

# Practical Methods for Vibration Control of Industrial Equipment

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**Abstract:** The generally accepted methods for vibration control of industrial equipment include; Force Reduction, Mass Addition, Tuning, Isolation, and Damping. This paper will briefly introduce each method, and describe practical methods for their application. Several scenarios and case studies will be presented, with emphasis on pragmatic solutions to industrial vibration problems.

**Keywords:** vibration control

**Notwithstanding** the replacement of worn or defective components, such as damaged bearings, 5 basic methods exist for vibration control of industrial equipment, as detailed below;

- **Force Reduction** of excitation inputs due to, for example, unbalance or misalignment, will decrease the corresponding vibration response of the system.
- **Mass Addition** will reduce the effect (system response) of a constant excitation force.
- **Tuning** (changing) the natural frequency of a system or component will reduce or eliminate amplification due to resonance.
- **Isolation** rearranges the excitation forces to achieve some reduction or cancellation.
- **Damping** is the conversion of mechanical energy (vibrations) into heat.

For the simple vibrating system (single degree of freedom), shown in Figure 1A, below, the force input is comprised of 3 distinct components; stiffness, damping, and inertia. The relationship of the component forces is shown as vectors in Figure 1B, and further described in Equation 1.

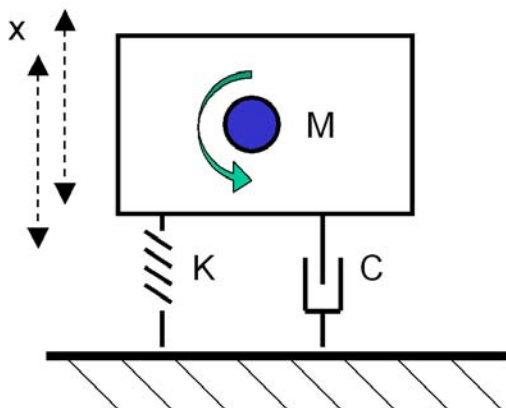


Figure 1A • simple vibrating system

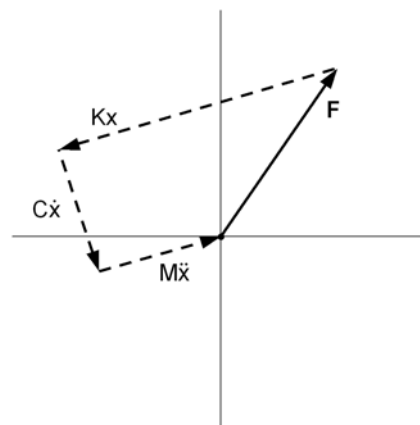


Figure 1B • force components

$$F = M\ddot{x} + C\dot{x} + Kx$$

[Equation 1 • force]

where;

F = force, lb [N]

M = mass, lb-sec<sup>2</sup>/in [g]

$\ddot{x}$  = acceleration, in/sec<sup>2</sup> [m/sec<sup>2</sup>]

C = damping, lb-sec/in [N-sec/m]

$\dot{x}$  = velocity, in/sec [m/sec]

K = stiffness, lb/in [N/m]

x = displacement, in [m]

As shown in Figure 2A, below, for systems operating well below the system natural frequency,  $f_n$ , input and response are in phase, and the system is essentially controlled by stiffness, K. Note that stiffness (spring) force,  $Kx$ , is 180° out of phase with response, O, and force input leads response by phase angle  $\phi$ .

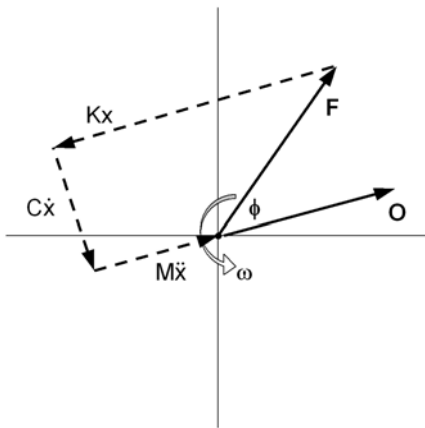


Figure 2A • force and response, below  $f_n$

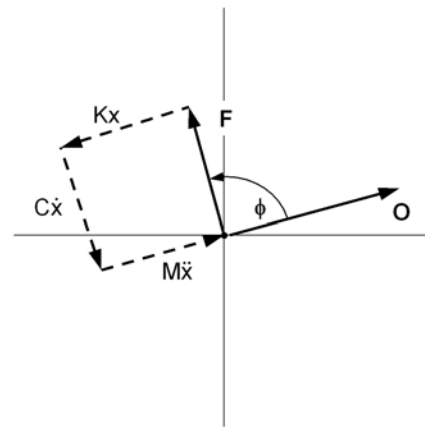


Figure 2B • force and response, at  $f_n$

For systems operating at or near the system natural frequency,  $f_n$ , (resonance), force input leads response by 90°, stiffness and inertia forces are 180° out of phase – and in effect cancel, and the system is essentially controlled by damping, C, as detailed in Figure 2B, above.

Similarly, for systems operating well above  $f_n$ , vibration response is 180° out of phase with force input, and the system is essentially controlled by inertia (mass), as shown in Figure 2C.

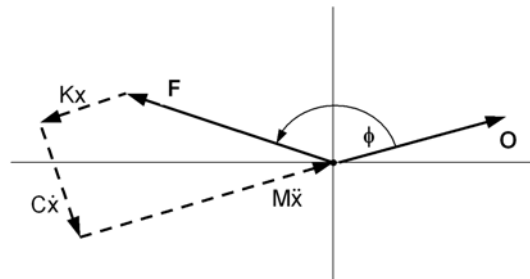


Figure 2C • force and response, above  $f_n$

Since it determines the dominant force component, it is essential that the frequency relationship between force input and system natural frequency,  $f_n$ , are evaluated prior to selecting a method for vibration control.

**Force Reduction** of inputs related to rotating components, such as unbalance, misalignment, looseness, and rubbing, will result in a corresponding reduction of vibration response. Typically, force input increases in proportion to the frequency (speed). For higher speed machines, balancing to specified tolerances and precision shaft alignment may be required to moderate the input force. As shown in Equation 2, below, force due to unbalance increases with the square of the speed. Conversely, on slower machines, residual unbalance may not necessarily result in unacceptably high force input and corresponding vibration response.

$$F = M e \left( \frac{f}{2\pi} \right)^2 \quad \text{[Equation 2 • unbalance force]}$$

where;

F = force, lb [N]  
M = mass, lb-sec<sup>2</sup>/in [g]  
e = eccentricity, in [m]  
f = frequency, Hz

For force inputs at or near the system natural frequency,  $f_n$ , (resonance), amplification of the vibration response is likely occur. This may cause otherwise acceptable (residual) force inputs to result in excessive vibrations. For well damped systems, force reduction may sufficiently control the vibration response. For lightly damped systems, force reduction is typically used in conjunction with Tuning.

One particular case of force reduction involved a variable speed drive connected to a speed reduction gearbox connected to a paper machine drive roll. The system was found to have a lightly damped torsional mode ( $f_n$ ) within its normal operating speed range. Amplified vibration response had resulted in several catastrophic failures. The problem was resolved using a “notch filter” to prevent steady-state motor operation within the relevant speed range, thereby controlling the 1X rotating speed force input.

**Mass Addition** applies Newton’s 2<sup>ND</sup> Law, shown in Equation 3, below, which implies that if the mass of a system is increased while the force input remains constant, acceleration (vibration response) will decrease. This approach to vibration control is especially useful for equipment that has inherent high vibrations or transient (impacting) forces, such as diesel engines, hammer mills, positive displacement pumps, etc.

$$F = M a \quad \text{[Equation 3 • Newton’s 2<sup>ND</sup> Law of Motion]}$$

where;

F = force, lb [N]  
M = mass, lb-sec<sup>2</sup>/in [g]  
a = acceleration, in/sec<sup>2</sup> [m/sec<sup>2</sup>]

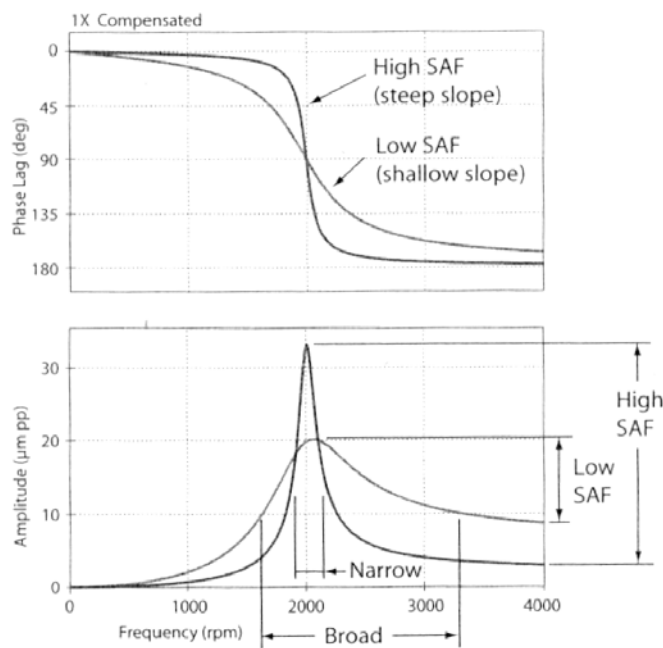
Typically, the mass of the system is increased at the equipment foundation. Therefore, to successfully apply this method for vibration control, machines must be firmly connected to the foundation.

From a machine design perspective, foundations that include a well designed sole plate, epoxy grouted to a concrete base, will help to achieve vibration control and maintenance free equipment operation. One rule of thumb states that the weight of the foundation should be 5X the machine weight.

Note that other vibration control techniques which include adding mass to change a system natural frequency,  $f_n$ , and/or the use of large “inertia block” foundations, are not considered Mass Addition, but rather Tuning and Isolation (rearrangement of force inputs). Each of these methods will be presented later in this paper.

**Tuning** is a process used to eliminate amplification due to resonance by changing a system or component natural frequency,  $f_n$ , so that it is no longer coincident with the frequency of a specific force input. Resonance of industrial equipment will amplify vibration response, in theory up to  $\infty$ , depending on system damping characteristics.

The Synchronous Amplification Factor (SAF) is a measure of how much 1X vibration is amplified when the system passes through a resonance. Systems with a high effective damping tend to have a low SAF, and systems with low effective damping have a high SAF. The Bode Plot, shown in Figure 3, below, indicates that systems with a high SAF (lightly damped) have a narrow range of resonance with high amplification; systems with low SAF (well damped) have relatively broad range of resonance with low amplification. Note that the range of resonance indicates the amount of tuning (change in the system natural frequency) required to eliminate resonance. Resonant frequencies that are nonsynchronous exhibit similar behavior.



**Figure 3 • Bode Plot**

Before attempting to detune a resonant system, the natural frequency (and damping characteristics) should be determined experimentally, typically by a forced response (bump) test. Once the frequency relationship between force input and system natural frequency,  $f_n$ , are evaluated, a decision must be made on whether to raise or lower the natural frequency. For a simple (sdof) vibrating system,  $f_n$  is proportional to the stiffness to mass ratio. Equation 4, below, indicates that adding stiffness will raise the  $f_n$ , and, conversely, adding mass will lower the  $f_n$ .

$$f_n \cong \frac{1}{2\pi} \sqrt{\frac{K}{M}}$$

[Equation 4 • natural frequency for simple vibrating system]

where;

$f_n$  = system natural frequency, Hz

$K$  = stiffness, lb/in [N/m]

$M$  = mass, lb-sec<sup>2</sup>/in [g]

Since increasing mass and/or decreasing stiffness (to lower the  $f_n$ ) may compromise the strength of a machine or its supporting structure, adding stiffness (to raise the  $f_n$ ) is the most common practical application of detuning. When adding stiffeners to a resonant system, the optimal location for end connections is the system antinodes, and installations at nodes will be ineffective. A modal or operating deflection shape analysis is useful for determining the location of antinodes. Care should be taken to ensure that; 1) the added mass of the stiffener does not cancel its stiffening effect (note that pipe has the best stiffness to mass ratio), 2) stiffeners do not introduce new component natural frequencies that are coincident with force input frequencies (ie resonance), and 3) stiffener end connections are as fixed (rigid) as practically possible.

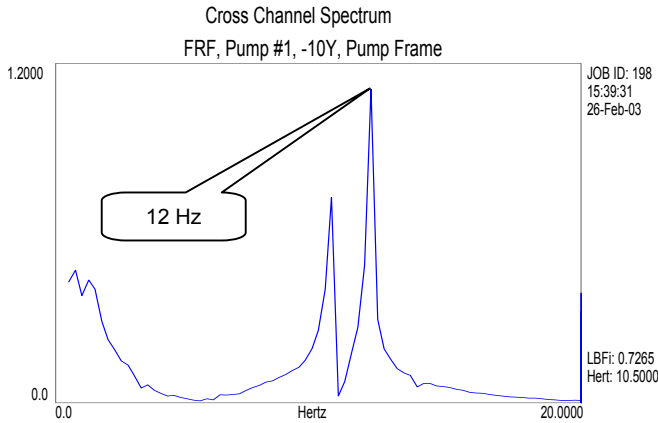


Figure 4A • forced response test of vertical pump

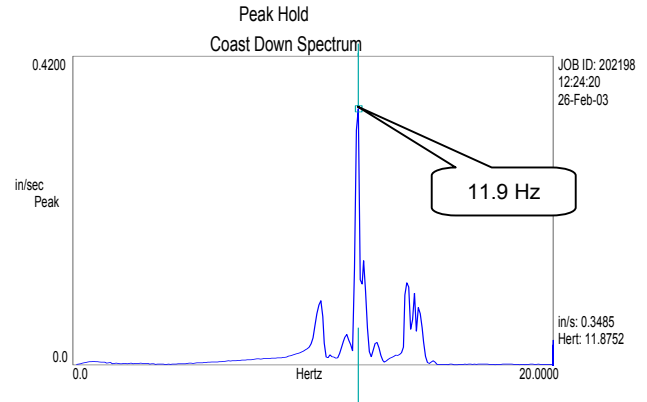


Figure 4B • peak hold coastdown of vertical pump

Figures 4A and 4B, above, show the evaluation of system  $f_n$  for a vertical pump, using both forced response testing and peak hold coastdown methods. In this case, 1X pump rotating speed, 700 rpm, is coincident with a system natural frequency (ie resonance). Note that by analyzing phase and coherence data, the 10.5 Hz peak shown in Figure 4A was found to be an external force input, rather than a system natural frequency.

A computer model identified system mode shapes (ie determine location of antinodes), and was used to evaluate several options for adding stiffness, as shown in Figure 5A, below. The installation of stiffeners, shown in Figure 5B, resulted in a significant reduction in vibration response.

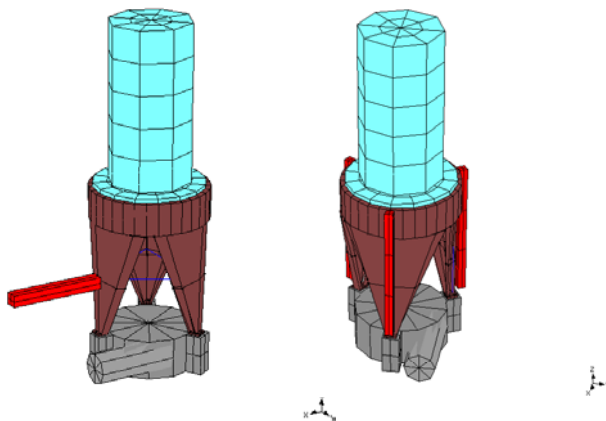


Figure 5A • computer model of structural modifications



Figure 5B • installed stiffeners

**Isolation** reduces the transmitted vibration response of a system by rearranging energy so that inertia (mass) opposes force (see Figure 2C). Resilient supports (isolators), typically elastomeric, spring, and/or pneumatic, decouple a system from force inputs, and cause the isolated system to be out of phase with the force inputs. Figure 6, below, shows various isolator designs. Isolators are sized according to; 1) the load on the isolators, 2) the  $f_n$  of the isolated system, and 3) the frequency of the force input.

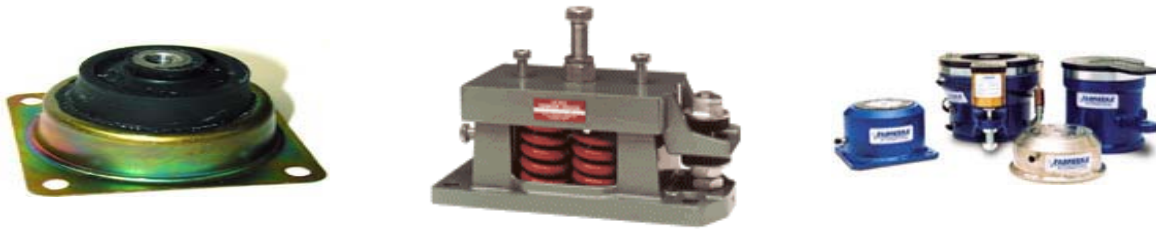


Figure 6 • various isolators

Referring to Figure 7A, below, the frequency ratio,  $f / f_n$ , must be greater than  $\sqrt{2}$  (isolation zone) for isolation to be successful. As the frequency ratio increases, force transmitted (vibration response) decreases. Systems where the frequency ratio is below  $\sqrt{2}$ , (amplification zone) are not suitable for isolation. One strategy for improving isolation is to add mass (inertia block), as shown in Figure 7B, thereby lowering the  $f_n$  of the isolated system (see Equation 4), and increasing the frequency ratio.

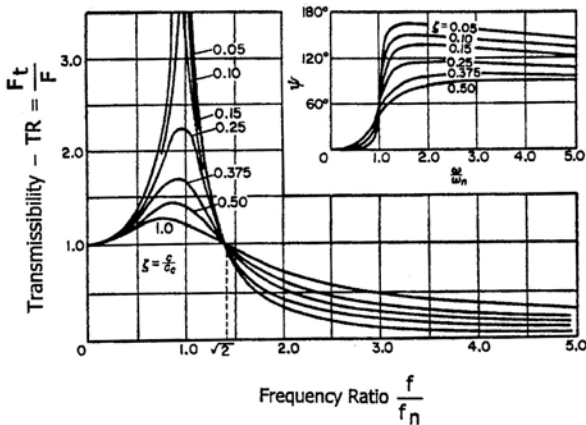


Figure 7A • transmissibility diagram

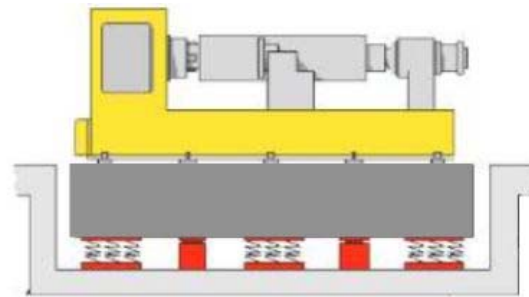


Figure 7B • installed isolators and inertia block

$F_T$  = force transmitted, lb [N]

$F$  = force input, lb [N]

$f$  = frequency of force input, Hz

$f_n$  = (isolated) system natural frequency, Hz

$\zeta = c/c_c =$  damping ratio

Care must be taken to ensure that the support structure and the equipment is rigidly mounted to the isolator. Any deflection of the support structure (or machine frame) will reduce the effectiveness of isolation.

A special case of isolation (energy rearrangement) is the **Dynamic Absorber**, which controls vibration by generating a force that opposes (and cancels) the excitation force of a resonant system. This is achieved using a spring-mass system that is tuned to have a system resonant frequency equal to the frequency of the excitation force.

Recalling Figure 2B, note that for a resonant system, the vibration response is 90° out of phase with the force input. The dynamic absorber may be considered as 2 resonant systems, A and B, as shown in Figure 8A, below. The vibration response of System A (90° out of phase with the force input) is the force input for System B (the dynamic absorber). It follows that the vibration response of the dynamic absorber is 180° out of phase with the original force input.

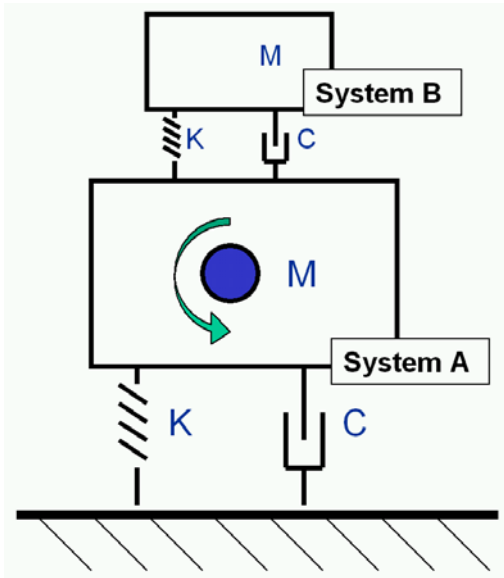


Figure 8A • model of Dynamic Absorber

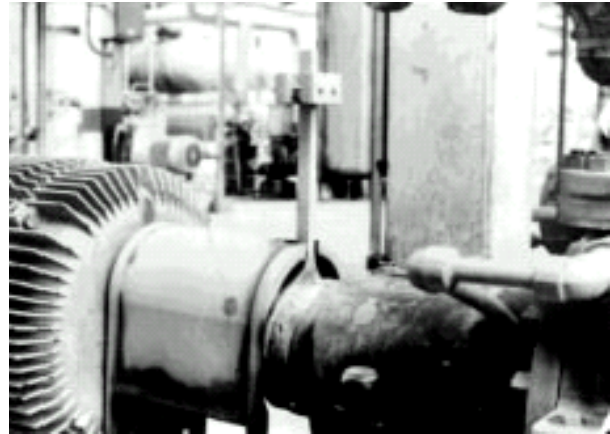


Figure 8B • installed Dynamic Absorber

Typically, as shown in Figure 8B, above, the dynamic absorber consists of an adjustable mass on a pipe, rigidly attached to the machine or component. One rule of thumb (to be used with discretion) is to size the mass in the range of 5% to 10% of the mass of the machine. This leaves only the length of the spring (absorber) to be determined, using Equation 5, below.

$$l = \sqrt[3]{\frac{3EI}{\left(\frac{f}{2\pi}\right)^2 M_a}}$$

[Equation 5 • length of dynamic absorber]

where;

$l$  = absorber length, in [m]

$E$  = modulus of elasticity, (29.5 X 10<sup>6</sup> psi for steel)

$I$  = section area moment of inertia, in<sup>4</sup> [m<sup>4</sup>]

$f$  = frequency, Hz

$M_a$  = absorber mass, lb-sec<sup>2</sup>/in [g]

Since dynamic absorbers are designed to cancel a specific resonant frequency, they are not well suited for variable speed machines, nor are they particularly well suited for well damped systems.

**Damping** effectively controls vibration at or near resonance through energy dispersion, usually as heat. 3 common types of damping forces are viscous, coulomb, and structural. Viscous damping forces are generated by masses moving through a fluid (eg dashpot or shock absorber). Coulomb damping forces are a result of sliding motion between 2 dry surfaces. Structural or material damping, the category most commonly applied for industrial vibration control, is caused by internal friction within a material.

Open cell (polyurethane or butyl rubber) foams encompass inherent damping (and elastic) properties that make them suitable for shock and vibration control of systems with low frequency vibration, high deflection, and large mass.

**Conclusion:** There are 5 basic and distinct methods available for vibration control of industrial equipment, as described in this paper. Prior to proceeding with a vibration control method, the vibrating system, including force inputs and system natural frequencies, must be evaluated.

#### **References:**

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