Middle-East Journal of Scientific Research 22 (9): 1405-1410, 2014
ISSN 1990-9233

DOI: 10.5829/idosi.mejsr.2014.22.09.22062

Investigation of a Wedge Shock Absorber Trucks Freight Cars Using Universal Mechanism

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Abstract: In this paper, a computer simulation of three-element wedge damper truck freight car and obtained the results as a force characteristics (hysteresis). Recently, in the study the problem of improving the safe operation of railway transport is widely used computer modeling, using appropriate software. With the advent of analog and then digital electronic computers became possible to solve the equations to the specific objectives and easier to consider the nonlinearity.

Key words: vibration absorber · The dynamics of the freight car · Horizontal force of friction · Vertical reaction

INTRODUCTION

Complex phenomena of the dynamics of the mechanical system "car - the way" or a system of cars in the train should be studied on the models in the laboratory. In the United States, Britain, Germany, France, Italy, Japan and other countries, this work is more than 25 years. The company «Mechanical Dynamics Incorporated», founded in 1977, has created a software system «ADAMS» (Automatic Dynamic Analysis of Mechanical Systems), which now holds about 60% of global market systems calculate the dynamics of mechanical structures. In Russia, the analysis of the dynamics of interaction between wheel and rail in a software system «ADAMS» engaged Sh.N. Shaydulin. D.Y. Pogorelov in the Bryansk State Technical University developed software package "Universal Mechanism". It includes a specialized module for modeling the dynamics of railway vehicles: locomotives, cars and track machines.

As you know the main and only damper in the standard trolley is 18-100 V-damper. It acts as a suppressor of both vertical and lateral vibrations. Study of friction shock absorbers are devoted I.I. Chelnokov, P.S. Anisimov, V.D. Husidov [1-4], etc. This paper describes the work of the V damper on two levels, depending on the frequency of oscillation bolster. Fluctuations in the beam are given as harmonic functions:

$$z = a(1-\cos(\omega_Z t)) \quad (1)$$

$$y = D\sin(\omega_Y t + \delta) \quad (2)$$

where z and y - vertical and lateral movements bolster; a = -A / 2;
A and D - the amplitude of vertical and lateral oscillations bolster;
ω_z and ω_y - circular frequency of vertical and lateral oscillations bolster;
δ - phase shift;
t - time, sec.

In the first phase the modeling vibrations bolster, separately, both in vertical and horizontal directions with the following initial data:

Vertical swing ω_z = 18.84 radians/sec; A = 0.05m;
Horizontal swing ω_y = 18.84 radians/sec; D = 0.01m; δ = 0.

The results of the simulation are shown in Figures 1-4.

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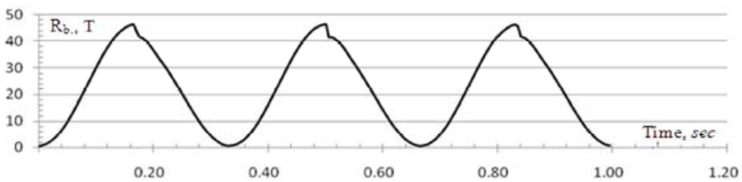

Fig. 1: The relationship between the vertical reaction of the wedge on the beam on time (ω_z = 18.84 radians/sec; A = 0.05m)

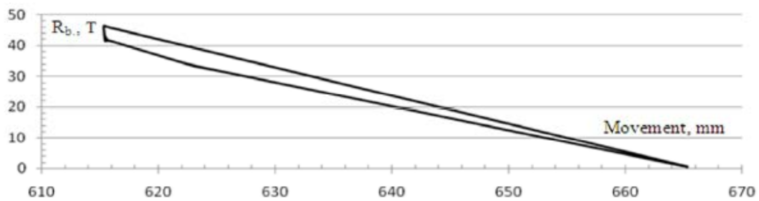

Fig. 2: The relationship between the vertical reaction of the wedge on the beam from the vertical position of the beam (ω_z = 18.84 radians/sec; A = 0.05m)

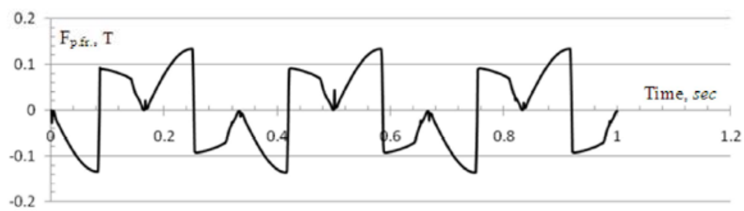

Fig. 3: The relationship between the horizontal force of friction on the friction plate in time (ω_y = 18.84 radians/sec; D = 0.01m δ = 0)

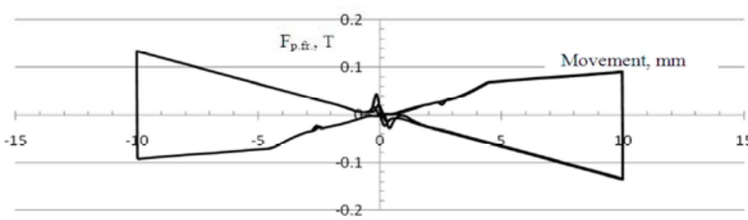

Fig. 4: The relationship between the horizontal force of friction on the friction plate on the cross beam deflection (ω_y = 18.84 radians/sec; D = 0.01m δ=0)

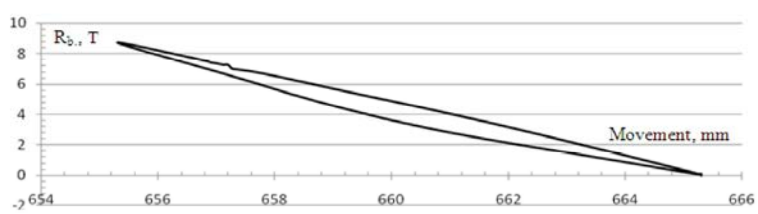

Fig. 5: The dependence of the vertical reaction of the wedge on the beam from the vertical position of the beam (ω_z = ω_y = 18.84 radians/sec; A = D = 0.01m δ=0)

In the second stage simulation of vertical and transverse vibrations of the beam produced at the same time. Testing has shown that the fluctuations bolster with the same amplitude A = D = 0.01m and frequency in the vertical and lateral directions ω_z = ω_y = 18.84 radians/sec and no phase shift (δ=0) the strength characteristics of the friction damper will take the form shown in Fig. 5-7.

If in the previous task to change only the frequency of horizontal vibration bolster, accept δ_y = 6.28 and the rest remain the same raw data, the dependence of the reactions on the beam vertical and horizontal forces of friction [5-9] on the friction plate in time to bolster movements take the form shown in Fig. 8-10.

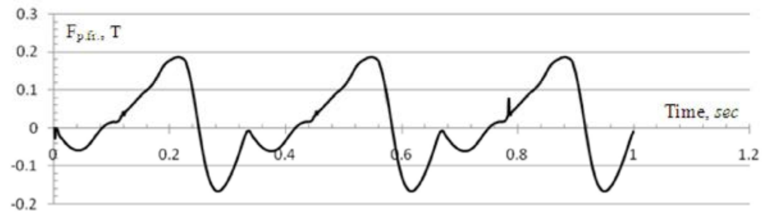

Fig. 6: The relationship between the horizontal force of friction on the friction plate in time ($\omega_z = \omega_y = 18.84$ radians/sec; $A = D = 0.01m$ δ=0)

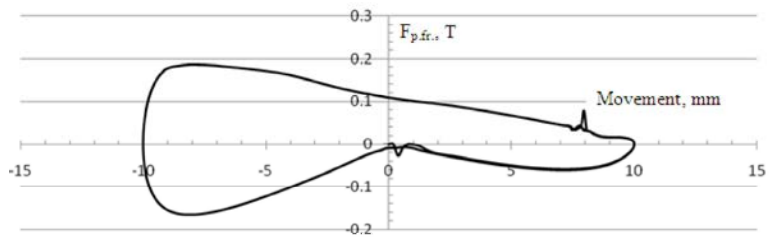

Fig. 7: The dependence of the horizontal force of friction on the friction plate on the cross beam deflection ($\omega_z = \omega_y = 18.84$ radians/sec; $A = D = 0.01m$ δ=0)

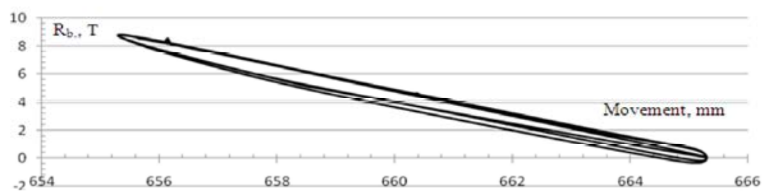

Fig. 8: The relationship between the vertical reaction of the wedge on the beam from the vertical position of the beam ($\omega_z = 18.84$ radians/sec; $\omega_y = 6.28$ radians/sec; $A = D = 0.01m$; δ =0)

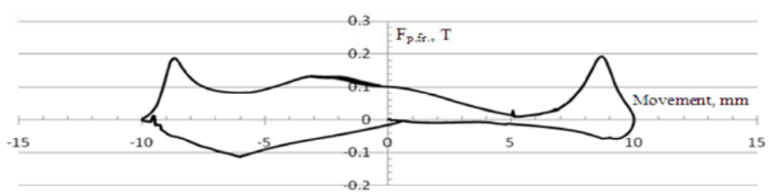

Fig. 9: The dependence of the friction force on the horizontal bar of the cross-beam deflection ($\omega_z = 18.84$ radians/sec; $\omega_y = 6.28$ radians/sec; $A = D = 0.01m$; δ=0)

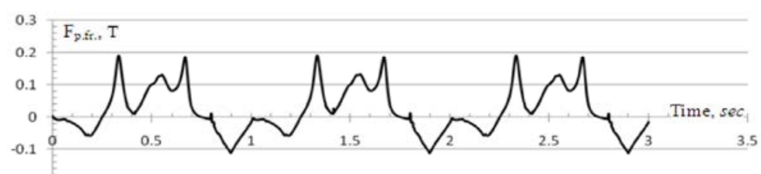

Fig. 10: The relationship between the horizontal force of friction on the bar in time ($\omega_z = 18.84$ radians/sec; $\omega_y = 6.28$ radians/sec; $A = D = 0.01m$; δ=0)

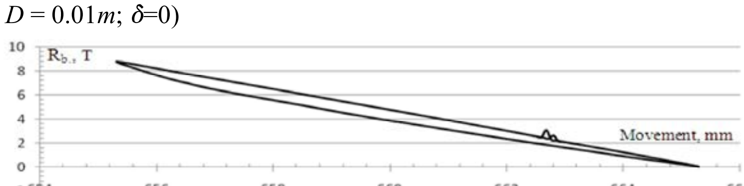

Fig. 11: The relationship between the vertical reactions of the wedge on the beam from the vertical position of The beam ($\omega_z = \omega_y = 18.84$ radians/sec; $A = D = 0.01m$ $\delta = \frac{\pi}{4}$)

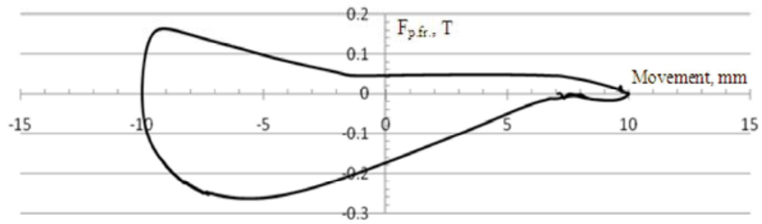

Fig. 12: The dependence of the friction force on the horizontal bar of the cross-beam deflection (ω_z = ω_y = 18.84 radians/sec; $A = D = 0.01m$ $\delta = \frac{\pi}{4}$)

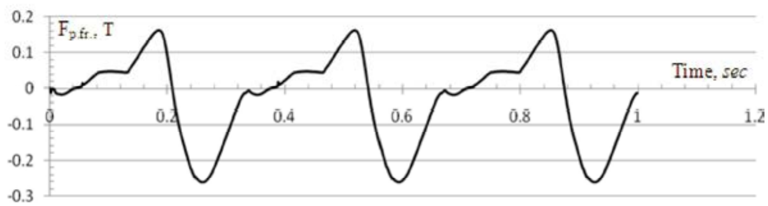

Fig. 13: The relationship between the horizontal force of friction on the bar in time (ω_z = ω_y = 18.84 radians/sec; $A = D = 0.01m$ $\delta = \frac{\pi}{4}$)

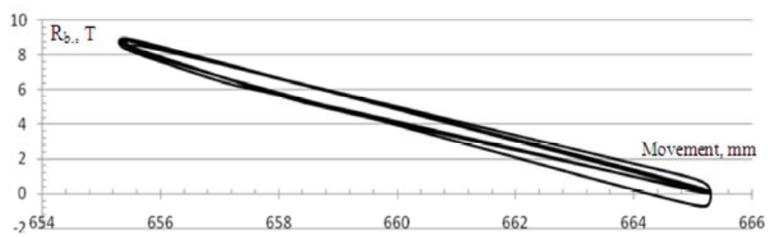

Fig. 14: The relationship between the vertical reaction of the wedge on the beam from the vertical position of the beam (ω_z = 18.84 radians/sec; ω_y = 6.28 radians/sec; $A = D = 0.01m$ $\delta = \frac{\pi}{4}$)

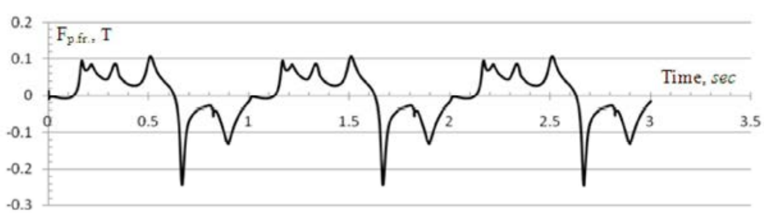

Fig. 15: The relationship between the horizontal force of friction on the bar in time (ω_z = 18.84 radians/sec; ω_y = 6.28 radians/sec; $A = D = 0.01m$ $\delta = \frac{\pi}{4}$)

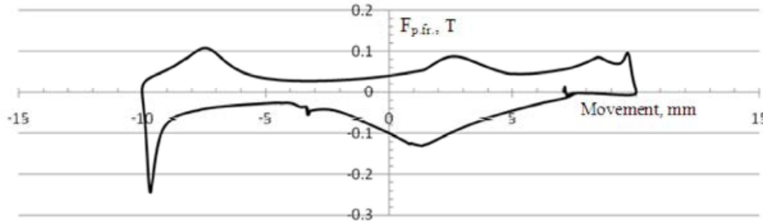

Fig. 16: The dependence of the friction force on the horizontal bar of the cross-beam deflection (ω_z = 18.84 radians/sec; ω_y = 6.28 radians/sec; $A = D = 0.01m$ $\delta = \frac{\pi}{4}$)

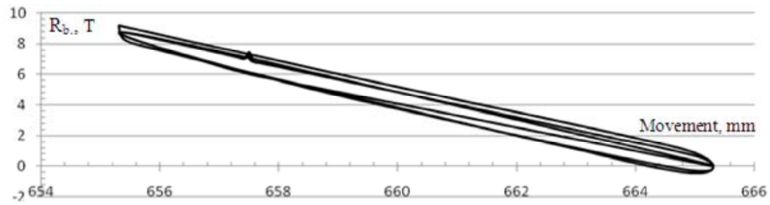

Fig. 17: The relationship between the vertical reaction of the wedge on the beam from the vertical position of the beam (ω_z = 18.84 radians/sec; ω_y = 6.28 radians/sec; $A = D = 0.01m$ $\delta = \frac{\pi}{2}$)

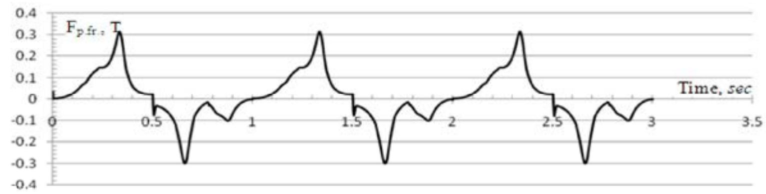

Fig. 18: The relationship between the horizontal force of friction on the bar in time (ω_z = 18.84 radians/sec; ω_y = 6.28 radians/sec; $A = D = 0.01m$ $\delta = \frac{\pi}{2}$)

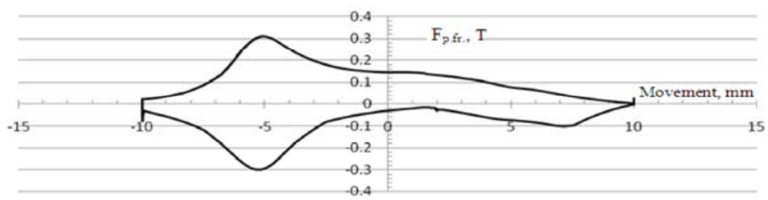

Fig. 19: The dependence of the friction force on the horizontal bar of the cross-beam deflection (ω_z = 18.84 radians/sec; ω_y = 6.28 radians/sec; $A = D = 0.01m$ $\delta = \frac{\pi}{2}$)

If at the same frequency $\omega_z = \omega_y$ = 18.84 radians/sec and the same source data to the phase shift $\delta = \pi/4$, the dependence of the vertical reaction of the wedge on the beam and the horizontal force of friction on the bar of the displacements bolster and time will form shown in Fig. 11-13.

Simulation results for $\omega_z = \omega_y$ = 6.28 radians/sec and $\delta = \pi/4$ and $\delta = \pi/2$ are given respectively in Figures 14-16 and 17-19.

CONCLUSION

The analysis carried out by computer simulation of the friction shock absorbers shows that the power characteristics of the friction wedge determined by the frequencies and phases of vertical and transverse vibrations of the beam and well corresponded with the results of the test trials [4], in connection with the PC "universal mechanism" can be recommended as a software apparatus for the study of oscillations of rail vehicles. The software package "Universal Mechanism" allows you to create a fully parameterized model: set with the identifiers or expressions of inertial and geometric parameters (including graphics elements, profiles, rails), as well as the main characteristics of the power components (for example, spring rates, damping coefficients, coefficient of friction in the contacts and so on.) Analysis of the results shows a good qualitative and quantitative convergence of the calculated and experimental values of the main indicators of the dynamics - the maximum difference in the test velocity range was 12-15%.

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