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The design of resilient mounting systems to control machinery noise in buildings

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Abstract: Vibration isolating systems have developed slowly over the last 70 years, but much of the knowledge of high frequency properties and the methods for designing and analysing high efficiency compound systems has only been gained in the last decade.

The specification of systems and devices does not often reflect this improved understanding. Greater information concerning material properties and the use of computer analysis methods has permitted the economical design of more efficient mounting systems making particular use of the specific properties of natural rubber and other resilient materials.

Introduction and history

Early resilient mountings were developed from leaf springs used on horse drawn carriages and early rail vehicles in the eighteenth century but it was not until the First World War that the need for good isolation of vibration became apparent for vehicles and aircraft. This was quickly followed by machinery in warships and fans and pumps in the better quality of buildings.

Between the two World Wars, steel coil springs and cork plates were the best resilient devices available. Although not widely used, rubber-to-metal bonded components had made an appearance. Spurred on by the rise in 1950s American skyscrapers and plate glass office blocks, air conditioning became more fashionable and building services noise became a problem to people other than just BBC sound engineers. All sorts of problems arose including electron microscopes shaken by tube trains and hotel guests unable to sleep because of the noise from bathroom extraction fans. New devices were developed like sprung inertia bases and air springs and even rubber in shear unit mountings, but little was understood of their application other than the basic theory[1] of vibration isolation at frequencies up to about four times the rigid body resonances.

Basic theory

Figure 1 shows a single degree of freedom model of an isolated system, as found in all the text books. This simple approach enables a response with frequency to be produced — Fig. 2. The height of the peak and its width are modified by the damping

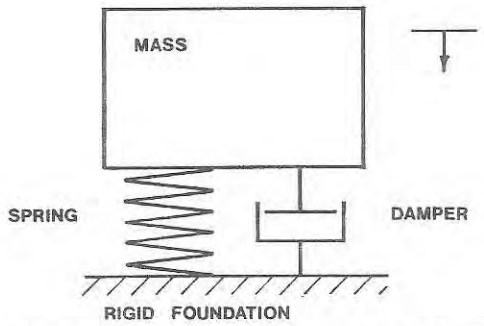


Fig. 1 Classic single degree of freedom mass-spring-damper system.

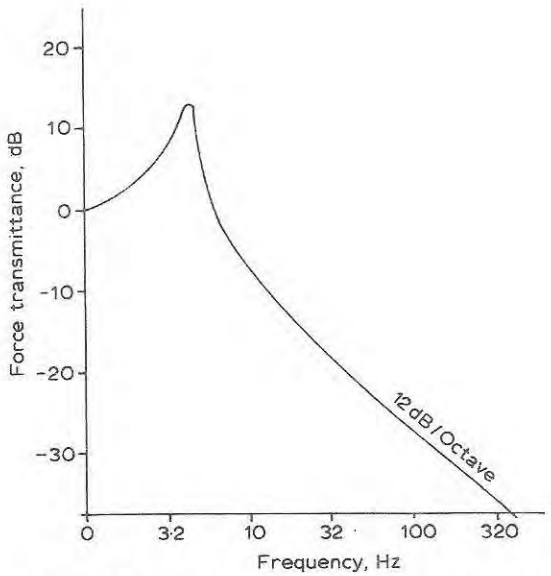


Fig. 2 Single degree of freedom frequency response, simple mounting system.

of the system. In general, increased damping provides less resonant vibration, but more force transmission at high frequencies. The damping may be produced by hysteresis within the material or by viscous dampers or, indeed, by the flow of air around the unit. In the case of a simple system on steel springs, there is very little damping and magnification of vibration at resonance is enormous.

Coupled modes

On a real body there are six degrees of freedom, the vertical, longitudinal and transverse and the rotational modes of pitch, roll and yaw. These may be coupled in pairs, threes or even all six coupled together depending upon the degree of symmetry of the machine. Coupling may be visualised as the effect of a force along one axis creating either motion along a perpendicular axis or a rotation. This is due to the spring forces not being balanced as can be seen in Fig. 3. The same principle is seen in a car cornering where the horizontal force at the tyres causes roll.

The coupled modes will exhibit a range of natural frequencies and Fig. 4 shows the single peak replaced by a number of discrete values resulting in a much wider frequency band in which problems can occur. The general approach of specifying only a vertical natural frequency for the system can clearly provide a very inaccurate assessment of the system performance and, in the case of machines which are tall and narrow and hence relatively unstable, very high natural frequencies can occur despite using low stiffness mountings.

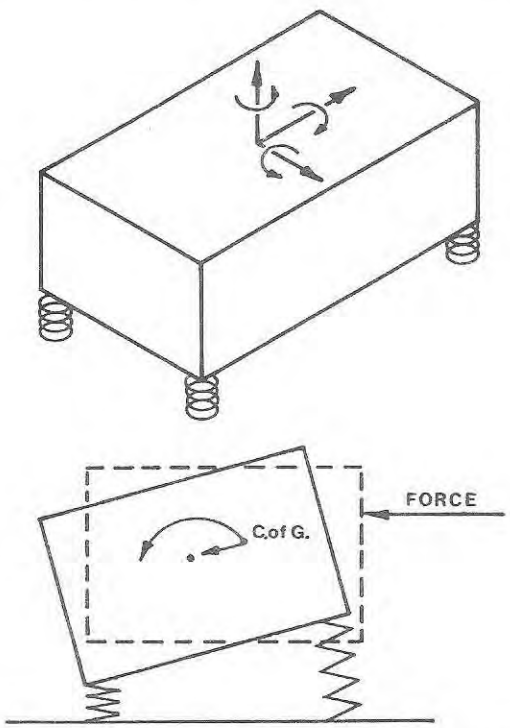


Fig. 3 The coupling of modes on a three-dimensional mass.

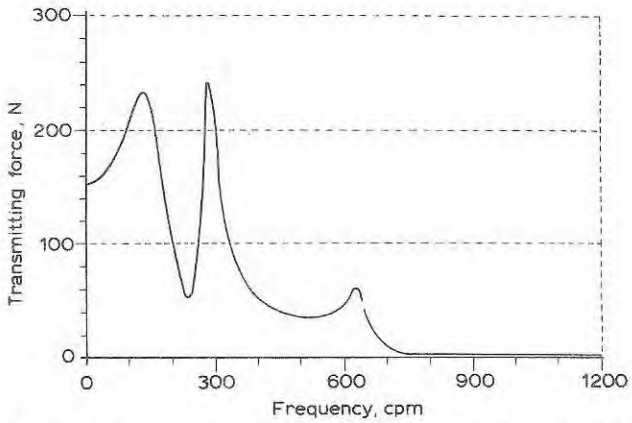


Fig. 4 A typical response with frequency of a mass with coupled modes.

Nowadays computer programs are used to calculate the natural frequencies and response to forces.

Coupled modes for two masses

These principles can be adapted to the case of two masses with springs between them, as for a machine sitting on a floor plinth which itself is connected to the building by the flexible floor. Occasionally, the machine may be mounted onto a secondary mass or raft as in Fig. 5. This raft might form a complete isolated plant room floor. Figure 6 compares vertical resonance for one and two masses using springs of similar stiffness. Although the high frequency performance looks much better with two masses there is a region between the lines where the lower mass resonance reduces the performance by, in this case, about 10 dB. Of course, a real floor may not resonate at the machine running speed and hence the reduction in performance may not be so great, but even so, the effect of the coupling of the various modes on each mass will often reduce the apparent performance of the mountings under the machine and this is difficult to calculate.

The Americans have developed some empirical ways of relating the mounting stiffness to the floor span and thickness to avoid such problems, this is often termed the Ashrae[2] method. In general, this holds that the thinner the floor and the larger the span, then the larger should be the static compression of the mounting. But these methods are undermined by yet more resonances which we will consider later.

FE analysis

There are cases where the building structure can have its resonant behaviour predicted and a well

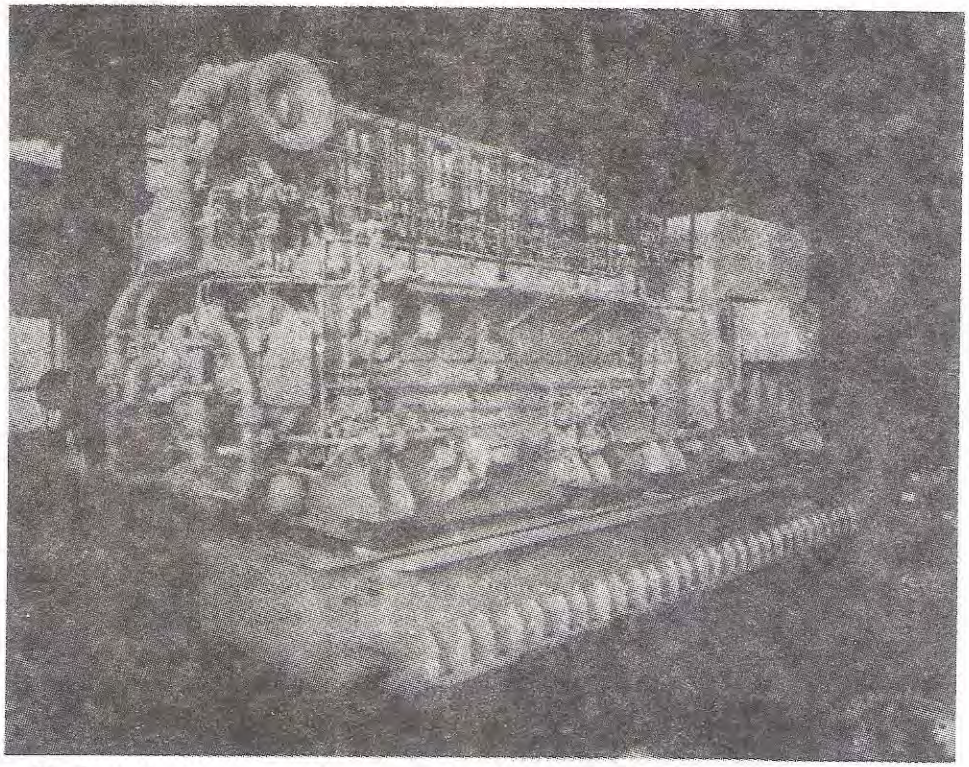


Fig. 5 A diesel generator and concrete and steel raft forming a compound mounting system.

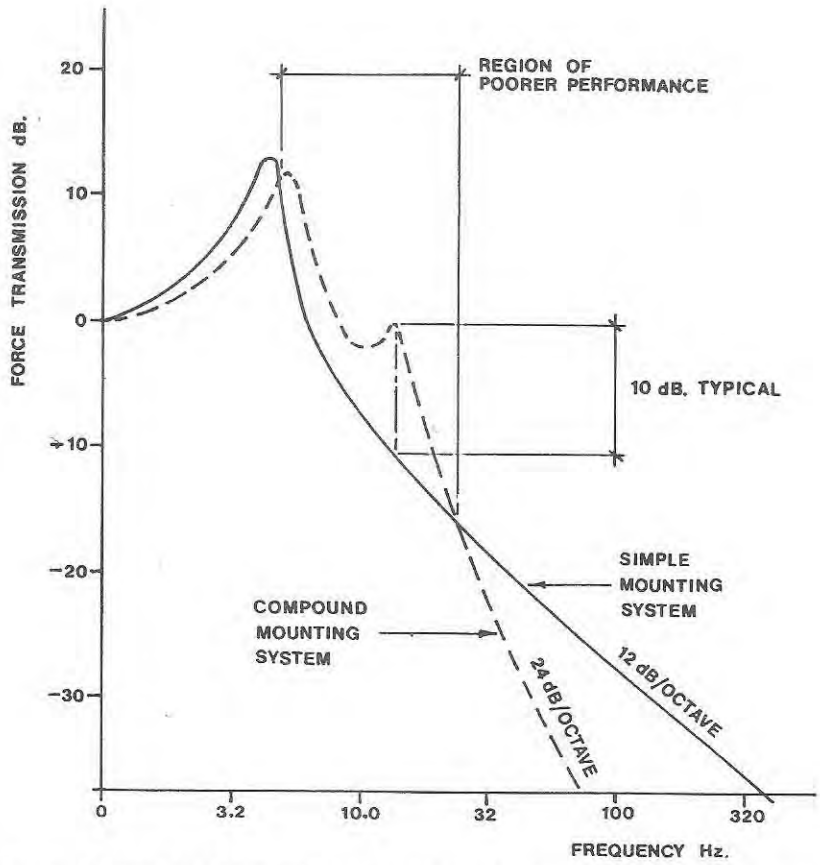


Fig. 6 Comparison of vertical responses for single and compound mounting system.

known, if expensive method, is finite element analysis. Complex grids of beams and springs and plates are modelled on the computer and estimates made of the local floor natural frequencies and the whole building bending and torsional modes. Unfortunately, these tend to show that so many resonances can exist over such a wide frequency range that they do not assist the engineer, as for example in selecting mountings for a pump which has running speeds very close to normal floor resonant frequencies.

The finite element methods tell us that mountings are never as efficient as predicted by the single degree of freedom models, and provide a qualitative guide to the effect of various design changes. At present, finite element analysis appears to be of most use in predicting response to transient effects such as earthquakes and for analysis of limited structures such as generator set frames and rafts.

Standing waves

Returning to the properties of flexible mountings, there is a feature of more importance than all the text book analysis so far considered. Any machine, spring or even 'rigid' support possesses standing waves within its material, excited by high frequency vibration forces. This topic is covered in more detail in an accompanying paper.[3]

In the case of steel coil springs, this effect is a surge of the coils which is akin to the transverse vibration of a violin string, but due to the shape of the spring, these surges are mainly axial.

Spring coils are relatively heavy with very little damping, with the result that resonances can create forces on the end of the spring at quite low frequencies. These may be larger than the exciting forces from the machine. Increases of more than 30 dB have been seen (and heard). For a spring of 25 mm static deflection with loads of 100 kg, the frequencies can be from as low as 150 Hz. Nor is there not just one resonance — harmonics exist greatly reducing the spring efficiency from the primary surge resonance to several kilo-Hertz.

So, the widespread method of specifying, say a 6 in deflection coil spring on a 6 m span suspended floor will often result in reasonable efficiency below about 60 Hz which happens to be a region where few fans or pumps produce much energy. At the awkward frequencies between 300 and 1000 Hz where the ear is most sensitive to machinery noise, such systems are at their worst. A number of methods have been used to try and improve matters. The most widely known is a 'Sound Pad' such as the ribbed Nitrile rubber seating pad of Fig. 7, which is designed to have a very different impedance from the spring itself and so not transmit much energy. Tests show that it is of only limited effectiveness.

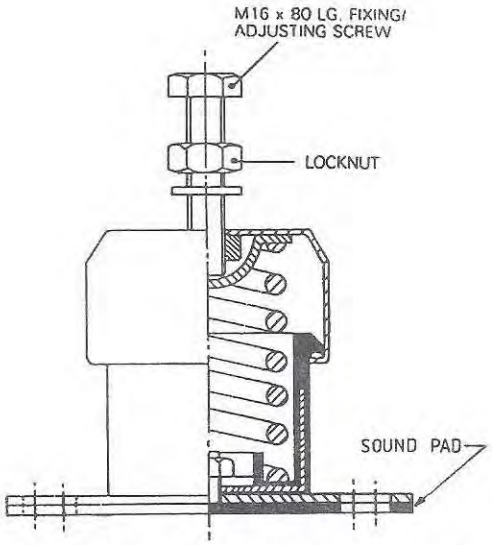


Fig. 7 An enclosed captive spring mounting.

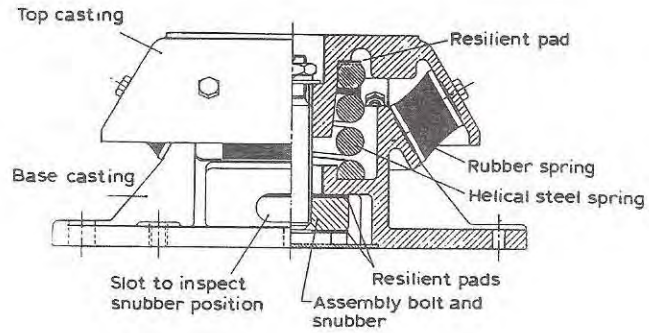
We have investigated a number of cases, particularly beneath roof mounted reciprocating chillers, where the solution to structural noise transmission has been to replace the steel springs with stiffer natural rubber mountings. Of course, these have standing waves just as the steel springs do, but these occur at higher frequencies and are more effectively damped by the hysteresis within the material.

Figure 8 shows a Christie & Grey combined rubber and steel spring mounting and a comparison of the spring and rubber surge resonances. The spring is slightly stiffer at low frequencies than the rubber but the overall effect in force transmission is that the rubber is much superior. Instead of a 20 dB magnification at 300 Hz we have about 15 dB magnification at 1200 Hz and this is a much less important region for machinery vibration.

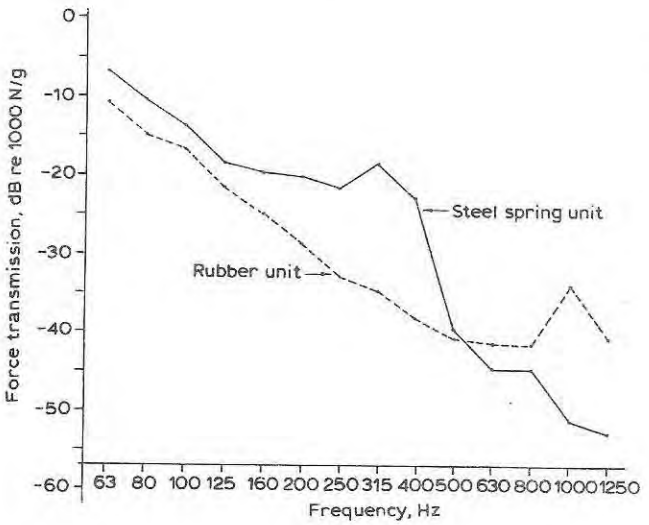
Typical applications of mountings

The theories of flexible mounting performance can be applied to all isolated systems, but they do not actually select the mountings for the engineer. Experience is needed to balance the various conflicting factors such as efficiency, cost, response to transient forces and durability. Typical design factors to be considered are listed below.

(a) Specification of force isolation.
  — as attenuation — dB
  — as force isolation percentage
  — as efficiency
  — as noise level in adjacent room
(b) Natural frequency range to achieve the isolation required.
(c) Load distribution of the machine.
  — is it equal on each mounting?



(a)



(b)

Fig. 8 (a) Christie & Grey type TSC steel spring and rubber mounting, and (b) high frequency transmission of the elements.

— is the centre of gravity low enough for stability?
— exposure to wind, snow, pipe forces, etc.

(d) Vibration amplitudes — low frequency.
— normal operation
— fault conditions
— starting and stopping
— is an inertia base needed?

(e) Higher frequency structure borne noise (100 Hz +).
— is there a specification?
— what is the building structure and use?
— is this really important?
— do we have data on the machine and probable mountings?

(f) Transient forces.
— are there shocks, earthquakes, machine transients?

(g) Environment.
— temperature
— humidity
— chemicals
— other

Example 1 — a refrigerant compressor

Consider the design of an isolator system for a typical reciprocating compressor in Fig. 9. Perhaps the machine is inexpensive with a poor standard of balance and little information is available from the manufacturers concerning the forces produced. Nor do the purchasers of the mountings wish to pay for extensive analysis or even for a sophisticated mounting system. Probably the first design requirement will be to limit the movement at start-up and when running. Many compressors run at less than 1000 rpm and steel springs with 20–25 mm deflection will provide natural frequencies of less than one-third of this, giving good attenuation of the low frequency harmonics. In the absence of a fully specified force isolation requirement this would usually be satisfactory.

The springs might be better with some additional damping, or the use of a concrete inertia base might provide sufficient limitation of movement that the lack of damping in the springs will not be of consequence. A computer calculation of vibration amplitudes is called for. For the machine shown with no inertia base, mountings with felt or rubber damping pads are used, but these are more expensive and need careful installation. Rubber mountings are not generally available for such large deflections and the final choice may be that which has been found to work best in the past.

Example 2 — air handling units

Machines which produce less out of balance forces at low frequency require a different emphasis. Figure 10 shows an air handling unit containing centrifugal fans and various filters and coils. The fan and motor impeller vibration will be mostly below 100 Hz and may be considerably reduced using steel springs with natural frequencies between 2 Hz and 8 Hz. The turbulent flow of the air will induce high frequency vibration in the air handling unit structure and this will be removed more efficiently by using rubber devices beneath the complete unit. These rubbers will also reduce any high frequency vibration caused by the standing waves in the spring isolators. Flexible bellows are used to prevent flanking transmission through the ductwork.

The types of machine in these examples use different isolation devices for sound engineering reasons. However, there are many cases where a fairly stiff rubber mounting with good high frequency performance is expected, when in fact steel springs and inertia bases are specified. We will now therefore turn to the benefits of rubber as against steel although it must be said that the final selections are not always as logical as might be hoped for in an engineering design process.

Rubber versus steel springs

We have introduced one case where the use of a rubber device of apparently poorer performance can replace a steel spring. Most rubber mountings are fairly simple devices to look at. The rubber can be chemically bonded to most metals and self-contained mountings of one piece are practical in most shapes and sizes. A wide range of synthetic materials is available but modern compounds of natural rubber accept most environments including those contaminated with salt water, mineral oil splash, radioactivity and a temperature range from −50 to about 80°C. By comparison, a building plant room should cause few problems. The synthetic rubbers can provide even better environmental performance, but unfortunately have been found to give disadvantages in terms of creep and high dynamic stiffness and so are not chosen as frequently in this country. In Germany and the USA the availability of natural rubber appears to be less and the synthetics have wider acceptance, but it must be said that in those countries, steel spring systems are also used extensively.

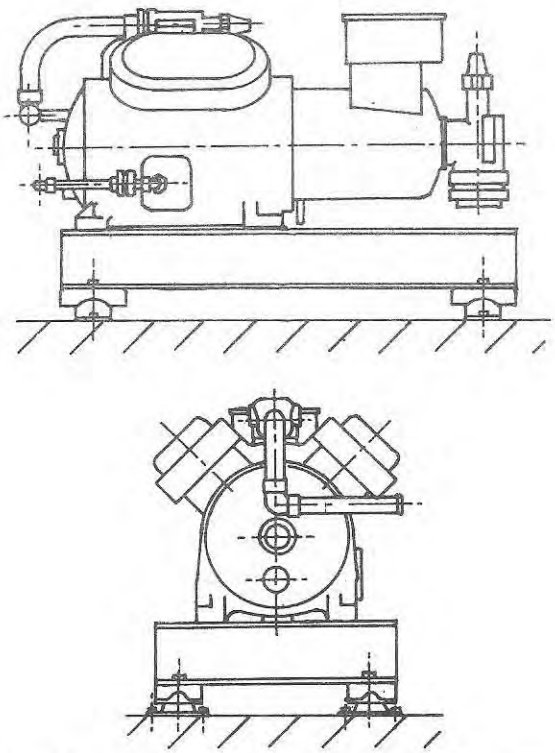


Fig. 9 A direct drive refrigerant compressor on captive spring mountings.

Steel springs may be used over a very wide temperature range and incorporated in mountings providing large static deflections. The effect of overloading can be easily viewed because the coils close up and until they are virtually solid there is no significant increase in stiffness.

An overloaded rubber mounting may be indistinguishable from one properly used, although the stiffness may have increased substantially and the subsequent creep rate may be excessive. Since steel springs do not creep significantly many consultants will specify them despite other disadvantages.

Table 1 lists some important properties for qualitative comparison.

Natural versus synthetic rubber

Now the disadvantages of rubber are also fairly well catalogued particularly for synthetic materials which have very high dynamic stiffnesses compared to their static values. This effect is apparent at all frequencies and leads to typically 50% more force transmission for a given condition than with a natural rubber product and perhaps 80% more than an equivalent steel spring. At the same time, the synthetics tend to show high creep rates and this will often prevent their choice when other factors are acceptable. For plant room applications which are

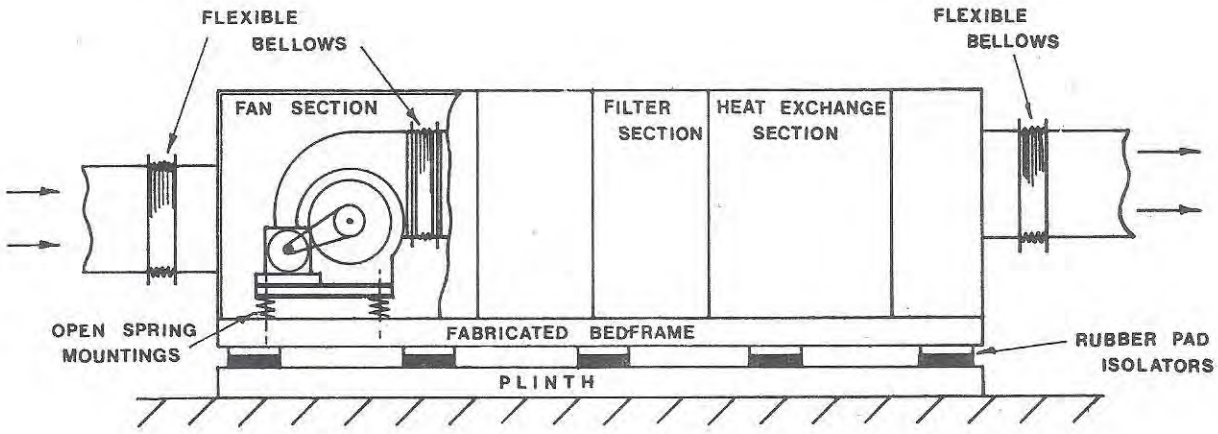


Fig. 10 The compound mounting system comprising the fan and frame of an Air Handling Unit.

Table 1. Comparison of mounting types

	Rubber		Steel springs	
	Good	Poor	Good	Poor
High frequency attenuation	*			*
Low frequency attenuation		*	*	
Ease of fixings to housings	*			*
Internal damping	*			*
Linear stiffness		*	*	
Dynamic: static ratio		*	*	
Manufacture of complex designs	*			*
Tolerance to high temperature (up to 200°C)		*	*	
Tolerance to low temperature (down to −50°C)	*		*	
Creep rate		*	*	
Resistance to hydrocarbons, sunlight, ozone, ageing		*	*	
Corrosion resistance	*			*
Ease of installation	*		*	

the main use of mountings in buildings, the synthetics are often ruled out even in areas when oils and other hydrocarbons would make them an obvious choice.

Special types

Special purpose isolators are available for cases where the standard units do not provide either sufficient performance or can interfere with the operation of the machine. Examples include large deflection steel springs with viscous dampers for use beneath generator sets to ensure very low levels of low frequency vibration.

Airsprings (in effect rubber bellows full of compressed air) with position sensors and servo-valves can provide a reasonably accurate maintenance of level to a machine which could be of assistance for laboratory instruments or for production test equipment.

Active mountings may become important one day, but as yet are still experimental.

As design methods and the products themselves have become better understood, the specialist products have found fewer uses since frequently the problems which have to be overcome can be handled by standard mountings at a great saving in cost. Indeed, the cost of the mounting system is very frequently an overriding factor and in a case where the specification is not very critical, there is considerable pressure on the engineer to provide a simple and cheap support which is a good compromise between the various factors which have to be met in the specification without necessarily providing the optimum solution to vibration control.

Inertia bases

We have mentioned the Ashrae methods which led to the selection of high deflection steel springs. Such systems can suffer excessive motion due to resonance or transient forces, leading to the use of added mass to control such movement.

In the 1960s most engineers disliked adding lumps of concrete or steel to a machine, reserving such systems for Power Station engines, compressors or large low speed fans where the forces made a directly isolated machine vibrate excessively. Reinforced concrete, cast on site, was often used with all the expense of customised reinforcement design, site shuttering and special jacking systems for installation.

Nowadays, inertia frames are used frequently for small machines — as on Fig. 11. These consist of a steel perimeter beam with brackets for the mountings and with a reinforcing mesh to ensure an adequate key for the concrete filling. Frames are very difficult to handle when larger than about 3 m long and, in many cases, they are not really necessary since the machines would sit quite happily on four or six rubber mountings. They find favour because installation is a builder's job (i.e. pouring the concrete) and the exact weight of the machine on top is unimportant, being relatively small. Even its position is not of great consequence. Pipe forces or duct pressures are also quite small compared with the

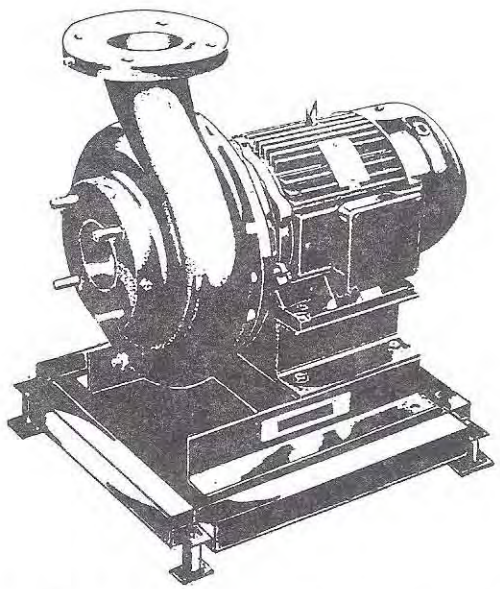


Fig. 11 The use of an inertia frame beneath a water pump.

weight of the inertia base and usually they can be discounted.

Bases are often found under small centrifugal pumps where £500 may be spent on the frame, isolators and concrete when four rubber mountings at £10 each would give at least as good a performance. Unfortunately, on a poor installation the static pressure in the pipes would flatten the rubber mountings and proper design would have to include tie bars on the pipe flexibles to avoid excessive distortion and flexible hangers on the pipes to reduce flanking transmission.

Pipe/duct noise

Whilst machinery mounting is reasonably well understood, the reduction of pipe and duct noise is still a black art, with most design based on experience. The conventional wisdom is to isolate a pipe for about 20 diameters away from the pump or perhaps for all pipes within a plant room. Little is properly understood about the relative importance of water borne as opposed to pipe wall borne vibration. It is clear, however, that resonances in both the pipe and the supporting structure can cause some installations to be much worse than others.

Most hangers consist of a frame, containing a steel spring or rubber mounting, which can be incorporated in a drop rod. As with machine mountings the effect of resonance of either the pipe or the supporting ceiling is very important, but few measurements exist. The use of finite element techniques could provide valuable information for designers, but the cost is prohibitive in most cases.

Environmental considerations

The main purpose of the vibration engineer's analysis is to select mountings to provide a proper reduction in noise and vibration force transmission, with due regard to the consultant's specification. Nonetheless, many other factors must be taken into account if the system is to provide a good engineering solution to the problem of supporting vibrating machinery. Questions must be asked:

(1) How easy is it to install the system?
(2) Will it last long enough?
(3) How much trouble will it be?

Most machinery areas are cool and dry, but some are very humid, some out of doors. We may need to allow for hurricanes and snowfall. A good paint finish and protection against oil contamination may become more important than selecting the optimum isolation performance.

Natural rubber lasts longer than synthetics for most applications, but it cracks in sunlight if not shrouded and can be badly affected by oil drips particularly from hydraulic pumps.

Steel springs may rust but they do not creep and are unaffected by most organic chemicals. Inside plant rooms, rubber mountings will usually last for 10–15 years whilst steel springs are good for 20 years and sometimes much longer. Frequently the superior properties of rubber as a vibration isolator outweigh the problems given by creep and chemical degradation.

Conclusions

We have seen that there are a number of practical devices available for the control of structure borne vibration and high frequency noise, the application of which has become a complex matter requiring specialist knowledge in order to provide a good solution to conflicting requirements.

In many cases, the materials and designs have been available for many years, but the methods of application and the understanding of the performance over a wide frequency range have only recently been properly understood. This has not yet been fully appreciated throughout the industry and many specifications are still written around devices which do not provide the optimum performance.

Calculation methods are available by which the engineer may ensure that the vibration performance of the machine and the effect it has upon the building are properly understood to give confidence that the design will meet the specifications.

As standards and expectations increase, the mounting system designer and the architects and building consultants need to learn more of both the limitations and capabilities of mounting systems.

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