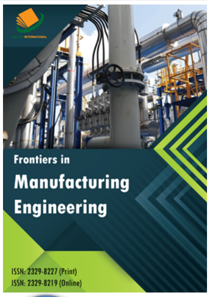
ZIBELINE INTERNATIONAL PUBLISHING

Frontiers in Manufacturing Engineering (FME)

DOI : http://doi.org/10.26480/fme.01.2018.11.14

ISSN: 2329-8227 (Print)
ISSN: 2329-8219 (Online)

CODEN : FMERA8



CrossMark

RESEARCH ON THE SHOCK ABSORBER PERFORMANCE FOR THE HYDRO-PNEUMATIC SUSPENSION IN CRANE

Shicai Feng

School of Mechanical Engineering, Changshu Institute of Technology, Changshu 215500, China
Corresponding Author E-mail: fengshc@cslg.cn

ARTICLE DETAILS

Article History:

Received 12 November 2017
Accepted 10 December 2017
Available online 06 January 2018

ABSTRACT

The performance of suspension directly affects travel property, safety, comfort and maximum velocity of vehicles. This is much more important for the special vehicles such as crane. In this paper, theoretical analysis have been performed for both the structure and the performance of hydro-pneumatic suspension system in crane. A nonlinear mathematical model for it is developed in which the oil compressibility is considered. With the language of MATLAB, Computer simulation has been used .With the full road experimental equipment produced by Company SCHENCK of Germany, the test has been done .The comparison between the simulative results and experimental results has proved that the nonlinear mathematical model developed and the program are believable .So some innovative conclusions have been obtained .This is beneficial to the design and modification of crane and its suspension .In addition ,many other related systems also can draw lessons from this research in the future.

KEYWORDS

Hydro-pneumatic suspension, absorbing shocking performance, nonlinear mathematic model, computer simulation, experiment

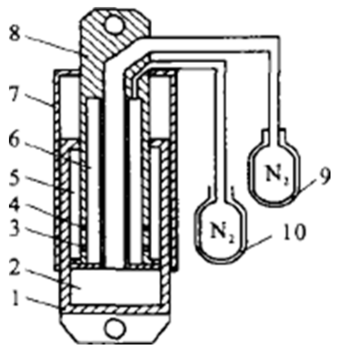
1. INTRODUCTION

For the last few decades, studies on the reduction of noise, vibration and harshness have been continued in automotive engineering. In order to reduce the interior noise and vibration of a vehicle body, the suspension system, which plays a main role in the riding safety, comfort, maximum velocity and stability of a vehicle, has attracted many engineer's attentions since it also transmits mechanical and acoustic vibration from the brake system or uneven road surface to the vehicle body. In general, we use leaf spring, rubber and hydraulic shock absorber as main elements of suspension, but their performance can't adapted to the request of crane traveling in uneven road surface. In contrary, the hydro-pneumatic suspension can [1].

Based on a study, hydro-pneumatic suspension is an advanced suspension [2]. In this paper, the complex nonlinear mathematical model is developed according to the specific structure and the oil flowing in the hydro-pneumatic suspension. The model is suitable for the analysis and parameter design of the products. The computer simulation results of the mathematical models are validated by experiments. Based on the computer simulation and experiment, the performance will be studied such as nonlinear stiffness, nonlinear damping and shock absorbing. In addition, the oil compressibility is considered and to verify the accuracy of the model, a quarter car test is made for the suspension system with a sine signal input which represents a typical road profile.

2. THE STRUCTURE OF HYDRO-PNEUMATIC SUSPENSION

Hydro-pneumatic suspension is mainly consist of suspension cylinder, accumulator and hydraulic control components. Its structure is shown in Figure 1. Suspension cylinder, which is a key component and comprises cylinder and piston assembly, can support the weigh and adjust the hight of vehicle body. In suspension cylinder, four damping pores and four one-way valves are set up skillfully. Accumulator, which is the elastic component and whose type is air chamber charged with nitrogen gas, can deposit and release energy by the elastic deformation of closed air and by using of air compressibility with help of the mediator of oil. When hydraulic cylinder 1 and piston 8 are compressed relatively, the air, which is in accumulator 10 linking inside ring chamber, would expand and make the oil flow from inside ring chamber 6 into outside ring chamber 5 through the one-way valve 3 and damping pore 4. In the same time, the oil in hydraulic chamber 2 would be compressed into the accumulator 9 linking the hydraulic 2 and the air in the accumulator 9 are compressed also. On this condition, the suspension plays elastic role in the vehicle. On the other hand, when the hydraulic cylinder 1 and the piston 8 are tensed relatively, the oil in outside ring chamber 5 would flow from outside ring chamber 5 to the accumulator 10 linking the inside ring chamber 6 through the damping pore 4. In the same time, the air in the accumulator 9 linking hydraulic chamber 2 would flow into hydraulic chamber 2. On this condition, the suspension plays damping role in the vehicle.



1. Hydraulic cylinder, 2. hydraulic chamber, 3.one-way valve, 4.damping pore, 5.outside ring chamber, 6.inside ring chamber, 7. outer cover, 8.piston, 9.accumulator, 10. aaccumulator

Figure 1: Structure of hydro-pneumatic suspension.

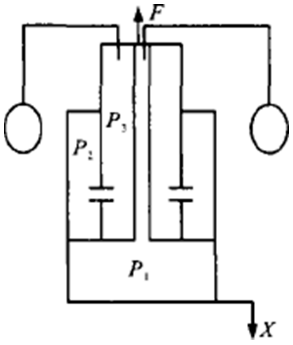


Figure 2: Physical model of the hydro-pneumatic suspension.

3. PHYSICAL MODEL OF THE HYDRO-PNEUMATIC SUSPENSION

According to a research, it is the first step to develop the physical model of the hydro-pneumatic suspension to be studied and the model should describe its main characteristics [3]. So it is necessary to grasp the main contradiction and abstractly simple the complex vehicle to a physical model of a quarter vehicle shown in Figure 2 which is consistent with its structure shown in Figure

1. In Figure 2, F is force acting on the top of the piston. F is "+" when the top of the piston is tensed and is "- " when compressed. X is displacement of suspension cylinder. When the position of suspension cylinder is under the balance position of vehicle, X is "+", otherwise X is "-".

4. NONLINEAR MATHEMATICAL MODEL

4.1 Development of the Mathematical Model

Assuming that the Coulob,s friction is relatively small and the velocity is "+" if its direction is downwards and the velocity is "-" if its direction is upwards. The mathematical model can be developed by applying some physical laws into the physical model shown in Figure 2 [4].

The force F can be described in the following:

$$F = p_2 A_2 - p_1 A_1 \tag{1}$$

Where A_1— Area of the hydraulic cylinder

A_2— Area of the outside ring chamber

p_1 — Pressure of the oil in hydraulic chamber

p_2 — Pressure of the oil in outside ring chamber

According to the theory of thin-walled pore, there are [5]:

$$p_2 - p_3 = \frac{\rho}{2c_d^2}\left[\frac{Q}{A_{01} + (0.5 - 0.5 sign\,\dot{x}) A_{02}}\right]^2 sign(\dot{x}) \tag{2}$$

Where p_3— Pressure of the oil in inside ring chamber

ρ — Hydraulic fluid density

c_d— Discharge coefficient of damping pore

Q — Flow rate of damping pore and one-way valve

A_{01}— Area of damping pore

A_{02}— Area of one-way valve

$\dot{x}$ — Velocity of the cylinder relative to the rod with time

Considering the oil compressibility, there are:

$$Q = A_2 \dot{x} - \frac{V_{20} dp_2}{\beta_e dt} - \frac{V_{30} dp_3}{\beta_e dt} \tag{3}$$

Where V_{20} — Initial volume of the oil in outside ring chamber

V_{30} —Initial volume of the oil in inside ring chamber

β_e— Bulk modulus of the oil

An adiabatic behavior is assumed for the accumulator, which yields the equation as following[6]:

$$pV^r = constant \tag{4}$$

Where p — Pressure of the air in accumulator

V — Volume of the air in accumulator

r — Adiabatic exponent of the air in accumulator

4.2 Test and Simulation

The experiment is carried out in a testing bench made in SCHENCK, Germany. The experiment object is CXP1032 crane produced in Germany. The experiment signal is as following:

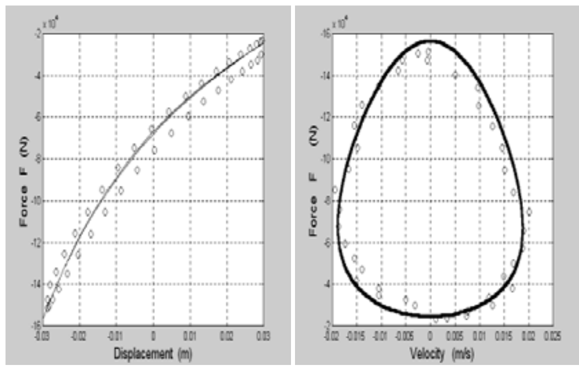
$$x(t) = x_{max}\sin(2\pi ft)$$

$$\dot{x}(t) = v(t) = 2\pi f x_{max}\cos(2\pi ft)$$

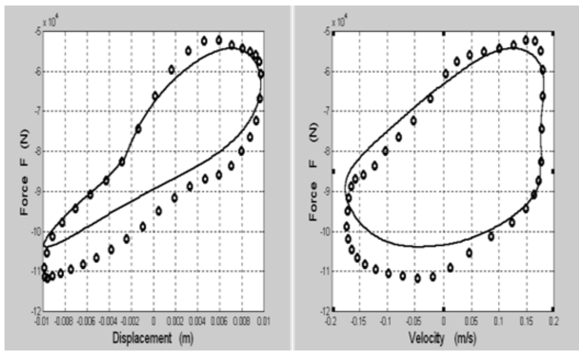
Where f — Frequency of input signal x_{max} — Magnitude of input displacement

The computer simulation software is developed using MATLAB/ SIMULINK5.0, which can solve the problem of nonlinear differential equations.

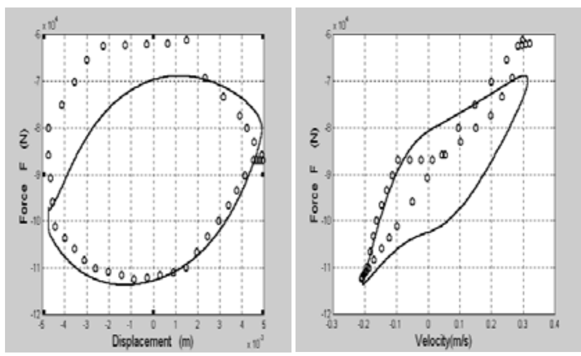
Figure 3 is the comparative diagram between the results of simulation and experiment. It needs say that the velocity in Figure 3 is the result of calculation, but the displacement is the results of experiment. Figure 3 show that the result of simulation and experiment are basically identical. The conclusion is that the nonlinear mathematical model developed above is basically right.



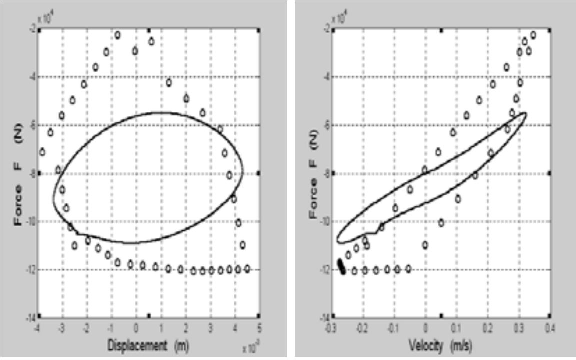
f=0.1 H, Xmax=30 mm



f=3 Hz, Xmax=10 mm



f=9 Hz, Xmax=10 mm



f=15 Hz, Xmax=5 mm (—Experiment, —Simulation)

Figure 3: Force—displacement and force—velocity diagram

5. NONLINEAR CHARACTERISTICS OF STIFFNESS AND DAMPING

Force *F* in Equation (1) comprises elastic force $F1$ and damping force $F2$. Force $F1$ is related to displacement *x(t)* and force $F2$ is related to velocity *v(t)*. With the mathematical model and the computer simulation program [9, 10], the performance can be predicated numerically. When the frequency of the input signal is high or the magnitude of the input displacement is large, it is difficult to carry out experiments. But computer simulation is easy. Figure 4 shows relations of $F1$—*x(t)* and $F2$—*v(t)*. Figure 5 shows coefficient of stiffness and damping.

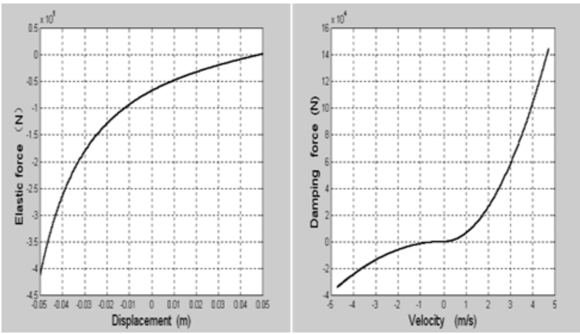


Figure 4: Curve of F1—x(t) and F2—v(t).

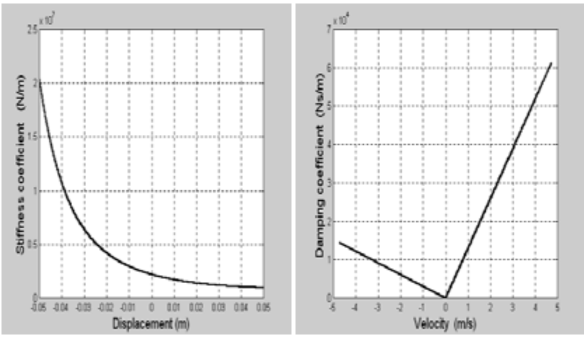


Figure 5: Coefficient curve of stiffness and damping.

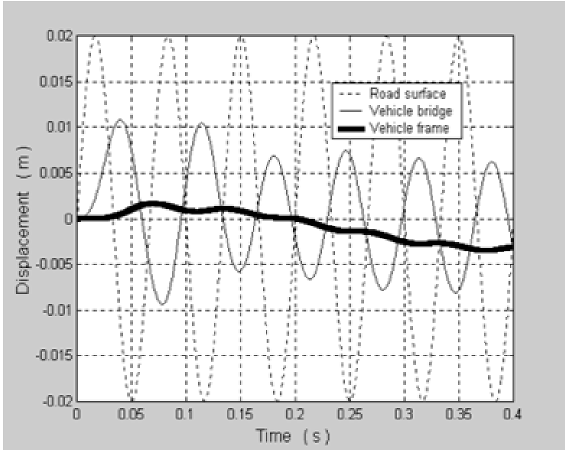


Figure 6: Displacement curve.

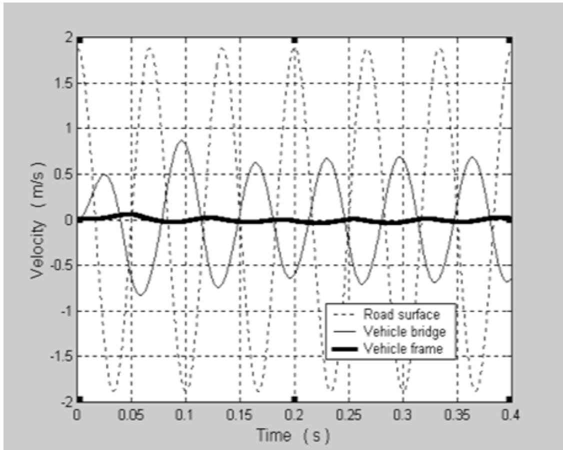


Figure 7: Velocity curve.

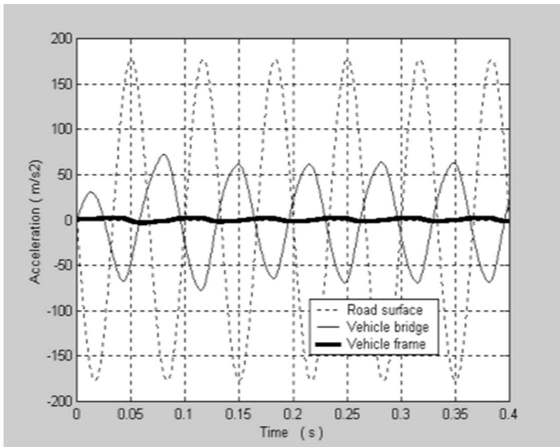


Figure 8: Acceleration curve.

Table 1: Data of simulation according to different input signal.

Simulation parameter	unit	value	value	value	value	value	value
Input signal Frequency of road surface	Hz	1	3	5	5	10	15
Input signal amplitude	mm	5	20	30	60	30	20
Displacement root-mean-square of road surface	mm	3.54	14.1	21.2	42.4	21.2	14.1
Displacement root-mean-square of vehicle bridge	mm	4.9	6.52	13.8	26.3	14.2	6.28
Displacement root-mean-square of vehicle frame	mm	6.6	11.6	5.2	9.7	1.6	0.56
Velocity root-mean-square of road surface	m/s	0.22	0.27	0.67	1.33	1.34	1.34
Velocity root-mean-square of vehicle bridge	m/s	0.03	0.12	0.43	0.8	0.87	0.53
Velocity root-mean-square of vehicle frame	m/s	0.05	0.16	0.12	0.25	0.08	0.03
Acceleration root-mean-square of road surface	m/s^2	0.14	5.02	20.9	41.8	83.5	125.1
Acceleration root-mean-square of vehicle bridge	m/s^2	0.61	4.3	17.4	33.3	54.6	44.7
Acceleration root-mean-square of vehicle frame	m/s^2	0.45	2.48	3.22	6.74	4.96	1.91
Displacement numerical decrement of vehicle frame relative to road surface	%	-87	18.3	75.6	77.9	92.5	96.1
Velocity numerical decrement of vehicle frame relative to road surface	%	-117	41.5	81.7	81.3	94.1	98.1
Acceleration numerical decrement of vehicle frame relative to road surface	%	-221	50.9	84.6	83.9	94.1	98.5

Table 1 shows the data of simulation according to different input signals.

6. PERFORMANCE OF SHOCK ABSORBING

According to the mathematical model, the results can be obtained which are shown in Figures 6-8 about the relations of displacement, velocity and acceleration to time. In Figures 6-8, the input signal frequency of road surface is 15Hz and amplitude is 20 mm. It is well known that the less root-mean-square of vehicle body,s acceleration is, the better comfort of vehicle is. Figures 6-9 show that there are a large numerical decrement in the displacement, velocity and acceleration of vehicle body relative to that of road surface. With the increase of frequency of signal input by road surface, the numerical decrement of displacement, velocity and acceleration of vehicle body becomes larger. When the frequency of input signal reach over 10 Hz, more than 90% numerical decrement can be reached of the displacement, velocity and acceleration of vehicle body relative to road surface [5-12].

7. CONCLUSIONS

The complex nonlinear mathematical model for a hydro-pneumatic suspension is developed in this paper. The fact that the results of computer simulation and real experiment are identical shows the mathematical model to be right basically. So the method to deal with the differential equation and the varying regular of stiffness and damping are reliable.

The research shows that the crane with hydro- pneumatic suspension has well and good shock absorbing performance in displacement, velocity and acceleration. This crane can provide a kind conform, safety and can be made to drive at a high speed on an uneven road surface because of its nonlinear stiffness and damping.

The model in this paper can be used in the parameter design. The effect of parameters on the performance by simulation and the parameter optimization will be studied in the future.

REFERENCES

[1] Moulton, A.E., Best. 1962. Hydragas Suspension, SAE. 790374, 1306-1327

[2] Kim, Y., Yoon, Y.S. 1995. Semi-Active Suspension with Preview Using a Frequency-Shaped Performance Index, Vehicle System Dynamics, 24, (10), 759-780.

[3] Moline, D., Floyd, S. 1994. Simulation and Evaluation of Semi- Active Suspensions, SAE Technical Paper 940864, 1120-1130.

[4] Warner, B., Rakheja, S. 1996. An Investigation of High Performance Dampers on the Suspension Performance of a Quarter Vehicle, SAE Technical Paper 962552, 503-517.

[5] Foste, A.W. 1978. A Heavy Truck Cab Suspension for Improved Ride, SAE Technical Paper 780408, 1-17.

[6] Duym, S., Stiens, R., Reybrouck, K. 1997. Evaluation of Shock Absorber Models, Vehicle System Dynamics, 27, (2), 109-127.

[7] El-Demerdash, S.M. 1996. Effect of nonlinear components on the performance of a hydro-pneumatic slow-active suspension system, Journal of Automobile Engineering, 210, (1), 23-34.

[8] Kumykova, T.M., Kh. Kumykov, V. 2013. Dynamics of mine hydro-pneumatic accumulator, Journal of Mining Science, 49, (5), 763-771.

[9] Emami, M.D., Mostafavi, S.A., Asadollahzade, P. 2011. Modeling and simulation of active hydro- pneumatic suspension system through bond graph, Mechanika, 17, (3), 312-317.

[10] Cao, D.P., Subhash, R., Su, C.Y. 2008. Dynamic analyses of roll plane interconnected hydro-pneumatic suspension systems, International Journal of Vehicle Design, 47, (1-4), 51-80.

[11] Łukasz, K. 2015. Damping characteristics of hydropneumatic suspension strut in function of car static load, Journal of Vibroengin eering, 17, (1), 74-81.

[12] Marin, D., Samoilescu, G., Nicolaie, S., Olaru, G., Cizer, L. 2009. Experimental model for an electric wave powered plant, in Annals of DAAAM and Proceedings, 521.