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Layers Disk Spring: Experiment And Simulation

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Abstract
In this paper one model layers disk spring is designed. In order to analyze effects of layers disk spring, based on the physical model and mathematical model presented in this study. Through experiment to determine the force-deformation relationship characteristics of the manufactured discs spring. Operating conditions such as impulse force, step force, speed of punching machine is analyzed to evaluate effects of dumping and isolation of vibration. The effect of isolation of layers disk spring is compared with effect of isolation of coil spring which has same characteristics
KEYWORDS: *layers disk spring, isolator, vibration, impulse force, step force…*

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I. INTRODUCTION

A conical spring as shown in Figure-1 absorbs more energy than a torsion spring in the same space. This type of spring is suitable when large loads and small deformations are required [4]. Springs are usually installed and arranged in layers. This type of spring has the ability to self-damp vibrations like leaf springs: Vibrations are quickly damped after compression. The conical disc of this type of spring has the cross-sectional dimensions and load as shown in Figure 1. The shape of the force-deformation curve depends mainly on the coefficient of free cone height h and thickness. The advantages of disc springs include small mounting space in the direction of the force, the ability to withstand lateral forces, and the force-deflection characteristics can be changed by adding or removing discs. Disadvantages include non-uniformity of the stress distribution, especially when the coefficient of difference between the inner and outer diameters is large [4].

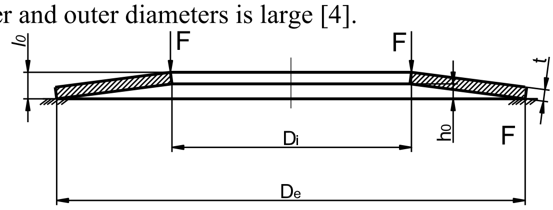

Figure 1. a single disk spring

TT	D_e (mm)	D_i (mm)	t (mm)	H (mm)	h_0 (mm)	h_0/t	μ	E (N/mm^2)
	50	25,4	1,25	2,85	1,8	1.44	0,3	$2{,}06*10^5$

Table1 – Parameter of a single disk sping

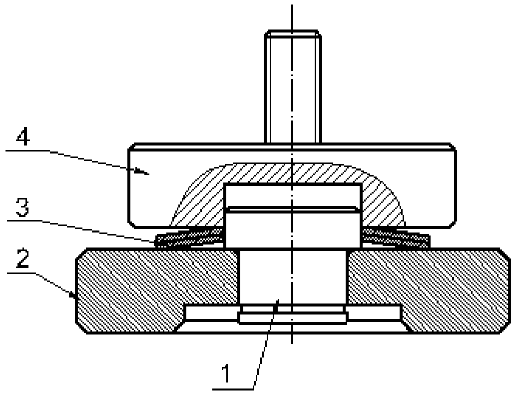

Figure 2. Experimental bearing model: 1. Core 2. Statics plate. 3. Disk spring. 4. Moving plate

L.j Zheng [20] proposed a formula to accurately calculate the load-displacement relationship of disc springs. Through theoretical analysis, Saini [21] studied the load-bearing capacity and deformation characteristics of disc springs with varying thickness. [22] An experimental study was conducted to demonstrate the damping characteristics of disc springs and showed that the damping capacity of disc springs is greater than that of conventional materials. G Curti [23] studied the effect of friction on the disc spring by finite element method and experiment. X.S Gong [24] proposed a method to build a dynamic model of the vibration damper by analyzing some types of vibration damper with nonlinear hysteresis characteristics.

F.Jia and F.Y.Xu designed a type of vibration damper using stacked disc springs that can slide in a guide core [26]. This type of bearing structure has a large load capacity, and the installation space is significantly reduced compared to the torsion spring type. Research results show that this type of combination bearing is 98% effective in reducing vibration. This type of bearing is suitable for machines such as punching machines, pressure machining...

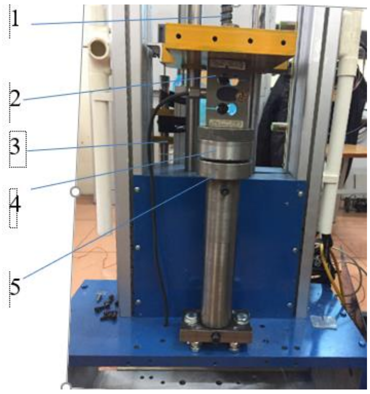

Figure 3 1. Drive screw 2. Loadcell 3. LVDT 4. Dynamic press 5. Static press

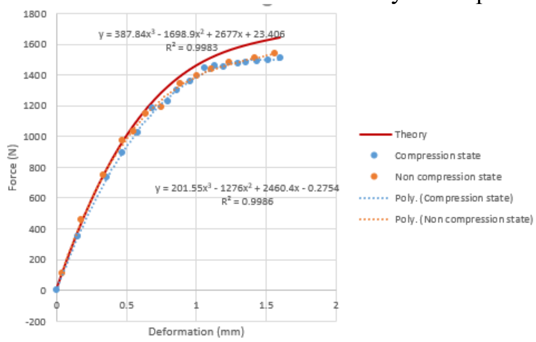

Figure 4. Force-deformation relationship graph

II. DYNAMIC MODEL

According [3] the mathematical model of the system is a single-degree-of-freedom motion model (figure-1):

$$m\ddot{z} + c\dot{z} + k.z = F(t) \qquad (1)$$

Where c is the damping coefficient, k is the stiffness of the layers disc spring. F_t is the excitation force.

If viscous damping is ignored, the vertical motion of the system, z(t) can be represented as follows:

$$z(t) = \frac{F_0/k}{1-r^2} sin(\omega.t) \qquad (2)$$

where: $r = \frac{\omega}{\omega_n}$ with $\omega_n = \sqrt{\frac{k}{m}}$, when $r=1$ the resonance phenomenon occurs, causing the oscillation amplitude to increase sharply even though the excitation force is small.

Natural frequency f_n:

$$f_n = \frac{\omega_n}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{k}{m}}(Hz) \qquad (3)$$

Critical frequency:

$$f = 60 f_n = \frac{60}{2\pi}\sqrt{\frac{k}{m}} \qquad (4)$$

Force transmitted to the floor:

$$F_T = k.x \qquad (5)$$

Transmission coefficient when considering the effect of damping:

$$\rho = \sqrt{\frac{1+(2\xi cr)^2}{(1-r^2)^2+(2\xi r)^2}} \qquad (6)$$

In which ξ is the vibration reduction coefficient.

$$\xi = \frac{c}{2m\omega_n}\frac{c}{c_r}; c_r = 2\sqrt{k.m} \qquad (7)$$

The characteristic value of the vibration damping coefficient is from 0.005 to 0.1 for steel and from 0.05 to 0.1 for rubber [2].
Assumptions:
The machine is modeled as a heavy object, mass M=340 kg.
The mass is evenly distributed on 04 bearings, so the mass acting on one bearing is 85kg.
The machine's legs are connected to an absolutely rigid foundation through a system consisting of a disc spring with stiffness *K* and a viscous damper with damping coefficient *c*.
The initial deformation of the spring is $s_0 = 0,25h_0 = 0,4$ *mm* when it bears the weight of the machine.
The stiffness of the coil spring is equivalent to the stiffness of the layers disc spring when it reaches deformation $s_0 = 0,25h_0 = 0,4$ *mm*, then the stiffness of the disc spring at deformation S_0 is equal to $K = 1607,78$ *N/mm* A quarter-vehicle dynamic model of punching machine is established to see the effect of the values of the damping coefficient of layers disk spring, as shown in Fig-3.

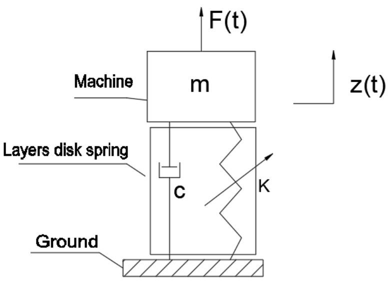

Figure 5**.** Quarter-machine dynamic model

In Fig-1, m is quarter- machine mass; k and c are the stiffness and damping coefficients of layers disk spring; *z(t)* is the vertical displacements of machine; and $F(t)=F_0 sin(\omega t)$ is the force excitation of the vibrating machine; F_0 is the amplitude of force excitation; ω is the angular frequency of the machine.
The stiffness equation of a single disc spring is obtained [6]

$$k = \frac{dF}{dx} = 1163,52x^2 - 3397,8x + 2677 \qquad (8)$$

According [4] The effect of vibration reduction is an important requirement in the design process isolator.
Usually, the vibration reduction efficiency needs to reach from 70% to 90%.
Transmission coefficient ρ determines the maximum transmission coefficient of the system based on the required vibration reduction efficiency
Minimum value of the excitation force frequency that ensures the vibration reduction efficiency of the bearing, according to [4] $f/f_n > \sqrt{2}$
In order to compare the vibration reduction characteristics of the layers disc spring and the helical spring with equivalent stiffness and the same damping coefficient, calculate the damping coefficient based on the model as shown in Figure 4:

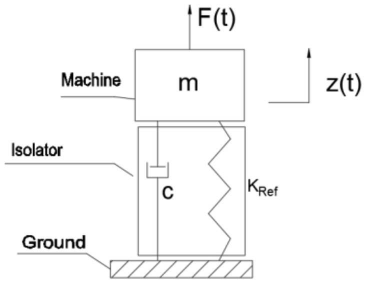

Figure 6. Vibration model using coil spring with stiffness K_{ref}

According [6]:

$$K_{ref} = \left.\frac{dF}{ds}\right|_{x^*=0,4} = 1163,52.(x^*)^2 - 3397,8.x^* + 2677$$

$$1163,52.(0,4)^2 - 3397,8.0,4 + 2677 = 1504.04\,N/mm \qquad (9)$$

$$= 1504,04\,N/mm$$

From (6) with the target of vibration damping efficiency being 90%, $\rho=0,1$ so we have:

$$r = \frac{\omega_{min}}{\omega_n} = 3.2 \qquad (10)$$

And have $c = 1357,94\,kg/s$ (11)

The natural frequency of the spring system with stiffness Kref:

$$\omega_n = \sqrt{\frac{K_{ref}}{m}} = \sqrt{\frac{1504,04.10\wedge 3}{85}} = 133,02\ (rad/s) \qquad (12)$$

$$r = \frac{\omega_{min}}{\omega_n} = 3.2 \Rightarrow \omega_{min} = \omega_n.r = 425.667(\,rad/s\,) \qquad (13)$$

Determine the amplitude F_0 according [27] the design working point of the disc spring is usually taken as $0,75h_0$. where h_0 is the height of the cone of the disc spring, the initial deformation is chosen as $s_0 = 0,25.h_0 = 0.4(\,mm\,)$ therefore the oscillation amplitude of the mass m in Figure – 1 $z_{max}=0,4mm$ so we have:

$$\frac{F_0}{K_{ref}} \le X_{max} \Rightarrow F_0 \le K_{ref}.X_{max} = 1504,04.10^3.0,0004 \qquad (14)$$

$$= 601,16N$$

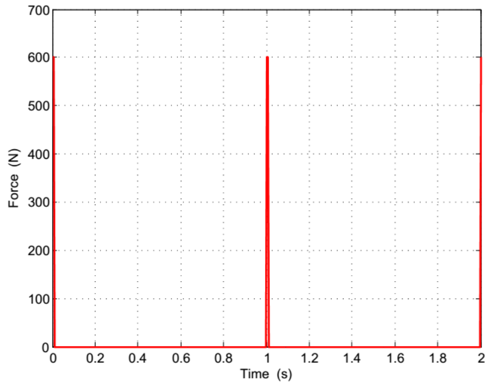

Figure 7. Impulse force

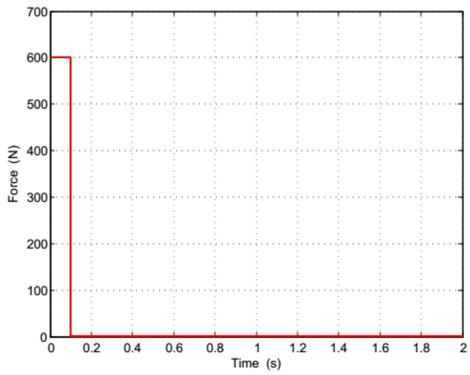

Figure 8. Step force

III. SIMULATION AND DISCUSSION

In order to solve the general dynamic differential equation of a punching machine presented in section 2, Matlab/Simulink software is used.

With exciting force have results as figure 6->figure 9

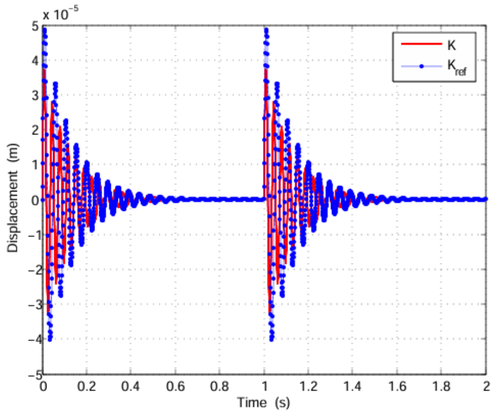

Figure 9. Vibration amplitude graph of mass M

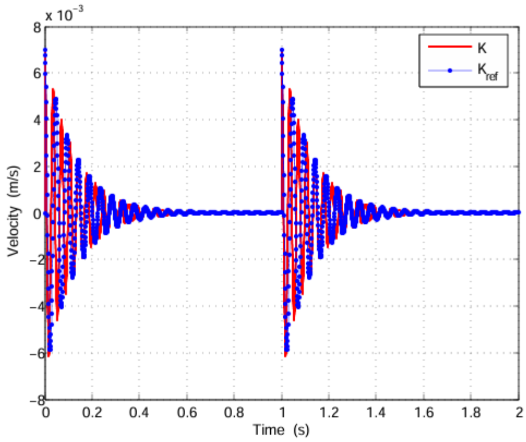

Figure 10. Vibration velocity of mass M

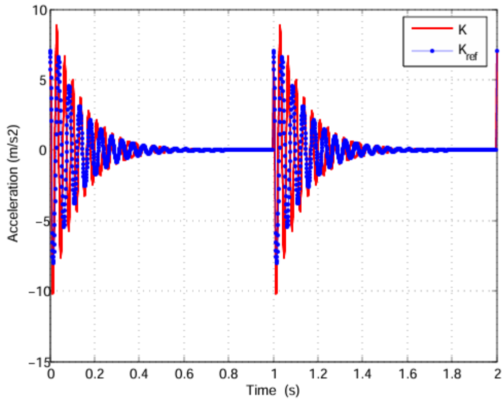

Figure 11. Vibration Acceleration of mass M

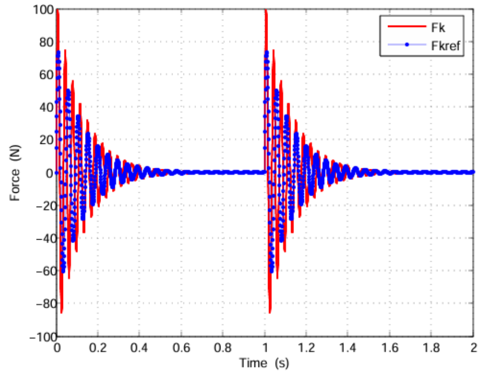

Figure 12. Force transmitted to the ground

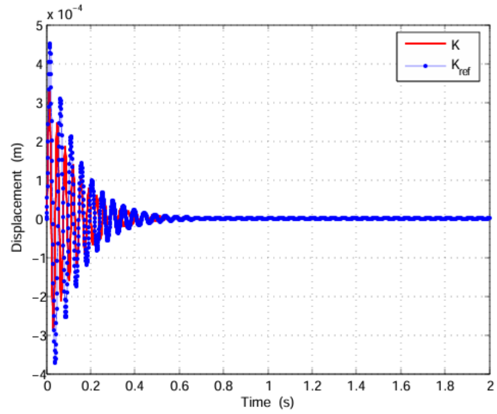

Figure 13. Vibration amplitude graph of mass M

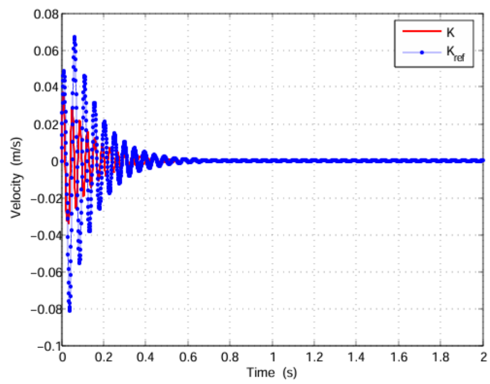

Figure 14. Vibration velocity of mass M

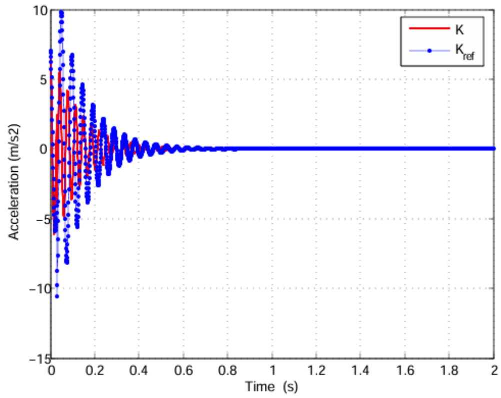

Figure 15. Vibration Acceleration of mass M

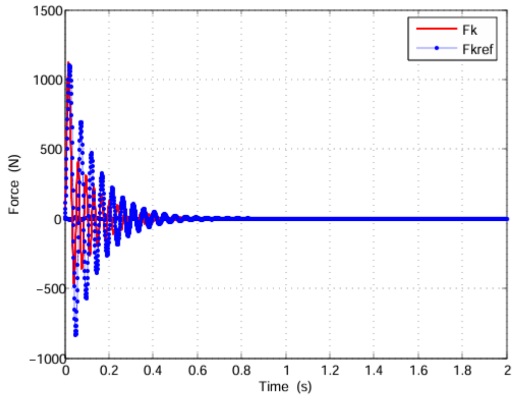

Figure 16. Force transmitted to the ground

IV. CONCLUSION

The study focuses on effect of the values of the damping coefficient of layers disk spring. A quarter-vehicle dynamic model of vibratory is established for analysis and evaluation. Parameters of small type punching machine are selected to apply on layers disk spring. The results of matlab/Simulink software are shown.
The major conclusions can be drawn from the analysis and evaluation results as follows:
Under impulse-type excitation force, the disc spring vibration damping bearing shows the ability to dampen vibrations faster than the torsion spring vibration damping bearing (the damping oscillation cycle of the torsion spring bearing is 1.3 times the cycle of the disc spring bearing). The response of this bearing model is quite suitable when used as a vibration damping bearing for machines such as medium and large punching and stamping machines when the pulse-type excitation force often appears in these types of machines.

With the excitation step type force in the form of vibration, the disc spring vibration damper shows clear advantages in limiting the oscillation amplitude of the system, reducing velocity and acceleration.

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