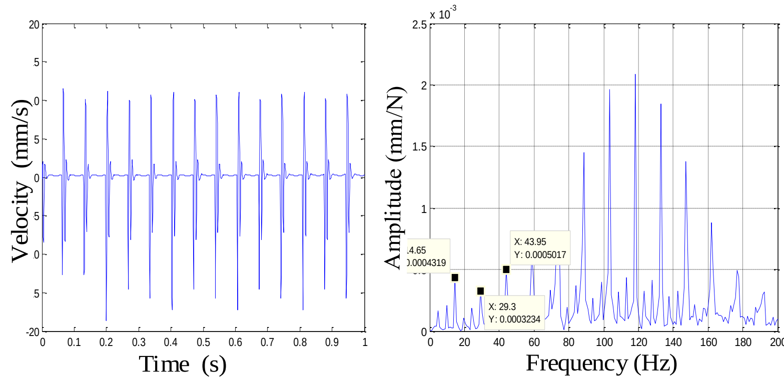


Fig.26. The time domain and spectrum diagram of the vertical velocity of the generator's foot

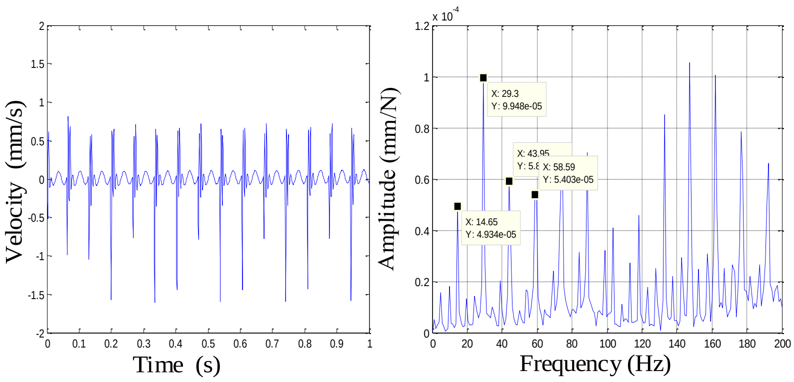


Fig.27. The time domain and spectrum diagram of the transverse velocity of the generator's foot

3.4 Analysis of simulation results

There are fundamental frequency 15Hz and 30Hz both in the spectrum of the shaft end and the feet position, and there are two combinated frequencies, such as 45Hz and 60Hz, etc.; With the reduction of the stiffness of the machine foot, the vibration amount of the shaft end and machine feet increases, and the spectrum characteristics change more significantly than the axis does; Changing the position of the feet has little effect on the vibration speed of the shaft end and the feet.

4 Field test and vibration analysis of diesel-generator set

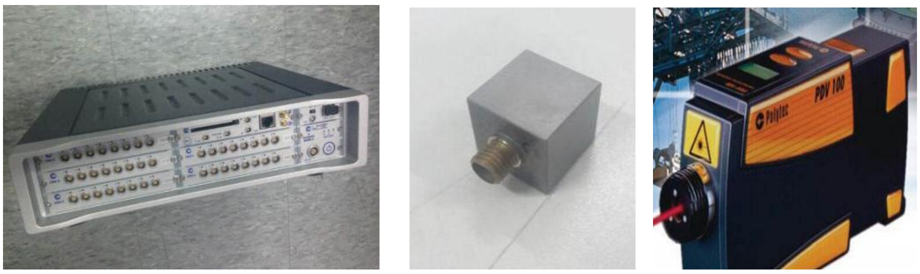
4.1 Field description of vibration test of diesel-generator set

The test was locomotiveried out under normal loading and running conditions of the diesel engine. The locomotive stopped on the track and did not run.

The main test instruments are shown in Table 2, including Siemens LMS-SCR205 data acquisition front-end with 40-channel high-performance high-precision function, PCB three-direction accelerometer and powerful LMS Test.lab signal acquisition and data analysis software. The LMS-SCR205 data acquisition front-end can satify users' requirements in data recording and management, acceleration test, modal analysis, and mechanical fault diagnosis, etc. This data acquisition front-end is also tightly integrated with LMS Test. Lab's data acquisition and analysis software, which meets the need for high-accuracy test and analysis of this vibration.

Table.2. The main instrument and quantity of the test

No.	Name	Main functional parameters	quantity
1	LMS SCR205 Front-end of number mining	40 channels, multi physical test	2
2	PCB 3713B	Three direction acceleration sensor	19
3	LMS.Test lab	Data recording and analysis	1
4	Polytech laser vibration meter	Single point long distance non-contact vibration measurement	1



(a) LMS 40 channels portable data instrument (b) PCBThree direction acceleration sensor (c) Laser Doppler acquisition vibration meter

Fig.28. Some measurement instruments

Adjust the rotating speed and load, and test the diesel-generator set. The operating speed was set 1800r/min, and the power was set 3300KW.

4.2 Measuring points arrangement

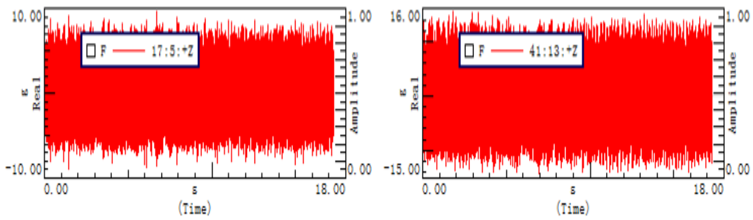
At the positions of the 6 machine feet and corresponding base plate of the diesel engine, and the 2 feet and corresponding base plate of the generator, web plate of generator end and bearing housing end cover position, measuring points are arranged.

4.3 Test data analysis

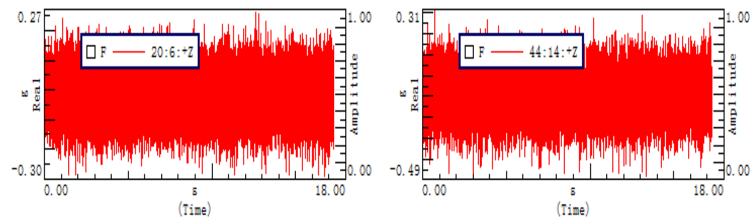
4.3.1 Vibration isolation ability of machine feet

The vibration tests of different locations are compared and analyzed. As shown in Fig. 29, the vertical and lateral vibration amount of the diesel engine feet and the corresponding base plate are compared at the rotating speed of 1800r/min.

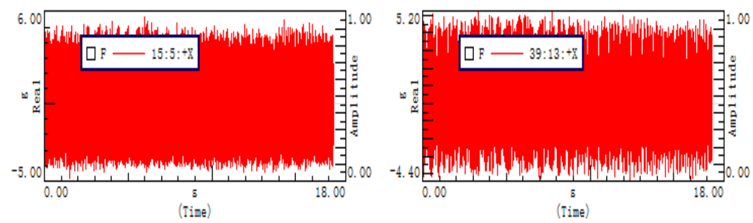
It is found that the vibration of the machine feet is 10 times higher than that of the base plate. And the frequency components in the spectrum are all related to the frequency conversion, and there are no other frequency components. It shows that the vibration isolator has a good vibration isolation effect, and the stiffness of the base plate meets the requirements.



(a) Vertical time domain diagram of the feet of eight gear diesel engines MP4 and MP3

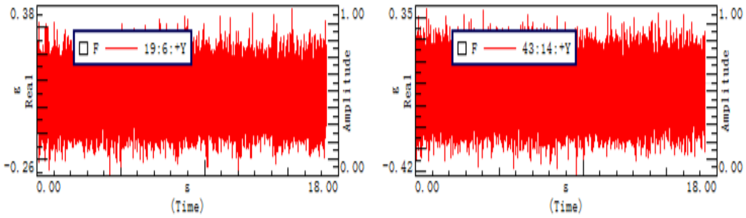


(b) Vertical time domain diagram of the floor of eight gear diesel engines MP4 and MP3



(c) Transverse time domain diagram of the feet of eight gear diesel engines MP4 and MP3

ISSN 2572-4975 (Print), 2572-4991 (Online)



(d) Transverse time domain diagram of the floor of eight gear diesel engines MP4 and MP3

Fig.29. Comparison of vibration between diesel engine foot and corresponding floor

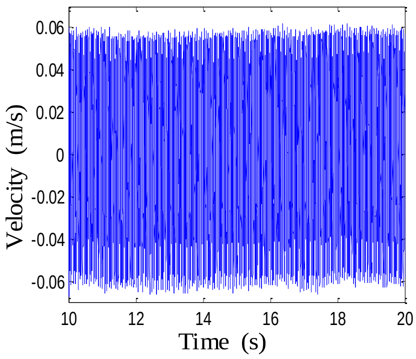
4.3.2 Vibration comparison between a vibration over standard locomotive and a normal locomotive

The acceleration signal of MP15 measuring point obtained by the test is integrated to obtain the speed signal and analyzed. Fig. 30 shows the vibration velocity time domain, and Fig. 31 shows the vibration velocity frequency domain. According to RMS value of vibration speed, the MP15 measurement point vibration speed of 4# locomotive is 35.06mm/s, and the MP15 measurement point vibration speed of 11# locomotive is 22.03mm/s. It can be found from the analysis of the spectrogram:

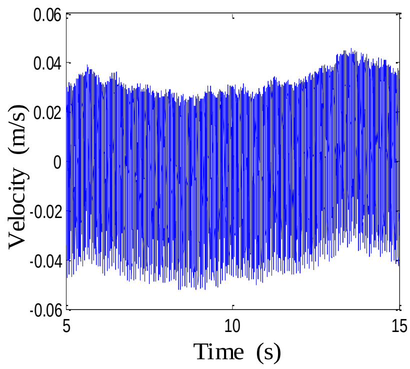
(1) The main frequency of the transverse vibration speed of the locomotive with excessive vibration is (45Hz, 42.5mm/s), which is nearly twice as that of the normal locomotive (45Hz, 22.47mm/s). (315Hz, 6.460 mm/s) is also much larger than the normal locomotive (315Hz, 1.889mm).

(2) The main frequency of the longitudinal vibration speed of the locomotive with excessive vibration is still 315Hz.

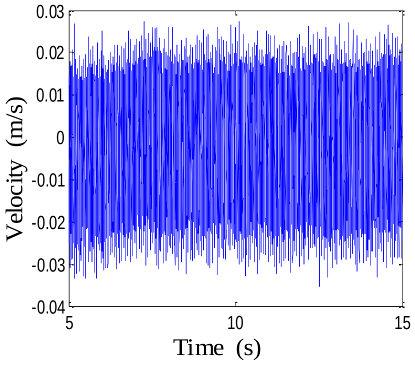
(3) The main frequency of vertical vibration speed is 45Hz.



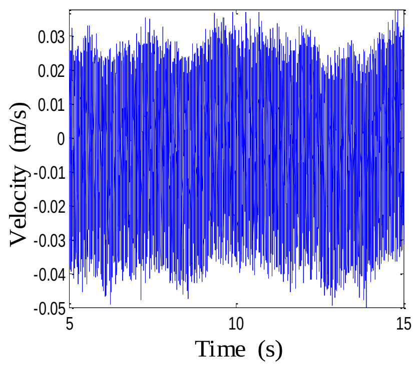
Transverse over standard locomotive (RMS: 35.06)



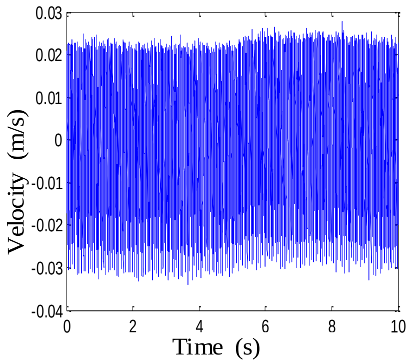
Transverse normal locomotive (RMS: 22.03)



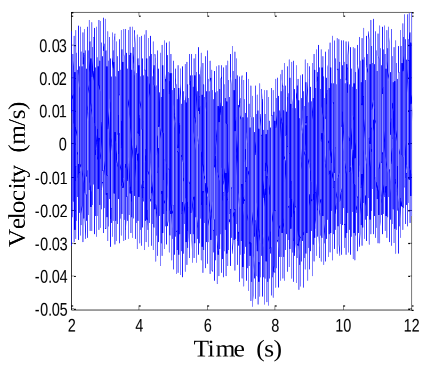
Longitudinal over standard locomotive



Longitudinal normal locomotive

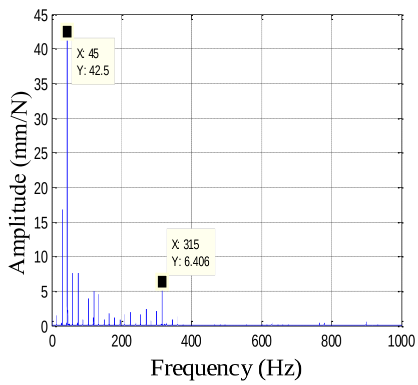


Vertical over standard locomotive

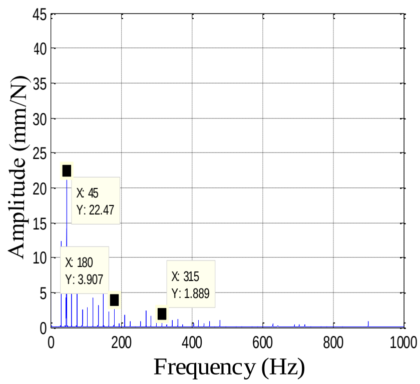


Vertical normal locomotive

Fig.30. Transverse, longitudinal and vertical Vibrational spectrum diagram and Velocity time domain diagram of MP15 measuring points for over standard locomotive and normal locomotive

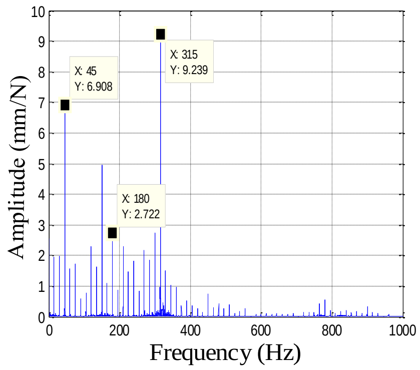


Transverse over standard locomotive

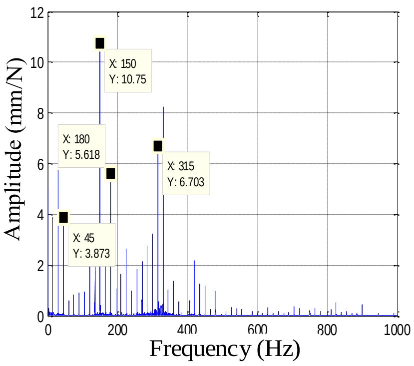


Transverse normal locomotive

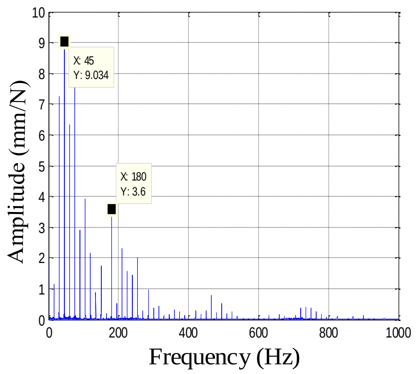
ISSN 2572-4975 (Print), 2572-4991 (Online)



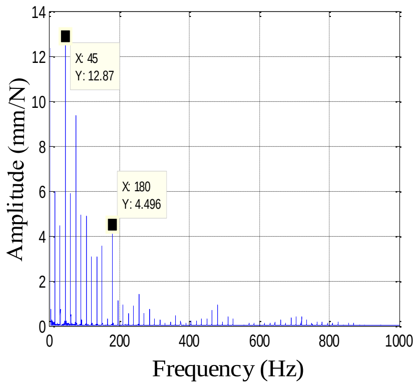
Longitudinal over standard locomotive



Longitudinal normal locomotive



Vertical over standard locomotive



Vertical normal locomotive

Fig.31. Transverse, longitudinal and vertical vibrational velocity spectrum of MP15 measuring points for over standard locomotive and normal locomotive

5 Conclusions

According to the actual structural parameters and characteristics of a certain type of diesel-generator unit, equivalent model is established. Based on the multi-body dynamics theory, the whole machine vibration analysis of this diesel-generator unit is performed via ADAMS simulation platform. And the vibration isolation effect of the vibration isolator is verified by performing live vibration test. At the end, the vibration of the locomotive beyond the standard vibration amount and that of the normal one are compared and analyzed. In this study, we mainly conclude:

(1) There are fundamental frequency 15Hz and 30Hz both in the spectrum of the shaft end and the feet position, and there are two combined frequencies, such as 45Hz and 60Hz, etc.

(2) With the reduction of the stiffness of the machine feet, the vibration amount of the shaft end and machine feet increases, and the spectrum characteristics change more

significantly than the axis does; Changing the position of the feet has little effect on the vibration speed of the shaft end and the feet.

(3) The vibration of the machine feet is 10 times higher than that of the base plate. And the frequency components in the spectrum are all related to the frequency conversion, and there are no other frequencies, indicating that the vibration isolator has a good vibration isolation effect, and the stiffness of the base plate meets the requirements.

(4) The main frequency of the transverse vibration speed of the locomotive with excessive vibration is (45Hz, 42.5mm/s), which is nearly twice as that of the normal one (45Hz, 22.47mm/s); longitudinal and vertical vibrations are not sensitive; the main frequency of the longitudinal vibration speed of the locomotive with excessive vibration is still 315Hz; the main frequency of vertical vibration speed is 45Hz.

References

1. Geng Z, Chen J, Hull J B, Analysis of engine vibration and design of an applicable diagnosing approach, ***International Journal of Mechanical Sciences***, 45.8(2003) 1391-1410.
2. Delvecchio S, D'Elia G, Mucchi E, et al., Advanced signal processing tools for the vibratory surveillance of assembly faults in diesel engine cold tests, ***Journal of Vibration and Acoustics***, 132.2 (2010) 021008.
3. Rui W, Hanhua Z, Xisheng Z, et al., Optimization of Vibration Isolation Parameters of Marine Engine Vibration Isolation System, ***Diesel Engine***, 6 (2015) 009.
4. Wenzhi G, Zhiyong H, Active control and simulation test study on torsional vibration of large turbo-generator rotor shaft, ***Mechanism and Machine Theory***, 45.9 (2010) 1326-1336.
5. R.L. Ma, Y.Q. Ma, H.Q. Liu, et al., Ambient vibration test and numerical simulation for modes of wind turbine towers, ***Journal of Vibration and Shock*** , 30.5 (2011) 152-155.
6. W. Zhang, X. Zhao, et al., Characteristics analysis of non-linear torsional vibration in engine and generator shafting system, ***Telecommunication Computing Electronics and Control***, 13.1 (2015) 41-54.
7. N. Xiao, R. Zhou, X. Xu, Vibration of diesel-electric hybrid propulsion system with nonlinear component, ***Journal of Vibration and Control***, (2018) 1077546317753010.
8. Zhe Y, Research and Improvement on Characteristics of Emergency Diesel Generating Set Mechanical Support System in Nuclear Power Plant, ***Conference Series: Materials Science and Engineering. IOP Publishing, 2017***, 1 (2017) 012074.
9. Kambrath J K, Alexander A A, Wang Y, et al., Dynamic analysis of diesel generator set under cylinder deactivation, ***Transportation Electrification Conference and Expo (ITEC), IEEE,*** (2017) 370-375.

ISSN 2572-4975 (Print), 2572-4991 (Online)

10. Y.H. Sun, D.W. Dong, B. Yan, etc., Stiffness Optimization and Experimental Analysis on Two-Stage Vibration Isolation System of Diesel Engine, ***Transactions of CSICE,*** 2 (2014) 186-191.