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Vibration Study of Fork-lift Truck Based on the Virtual Prototype Technology

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Abstract: The forklift truck is one of important equipments of the modern logistics system. As the forklift truck is running, the driver seat and steering wheel of a certain type of fork-lift truck vibrate strongly, virtual prototyping technology and multi-body dynamics are used to make simulation of dynamic performance of fork-lift truck in this paper, and then the test result is compared with time course load that obtained from frame junction with the annex. We should repeatedly modify the simulation model based on test results, which is consistent with the actual results. Based on this model, so we put forward measures for improving design: Firstly, the axis of rotation of oval steering axle is implied; Secondly, the overhead guard is connected with the frame by the rubber cushion blocks at four different locations; Thirdly, the engine is fixed on the frame by the rubber cushion blocks (shock mount) in two different position. The improved simulation and experimental verification are carried out under the same conditions, and the results show that the fundamental frequency of seat of the improved fork-lift truck and vibration energy are lower. The result proves the practical value of this method in the research of the vibration characteristics of complete engineering machine. *Copyright © 2014 IFSA Publishing, S. L.*

Keywords: Virtual prototype, Fork-lift truck, Seat vibration, Multi-body dynamics.

1. Introduction

The fork-lift truck, which is one of important equipments of the modern logistics system, has become highly efficient equipment for mechanized loading and unloading, stacking and short distance transporting. It is widely used in stations, ports, airports, warehouses and other various departments of the national economy.

Vibration is an inevitable phenomenon of the fork-lift truck at work, and it causes noise and make driver fatigue, Besides, it reduce systematic efficiency, furthermore, it will damage the driver's health. In this case, with little or no loss of power, reducing the noise and vibration has become a hot research topic that optimizing the fork-lift truck dynamic performance, improving the reliability and the comfort of the driver's job.

The dynamic characteristic of the fork-lift truck is studied by domestic and foreign scholars with methods of multi-body dynamics, finite element method, boundary element method, experimental modal and vibration testing, and it has made a series of important advances. Wei Liangbao and other authors [1] have studied the vibration characteristics of the fork-lift truck seat and analyzed the relationship of the dynamic parameters. Yang Yi, Li Zhiyuan and other authors [2] have diagnosed the vibration type of the overhead guard with the method of modal testing. Zhao Jian and other authors [3] have analyzed the vibration of overall gantry system of fork-lift truck with modal synthesis technique. Ma

Qingfeng [4] and other authors have researched on the steering wheel vibration with CAE methods. Wang Yong and Ding Weiping [5] have carried on research of high-speed forklift driving performance. The above study has played role in guiding the current technology development, but the key components of the forklift is still confined and the overall dynamic characteristics of the forklift can not be fully reflected. It is a new attempt to analyze the entire forklift vibrations that combining of multi-body dynamics rigid-flexible coupling system with contact constraints and finite element method.

The research thinking of this thesis helps us understand the machine vibration performance of the forklift on the basis of test, using virtual prototype technology analyzes the vibration characteristics of its space motion and abstracts its various parts to a kind of rigid or flexible bodies, using various methods revise the finite element model to obtain the simulation model of consistent with experimental results, then the finite element model can be used in re-design and re-analysis of forklift structure.

2. Testing and Results Processing

2.1. Test System

Forklift machine vibration performance test system shown in Fig. 1.

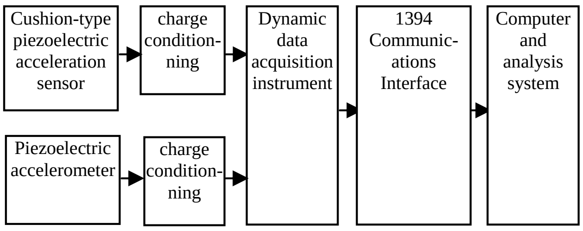


Fig. 1. Test system.

The prototype shown in Fig. 2. The measurement points are arranged at: the upper surface of driver's seat, steering wheel, driving axle body, steering bridge body, the driver's seat cover and pedal of overhead guard frame, foot pedal, the chassis of engine bracket stent, junction of overhead guard, and other areas. The test site selected concrete floor, the forklift run an effective distance of 25 m in about six seconds. The forklift test load is 0.7 times of the rated load of the standard test block, and the fork-lift truck test 8 times at a speed of 10~13 km/h. The test site is shown in Fig. 3.

In all the recorded data, select 5 data to processes [6], each data only select 25 m and 6 seconds in the effective section to process. Respectively, Fig. 4 to Fig. 6 is respectively right self-spectrum of engine seat, acceleration of seat upper surface power spectrum, acceleration of seat upper surface power spectrum in operating process and driver's seat slide rail left self-spectrum.

Fig. 2. The tested forklift.

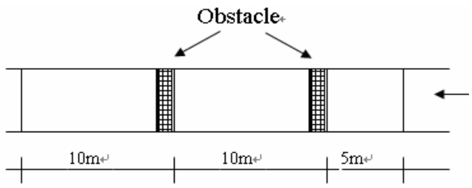


Fig. 3. Test site diagram.

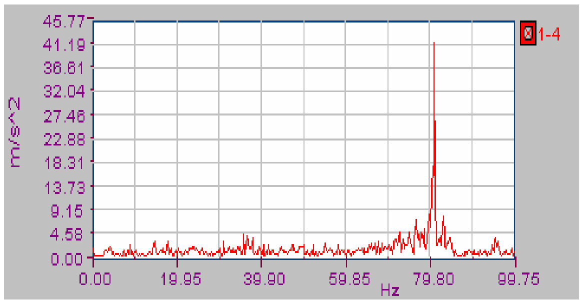


Fig. 4. Right self-spectrum of engine seat.

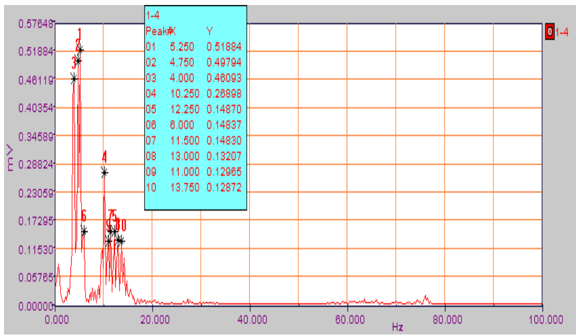


Fig. 5. Acceleration of seat upper surface power.

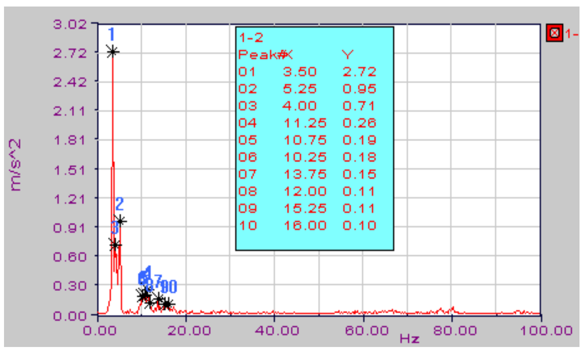


Fig. 6. Driver's seat slide rail left self-spectrum.

2.2. Result Processing

We should analyze the frequency spectrum of the measured acceleration in order to gain acceleration amplitude of different frequency. First of all, for a sample of the power spectrum, we should divides frequency band by 1/3 octave band, then obtain the vibration amplitude of 1/3 octave, and solve the weighted rms of the acceleration, values are shown in Table 1.

Table 1. Vibration acceleration of seat upper surface.

Admeasurements	The number of Test					$\overline{A}_{rsm}$
	1	2	3	4	5	
Acceleration	1.96	2.05	2.06	2.21	1.87	2.03

The validity of the test results, solve standard deviation σ for the five results [6].

$$\sigma = \sqrt{\frac{\sum_{i=1}^{M}(A_{rsmi} - \overline{A}_{rsm})^2}{M-1}} = 0.145 m/s^2 \qquad (1)$$

where A_{rsmi} is the weighted rms of the *i*th time, *M* is the measurement times, $\overline{A}_{rsmi}$ is the average of weighted rms value of *M* times.

The smaller standard deviation is, the more reliable. Deviation factor calculated by the standard deviation is not greater than 0.15 in the standard.

Its deviation coefficient is,

$$Cv = \sigma / \overline{A}_{rsm} = 0.062 < 0.15$$

The test results are effective.

3. Forklift Dynamic Model

3.1. Rigid-flexible Coupling Model Theory

Newton - Euler equations of motion for space constrained mechanical systems:

$$\begin{cases} M\ddot{r} + \boldsymbol{\Phi}_r^{\mathrm{T}}\lambda = F^A \\ J'\dot{\omega}' + \boldsymbol{\Phi}_{r'}^{\mathrm{T}}\lambda = n'^A - \tilde{\omega}' J'\omega' \\ \boldsymbol{\Phi}_r\ddot{r} + \boldsymbol{\Phi}_{\pi'}\dot{\omega}' = \eta \end{cases} \qquad (2)$$

where η is the acceleration, $\eta(q,\dot{q},t) = -\Phi_q(q,t)\ddot{q} - 2\Phi_{qt}\dot{q} - \Phi_{tt}$, Φ_q is the constrained Jacobian matrix, $q,\dot{q}$ are the system position and velocity vector, *t* is the time, F^A is the External force, n'^A is the External torque.

The differential equation of motion of the flexible body is:

$$M\ddot{\xi} + \dot{\boldsymbol{M}}\dot{\xi} - \frac{1}{2}\left[\frac{\partial \boldsymbol{M}}{\partial \xi}\dot{\xi}\right]^T \dot{\xi} + \boldsymbol{D}\dot{\xi} + \boldsymbol{K}\xi + G + \left[\frac{\partial \boldsymbol{\Psi}}{\partial \xi}\right]^T \lambda = Q \qquad (3)$$

where $\dot{\xi}$ is the time derivative of the generalized coordinates, M is the flexible body mass matrix, $\dot{M}$ is the time derivative of flexible body mass matrix, K is the generalized stiffness matrix of the structural components of the corresponding modal coordinate θ, G is the Gravity, D is the constant symmetric matrix contained the damping factor c.

Forklift dynamics model of rigid-flexible coupling system is expressed by the mixed differential equations composed by equation (2) and (3). We can get the position, velocity and acceleration in any time at any position in the system after solved the mixed equations.

3.2. Simulation Model

The model has been simplified on the premise of reflecting the main mechanical characteristics of true forklift structure as much as possible

1) Body modeling.

The solid models that contain the frame, overhead guard, front, rear, steering axle, engine, balance weight, the door frame, left and right box and bridges about 11 part solid model, are created in the three-dimensional design software SolidWorks by 1:1 ratio according to the physical vehicle size, and then imported into the multi-body dynamic simulation software ADAMS.

2) Tire Modeling.

ADAMS provides five kinds of tire model. We should select the UA tire model, and establish the tire property file (.tpf) of The ADAMS / view [7].

3) Road model.

According to the selected working condition and the field test road condition, we establish appropriate level road after modifying the parameters of road spectrum that ADAMS software carries.

4) Simplified model of drive system.

Simplify the transmission system, and retain the mass.

The forklift rigid-flexible coupling system dynamics model is shown in Fig. 7.

3.3 Dynamics Simulation

The models that importing into ADAMS, assembling with other rigid parts, the transient response that happened in the mechanical structure system in the complex dynamic loads, has been calculated by a variety of numerical calculation of Finite element method and structural dynamics.

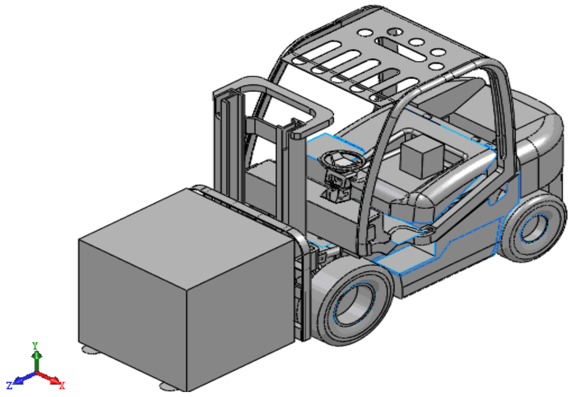
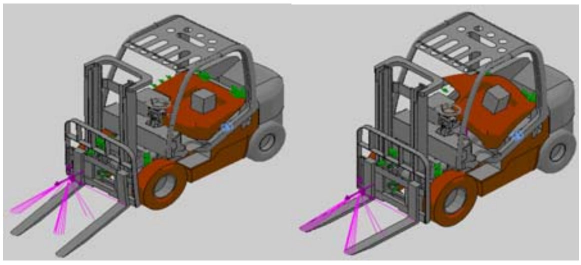


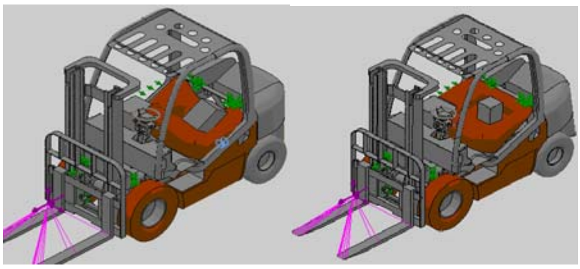
Fig. 7. Forklift rigid-flexible coupling system dynamics model.

1) Forklift vehicle system modal analysis.

The parameters of its preceding 12 order models can be calculated, shown in Table 2. The corresponding 1-4th order mode shape shown in Fig. 8.



(a) (b)



(c) (d)

Fig. 8. 1-4th order mode shape of frame.

2) Rigid-flexible coupling dynamic response of the road excitation [8].

Virtual prototype travels on the B-Class road at the speed of 10 km/h. The road random excitation is imposed in the form of the power spectrum and spectral density. According to the speed, the road power spectrum is transformed from space spectral density into the form of time spectral density, and generates road loading files.

The curves of acceleration power spectral density and acceleration vertical vibration of the seat upper surface are shown in Fig. 9 to Fig. 10.

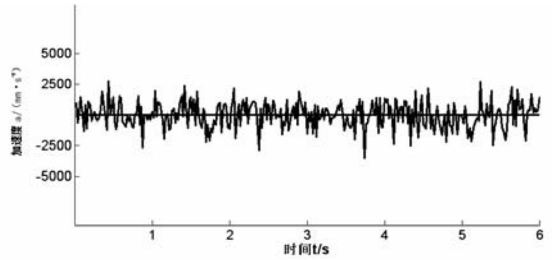


Fig. 9. The curves of acceleration of the seat upper surface vertical vibration.

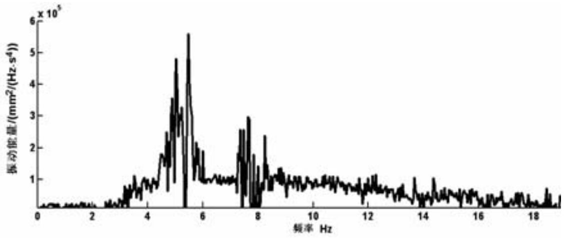


Fig. 10. Acceleration power spectral density of the seat upper surface vertical vibration.

Table 2. The whole models parameter before improved.

Order number	**Natural frequency (Hz)**	**Mode shape**
1	0.46443	Forklift seat vibrated upper and lower in the vertical direction around the pin
2	3.4483	The entire seat vibrated around the front and behind central axis
3	4.2512	Goods fork vibrated upper and lower
4	5.0813	Seat vibrated left-right
5	17.236	The Vibrated upper and lower of weight forklift caused the vibration from frame to the gantry system.
6	19.386	The entire seat vibrated around the left-right central axis
7	20.425	The entire seat twisted around the front and behind central axis
8	20.728	The vibrated left-right of overhead guard caused left-right vibration of the steering wheel
9	20.819	The swung left-right goods fork caused swing of the Steering wheel and complete vehicle, but the seat does not move
10	21.417	Forklift overhead guard twisted around front and behind central axis, and it caused the twist of box and Swing of the steering wheel
11	23.683	The swung backward and forward steering wheel caused light vibration of Overhead guard
12	27.642	The seat swung around the pre and post axis

3.4. Result Analysis

1) The experiment of the pavement driving shows that the maximum of seat amplitude frequency is 3.5 Hz. In comparison with several mode shapes, it has nearly mode of 3.4483 Hz, 4.2512 Hz, 5.0813 Hz. The shape of 3.4483 Hz has the biggest influence in the different measured points of Transfer Function peak. The influence of 4.2512 Hz, 5.0813 Hz take second place. Therefore, the vibration of 3.5 Hz may be the comprehensive between 3.4483 Hz and 5.0813 Hz in the experiment of pavement driving. Thus, the key mode should be the 3.4483 Hz order from the perspective of influence seating comfortable. The appropriate measurement of reducing vibration of the order mode should be adopted.

2) The seat in vertical direction has the resonance zone at the frequency of about 5 Hz. The vibration acceleration rms value is 2.14 m/s^2, vibration energy is 5.8×105 mm/(Hz·s4).

The analysis results of the Table 1 show that vibration acceleration rms value of the seat surface is 2.03 m/s^2. From the experimental and simulation results, it is found that the relative errors of finite element results and test results are less than 6 %, so finite element simulation model preferably reflect the main mechanical properties of the forklift structure.

The 2nd and 4th order modal shape mainly reflected the seat vibration, the size is between 4 to 8 Hz, and it belongs to the sensitive frequency of body, so it has a strong influence on the human body.

3) Forklift seat excitation is determined by the similar degree of the suffered exciting force and a certain modal [9], and the possible excitation may come from the engine and road.

Minimum modal frequency is much smaller than the engine frequency 70 Hz to 80 Hz, it shows that the seat resonance vibration is not caused by the engine excitation, because the damping pad between the engine and the frame has played a good role.

From this we can infer that the engine has little effect on frame vibration, and forklift vibration is caused by road excitation, so we can take measures to improve it.

3.5. Analysis of Body Dynamic Characteristics Caused by Road Roughness

Forklift drives on the road that its roughness wavelength is λ at a speed of v, and the time frequency is [10],

$$f = v/(3.6\times\lambda) \quad (4)$$

If the forklift inherent frequency consistent with the time frequency, it will cause resonance.

The incentive of road roughness is related to running speed. A variety of road roughness wavelengths shown in Table 3.

Table 3. Different road roughness wavelength.

Pavement	No Pavement Road	Gravel	Washboard Road	Flat road
Pavement wavelength	0.7~2.5	0.32~6.5	0.74~5.6	1~6.3

The highest excitation frequency of the road may be solved at about speed of 10~13 km/h, if we take the road minimum roughness wavelength 1 m

$$f = v/(3.6\lambda) = 13/(3.6\times1) = 3.61Hz$$

The lowest excitation frequency may generate by the road:

$$f = v/(3.6\lambda) = 10/(3.6\times6) = 0.46Hz$$

The highest excitation frequency closely coincides with the 2nd order frequency 3.4483 Hz (See Table 2). The lowest excitation frequency is close to baseband 0.46443 Hz (See Table 2).

4. Structural Improvement and Experimental Verification of the Forklift Truck System

4.1. Structural Improvements

Corresponding improvement is made according to the above analysis.

1) Steering axle adopts a flexible hinge that shaft is oval-shaped and the rubber pad is in the bearing, and flexible rubber gaskets are added in the corresponding structure junction

2) Dual suspension system is used in overall engine and cockpit.

3) The instrument front panel connects the overhead guard lift and right framework, the rear beam installs the engine cover, then the stiffness of them should be strengthened and the plate thickness increased, so the rigidity of the bottom of the overhead guard is improved and certain parts reduced.

4) When steering wheel in place, it can fix firmly with instrument panel bracket and cause synchronous vibration with overhead guard.

5) Reelect suspension seat with damping performance.

6) Stopper between steering axle and vehicle frame should be selected the rubber elastic material, and the contact clearance of that position should be reduced, steering axle swinging scope should be guaranteed by elastic deformation of the rubber pad.

7) Stiffeners was set on the roots of engine mounting belongs to the frame, it improve the torsional vibration stiffness.

4.2. Experimental Verification After Improved

The improved vehicle system was simulated at the same conditions, and the results shown in Table 4.

Table 4. The whole models parameter after improved.

Order number	Natural frequency (Hz)	Mode shape
1	9.7003	Goods fork vibrated upper and lower
2	10.086	The entire seat vibrated around the front and behind central axis
3	15.052	Seat vibrated left-right
4	19.421	The Vibrated upper and lower of weight forklift caused the vibration from frame to the gantry system.
5	20.315	The entire seat vibrated around the left-right central axis
6	20.724	The vibrated left-right of overhead guard caused left-right vibration of the steering wheel
7	20.819	The swung left-right goods fork caused swing of the Steering wheel and complete vehicle, but the seat does not move
8	21.373	Forklift overhead guard twisted around front and behind central axis, and it caused the twist of box and Swing of the steering wheel
9	21.547	The entire seat twisted around front and behind central axis
10	23.683	The swung backward and forward steering wheel caused light vibration of Overhead guard
11	30.791	The seat swung around the front and behind axis
12	34.634	Shelf fence twisted left-right

4.3. Vehicle Performance was Compared After Structural Improved

It can be seen from Table 1 and Fig. 11 that simulation results show that its fundamental frequency and seat vibration frequency are 9.73 Hz and 10.09 Hz after the structural modifications of the forklift truck system, and they are all greater than the sensitive frequency of body, vibration energy reduced from $5.8\times10^5\,\mathrm{mm}/(\mathrm{Hz}\cdot\mathrm{s}^4)$ to $4.9\times10^5\,\mathrm{mm}/(\mathrm{Hz}\cdot\mathrm{s}^4)$.

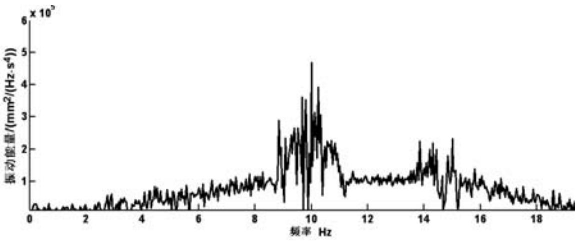


Fig. 11. Curve of acceleration power spectral density after structure improved.

5. Conclusion

On the research of forklift system dynamic characteristics, simulation calculation and experiment verification are shown on the paper, through testing and judging the motivation status and establishing the corresponding dynamic model. Conclusions are drawn as follows:

A certain type of forklift seat vibrated quite strongly upper and lower. This forced vibration is caused by the road motivation.

After the forklift structure modified, simulation results display fundamental frequency and seat vibration frequency more than one that arouses body sensitive frequency region, and vibration energy become smaller, parts improved accurately and measures suitable are showed, after modified structure of anti-vibration performance is improved.

Acknowledgments

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