

Computer Aided Design of A Vibration Isolation Damper for 1-1.5 Ton Vehicles.

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ABSTRACT

Vehicles are prone to vibration as the surfaces on which they ply on are not necessarily smooth. This paper explains the design procedure to be followed while designing a vibration isolation damper in the case of 1-1.5 ton vehicles. This damper helps in isolating the vibrations so that the ride can be smooth enough for the passenger. Vibrations can cause serious harm to the other mechanical devices and components present inside the vehicle. Undue vibration can cause loosening of bolts and failure of joints. Vibration needs to be kept under strict limits in order to prevent failure. The damper comprises of a cylinder (piston housing), piston, connecting rod (shaft), spring, rubber pad or silencer rubber, plate (load carrier), fluid and rubber seal. The damper was designed using virtual engineering through Computer Aided Design and simulation of the dashpot was done using the ANSYS Structural software. Simulation provides results showing where the maximum and minimum stresses will act and hence the location of the critical areas where failure can occur.

Keywords- vehicle, vibration, piston, isolation, simulation, damping, suspension.

I. INTRODUCTION

Vibration isolation is the process of isolating an object, such as a piece of equipment, from the source of vibrations. "Passive vibration isolation" refers to vibration isolation or mitigation of vibrations by passive techniques such as rubber pads or mechanical springs, as opposed to "active vibration isolation" or "electronic force cancellation" employing electric power, sensors, actuators, and control systems. Passive vibration isolation is a vast subject, since there are many types of passive vibration isolators used for many different applications. A few of these applications are for industrial equipment such as pumps, motors, HVAC systems, or washing machines; isolation of civil engineering structures from earthquakes (base isolation) sensitive laboratory equipment, valuable statuary, and high-end audio.

Unless a dampening structure is present, a car spring will extend and release the energy it absorbs from a bump at an uncontrolled rate. The spring will continue to bounce at its natural frequency until all of the energy originally put into it is used up. A suspension built on springs alone would make for an extremely bouncy ride and, depending on the terrain, an uncontrollable car.

The shock absorber, or snubber, a device that controls unwanted spring motion through a process known as dampening. Shock absorbers slow down and reduce the magnitude of vibratory motions by turning the kinetic energy of suspension movement into heat energy that can be dissipated through hydraulic fluid. When the vehicle gets into high and low speed bumps and rebound situations. Basically, the characteristics of the damper may need to be different depending on the speed the suspension compresses at. For example, on smooth sweeping fast roads, you may want the suspension to load up progressively as the car rolls. But on bumpier roads, you may want faster reactions as the wheels will be travelling up and down much faster. The way the damper is designed can create different ratios between bump and rebound settings at different speeds, ranging anywhere between 1:1 to 3:1 rebound: bump.

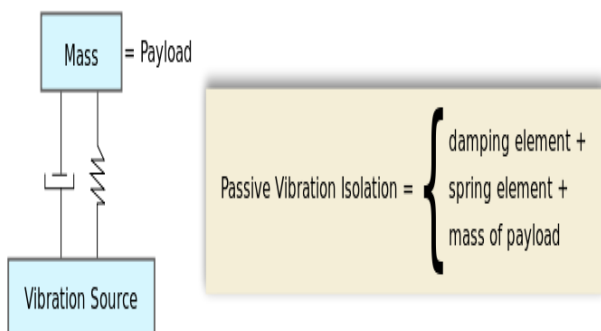


Fig 1: Basic concept

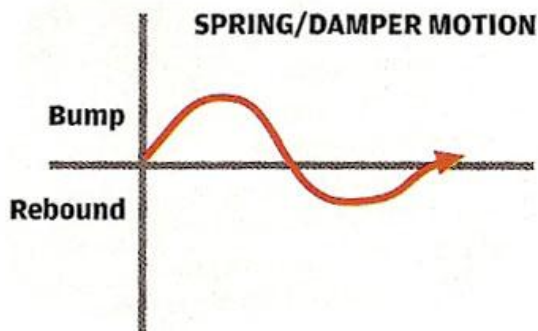


Fig 2: Damper graph

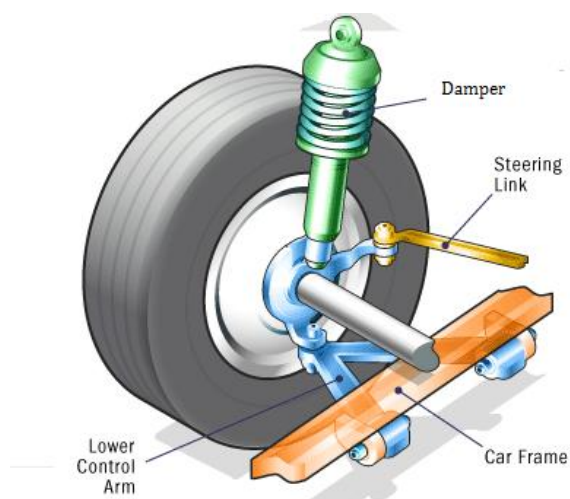


Fig 3: Location on the vehicle

II. DESIGN CALCULATION

Any vibration isolation device has an interface to the structure being isolated from vibration sources. This interface can be approximately taken as a single point in many cases either for its sufficiently small area or for its high-enough stiffness that the interface can be considered as rigid body. In such cases, the single degree of freedom model is the most widely used approach to represent a vibration isolation system (Liang and Wenhui, 2008). The design is based on the consideration of a single degree freedom system figure 3 formed by a rigid mass in resting on a spring of stiffness K and a viscous damper having damping coefficient C . a system is said to be of a single degree when its motion is constrained to one direction Let the system be set in motion by giving an initial velocity to the mass, thus the equation of motion for the free vibration of the system is

$$M\ddot{x} + C\dot{x} + Kx = 0 \quad (1)$$

(Inertia Force) (damping Force) (spring Force)

Where

x = displacement

\dot{x} = the velocity of the mass

\ddot{x} = acceleration of the mass (Adeyeri, 2001)

In equation 1, the right hand side equals zero since there is no external force on the system during

vibration. However, if the system is subject to a loading force, equation (1) now becomes

$$M\ddot{x} + C\dot{x} + Kx = F \quad (2)$$

(Inertia Force) (damping Force) (spring Force) (Loading Force)

These fundamental representations could be used for a variety of practical modeling and design exercises. In particular, one can frequently obtain helpful initial information on structural and machine natural frequencies, vibration transmissibility, effectiveness of simple dampers and absorbers (Mathew, 1990).

Design for Piston Thickness

In designing for piston thickness, it is assumed that the diameter of the piston is that of the cylinder with a considerable length. In order to withstand the load, the equation,

$$t = \frac{3F}{4S} \quad (3)$$

Where t = thickness of piston, (m); F = force of load, (N) and S = yield stress of the material

Design for the Piston Rod Diameter

The piston rod diameter is got through the expression given by Lingaiah and Narayan (1973)

$$D_{ip} = 1.9 \text{ to } 2.5D_c \quad (4)$$

Where D_{ip} = the piston inside diameter

D_c = diameter of the connecting rod

Design of Piston's Length

Discretion is used to ascertain as expressed in equation (5).

$$L = 0.6 \text{ to } 0.7D \quad (5)$$

Where L = piston length

D = diameter of the cylinder bore

Design for Thickness of Piston Grooves

This thickness is connected by equation 6 below;

$$h = 0.025D \text{ to } 0.03D. \quad (6)$$

Design for the Cylinder Diameter

The diameter D , of the cylinder bore is chosen with respect to the piston's diameter plus clearance to maintain efficient damping. Mathematically,

$$D = D_{op} + 1 \text{ mm (clearance)} \quad (7)$$

Where,

$$D_{op} = (D_{ip} + 2 \times \text{thickness of the piston rod})$$

$$\therefore D = (D_{ip} + 2 \times \text{thickness of the piston rod}) + 1$$

and D_{op} = the piston outside diameter

Design for the Cylinder Depth

The depth of the cylinder bore is dependent on the maximum amplitude of displacement plus half of the piston's length. This statement is therefore expressed mathematically through equation 8.

$$h_c = X_a + 0.5L \quad (8)$$

Where h_c = the cylinder depth
 L = as previously defined

Design for the Shaft Diameter

The Euler's equation 9 below is used

$$F_{cr} = \frac{C\pi^2 E_{steel} A}{(L/K)^2} \quad (9)$$

Where F_{cr} = critical load to cause buckling
 C = constant (depending upon the end condition)
 E = Modulus of elasticity MN/m²
 A = area; L = length of column (meter)
 K = minimum radius of gyration

Spring Constant Calculation

$$w_n^2 = \frac{k}{m} \quad (10)$$

But, $w_n = \frac{2\pi N}{60}$

Taking $w_n = 600$ rpm

$$w_n = \frac{2\pi \cdot 600}{60}$$

And $m = 1.5\text{ton} = 1500$ kg

Thus $K = w_n^2 m$

Vibration Isolation

A convenient expression for the statement of the performance of vibration isolation system is that of transmissibility. In case of force excitation this is expressed as force transmissibility T_r , the ratio of the force supplied to the support compared with the force applied to the mounted item (F_{sp}/F).

This transmissibility is defined as

$$T_r = \frac{1}{(\frac{f_d}{f_n})^2 - 1} = \frac{F_{sp}}{F} \quad (11)$$

Where T_r = transmissibility

f_d = driving force frequency (Hertz)

f_n = Natural frequency (Hertz)

F_{sp} = The supplied force to the support (N)

F = the applied force to the mounted item (Albert and Richard, 1976)

From equation 11, the following theoretical and practical points become apparent for:

- i.) vibration isolation to occur, $f_d > f_n \cdot 2$
- ii.) for $f_d < f_n \cdot 2$, amplification of force or motion occurs.
- iii.) the higher f_d (or alternatively the lower f_n) the higher is the degree of isolation obtained.
- iv.) for $f_d = f_n$, $T_r = \alpha$ [in practice, force of motion is never infinite because of damping and restraint inherent in practical systems]

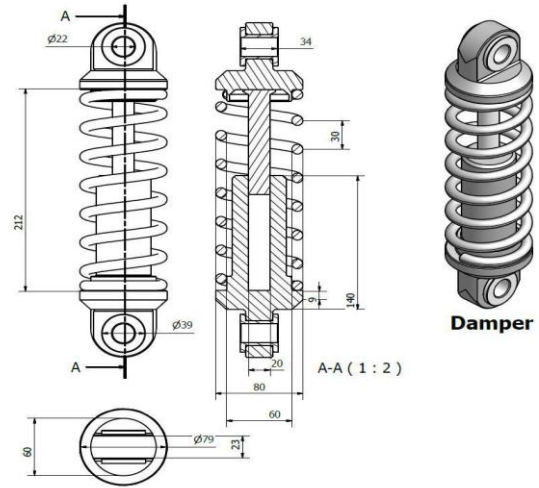


Fig 4: Designed damper

III. METHODOLOGY

The damper is designed using Autodesk INVENTOR software and the simulation is carried out using ANSYS 14.5 Static Structural. The damper is simulated with a load of 15000 N and the results have been shown below in the form of images. These images shown where the maximum and minimum forces act on the overall assembly of damper. With the help of these results it can be seen where the maximum and minimum forces act and which are the critical sections. The simulation is carried out to find the following:

- 1) Total Deformation
- 2) Equivalent (Von-Mises) Stress
- 3) Maximum Principal Stress
- 4) Equivalent Elastic Strain

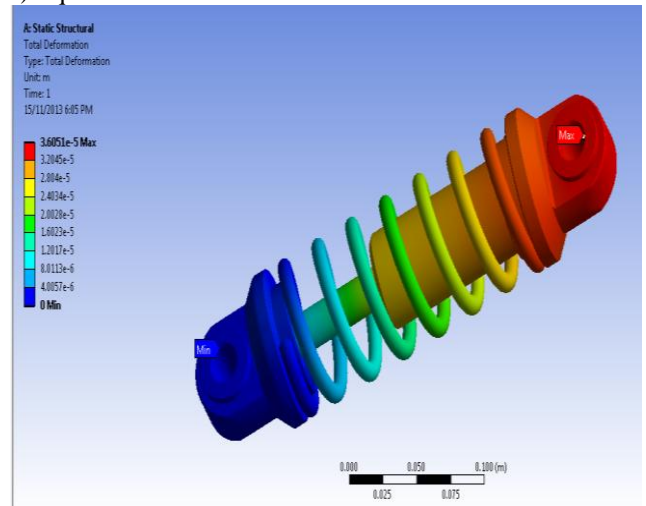


Fig 5: Total deformation

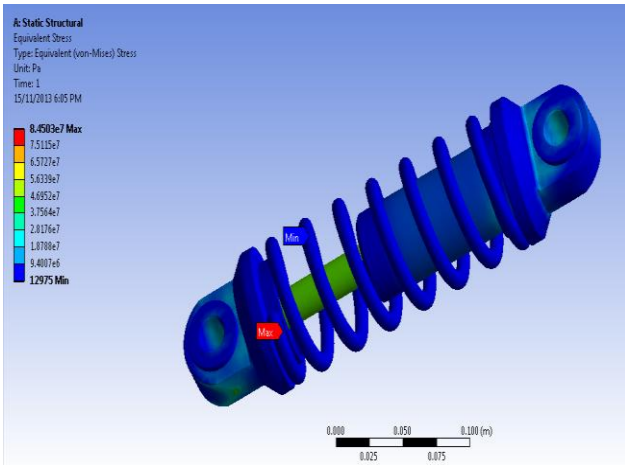


Fig 6: Equivalent stress

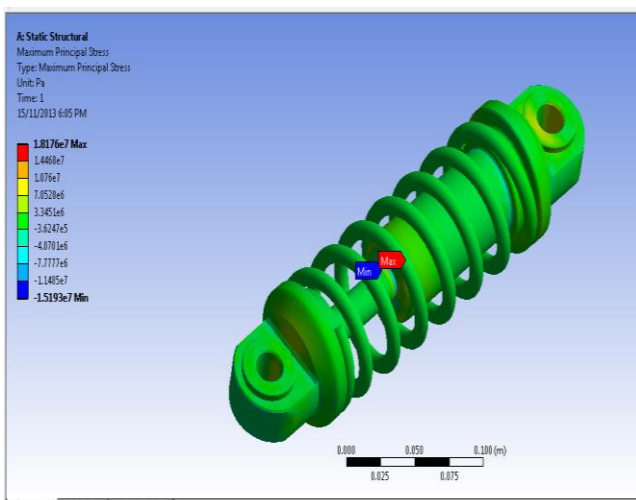


Fig 7: Maximum principal stress

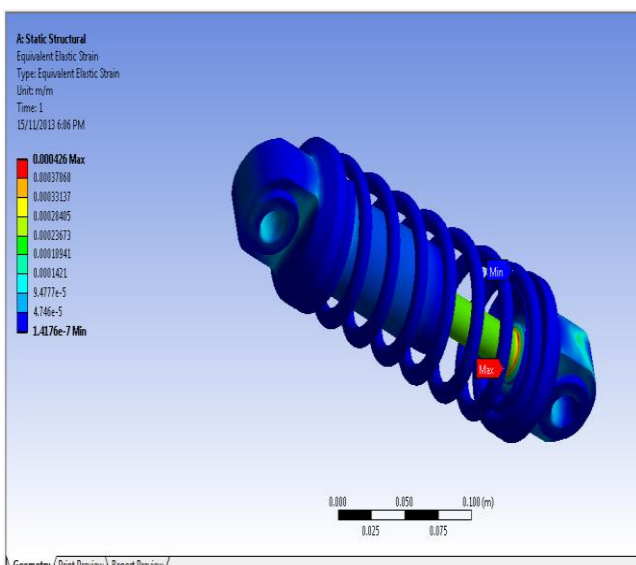


Fig 8: Equivalent elastic strain

IV. RESULTS AND DISCUSSION

The designed dashpot was tested for through simulation as can be seen from the figures above. The critical sections can be viewed with the help of the

simulation results. The maximum and minimum values of the forces can be analyzed and the material to be chosen should be such that its ultimate and tensile stress limit should be within these values.

V. CONCLUSION

This paper thus explains the design procedure for a vibration isolation damper, the materials to be used and shows the simulated results. The areas where failure can occur are also shown and hence it will be useful while designing the piston and also for any design modifications in the future.

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