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Proceedings of The Institute of Acoustics

COMMON PROBLEMS IN VIBRATION ISOLATION

P. H. ALLAWAY

PARTNER, GROOTENHUIS ALLAWAY ASSOCIATES
MANAGING DIRECTOR, ALLAWAY ACOUSTICS LTD.

The practical problems in designing vibration isolation systems for mechanical services plant include economics as well as vibration engineering. Even as recently as 20 or 30 years ago it was conventional in the mechanical services industry to mount machines on a concrete block supported "on 2" thick builders cork"; this technique was not too unsatisfactory where machines were mounted at ground level on solid foundations but the modern trend towards mounting mechanical servicing plant high up in buildings and the trend towards light weight construction are against such a simple solution. The services engineers who, on the drawing board, were showing such simple systems are - in some cases - now in senior positions in mechanical services consultants and contractors and some still yearn for the simplicity of the old specification. Most vibration engineers would agree that we are winning the battle to get properly engineered vibration isolation systems, but it is a long, hard battle and there are potential trip-wires which can floor the unsuspecting practitioner who incautiously advances. Let us consider some of these trip-wires and see how high one must raise one's feet to step over them.

1. Multiple spring-mass systems, or "Floppy floors and floating structures".

A simple vibration isolation system is shown as a mass supported on a spring and damper from a rigid base (Fig. 1). In practical terms if a base <u>is</u> rigid then no vibration isolation is necessary because no amount of force transmission will move it; we must therefore deal with at least two spring-mass systems in series and more than two in many cases. A simple, but real, case would be a machine supported on springs mounted on a suspended floor. The suspended floor has stiffness and an effective mass; can we quantify these factors at the design stage of a project when decisions must be taken and costs established ?

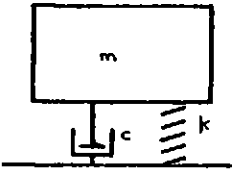

Fig. 1.

The stiffness of the floor will control its natural frequency and estimates of the floors mid-span deflection under self and imposed loads can be obtained from the structural engineer or made by assuming that an allowable deflection of, say, 1/325 of the span will be reduced to 1/650 of the span by the inherent conservatism of structural engineers who, happily, do not like floors to collapse. Given this deflection, the fundamental natural frequency of the floor can be assessed by :-

$$f_{nat} = \frac{1}{2\pi}\sqrt{\frac{g}{\delta} \times \frac{4}{\pi}} \quad \text{(Hz.)}$$

$$\text{or} \quad = 0.56243\sqrt{\frac{1}{\delta}} \quad \text{(Hz.)}$$

with δ = deflection (m)

Now, for real floors supporting mechanical services plant, the fundamental natural frequency is unlikely to be less than 10 Hz. or more than 35 Hz (deflections of about 3.0 and 0.3mm respectively) and this range includes the frequency, or rotational speed, of many typical machines used in mechanical services. If the isolators under the machine have, relatively, very low stiffness the impedence seen by the floor is very small and the floors dynamic response will be little affected by the presence of the machine.

Suppose now that the machine speed is the same as the floors fundamental natural frequency, what is the consequence ? However effective the low stiffness springs under the machine may be, the residual force which is transmitted to the floor will be at the floors natural frequency and the floor must respond in resonance to an extent controlled by its damping and mass. For good concrete, the damping is small (Q factor of 15 to 25, say) and the mass is not generally adjustable at the behest of the vibration engineer.

It is clear that even if the very large spring deflections (100 - 125mm) proposed by some vibration isolator manufacturers are adopted, there will still be resonance and, probably, a vibration or structure-borne noise problem. A large spring deflection is still beneficial but introduces stability problems.

Solutions may lie in persuading the structural engineer to stiffen the floor so that resonance is avoided or in increasing the damping (floating a screed on bitumen, perhaps). An elegant, theoretical solution would be to interpose a third spring mass system between the machine and the floor. This might be a "Floating Floor" (1) If it is designed so that, as a "single degree of freedom" unit, it would have the same natural frequency as the floor, then the floor and our new mass would be strongly coupled so that the natural frequencies would shift; one would be higher than the original floor frequency and the other would be lower. Resonance with the machine speed would thus be avoided. The spread between the two, new, coupled natural frequencies is greatest when the effective masses of the structural floor and the "floating floor" are equal. See Fig. 2.

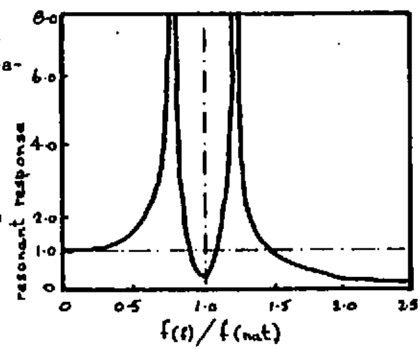

Fig. 2.

The effective mass of the structural floor is difficult to estimate. It can be formed by an area of floor up to 50% of the floor dimension in both directions; if the machine is small relative to the plan area of the floor (say less than 10%) the effective mass may be estimated by (this is guess work) :-

m = a x b x surface density of floor x x/10
where a and b = floor plan dimensions
x = percentage of total floor area occupied by machine.

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When running up to speed, the machine will pass through a resonant condition with the lower of the two coupled natural frequencies. If this system is only lightly damped the displacement amplitude and transmission will be large; it is however a consoling thought that to reach an infinitely large amplitude takes an infinitely long time ! Remember, however, that to absorb an alternating force it must be made to move a mass and an alternating moment must move an inertia (2). Rocking resonances must also be avoided.

2. Isolators-in-series or "Isolating the isolators".

It is commonly thought that if two sets of vibration isolators are used in series so that a dynamic system response of the type shown in Fig. 2 is obtained, then it is unwise to select these isolators to have the same single-degree-of-freedom frequency. Fig. 2 however, shows that there will not be a resonant response at that frequency but that the upper of the two, new, coupled natural frequencies will be relatively higher than the single-degree-of-freedom system; this must be considered when selecting the vibration isolators.

If the isolators are chosen to have very different single-degree-of-freedom frequencies there will still be a shift in the values of these frequencies due to the coupling of the systems and the higher of the two single-degree-of-freedom frequencies will move to a still higher value. There is then a greater danger of resonance between machine speed and this upper coupled natural frequency.

3. Fluid Pressure Reaction or "Stop shoving".

A machine whose operation causes fluid to flow in pipes or ducts, for instance fans or pumps, will generate a positive pressure increment in its delivery lines; conversely it will normally generate a, relatively, negative pressure in its suction lines. The machine will be acted upon by a force related to the pressure differences and the surface areas over which these pressure differences act. These forces can be quite large and, for a properly vibration isolated machine on low stiffness springs, must be resisted by the vibration isolators. The isolators must be capable of permitting the additional deflection caused by these pressure differences.

4. Pipe and duct connections or "Don't let your arteries harden".

Pipe and duct connections to vibration isolated machines add stiffness to the supports and cause increases in natural frequencies. The quality of vibration isolators will thus be adversely affected. "Canvas" duct connections have low stiffness provided that they are loose but noise-insulating-duct-connections of proprietary materials can have high stiffnesses.

Pipe connections can be very stiff especially if "control units" are used to resist stretching due to fluid pressure reaction. Some flexible pipe connection manufacturers quote values of "force per unit-deflection" so that reasonably accurate stiffnesses can be calculated.

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5. Spring surge frequency or "Belly-dancing dampers".

Helical coil springs have a resonant mode of vibration in which the two ends are substantially still but the centre coils vibrate axially - hence the belly dancing of the title. At this frequency the isolation quality will be poor whatever fundamental natural frequency has been provided. This frequency must be designed-out of any critical region. Its value is about :-

$$f\ surge = \frac{3.5 \times 10^5 \times d}{N.D^2} \quad (Hz.)$$

for circular steel wire (approximate value).
where d = wire diameter, mm
N = number of active coils
D = mean coil diameter, mm

Internal wave effects in rubber isolators can have similar effects.

6. Structure-borne noise or "Never mind the vibration-feel the noise".

Low frequency vibration in air, say less than 35 Hz, is as much felt by the recipients as heard by them. It is none the less objectionable for being below the frequency range covered by the NC and NR curves. The thresholds for perception of vibration are quite well established on a statistical basis (3)(4) but large variations between individuals can be expected. Pressure waves in air need only minute amplitudes in order to be detected by us as noise; for instance 34 dB sound pressure level is produced by pressure waves of only 0.001 N/m^2 pressure. At the right frequency and in the wrong environment this can be a pretty penetrating noise but the particle amplitude is hardly more than of molecular dimensions and much less than is perceptible by touch. The quality of vibration isolation must therefore be of a higher standard for the control of structure-borne noise than for the control of the sensation of vibration in structures.

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