ANTI-VIBRATION TECHNIQUES

MASS SPRING SYSTEM

A mass spring system may be represented by a mass "M", excited by a force "F" and supported on an elastic stiffness element "K" with a dampening factor "C".

The frequency of the mass spring system is equal to:

$$f_o = \frac{1}{2\cdot\pi}\sqrt{\frac{k}{M}}$$

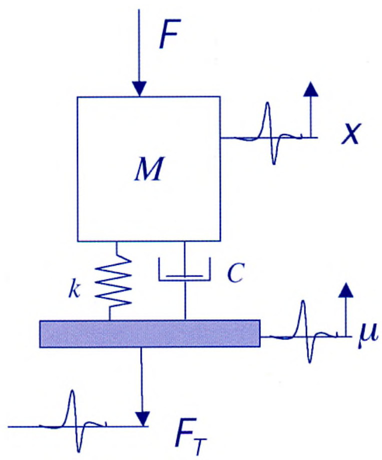


figure 3

K = N/m
M= in Kg
Fo in Hz
C in Ns/m

The effectiveness of the suspension may be measured by transmissibility, i.e. by the force which is transmitted by the machine to the ground or floor. It is defined as the ratio between the force transmitted to the ground, and the original force produced by the vibration.

Another practical term is often used to describe the efficacy of an anti-vibration mount, namely the degree of insulation, which is:

E = (1-T) x 100%

Transmissibility equation:

Taking the following parameters into account:

Excitation

$X=X_o\sin(\omega t+\theta)$

$F=F_{To}\sin(\omega t+\theta)$

Response

$\mu=\mu_o\sin\omega t$

$F=F_o\sin\omega t$

Own Pulsation: $\omega_o = \sqrt{\frac{k}{M}}$ for $C \cong 0$

and natural frequency of $f_o = \frac{1}{2\cdot\pi}\sqrt{\frac{k}{M}}$

The damping parameters are: $C_C = 2\cdot\sqrt{kM}$

Where Cc is the critical damping and ξ the damping coefficient.

$$\xi = \frac{C}{C_C}$$

For this system we obtain a transmissibility T and a magnification factor A:

$$T = \frac{X_o}{\mu_o} = \frac{F_{To}}{F_o} = \sqrt{\frac{1+\left[2\cdot\xi\cdot\frac{\omega}{\omega_o}\right]^2}{\left[1-\frac{\omega^2}{\omega_o^2}\right]^2+\left[2\cdot\xi\cdot\frac{\omega}{\omega_o}\right]^2}}$$

For the case of active $T = \frac{F_{To}}{F_o}$ and

passive isolations, we will have $T = \frac{X_o}{\mu_o}$

Figure 5 represents the transmissibility curve of the schematic mass spring system of figure 3.

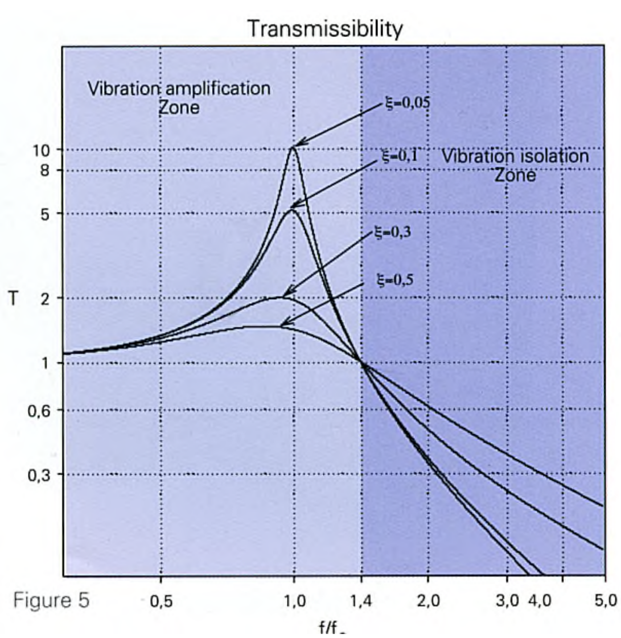


Figure 5

Examining this curve allows us to reach basic conclusions for effective isolation.

If the frequency of excitation is $\sqrt{2}$ times less the natural frequency, transmissibility is greater than one, then the force transmitted is greater than the excitation force, there is magnification of the vibrations. When we work in this area, the existing damping in the system is important. The greater the latter, the smaller the magnification of the vibrations will be.

If the frequency of excitation is $\sqrt{2}$ times greater than the natural frequency, transmissibility is less than one, or in other words the force transmitted is less than the force originated in the system, then we are in the damping area.

In order to achieve the greatest isolation, the lowest possible natural frequencies should be sought. There are two ways of doing this:

- By increasing the system mass.
- By reducing the stiffness of the anti-vibration mount.

To increase the efficiency of the isolation in the damping area, it is advisable to have low damping, although weak damping generates greater displacement when passing through resonance, it is advisable to use a damping coefficient t so that passage through resonance does not give rise to inadmissible displacement for the machine.

STATIC AND DYNAMIC STIFFNESS

The stiffness of a rubber anti-vibration mount changes when a dynamic force is applied to it. This parameter depends on architecture, the compound used and even the frequency of excitation.

Generally speaking, dynamic stiffness is always greater than static stiffness, so calculations based on static stiffness may lead to wrong conclusions.

In some cases it is possible to reach limits of dynamic stiffness which are two and even three times greater than the static stiffnesses.

DAMPING

The damping coefficient depends basically on the compound used in manufacturing of the anti-vibration mount. It is a crucial parameter that must be addressed when designing anti-vibration suspensions.

CREEPING AND LONG-TERM BEHAVIOUR

If an elastomeric element is under a static load, this load produces a progressive increase in deformation. This phenomenon may be important in a wide variety of applications, from mounts for buildings to engine mounts.

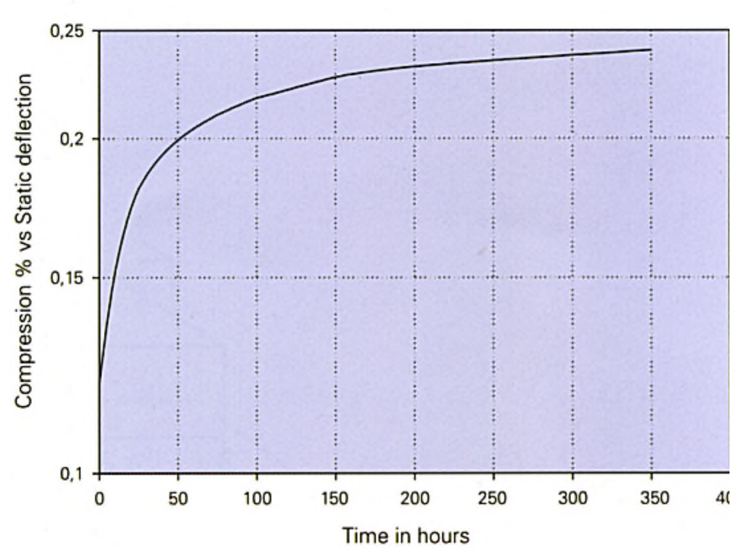
Creeping at a given time t is calculated as:

$$t = \frac{X_1 - X_o}{X_o} \times 100\%$$

And is expressed as a percentage (%) of the initial deformation. This value depends on the geometry of the mount, and above all on the way the rubber is worked.

Custom Audio Designs

Creeping



Designs that use rubber in shear are more conducive to “Creep” than designs which use rubber in compression or shear and compression.

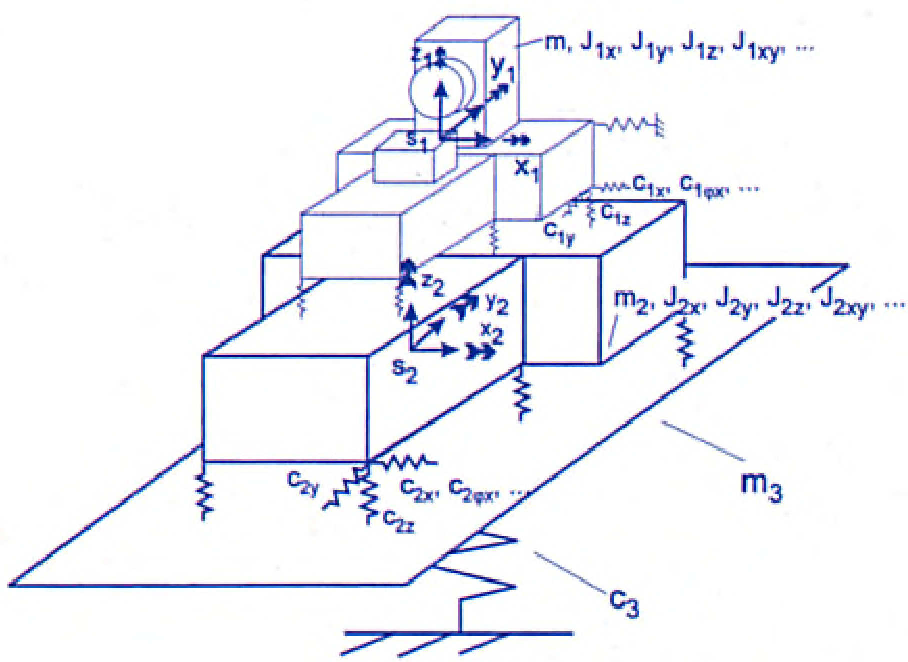
DYNAMIC TESTING MACHINE

Dynamic stiffness can only be established by measurement on a dynamic test bench. Similarly, the damping coefficients of compounds are further values that can be measured with this type of machines.

One concept that must be taken into account when designing an anti-vibration mount is its durability. A dynamic testing machine allows us to conduct fatigue tests that reproduce the real working conditions of the part so that its useful life can thus be predicted accurately.

ANALYSIS OF SYSTEMS OF MORE THAT ONE DEGREE OF FREEDOM

In actual fact, there are cases where the model of 1 degree of freedom cannot correctly define the behaviour of the equipment to be isolated. In such cases we have analysis tools that enable more elaborate models to be made taking into account the 6 Degrees of Freedom rules.



The latest computing tools can also generate virtual models of solid rigid multiples and study how they interact with each other and with the environment.

As a result, we can ascertain the natural frequencies of the system which are really important to prevent them from coinciding with the excitation frequencies so as not to have resonance problems.