MACHINE MOUNTING FOR VIBRATION ATTENUATION
Revision B

By Tom Irvine
Email: tomirvine@aol.com

April 1, 2000

Introduction

Machines produce vibration. This vibration can have a number of consequences. The question arises, "How should a machine be mounted to the floor to minimize vibration effects?"

The optimum solution depends on a number of factors as outlined in the following text. The text draws from descriptions in References 1 through 3.

Vibration Consequences

Types

An understanding of vibration consequences is a prerequisite to determining the best isolation strategy for a particular machine. One problem type is the transmission of excessive vibration from the machine to adjacent structures. The second type is excessive vibration within the machine itself.

Transmitted Vibration

Machinery dynamic forces can excite building resonant frequencies. This vibration could disturb production processes. For example, consider an air compressor which excites a shop floor. Now consider a machine tool which is mounted nearby. The machine tool may produce parts which are out of tolerance due to the unplanned motion of the floor.[1]

In some cases, the machine-induced vibration could cause fatigue and cracking in beams, floors, walls, and other structural members. The load-bearing capability of a structural member may thus be compromised.

This vibration could have a number of physiological and psychological effects on humans.[2] The vibration might produce acoustic noise that is a nuisance to nearby employees. In some cases, the sound levels may be severe enough to damage hearing. The vibration could also cause suspended light fixtures to oscillate and provoke other

---

[1] In this example, isolation of the generator would be more efficient than isolation of the machine tool, particularly since the generator might be exciting additional machines.

[2] In addition, people who handle chain saws and other power tools can develop a numbness condition called "White Finger." Vibration attenuation for hand-held power tools will be covered in another report.

optical effects which might annoy employees. In addition, sensitive employees might feel somewhat insecure walking across an oscillating floor, particularly at an upper story of a building.

Internal Vibration

Vibration generated within a machine may cause structural fatigue, fastener loosening, and other problems. Furthermore, a machine tool may be unable to meet precise tolerances if its own vibration levels are excessive.

Machine tools with automatic controls have a particular susceptibility to vibration. The vibration modes could interfere with control frequencies, thereby causing instability.

The responsibility of reducing internal vibration to acceptable levels belongs primarily to the machine manufacturer.[3] Nevertheless, internal vibration limits are a consideration in mounting method selection.

<u>Machine Vibration Output</u>

The vibration output of the machine must be characterized prior to mounting selection. Ideally, the manufacturer gives the following information:

1. The mass of the machine
2. The dimensions of the machine
3. The center of mass
4. A description of all static forces
5. A detailed description of all dynamic forces, including start-up and shutdown transients
6. A description of any force changes due to temperature changes
7. A description of closed-loop control system frequencies, if applicable

Measurements are necessary if this data is unavailable. Measurements may be needed anyway to verify the manufacturer's data and to verify that the machine is still operating in its nominal condition.[4]

The dynamic forces could be sinusoidal, random, or transient. Examples of machines and their vibration outputs are given in Table 1.

---

[3] Ideally, the manufacturer has designed the machine to produce as little vibration as possible since vibration usually represents wasted kinetic energy. An exception would be a machine which uses vibration as part of its normal function, such as certain sorting machines.

[4] Machine vibration is also monitored for fault detection. Typical faults are shaft imbalance, worn gears, damaged bearings, and loose drive belts. Early detection of developing faults allows maintenance to be scheduled in a cost-effective manner. This is a subject for another report.

Table 1. Machine Vibration Examples		
Machine	Primary Motion	Vibration Type
Fans Centrifugal Pumps Compressors Generators Lathes Turbines Washing Machines	Rotation	Sinusoidal[5]
Piston Engines Reciprocating Pumps Screening Machine Weaving Machines	Reciprocation	Sinusoidal
Forging Hammers Molding Presses Punching Machines	Impact	Transient

Obviously, some machines have several types of motion. For example, some engines have both reciprocating pistons and a rotating shaft.

The goal of this step is to characterize the machine's dynamic force in terms of a single, dominant frequency. This frequency is called the operating frequency.

Tuning

Octave Rule

The primary goal of machinery isolation is to attenuate the vibration energy flow from a machine to adjoining areas.[6] This is accomplished most effectively by "tuning."

Tuning is the process of separating the machine operating frequency from any structural frequencies. Note that the machine has a structural frequency.[7] This structural frequency

---

[5] Mass unbalance is the classic cause of vibration in rotors. This occurs when the center of mass of a rotating part does not coincide with the axis of rotation.

[6] A secondary goal might be to limit the machine's vibration amplitude caused by its own dynamic forces. A tertiary goal might be to attenuate the vibration flowing from external sources into the machine. This tertiary goal is a concern where automobile traffic and seismic excitation might disturb a machine.

[7] Consider a machine with all power turned off. The structural frequency is the natural frequency at which the machine would vibrate if it were disturbed from its resting position by a sudden, impulse force.

shall be referred to as the "isolation frequency" throughout the remainder of this report. In addition, the floor or foundation has its own natural frequency.

The frequency separation should satisfy the "octave rule" as a minimum requirement. A one-octave separation means that the higher frequency is twice the value of the lower frequency.

A case of "low tuning" exists if the isolation frequency is at least one octave below the machine operating frequency.

A case of "high tuning" exists if the isolation frequency is at least one octave above the machine operating frequency.

Measurement of the floor natural frequency is necessary for either of these methods.

Low Tuning

Certain conditions must be satisfied for low tuning to be effective.

1. The machine operating frequency should be above 4 Hz.
2. The soft support should not cause manufacturing problems in the case of a machine tool.
3. Machine start-up and shutdown transients should not cause excessive deflections or velocities.
4. The support must withstand both the static and dynamic forces of the machine.
5. The isolation frequency should be at least one octave below the machine operating frequency.

There are numerous methods for implementing low tuning. The most common methods are summarized in Table 2.

Another possible requirement is that the isolation frequency be at least one octave below the floor frequency. This would prevent dynamic coupling between the floor and the machine, with respect to vibration sources external to the machine. An exception to this possible requirement would be the hard-mount case shown in Table 2 where the floor natural frequency is used as the isolation frequency.

Table 2. Low Tuning Methods		
Method	Isolation Frequency Range	Notes
Hard-mount to Floor	2 to 3 Hz	This is only practical if the floor natural frequency is 2 to 3 Hz. Typically, a floor has a natural frequency greater than 8 Hz, however.
Helical Steel Springs	4 to 10 Hz	Springs have linear stiffness. Separate damping elements may be required.
Air Cushions	0.5 to 3 Hz	Air cushions are very effective, but the air pressure must be maintained.
Rubber Mat or Mounts	5 to 10 Hz	Rubber has nonlinear stiffness. It usually provides good damping, however.
Stabilizing Mass with Isolation Springs	4 to 10 Hz	Please see Note 1 below.

Further Notes:

1. A stabilizing mass is often required for precision machine tools. The stabilizing mass should be greater than the machine mass. A higher stabilizing mass allows stiffer springs to be used to maintain a given isolated frequency. Stiffer springs reduce the machine's own vibration amplitude, as shown by the equations in Appendix A.

High Tuning

High tuning is used for situations where low tuning is impractical. For example, high tuning might be used for a case where the vibration amplitude of a machine tool needed to be limited so that it could maintain precise tolerances.[8]

High tuning requires a rigid connection between the machine and the floor. The floor's natural frequency effectively becomes the isolation frequency.

The floor natural frequency must be at least one octave higher than the machine operating frequency to achieve high tuning.

High tuning is somewhat similar to the low tuning hard-mount method, except that the floor frequency is higher than the machine operating frequency for the high tuning technique.

[8] Again, the low tuning method of an isolated, stabilizing mass may also be used to minimize the machine's own vibration displacement amplitude. The high tuning example assumes that the stabilizing mass method was ruled out for some reason.

Conclusion

The text has given a qualitative explanation of machinery isolation. Extensive mathematical theory related to this subject is given in References 2 and 3. Some basic theory is included in Appendix A.

In many cases, the best method to attenuate a machine's vibration power output is by low tuning. This is particularly effective if the machine has a number of significant harmonic frequencies in addition to its fundamental frequency.

Furthermore, the best low tuning technique tends to be the method of mounting the machine on a stabilizing mass with isolation springs.[9] This allows stiffer springs to be used which reduces dynamic deflection.

Nevertheless, the best method for a given situation depends on a myriad of details which must be carefully considered before a mounting design is chosen. A series of measurements are required as part of this consideration.

Finally, note that a number of standards have been written relating to this subject. A summary is given in Appendix B.

References

1. H. Bachmann, et al., Vibration Problems in Structures, Birkhauser Verlag, Berlin, 1995.
2. C. Harris, Shock and Vibration Handbook 4th ed., McGraw-Hill, New York, 1996.
3. W. Thomson, Theory of Vibration with Applications 2nd ed., Prentice-Hall, New Jersey, 1981.

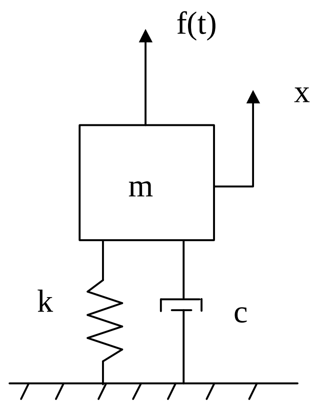
---

[9] In addition, the use of stabilizing mass offers an advantage for asymmetric machines. The stabilizing mass can be designed to co-locate the system center of mass with the geometric center of the isolation springs. This step prevents coupling between translational and rotational vibration modes. This de-coupling would be important if the isolated system behaves as a two-degree-of-freedom-system.

APPENDIX A

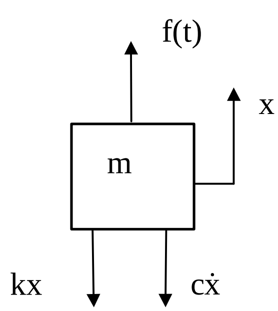
Displacement Response to Applied Force

Consider a single-degree-of-freedom system:



where m equals mass, c equals the viscous damping coefficient, and k equals the stiffness. The absolute displacement of the mass equals x, and the applied force is given by f(t). The applied force could arise from a rotor imbalance if the model represents a machine.

Free-body diagram:



Next, apply Newton's law. Summation of forces in the vertical direction

$$\sum F = m\ddot{x} \tag{1}$$

Note that the double-dot denotes acceleration. A single-dot denotes velocity.

$$m\ddot{x} = -c\dot{x} - kx + f(t) \tag{2}$$

The governing differential equation of motion is thus,

$$m\ddot{x} + c\dot{x} + kx = f(t) \tag{3}$$

Dividing through by mass,

$$\ddot{x} + (c/m)\dot{x} + (k/m)\,x = (1/m)f(t) \tag{4}$$

By convention,

$$\omega_n{}^2 = \frac{k}{m} \tag{5}$$

$$2\,\xi\,\omega_n = \frac{c}{m} \tag{6}$$

Note that ξ is the damping ratio, and that ω_n is the natural frequency in radians per second.

By substitution,

$$\ddot{x} + 2\xi\omega_n\dot{x} + \omega_n{}^2 x = \left[\frac{1}{m}\right] f(t) \tag{7}$$

The mass term on the right-hand-side can be replaced using the relation in equation (5).

$$\ddot{x} + 2\xi\omega_n\dot{x} + \omega_n{}^2 x = \left[\frac{\omega_n{}^2}{k}\right] f(t) \tag{8}$$

Now consider the case where the force f(t) is sinusoidal.

$$f(t) = F\sin(\omega t) \tag{9}$$

The forcing amplitude is F. The forcing frequency is ω.

By substitution,

$$\ddot{x} + 2\xi\omega_n\dot{x} + \omega_n{}^2 x = \left[\frac{\omega_n{}^2}{k}\right] F\sin(\omega t) \tag{10}$$

The differential equation can be solved using Laplace or Fourier transforms. The intermediate steps are omitted for brevity.

The resulting steady-state transfer function magnitude is

$$\left|\frac{kx}{F}\right| = \frac{{\omega_n}^2}{\sqrt{\left({\omega_n}^2 - \omega^2\right)^2 + (2\xi\omega_n\omega)^2}} \tag{11}$$

This equation can also be expressed as

$$\left|\frac{kx}{F}\right| = \frac{1}{\sqrt{\left(1-\left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\xi\left(\frac{\omega}{\omega_n}\right)\right)^2}} \tag{12}$$

Now consider a system with a given ξ, ω, ω_n, and F. Equation (11) shows that a greater stiffness k will yield a smaller displacement x. Note that the mass m would also need to increase to maintain a constant natural frequency ω_n, per equation (6).

This is one of the reasons why a stabilizing mass on stiff springs can be an effective vibration attenuation solution. Both the extra mass and stiffness serve to limit the displacement amplitude of the isolated machine.

A graph of equation (12) is given in Figure A-1 for five damping cases.

In addition, note that the displacement x has a phase lag θ with respect to the force f(t).

$$\theta = \arctan\left[\frac{2\xi\left(\frac{\omega}{\omega_n}\right)}{1-\left(\frac{\omega}{\omega_n}\right)^2}\right] \tag{13}$$

Note that the response amplitude in equation (12) reaches its highest value at resonance. This occurs when

$$\omega = \omega_n\sqrt{1-\xi^2} \tag{14}$$

For the case of low damping, resonance occurs when

$$\omega \approx \omega_n \tag{15}$$

Steady-state Response of a Single-degree-of-freedom System Subjected to an Applied Sinusoidal Force

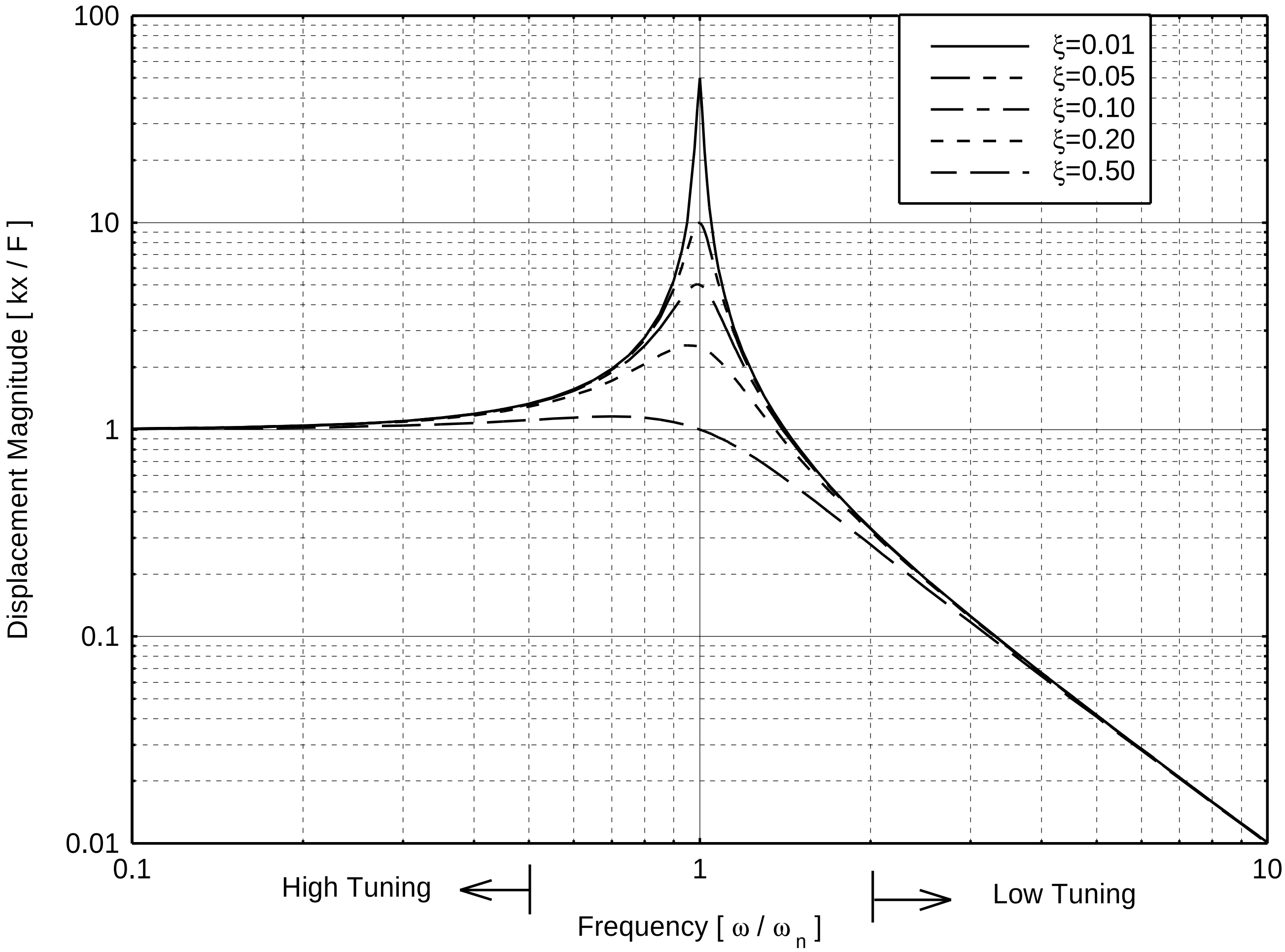


Figure A-1.

Low tuning is achieved when the operational forcing frequency ω is at least twice the value of the isolated natural frequency ω_n. In other words, the isolated frequency is at least one octave below the operating frequency.

High tuning is achieved then the operational forcing frequency ω is less than one-half the value of the isolated natural frequency ω_n. In other words, the isolated frequency is at least one octave above the operating frequency.

Transmitted Force

The magnitude of the force F_t transmitted through the spring and damper to the floor is

$$\left| \frac{F_t}{F} \right| = \frac{\sqrt{1+(2\xi\rho)^2}}{\sqrt{\left[1-\rho^2\right]^2+[2\xi\rho]^2}} \quad , \tag{16}$$

where $\rho = \left(\frac{\omega}{\omega_n} \right)$

APPENDIX B

Some relevant vibration standards are given in Table B-1.

Table B-1. Vibration Standards	
Abbreviation	Title
BS CP 2012/1	“Code of Practice for Foundations for Machinery: Foundations for Reciprocating Machines.” British Standard Code of Practice, 1974.
ISO 2372*	“Mechanical Vibration of Machines with Operating Speeds from 10 to 200 rev/s – Basis for Specifying Evaluation Standards.” International Standards Organization, Geneva, 1974. Amendment 1, 1983.
ISO 2373	“Mechanical Vibration of certain Rotating Electrical Machinery with Shaft Heights between 80 and 400 mm – Measurement and Evaluation of the Vibration Severity,” International Standard Organization, Geneva, 1987.
ISO 2631/1	“Evaluation of Human Exposure to Whole-body Vibration: General Requirements.” International Standard Organization, Geneva, 1985.
ISO 2631/2	“Evaluation of Human Exposure to Whole-body Vibration: Continuous and Shock-induced Vibration in Buildings (1 to 80 Hz).” International Standard Organization, Geneva, 1989.
ISO 3945	“Mechanical Vibration of Large Rotating Machines with Speed ranging from 10 to 200 r/s – Measurements and Evaluation of Vibration Severity in Situ.” International Standard Organization, Geneva, 1985.
ISO/DIS 4866	“Mechanical Vibration and Shock – Measurement and Evaluation of Vibration Effects on Buildings – Guidelines for the use of Basic Standard Methods.” International Standard Organization, Geneva, 1986.
ISO/DIS 10137	“Bases for Design of Structures – Serviceability of Buildings against Vibration.” International Standard Organization, Geneva, 1991.

* Replaced by ISO 10816-1