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THE DYNAMIC VIBRATION VALIDATION ON THE DRUM BRAKE OF CRANE BASED ON MULTI-BODY DYNAMICS & EXPERIMENT RESULT

Fengqi Wu, Haixiang Xu,Yiwen Yuan and Peng Hu

Shanghai Institute of Special Equipment Inspection & Technical Research Room 305,No.399 North Nujiang Road, Putuo District,Shanghai, 200333, China
e-mail: wufq@ssei.cn

According to the actual working condition of the drum brake experiment system of crane, Based on the multi-body system dynamics in the ADAMS environment, a vibration prototype of the drum brake of crane was developed. In this model, the virtual running environment was established according to the actual crane operation cases, which is designed by the brake working process. It is the key step to build the drum brake rigid vibration dynamics model for performance analysis of system dynamics, which is the basis for the optimization design of the drum brake of crane. The simulation results show that the response value of the braking time & the braking torque, these curves demonstrate that the rotate angular velocity has little relation with the braking torque, but it is proportional to the braking time. The vibration signal of the drum brake experiment system is sampled and analyzed. These results show that the model reflects the actual dynamics of the drum brake of crane, and also presents that some of the theoretical analysis results cannot be usually confirmed. These simulation results based on ADAMS & experiment vibration result will be applied to prototype design and evaluation, and save a lot of manpower and material resources. At the same time, the method has an advantage for dynamics analysis to simulating some dangerous movement conditions, which is hard to be replayed or simulated at the test actual working condition site. In particular, some cases cannot be recurrence in the accident analysis on brake.

1. Introduction

Brake is one of the most important safety components of lifting appliances, with the realization of the parking brake and preventing lifting from falling. It is the control device to realize the normal working of mechanism, and the protection device to ensure the safe operation of hoisting machinery. The braking performance directly affects the operation accuracy and safety of heavy machinery. According to relevant data shows, mechanical failure caused by crane parts is gradually increasing, which caused by mechanical failure due to brake is particularly prominent, the total of 22% common types of crane fault. Therefore, it is important to improve the overall reliability and stability of brake from an economic point of view.

The performance test and detection of brake is always paid much attention by the domestic and foreign manufacturing enterprises. Special brake test system for various hoisting machinery and industrial vehicles was set up by related enterprises.

However, limited by funding, venues, technical and other adverse conditions, test equipment and test method of most brake products used are relatively backward at present; it is much more especial in the aspect of the dynamic braking moment capacity of the brake test.

Lack of brake performance testing and test platform has been plaguing all manufacturing enterprises, which greatly restricts the progress and development of the enterprise.

In addition, with the development of the crane industry, brake needs to be more precise and wider measurement parameter range test bench to meet the existing requirements. At the same time, the test bench also plays a decisive role in the field of brake principle research, the stability of product inspection and safety accident analysis. In the process of new product development, it also can simulate operating condition by means of brake test bench, obtain the actual performance parameters, and ensure that the new products on the market completed the tasks with the security and stability.

Brakes serve in the complex work environment, and their performance is affected by many factors. A lot of manpower and material resources are spent on the whole experimental research, and some of the phenomenon is difficult to reproduce. But all kinds of large CAD/CAM/CAE analysis software such as Msc.adams, Patran or Marc are applied [1~3]; some research has been done for the drum brake virtual prototyping technology internally. Reference [4] established flexible multi-body dynamics model of bus drum brakes and analyzed braking self-lock, stuck and vibration phenomenon. Models established on this basis are accurate and perceptive, which can reflect the actual situation of the virtual prototype, and at the same time increasingly sophisticated conditions can better guarantee the credibility of the results. Therefore, virtual prototype provides an effective means for the study of the drum brake problem.

In this paper, on the basis of theoretical calculation of brake factor and brake moment, it focuses on exploring the method of establishing the rigid virtual prototype model with combination of MSC.ADAMS software, The virtual model of brake & its experiment system is built up by using the virtual prototype technology and being based on the contact analysis about the kinematics and dynamics of rigid body. On this basis, the multi-body dynamic simulation is carried out and compared with the experiment result such as postprocessor’s data and its vibration signal etc, which lays a good foundation for the design of drum brakes, shortening the product design cycle, improving design quality and efficiency.

2. Theoretical calculation and analysis of brake performance

2.1 Test simulation principle of brake test rig

Crane is a kind of intermittent movement machine and its working characteristics are frequent start-up and stop. Brake in the mechanism is mainly to the slowdown and the parking brake function.

Crane in the deceleration & parking process, based on the principle of energy conservation, the movement of objects in the process of consumption of mechanical energy is equal to the work done of the force acting on the object. These energy absorption of brake ultimately is sent out with the heat way.

Its expression is:

$$T_Z\theta_Z = \frac{1}{2}m_z v^2 + m_z gh = \frac{1}{2}J_Z\omega^2 \tag{1}$$

According to the theorem of moment of momentum:

$$T_Z t = J_Z\omega \tag{2}$$

Then:

$$T_Z = J_Z \frac{\omega}{t} = J_Z \dot{\omega} \qquad (3)$$

Where in:

T_Z——The brake moment

θ_Z——Brake angle

m_z——The weight of the load

v——Initial braking speed

h——Loading height variation value

t——The braking time

J_z——Moment of inertia

ω——Brake initial angular velocity

The test system simulates the braking process of the different size of brake through changing in different combinations of J and ω with Tz and t. the different size of brake was used in the different levels' crane. According to the crane design manual, the general control time of T within 3s. This is the dynamic simulation principle of crane brake inertia test system.

2.2 The realization method of brake inertia test system

（a）Brake is the measured object

（b）The energy of test system of rotating body (loading system flywheel) is for testing the energy simulation of braking process. And test condition factors also can change. And the change in value range is derived from the simulation of the actual working conditions of various cranes, which can be realized by change the combination of the system inertia moment of J and rotate angular acceleration of Ɛ.

（c）The required dynamic braking performance is obtained through testing the system, such as dynamic braking moment, braking time etc.

（d）Moment of inertia（ΣJ）of test rig is set well and the brake is released. The drive motor of test system is switched on and gradually accelerating drive. The drive motor power is turned off when the speed reaches the setting rotate speed. At the same time, the brake driving device power is turned off before the brake begins to close. The rotate speed in the whole test process can be detected by the pulse encoder and continuously collected by the computer with the same sampled time interval. The value of rotate speed $n_0, n_1, n_2, \ldots, n_{i-1}, n_i$ with the different time obtained, The calculation formula is as follows:

$$M_{ed1} = \frac{(n_0 - n_1)\sum J}{9.55t}\eta$$

$$M_{ed2} = \frac{(n_1 - n_2)\sum J}{9.55t}\eta \qquad (4)$$

$$\cdots\cdots\cdots$$

$$M_{edi} = \frac{(n_{i-1} - n_i)\sum J}{9.55t}\eta$$

Where in:

M_{edi}- the average moment of ith paragraph, N.m

n_0- the braking initial speed, revolution per minute (rpm)

n_i- the end speed of ith paragraph period, rpm

t- the sample time interval of speed, s

η- the mechanical efficiency of the transmission system.

3. First page of the manuscript Establishment of virtual prototyping

Brake and its experiment bench are built by three-dimensional solid model. Establishment of its assembly is shown in Fig.1. The main components are including the brake disc, brake plate, friction plate, inertia flywheel, compression spring rod and clearance adjusting screws.

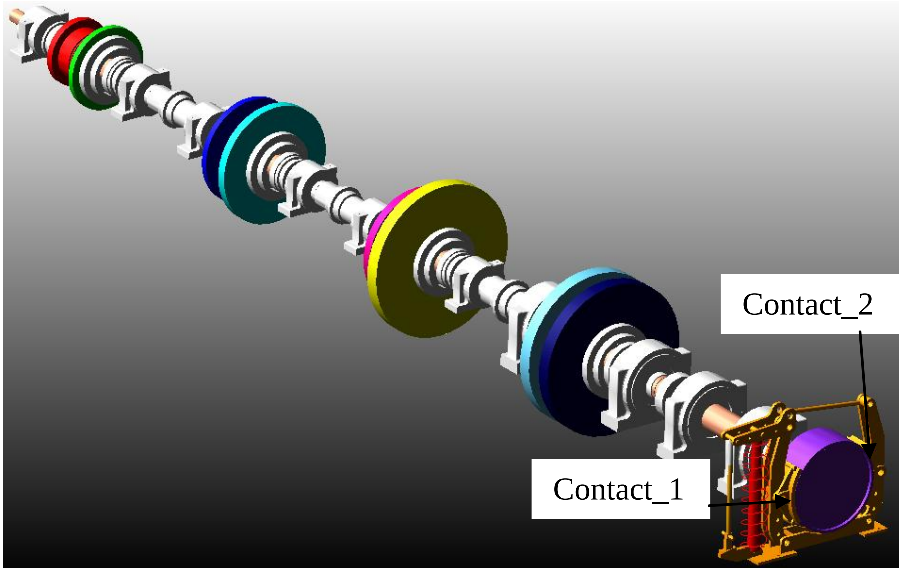

Figure 1. Three-dimensional solid model of brake test system

The Establishment of Multi-rigid body model is as follow:

The model established in this article is the brake test bed assembly, which simulates the brake dynamic model of lifting appliances. In the model, the brake rig is fixed on the ground, that is to say there is no relative motion between the brake rig and the ground; the lifting inertia is converted to inertia flywheel, and the inertia flywheel and the brake disc is fixedly connected rotate axis with a fixed hinge; Brake disc rotates relative to the brake rig, and adds the rotation pair between them; The motion between pusher and the brake rig is the translational motion and add translation pair; brake disc and friction plate are connected through contact force; the return spring is added between the base and the brake rig to achieve the automatic back.

According to equation (3) of the second part, the lifting inertia is converted to inertia flywheel, and the equivalent moment of inertia added to the single brake disc is 60.21kg m^2, and then modify the parameters of the inertial flywheel with it. The inertial flywheel centre coincides with the brake disc centre, and is connected to bolt hole of brake disc with fixed pairs. Thereby it achieves the conversion from the lifting inertia to the flywheel. Thus, multi-rigid body virtual prototype model of brake test rig is established.

Seen from Fig.2, the two curves demonstrate the rotate angular velocity varies with the braking torque, there is a semicircle shoulder at 0~0.3s of the braking torque curve, which is the effect that the rotate angular velocity increases from 0 rpm to 600 rpm. When the rotate angular velocity slows down from 600 rpm to 0 rpm, the test system begins to brake at 0.54s, stops at 1.33s, so the braking time interval is 0.8s.

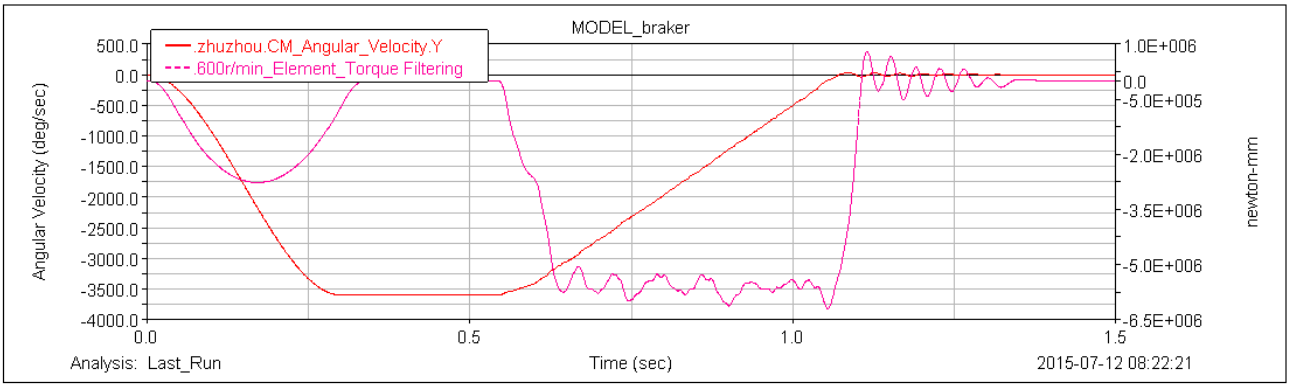

Figure 2. The rotate angular velocity changes with the braking torque

Fig.3 is shown the braking torque variation with different rotate angular velocity of 200 rpm, 300 rpm, 400 rpm, 500 rpm, 600 rpm.These curves demonstrate that the rotate angular velocity has little relation with the braking torque, but it is proportional to the braking time.

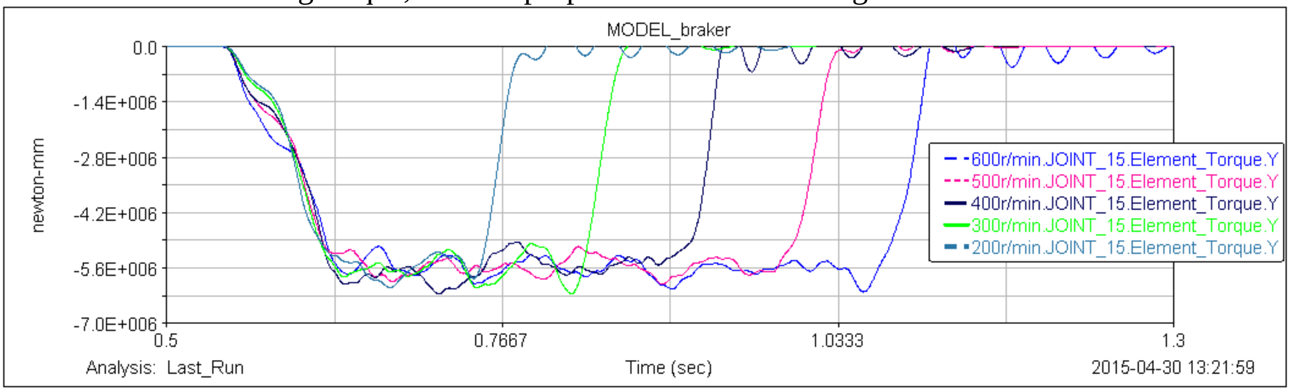

Figure 3. The braking torque variation with different rotate angular velocity

Seen from Fig.4, which shows that the contact force between the brake disc and the friction plate, the contact force of two friction plates is not same, which is due to the asynchrony mechanism movement and the different loading path of two friction plates.

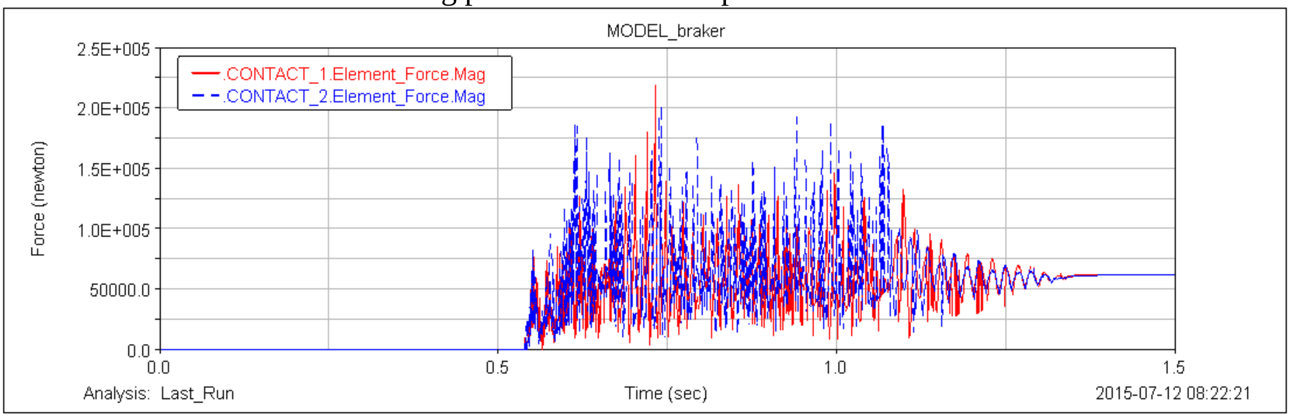

Figure 4. The contact force between the brake disc and the brake plate

As shown in Fig.5 & Fig.6, Fast Fourier Transform (FFT) of the vibration curve of brake plate contact force is implemented with hamming window filtering. The feature frequency is 8.137Hz and 7.738Hz, which is used to be compared with the feature frequency of experiment.

In the process of braking, the shaft and the braking disc deflection increases, the braking torque tries to slow down the system. Because of these effects, a variable torque is produced, the excitation can be 1X or more complex, and it will appear as torsional and radial vibration of the rotate system; torque variations will attempt to transfer rotational energy into radial vibration energy.

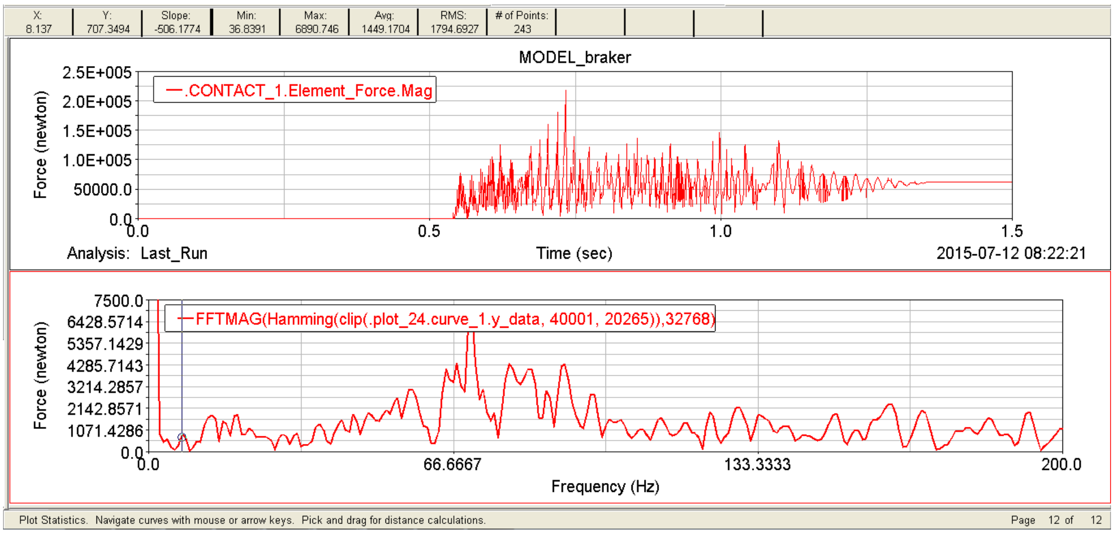

Figure 5. The vibration curve of brake plate contact force 1 and its Fast Fourier Transform

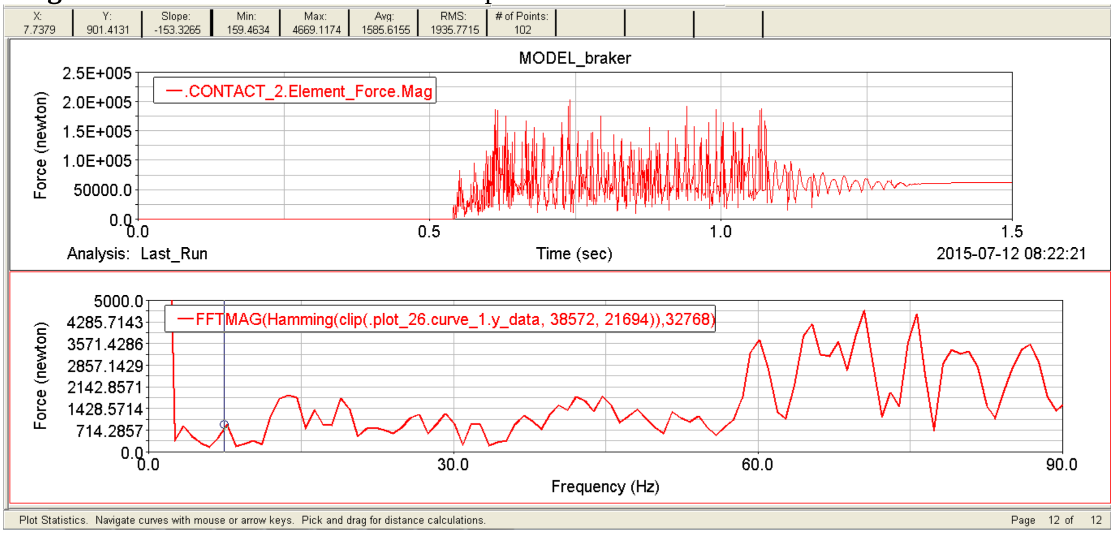

Figure 6. The vibration curve of brake plate contact force 2 and its Fast Fourier Transform

4. The Dynamic Vibration Validation

The dynamic braking torque is carried out in the brake experiment bench, the test system and measuring points are shown in Fig.7.

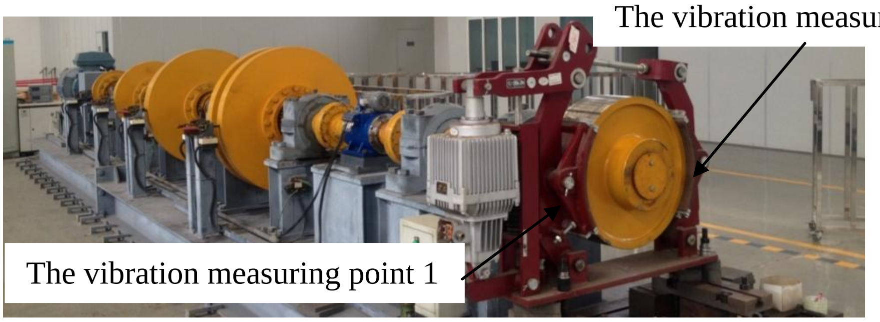

Figure 7. The test system and measuring points

Seen from Fig.8, the curve demonstrates the speed change during the braking process, the sample interval is 50ms,therefore ,the actual braking time is 0.8s when the rotate angular velocity decreases from 600 rpm to 0 rpm.The rotate angular velocity is measured by the speedometer motor. The main actual value in the curve are 600, 586, 442, 270, 91 rpm respectively. The maximum value is

600 rpm.The 1X (the same as rotor speed. The “X” is equivalent to a mathematical symbol. Thus, 1X can be read as”1 times rotor speed”) frequency of rotation axis is 10Hz. the brake torque is obtained through measurement speed change (acceleration) indirectly. The data vibration exists because the transducer is disturbed by the current magnetic field of variable-frequency drive. The average value of the experiment moment is 5867N.m. The simulated torque is basically consistent with the experiment result, which is the result of the influence factors of mechanical efficiency. It proves that the established drum brake virtual prototype model is reasonable, and that the model can truly reflect the actual braking situation.

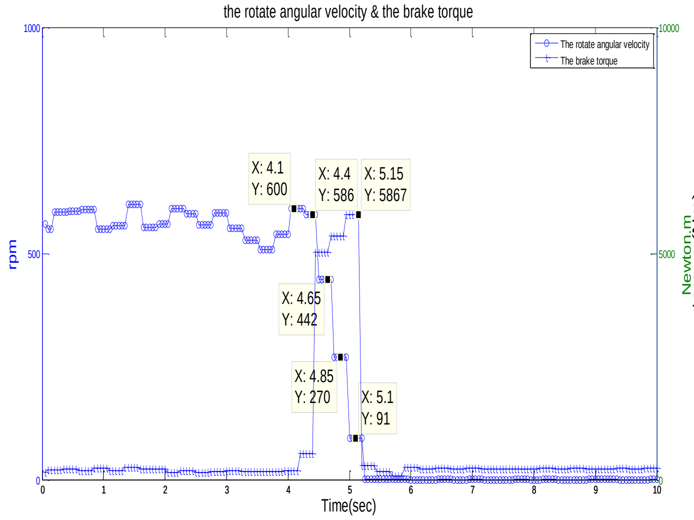

Figure 8. The speed change during the braking process

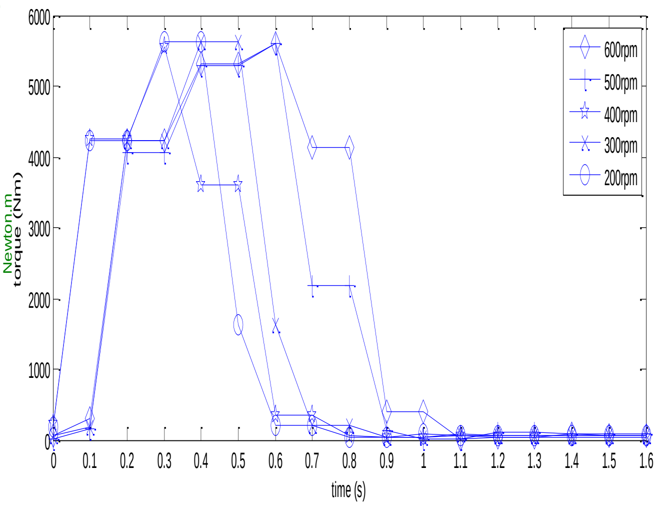

Figure 9. The brake torque with different speed

Seen from Fig.9, the brake torque with different rotate angular speed, the result is showen a similar trend with the result of Fig.3. The highest torque among different rotate initial speed is almost equal at about 5800N.m, and the faster rotate initial speed is accompanied with the longer duration of dynamic torque.

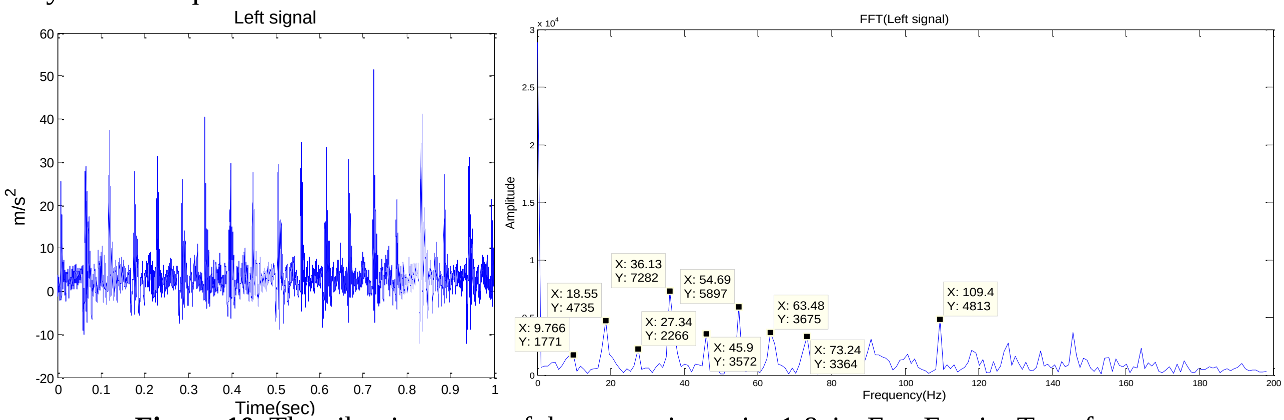

Figure 10. The vibration curve of the measuring point 1 & its Fast Fourier Transform

Fig.10 is shown the vibration curve of the measuring point 1, during the braking process ,the impact between the rotate disc and the brake happens with same interval time, which is the *obvious* phenomenon of rub vibration, the 1X frequency of vibration signal is 9.766Hz ,which is close to 10Hz,the actual rotate velocity is 9.766×60=585.96 rpm at the moment. Other distinct frequency exists in 18.55Hz(2X), 27.34Hz(3X),36.13 Hz(4X),45.9 Hz(5X),73.24 Hz(8X),82.03 Hz(9X),which are the feature extraction frequency in rub vibration.

Fig.11 is shown the vibration curve of the measuring point 2, during the braking process ,the impact between the rotate disc and the brake doesn’t happen with a distinct same interval time, which is not an obvious phenomenon of rub vibration, but the 1X frequency of is 8.799Hz,which is close to 9.766Hz and 10Hz,the actual rotate velocity is 8.799×60=527.34 rpm at the moment. Other distinct frequency exists in 18.55Hz(2X), 27.24Hz(3X), 36.12 Hz(4X),45.9 Hz(5X),73.24 Hz(8X),82.03 Hz(9X),which are the feature extraction frequency in rub vibration. The experiment result is consistent with the simulate analysis of Fig.5 & Fig.6.

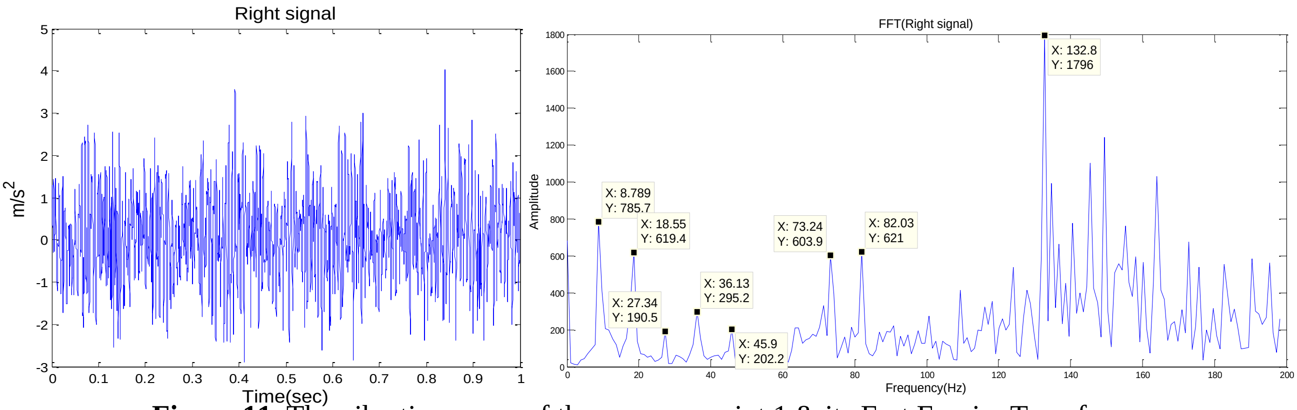

Figure 11. The vibration curve of the measure point 1 & its Fast Fourier Transform

5. Conclusion

Based on the multi-body system dynamics in the ADAMS environment, a virtual prototype of the brake test system of crane was developed. The simulated brake moment is consistent with the experiment result, which reflects the brake situation of drum brakes, and can be used for performance prediction and structural optimization of drum brakes. Virtual prototype rigid body simulation results based on ADAMS & experiment result will be applied to prototype design and evaluation, and save a lot of manpower and material resources. At the same time, the method has an advantage for dynamics analysis to simulate some dangerous movement conditions, which is hard to be replayed or simulated at the test actual working condition site. In particular, some cases cannot be recurrence in the accident brake handling process. Next work is using PATRAN software to generate brake modal neutral file, and it imports MNF file through the ADAMS/Flex interface to establish flexible body, and sets up the rigid-flexible coupling virtual prototype model by the addition of contact force.

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