ACTOM

MECHANICAL EQUIPMENT

Engineering

Theory & Practice

ARNOT

Vibration & Rubber
Engineering Solutions

Anti-vibration Mounts – Theory & Practice

The characteristic of an anti-vibration mounting that mainly determines its efficiency as a device for storing vibration energy, as opposing to transmitting it, is the amount that it deflects (usually meaning the amount it compresses) under load. The first step when selecting an AV mounting is to decide what load it must carry and how much it must deflect.

1. NATURAL FREQUENCY

If a machine such as a compressor and an electric motor were standing together on a steel base and were isolated from the floor by AV mountings at the four corners, and you placed your foot on the base to bounce it up and down and get the mountings oscillating freely, they would do so at their Natural Frequency.

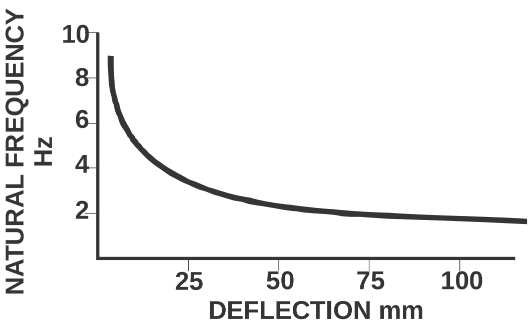
In the case of a steel coil spring (which has little damping) its natural frequency when compressed is independent of its size or shape, or o f the wire diameter or number of coils, but is determined solely by the amount by which it is compressed, according to the following simple equation in which d is in mm and f_n is in *Hz*.

$$f_n = \frac{15.8}{\sqrt{d}}$$

The curve for this equation is shown below and some approximate solutions are given in Table (1).

Table (1)

Deflection 'd' [mm]	Natural Frequency [Hz }
2	11
3	9
5	7
10	5
25	3
40	2.5
60	2
110	1.5

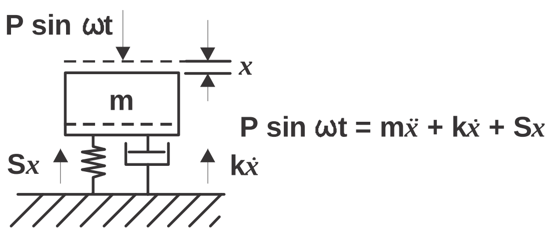


With rubber mountings deflections somewhat higher than those shown above are necessary to produce the same natural frequencies.

2. EFFICIENCY AND TRANSMISSIBILITY

The Efficiency of an AV mounting is the percentage of the unbalanced force that it isolates. Conversely the amount that gets past is called the Transmissibility, with 1.0 representing full transmissibility or zero efficiency. Alternatively transmissibility can be expressed as a percentage so that, for example, 90% efficiency equals 10% transmissibility.

In every vibration textbook the simplest model of forced vibration by an unbalanced machine is represented by the following single-degree-of-freedom system.



The spring supported mass m is displaced a distance x by an exciting or disturbing force P which varies harmonically with time. The spring, assumed to be standing on a rigid floor, exerts a restoring force S_x where S is the spring stiffness. Internal friction or damping exerts a further force which is assumed to be proportional to velocity.

From the well known differential equation of forces shown above it can be derived that transmissibility, the ratio of the transmitted force to the exciting force, is given by :

$$T = \sqrt{\frac{1+4D^2Z^2}{\left(1-Z^2\right)^2+4D^2Z^2}} \quad \text{--(2)}$$

where $Z = f_d / f_n$, the ratio of the disturbing frequency of the rotating machine to the natural frequency of the mounting, called the frequency ratio or sometimes the tuning ratio, and D is a damping factor dependent mainly on the spring material.

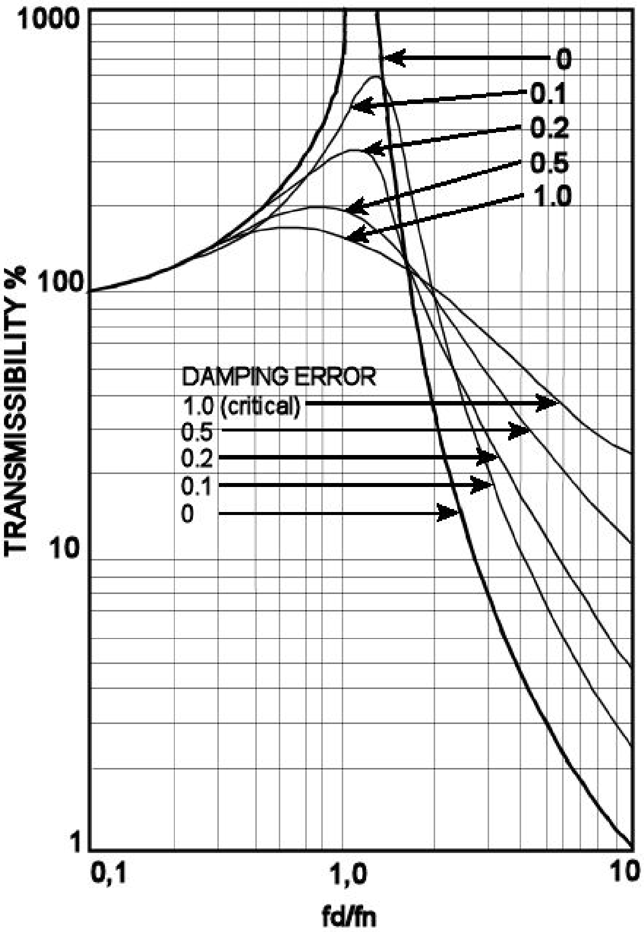
If the damping factor is zero or very small so that the term $4D^2Z^2$ can be ignored then, expressed as a percentage.

$$T = \sqrt{\frac{100}{\left(f_d/f_n\right)^2-1}} \quad \text{--(3)}$$

The efficiency can be expressed as :

$$E = 100\left(1-\frac{1}{\left(f_d/f_n\right)^2-1}\right) \quad \text{--(4)}$$

The transmissibility equations (2) and (3) can be better appreciated if show graphically. The light lines represent damping factors of 0.1; 0.2; 0.5 and 1.0 as in equation (2) and the heavy line represents zero damping as in equation (3).



These curves shows that:-

2.1 At frequency ratios < √2, that is 1.414, the transmitted force is more that if there were no AV mountings. At or near a ratio of 1, the condition called Resonance, the force transmitted with zero damping is theoretically very large, but is progressively reduced as the damping factor increases. Amplitude response is similar to force response.

2.2 Vibration isolation only occurs if the frequency ratio exceed √2.

2.3 At frequency ratios > √2 damping reduces an AV mounting's efficiency. That means when there is damping more deflection is needed than when there is no damping to achieve the same efficiency.

2.4 The function of AV mountings is to reduce the value of f_n so as to produce a frequency ratio > √2, the higher the ratio the better (within limits).

3. DEFLECTION

It follows that for any disturbing frequency (speed of unbalanced machine) and any required transmissibility the AV mountings must, in order to provide the necessary frequency ratio, have a particular deflection under load (or in practice that must be the minimum deflection).

Typically about 33% transmissibility (frequency ratio 2.0) is at the lower end of industrial acceptability, 12% (frequency ratio 3.0) near the upper end. Transmissibility of 5% is quite a high standard, at or close to level of acceptability of vibration from air conditioning equipment, pumps, fans, and compressors operating in a hospital or superior office block. Transmissibility of 1% or 2% represents the standard for a broadcasting or sound recording studio.

The following table shows the mounting deflections required to produce the transmissibility referred to in the previous paragraph at machine speeds of 3000, 1500, 1000 and 500 RPM according to equations (1) and (3), i.e. for zero damping.

Table (2)

Required Transmissibility [%]	Necessary Frequency Ratio	Required Natural Frequency cpm [Hz]				Min.Mounting Deflection [mm]			
		Machine Speed [rpm]				Machnie Speed [rpm]			
		3000	1500	1000	500	3000	1500	1000	500
33	2	1500 (25)	750 (12.5)	500 (8.3)	250 (4.2)	< 1	1.5	3.5	14
12	3	1000 (16.6)	500 (8.3)	333 (5.6)	167 (5.6)	< 1	3.5	8	32
5	5	600 (10)	300 (5)	200 (3.3)	100 (1.7)	2.5	10	23	90
2	7	430 (7.1)	215 (3.6)	140 (2.4)	71 (2.4)	5	20	44	175
1	10	300 (5)	150 (2.5)	100 (1.7)	50 (0.8)	10	40	86	360

4. GETTING FROM THEORY TO PRACTICE

4.1 We have already established that transmissibility is influenced by the degree of damping. Damping is a dissipation of energy by conversion of internal friction to heat with every cycle, as can be represented by a hysteresis loop. Hysteresis is a measure of energy lost and is the opposite of resilience which is a measure of energy stored and then returned. If the exciting force were to be removed damping would cause the amplitude of oscillation to decay to zero over a number of cycles. The term Critical Damping is used to express total decay occurring in only one cycle and it is then convenient to express other levels of damping, represented by slower rates of decay, as a ratio or percentage of critical damping.

Critical damping is very heavy damping. Natural rubber mountings under moderate strain, depending on the hardness and composition of the compound, usually have up to about 10% of critical damping, and synthetic rubbers somewhat more. For practical purposes steel springs operate close to zero damping.

4.2 The dynamic stiffness of a spring is higher than its static stiffness. The difference is higher for a rubber spring than for a steel spring, higher for a hard rubber than for a soft rubber and higher for a synthetic rubber than for a natural rubber.

In broad practical terms the dynamic ratio is negligible for steel springs and, very much depending on the shape of the mounting, usually about 1.3 to 1.4 for 40 IRHD hardness natural rubber and about 1.5 to 2.0 for 60 to 70 IRHD. For synthetic rubbers the ratios are higher still.

Again the fact is that to achieve a desired dynamic natural frequency (hence transmissibility) a rubber mounting's deflection must be more that the theoretical deflection by some factor.

4.3 The theory assumes that the floor or structure on which the AV mountings stand is rigid. In fact it is because it is not rigid that we have a vibration problem. Lack of rigidity is particularly the case in modern building construction. A high-rise building may consist of a central tower for lifts and services from which the floors are cantilevered with no perimeter support. The walls may be glass or aluminium panels and with open offices there may be few if any structural walls! It does not take much vibration from equipment in such flexible buildings to disturb the people who work or live in them.

Depending in the type of construction, on the floor span and on the location of the equipment (in the centre of the span or close to the structural wall) the floor deflection under a machine may be considerable, may even exceed the theoretical mounting deflection. Obviously the deflection of AV mountings in these circumstances has to be increased by some factor to make them softer than the floor.

4.4 Table 2 reveals the wide range of theoretical deflections necessary for AV mountings to achieve different transmissibilities at different machine speeds with zero damping. For example it shows that 5% transmissibility theoretically requires 2,5mm deflection at 3000rpm, or 10mm at 1500rpm, or 23mm at 1000rm, or 90mm at 500rpm. Reducing transmissibility from 12% to 5% requires deflection to be roughly trebled, and reducing further to 1% requires deflection to be increased again about 4 times. In relation to such large differences it becomes necessary (unless the situation demands further analysis and can support the cost) to make allowances for damping, dynamic stiffness and floor flexibility by increasing the static deflection above the theoretical deflection without knowing with precision what the allowances should be.

This is particularly the case when even the required transmissibility may be difficult to decide. In terms of energy transmitted to a building a large machine running at a particular speed is not the same energy problem as a small machine running at the same speed. The large machine will require a lower transmissibility (more deflection) to reduce transmitted energy to the same level. On the other hand if the small machine was in a room next to a recording studio and the large machine was far away in the basement perhaps the smaller machine would need the extra deflection. Transmissibility is no more than a ratio.

Calculations arriving at precise solutions should therefore be treated with suspicion.

In practice the decision as to the mounting deflection required for the satisfactory isolation of a particular machine in a particular location of a particular building is usually done empirically based on experience, and proven practice. The decision (how much deflection is required considering all factors to achieve a desired transmissibility) is made easier by the fact that AV mountings fall into a few classes, and the options are fairly clear cut

Steel springs are commercially available in three sizes giving up to 25, 50 or 75mm deflection, with natural frequencies ranging from 4Hz down to less than 2 Hz.

Rubber mountings generally give between 2 and 10mm deflection, although a few large and expensive types are capable of 20mm or more. It is not too difficult to decide whether to choose the 2 to 3mm category (the lowest level of vibration isolation), or the intermediate 4 to 6mm category, or whether to step up to 7 to 10mm (at which deflection dynamic natural frequency is around 6 to 12Hz depending on shape and hardness).