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Vibration & Noise
Control Systems

Isolation Technology

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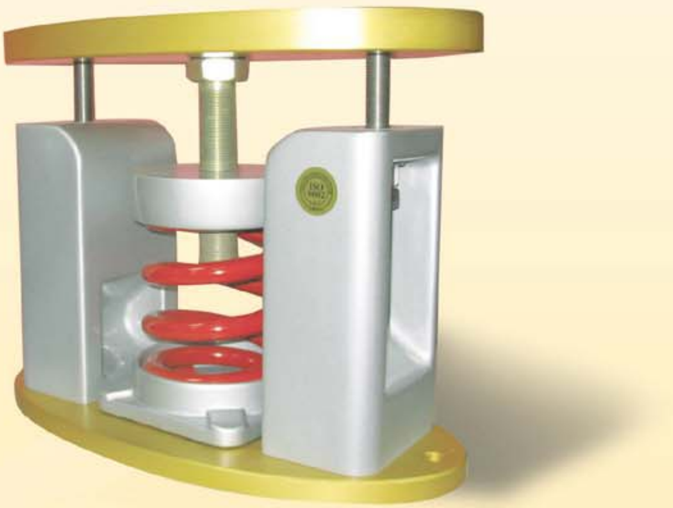

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JSC Scientific Controls Co., Ltd.

ISO-9001 Recognize

Glossary of Common Vibration-Related Terms

Vibration:
Vibration is a periodic motion about a position of equilibrium.

Random Vibration:
Random vibration is vibration whose magnitude is not specified for any given instant of time.

Steady State Vibration:
Steady state vibration exists in a system if the velocity is a continuous periodic quantity.

Transient Vibration:
Transient vibration is temporarily sustained vibration of a mechanical system. It may consist of forced vibration.

Free Vibration:
Free vibration is the periodic motion occurring when an elastic system is displaced from its equilibrium position.

Forced Vibration:
Forced vibration is the vibration resulting from the application of an external periodic force.

Periodic Motion:
Periodic motion is a motion that repeats itself at definite intervals of time.

Frequency:
Frequency is the number of times the motion repeats itself per unit of time. (The unit cycle per second is called Hertz (Hz.))

Natural Frequency:
Natural Frequency is the frequency of free vibration.

Resonant Frequency:
Resonant frequency is a frequency at which resonance exists.

Resonance:
Resonance is the frequency match between the natural frequency of the system and the external forced vibration frequency. Very large amplitudes will occur.

Amplitude:
Amplitude is the maximum value of a sinusoidal quantity (i.e. acceleration, displacement).

Damping:
Damping is dissipation of energy in an oscillating system. Limits maximum amplitude at isolator natural frequency.

Spring Rate:
Force necessary to produce a unit deflection in an elastic element.

Transmissibility:
Percentage of vibratory force or motion transmitted to its support.

Acceleration:
Acceleration is a vector quantity that specifies the time rate of change of velocity.

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Glossary of Common Vibration-Related Terms

Velocity:
Velocity is a vector quantity that specifies the time rate of change of displacement with respect to a reference time.

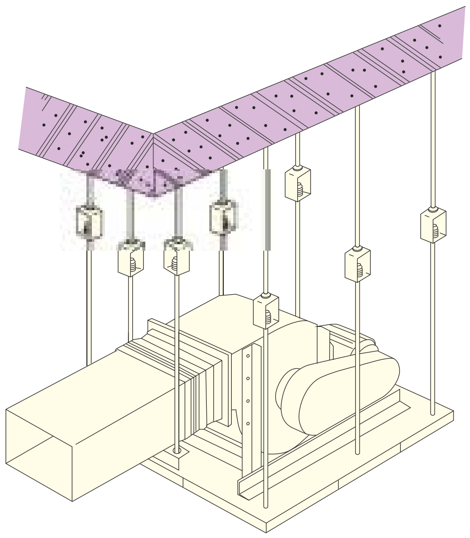
Foundation (Support):
A foundation is a structure that supports the gravity load of a mechanical system.

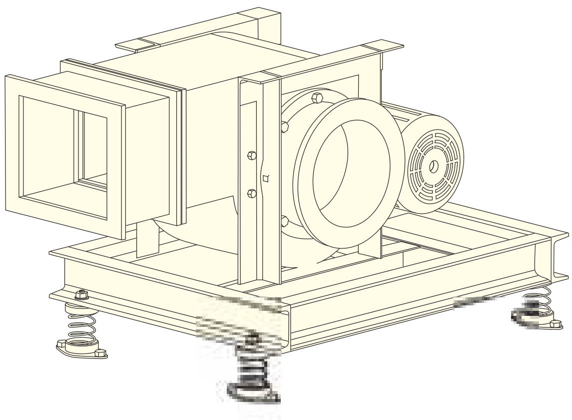
Isolation:
Isolation is a reduction in the capacity of a system to respond to an excitation. This is attained by the use of a resilient support.

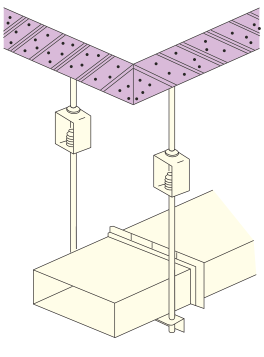
Shock Absorber:
A shock absorber is a device which dissipates energy to modify the response of a mechanical system to applied shock.

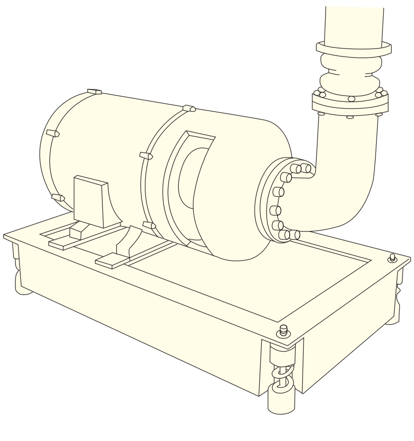
Shock Isolator (Mount):
A shock isolator is a resilient support that tends to isolate a system from shock motion (excitation).

Vibration Isolator:
A resilient support that tends to isolate a mechanical system from steady state excitation









Vibration

This outline of basic vibration theory is intended to present a simplified approach to application and sizing of isolators.
It will enable the design engineer to select the proper isolator to reduce the harmful effects of vibration. Obviously, real life situations are more complex than this simplified approach indicates.
Vibration is defined as a magnitude (force, displacement, or acceleration) which oscillates about a reference point.
Vibration is commonly expressed in terms of frequency, cycles per second or Hertz (Hz).
Vibration problems generally fall into two classes.

Force excitation: If the equipment is the source of the vibration and/or shock, the purpose of the isolator is to reduce the force transmitted from the equipment to the support Structure.
The direction of force transmission is from the equipment to the support structure. This is illustrated in Figure 1, where M represents the mass of a motor which is the vibrating source, and K, which is located between the motor and the support structure, represents the isolator.

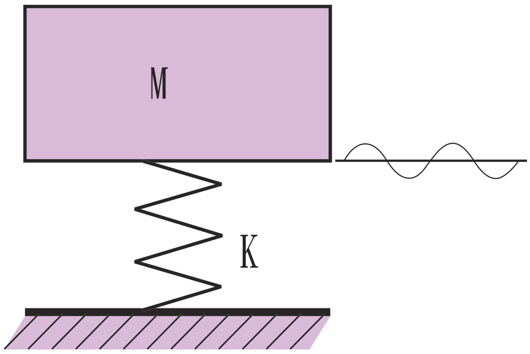


Figure 1. Schematic diagram of a dynamic system where the mass, M, is the vibratory source.

2. Motion excitation: If the support structure is the source of the vibration and/or shock, the purpose of the isolator is to reduce the dynamic disturbance transmitted from the support structure to the equipment. The direction of motion transmission is from the support structure to the equipment. This occurs, for instance, in protecting delicate measuring instruments from vibrating floors. This condition is illustrated in Figure 2, where M represents the mass of a delicate measuring instrument which is protected from a vibrating floor by an isolator signified as K.

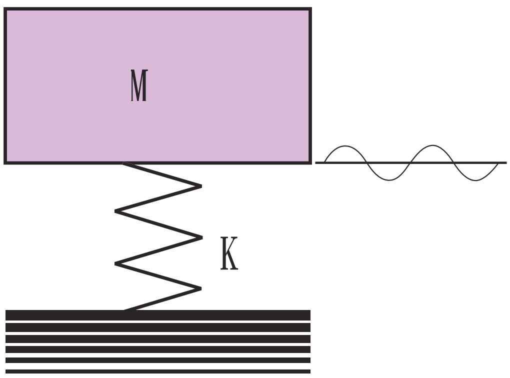


Figure 2. Schematic diagram of a dynamic system where the floor is the vibratory source.

VIBRATION

A magnitude (force, displacement, or acceleration) which oscillates about some specified reference where the magnitude of the force, displacement, or acceleration is alternately smaller and greater than the reference. Vibration is commonly expressed in terms of frequency (cycles per second or Hz) and amplitude, which is the magnitude of the force, displacement, or acceleration. The relationship of these terms is illustrated in Figure 3.

Frequency

Frequency may be defined as the number of complete cycles of oscillations which occur per unit of time.

$$\text{Frequency} = \frac{\text{cycles}}{\text{second}} \text{ (cps)} = \text{Hertz (Hz)}$$

PERIOD

The time required to complete one cycle of vibration.

$$\text{Period} = \lambda = \frac{1}{f}$$

FORCING FREQUENCY

Defined as the number of oscillations per unit time of an external force or displacement applied to a system.

$$\text{Forcing frequency} = fd$$

NATURAL FREQUENCY

Natural frequency may be defined as the number of oscillations that a system will carry out in unit time if displaced from its equilibrium position and allowed to vibrate freely. (See Figure 3)

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \text{ ------------------- Eq.1}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{Kg}{W}} \text{ ------------------- Eq.2}$$

$$f_n = 15.76\sqrt{\frac{K}{W}} \text{ ----------------- Eq.3}$$

$$f_n = 15.76\sqrt{\frac{1}{\delta}} \text{ ---------------- Eq.4}$$

δ Unit = mm

When damping is considered, Equation 2 becomes:

$$f_n = \frac{1}{\delta}\sqrt{\frac{kg}{W}\left[1-\left(\frac{C}{Cc}\right)^2\right]} \text{ ----------------- Eq.5}$$

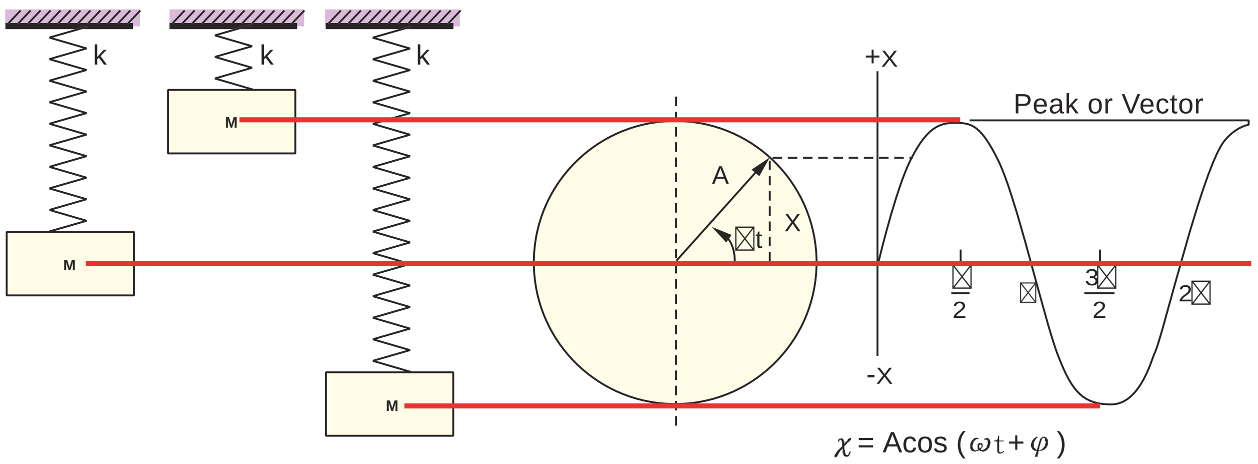


Figure 3. Schematic of oscillating spring mass system and graphical representation of vibratory responses

SPRING STIFFNESS

Described as a constant which is the ratio of a force increment to a corresponding deflection increment of the spring.

$$\text{Spring Stiffness} = K = \frac{F}{\delta} = \frac{\text{Force}}{\text{Deflection}} = \frac{\text{Kg}}{\text{mm}} \text{ ------------------ Eq.6}$$

DAMPING

Damping is the phenomenon by which energy is dissipated in a vibratory system,. Three types of damping generally encountered are: coulomb, hysteresis and viscous.

DAMPING COEFFICIENT

Damping for a material is expressed by its damping coefficient.

$$\text{Damping coeff.} = C = \frac{\text{Kg.sec}}{\text{cm}}$$

CRITICAL DAMPING

A system is said to be critically damped when it is displaced from its static position and most quickly returns to this initial static position without any over-oscillation. The damping coefficient required for critical damping can be calculated using:

$$Cc = 2\sqrt{KM} \text{ -------------------------------- Eq.7}$$

DAMPING FACTOR

The non-dimensionless ratio which defines the amount of damping in a system.

$$\text{Damping factor} = \frac{C}{Cc} = \zeta \text{ -------------- Eq.8}$$

RESONANCE

When the forcing frequency coincides with the natural frequency of a suspension system, this condition is known as resonance.

TRANSMISSIBIL1TY

Defined as the ratio of the dynamic output to the dynamic input.

$$T = \sqrt{\frac{1+\left(2\frac{f_d}{f_n}\,\frac{C}{Cc}\right)^2}{\left(1-\frac{f_d^2}{f_n^2}\right)^2+\left(2\frac{f_d}{f_n}\right)^2}} \text{ ---------------- Eq.9}$$

For negligible damping (C/Cc =0), T becomes:

$$T = \left|\frac{1}{1-\left(\frac{f_d}{f_n}\right)^2}\right| \text{ ------------------------------ Eq.10}$$

When resonance occurs, fd/fn=1 and C/CC = any value, T is at its max and Equation 10 becomes:

$$T_{max} = \frac{1}{2\frac{C}{Cc}} \text{ ------------------------------ Eq.11}$$

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Referring to Figure 4, it can be seen that when the ratio of the disturbing frequency fd over the natural frequency fn is less than $\sqrt{2}$ or 1.414, the transmissibility is greater than 1, or the equipment experiences amplification of the input.
Simply expressed, when: FREQUENCY RATIO $u = \frac{f_d}{f_n}$

$\frac{f_d}{f_n} \leq \sqrt{2}$ ---------------- TRANSMISSIBILITY ≥ 1

$\frac{f_d}{f_n} = \sqrt{2}$ ---------------- TRANSMISSIBILITY $= 1$

$\frac{f_d}{f_n} > \sqrt{2}$ ---------------- TRANSMISSIBILITY < 1

Transmissibility (T) is the ratio of the output to the input. If the input amplitude is 0.1 inches, and the output is 0.03 inches, the transmissibility will be:

$$T = \frac{\text{output}}{\text{input}} = \frac{0.03}{0.1} = 0.3$$

The percent of isolation can be expressed as: % Isolation = (1-T) x 100
or in this case: % Isolation = (1-0.3) x 100= 70%.

Typical Transmissibility For Viscous Damping

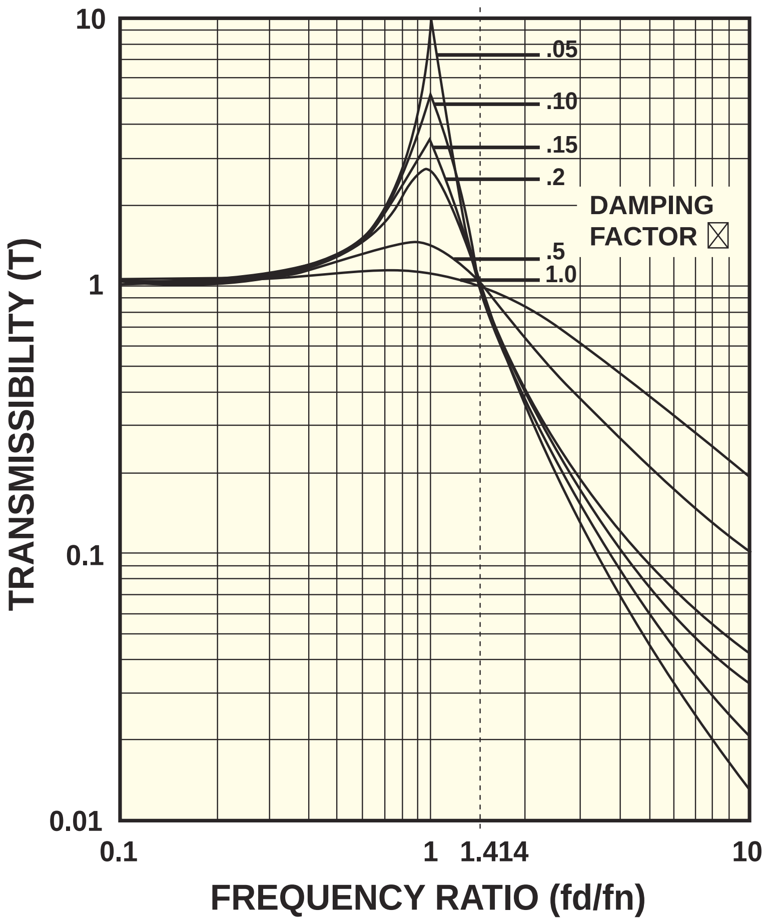

Figure 4

The concept of static deflection often is used to define the characteristics of an isolator. Static deflection is the deflection of the isolator under the static or deadweight load of the mounted equipment. Referring to Equation 2 and substituting g = 981 cm/sec, W/K= , the following expression is obtained for natural frequency in terms of static deflection:

$$f_n = 15.76\sqrt{\frac{1}{\delta}} \quad \text{--------------------------------------} \quad \text{Eq.4}$$

A graphic portrayal of Equation 4 is given in Figure 5. It thus appears possible to determine the natural frequency of a single-degree-of-freedom system by measuring only the static deflection. This is true with certain qualification. First, the spring must be linear its force vs. deflection curve must be a straight line. Second, the resilient material must have the same type of elasticity under both static and dynamic conditions.

Metallic springs generally meet this latter requirement, but many organic materials used in isolators do not. The dynamic modulus of elasticity of these materials is higher than the static modulus; the natural frequency of the isolator is thus somewhat greater than that calculated on the basis of static deflection alone.

Dynamic stiffness may be obtained indirectly by determining the natural frequency when the isolator is vibrated with a known load and calculating the dynamic stiffness from Equation 2. The various organic materials have certain peculiarities with respect to dynamic stiffness which will be discussed later in connection with the specific materials.

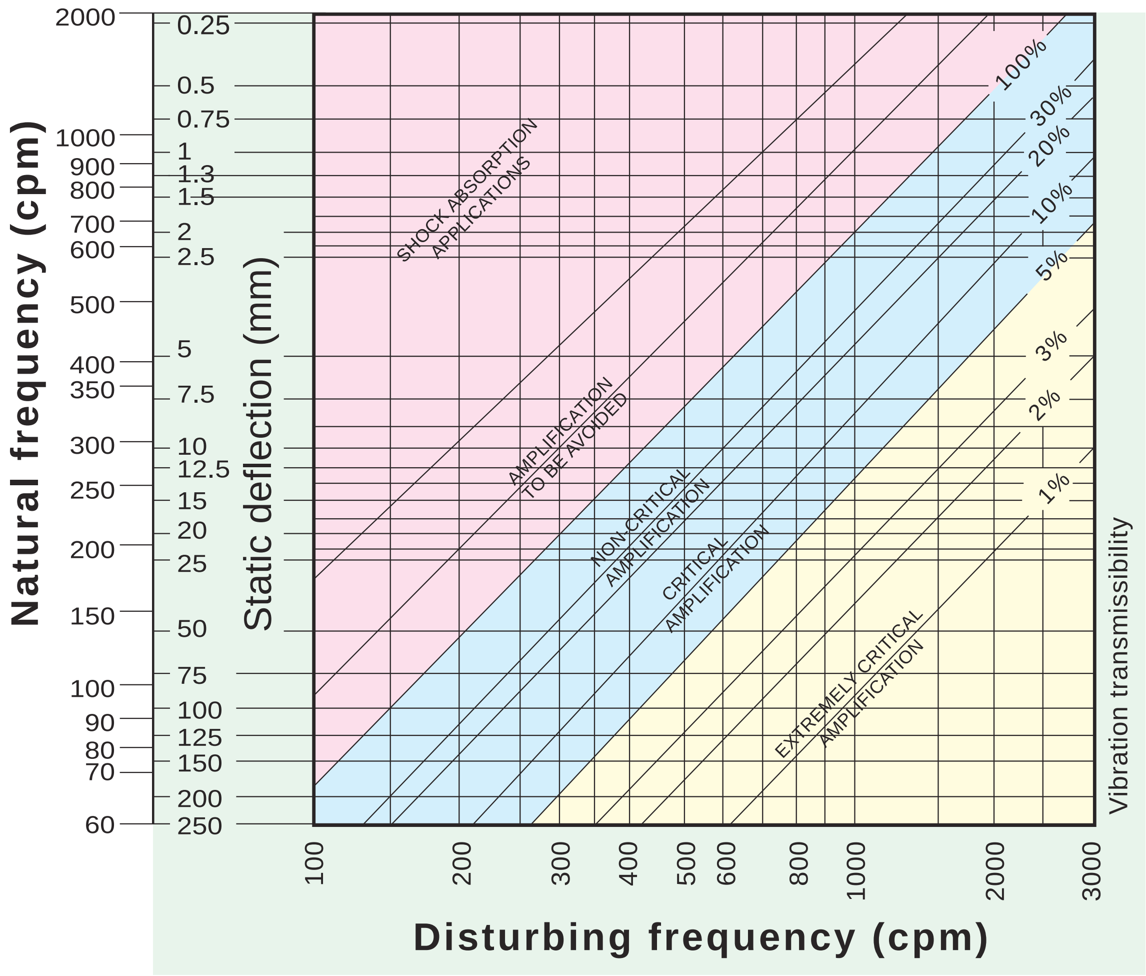

Figure 5. Relation of natural frequency and static deflection of a linear, single-degree-of-freedom system.

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Design Examples

This section deals with the selection and application of vibration isolators. For the proper selections of isolators, it is desirable to obtain, where possible, pertinent information relating to the equipment, input and output requirements, and the general environment.

Examples of the type of information or data required are:

Relating to the equipment:

- Weight
- Dimensions.
- CG location.
- Number and location of isolators.
- Available space for isolators.
- Fragility level of the equipment.

Relating to the dynamic inputs and outputs:

- Level of vibration.
- Space Limitations.

All of the above information is not always readily available nor is it always completely required in some applications. This will be further clarified in the following problem examples.

Practical vibration isolating design

1. Determination of positions

   The support positions must be decided so that the static loads at the support points are nearly equal. In the case of general industrial machines, the anchor bolt positions are used as the support points but it is necessary to re-check the positions when the load is unevenly distributed.

2. Determination of natural vibration frequency and spring rate

   Calculate the natural frequency and spring rate according to the aforementioned

$$f_n = 15.76\sqrt{\frac{1}{\delta}} \text{ -------------------------------- Eq.4}$$

$$T = \left|\frac{1}{1-\left(\frac{f_d}{f_n}\right)^2}\right| \text{ ------------------------------------ Eq.10}$$

3. Selection of isolator

   Select appropriate isolator taking into account the spring rate, allowable load, allowable deflection and forcible power direction.

Design requirements

Type of machine：Air compressor: 20 HP

Weight of machine

- Machine body: 740kg
- Added base 970 kg
- total: 1,710 kg

Rotating speed

- Compressor. 1,170 rpm.
- Motor: 1 ,750 rpm.

Support positsons 6 point

Calculation of spring rate

2-1 Weight per mounting point

W = P/N = 1,710/6 = 285 (kg)

2-2 Determination of natural frequency

If vibration transmissibility Tr=5%or less is intended, then the frequency ratio u=fd/fn=4.58

from Eq 10. and the natural frequency f n =fd/u = (l,170/60)/4.58 = 4.26Hz

2-3 Calculation of spring rate

Since the static load per point is 285 kg, the required spring rate is as follows from Eq 3.

$$f_n = 15.76 \sqrt{\frac{K}{W}} \text{ ------------------------- Eq.3}$$

$$4.26 = 15.76 \sqrt{\frac{K}{W}}$$

$$4.26 = 15.76 \sqrt{\frac{K}{285Kg}}$$

K = 20.96Kg / mm

Determination of mounts

3-1 Selection of mountings：Select the mountings which satisfy the following conditions.

(1) Spring rate K= 20.96 kg/mm or less

(2) Load W = 285 kg

(3) Deflection = 285/20.96 = 13.6mm

The JA spring mounts JA-1-500 can satisfy these conditions.

3-2 Check of vibration transmissibility

JA-1-500 Spring rate K=19.2 (kg/mm)

Allowable load W=470~530 kg

Natural frequency from Eq.3

$$f_n = 15.76 \sqrt{\frac{K}{W}}$$

$$= 15.76 \sqrt{\frac{19.2}{285}} = 4.09 \text{ Hz}$$

Vibration frequency ratio from.

FREQUENCY RATIO $u = \frac{f_d}{f_n}$

- fn =fd/u = (l,170/60)/4.09 = 4.77

against compressor rotating speed;

- fn =fd/u = (l,750/60)/4.09 = 7.13

against motor rotating speed.

Vibration transmissibility from Eq 10

$$Trl = \frac{1}{\left|1-\left(\frac{1170/60}{4.09}\right)^2\right|} = 0.0265 = 2.65\%$$

against compressor rotating speed;

$$Tr2 = \frac{1}{\left|1-\left(\frac{1750/60}{4.09}\right)^2\right|} = 0.016 = 1.6\%$$

against motor rotating speed.

The vibration transmissibility as a whole is 2.65% or less.

3-3 Determination of isolator

Type . JA Spring Mounts

Product No. : JA-1-500

Quantity . 6 pcs

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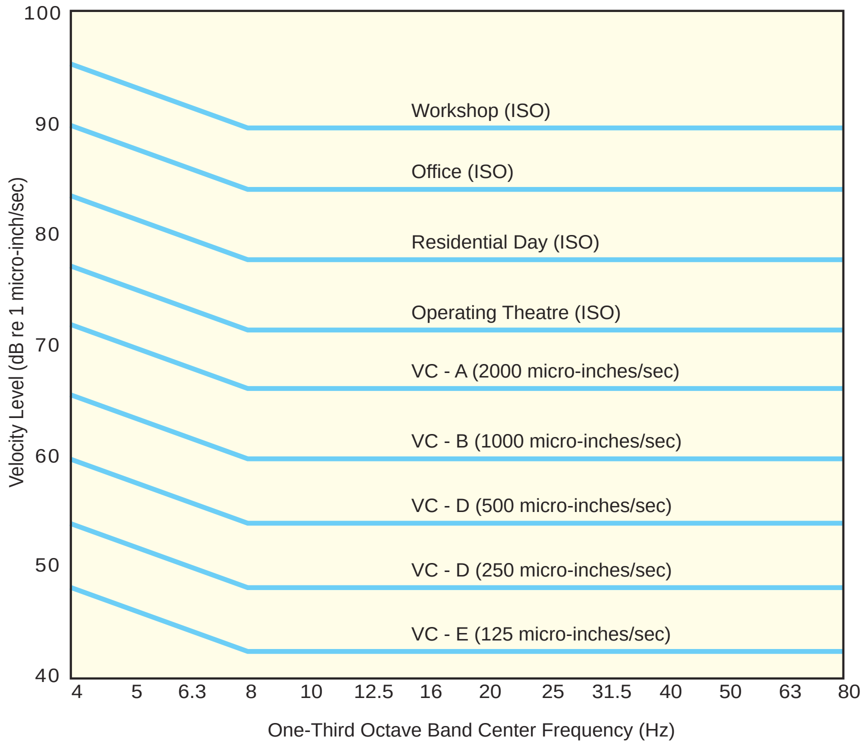
Environmental Vibration Criteria



Generic Vibration Criteria Curves

Criterion Curve	Max Level(1) micro-in./sec (dB)	Detail Size (2) microns	Description of Use
Workshop (ISO)	32,000 (90)	N/A	Distinctly felt vibration. Appropriate to workshops and non-sensitive areas.
Office (ISO)	16,000 (84)	N/A	Felt vibration. Appropriate to offices and non sensitive areas
Residential Day (ISO)	8000 (74)	75	Barely felt vibration. Appropriate to sleep areas in most instances. Probably adequate for computer equipment, probe test equipment and lower-power (to 20X) microscopes.
Op. Theatre (ISO)	4000 (72)	25	Vibration not felt. Suitable for sensitive sleep areas. Suitable in most instances for microscopes to 100X and for other equipment of low sensitivity.
VC-A	2000 (66)	8	Adequate in most instances fo optical microscopes to 400X, microbalances, optical balances, proximity and projection aligners, etc.
VC-B	1000 (60)	3	An appropriate standard for optical microscopes to 1000X, inspection equipment (including steppers) to 3 micron line-widths.
VC-C	500 (54)	1	A good standard for most lithography and inspection equipment to 1 micron detail size.
VC-D	250 (48)	0.3	Suitable in most instances for the most demanding equipment including electron microscoped (TEMs and SEMs) and E-Beam systems, operation to the limits of their capacity.
VC-E	125 (42)	0.1	A difficult criterion to achieve in most instances. Assumed to be adequate for the most demanding of sensitive systems including long path, laser-based, small target systems and other systems.

The information given in this table is for guidance only. In most instances, it is recommended that the adveice of someone knowledgeable about applications and vibration requiremints of te equipment and process be sought.

(1) As measured in one-third octave bands of frequency over the frequency range 8-100 Hz. The dB scale is referenced to 1 micre-in./sec.

(2) The detail size refers to the line widths for microelectronics fabrication, the particle (cell) size for medical and pharmaceutical research, etc. The values given take into account the observation that the vibration requirements of many items depend upon the detail size of the processl.

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Selection Guide Vibration Isolation for HVAC Equipment

Equipment Type	Equipment Category	Horsepower and other (Shaft Power, kW and Other)	RPM	Grade Supported Slab		
				Base Type	Isolator Type	Min. Deflection (mm)
Refrigation Machines and Chillers	Reciprocating	All	All	4	1	6
	Centrifugal	All	All	4	1	6
	Open Centrifugal	All	All	7	1	6
	Absorption	All	All	4	1	6
Air Compressors and Vacuum Pumps Large	Tank-Mounted	up to 10 (7.5)	All	4	2	19
		15 (11) and over	All	7	2	19
	Base-Mounted	All	All	7	2	19
	Reciprocating	All	All	7	2	19
Pumps	Close Coupled	up to 7.5 (5.6)	All	5/6*	1	6
		10 (7.5) and over	All	7	2	19
	In Line	5 to 25 (3.7 to 19)	All	4	2	19
		30 (22) and over	All	4	2	44
	End Suction	up to 40 (30)	All	7	2	19
		50 to 125 (37 to 93)	All	7	2	19
		150 (110) and over	All	7	2	19
	Split Case	up to 40 (30)	All	7	2	19
		50 to 125 (37 to 93)	All	7	2	19
		150 (110) and over	All	7	2	19
Cooling Towers	Centrifugal	All	up to 300	4	1	6
			301 to 500	4	1	6
			500 and over	4	1	6
	Propeller	All	up to 300	4	1	6
			301 to 500	4	1	6
			500 and over	4	1	6
Boilers	All	All	All	4	1	6
Axial Flow Fans and Fan Heads	up to 22"Ø	All	All	4	1	6
	24" Ø & over	up to 2" (500 Pa) static pressure	up to 300	5/6	2	64
			301 to 500	5/6	2	19
			501 and over	5/6	2	19
		2.1 (501 Pa) static pressure and over	up to 300	7	2	64
			301 to 500	7	2	44
			501 and over	7	2	19
Centrifugal Fans	up to 22"Ø	All	All	5/6	1	6
	24" Ø & over	up to 40 (30)	up to 300	5/6	2	64
			301 to 500	5/6	2	44
			501 and over	5/6	2	19
		50 (37) and over	up to 300	7	2	64
			301 to 500	7	2	44
			501 and over	7	2	25
Propeller Fans	Wall-Mounted	All	All	4	1	6
	Roof Exhauster	All	All	4	1	6
Heat Pumps Condensing	All	All	All	4	2	19
	All	All	All	4	1	6
AH, AC and H & V Units	All	up to 10 (7.5)	All	4	2	19
		15 (11) and over up to 4" (1 kPa) static pressure	up to 300	4	2	19
			301 to 500	4	2	19
			501 and over	4	2	19
		15 (11) and over 4" (1 kPa) static pressure and over	up to 300	5/6	2	19
			301 to 500	5/6	2	19
			501 and over	5/6	2	19
Packaged Rooftop Equipment	All	All	All	4/8	1	6

Products Meeting Selection Criteria

Type 1 - Elastomer Isolation Pad, Isolation Mount, Machinery Mount, Vibration Isolation Mount, Isolation Hanger,	Model PS Model JN,DR Model MT,JRA Model JM Model JH-R,JH-A,JH-E
Type 2 - Free-standing Steel Spring, Isolation Hanger,	Model JA,JB,JC,JS Model JH TYPE
Type 3 - Restrained Spring Isolator,	Model JG TYPE

Type 4 - No Base Required	
Type 5 - Structural Rail Base,	Model CN,CN-A
Type 6 - Integral Structural Beam Base,	Model CF,CFS
Type 7 - Concrete Inertia Base,	Model CE
Type 8 - Roof Curb Rail	Model CH

Selection Guide Vibration Isolation for HVAC Equipment

Up to 6 m Floor Span			6 - 9 m Floor Span			9 -12 mFloor Span			Reference Notes
Base Type	Isolator Type	Min. Deflection	Base Type (mm)	Isolator Type	Min. Deflection	Base Type (mm)	Isolator Type	Min. Deflection (mm)	
4	3	19	4	2	44	4	3	64	-
4	3	19	4	2	44	4	2	44	-
7	3	19	7	2	44	7	2	44	-
4	3	19	4	2	44	4	2	44	-
4	2	19	4	2	44	4	2	44	-
7	2	19	7	2	44	7	2	44	-
7	2	19	7	2	44	7	2	44	-
7	2	19	7	2	44	7	2	44	6
7	2	19	7	2	19	7	2	19	7
7	2	19	7	2	44	7	2	44	5, 7
4	2	44	4	2	44	4	2	44	-
4	2	44	4	2	44	4	2	64	2.5
7	2	19	7	2	44	7	2	44	7
7	2	19	7	2	44	7	2	64	5, 7
7	2	44	7	2	44	7	2	64	5, 7
7	2	19	7	2	44	7	2	44	7
7	2	19	7	2	44	7	2	64	5, 7
7	2	44	7	2	44	7	2	64	5, 7
4	3	89	4	3	89	4	3	89	1,3
4	3	64	4	3	64	4	3	64	1
4	3	19	4	3	19	4	3	44	1
4	3	89	4	3	89	4	3	89	1,3
4	3	64	4	3	64	4	3	64	1
4	3	19	4	3	19	4	3	44	1
5/6*	3	19	5/6*	3	44	5/6*	3	64	1
4	2	19	4	2	19	7	2	19	1,3
7	2	89	7	2	89	7	2	89	1,3,4
5/6	2	44	7	2	64	7	2	64	1
5/6	2	44	5/6	2	44	5/6	2	44	1
7	2	89	7	2	89	7	2	89	1,3,4
7	2	44	7	2	64	7	2	64	1,4
7	2	44	7	2	44	7	2	64	1,4
5/6	2	19	5/6	2	19	7	2	44	3,4
5/6	2	89	5/6	2	89	5/6	2	89	3
5/6	2	44	5/6	2	64	5/6	2	64	3
5/6	2	19	5/6	2	19	5/6	2	44	3
7	2	89	7	2	89	7	2	89	3,4
7	2	44	7	2	64	7	2	64	3,4
7	2	44	7	2	44	7	2	64	3,4
4	1	6	4	1	6	4	1	6	1
4	1	6	5/6*	3	44	8	3	44	1
4 4	2 3	19 19	4 4	2 3	19 44	4/8 4/8	2 3	44 44	- -
4	2	19	4	2	19	4	2	19	-
4	2	89	4	2	89	7+	2	89	1,4
4	2	64	4	2	64	4	2	64	4
4	2	44	4	2	44	4	2	44	4
7	2	89	7	2	89	7	2	89	1,3,4
7	2	44	7	2	64	7	2	64	1,4
7	2	44	7	2	44	7	2	64	1,4
8	2	19							8

Notes:
1. Provide Type 5 or 6 base if required to support equipment properly.
2. Provide Type 5, 6 or 7 base if required to stabilize supported equipment.
3. Isolator natural frequency to be 40% of the lowest equipment operating speed.
4. Provide HSR thrust restraints for air-moving equipment operating at 53mm. (501 Pa) static pressure and above.
5. Provide 305 mm thick Type 7 inertia base for 75 HP (56 kW) and over pumps.
6. Provide Type 7 inertia base weighing a minimum of 10 times the maxi-mum equipment unbalanced forces.
7. Provide Type 7 inertia base large enough to provide elbow support.
8. If RTU weight causes additional roof deflection >6 mm, stiffen roof structure or relocate RTU so additional deflection is <=6 mm RTUs over noise-sensitive areas may require additional acoustical treatment too reduce airborne noise below.

* Reference notes do not apply.
\+ Reference note #1 does not apply.

ISO 2372-Vibration Severity

<table>
<tr><th colspan="2">Vibration Severity Range Limits (Velocity)

From ISO 2372</th><th colspan="4">Vibration Severity Ranges for Machines Belonging to:</th></tr>
<tr><th>In/Sec (PK)</th><th>MM/Sec (RMS)</th><th>Class Ⅰ
< 20 HP</th><th>Class Ⅱ
20-100 HP</th><th>Class Ⅲ
> 100 HP</th><th>Class Ⅳ
> 100 HP</th></tr>
<tr><td>0.015</td><td>0.28</td><td rowspan="2">A</td><td rowspan="3">A</td><td rowspan="4">A</td><td rowspan="5">A
(Good)</td></tr>
<tr><td>0.025</td><td>0.45</td></tr>
<tr><td>0.039</td><td>0.71</td><td rowspan="2">B</td></tr>
<tr><td>0.062</td><td>1.12</td><td rowspan="2">B</td></tr>
<tr><td>0.099</td><td>1.8</td><td rowspan="2">C</td><td rowspan="2">B</td></tr>
<tr><td>0.154</td><td>2.8</td><td rowspan="2">C</td><td rowspan="2">B
(Allowable)</td></tr>
<tr><td>0.248</td><td>4.5</td><td rowspan="7">D</td><td rowspan="2">C</td></tr>
<tr><td>0.392</td><td>7.1</td><td rowspan="6">D</td><td rowspan="2">C
(Tolerable)</td></tr>
<tr><td>0.617</td><td>11.2</td><td rowspan="5">D</td></tr>
<tr><td>0.993</td><td>18</td><td rowspan="4">D
(Not
Permissable)</td></tr>
<tr><td>1.54</td><td>28</td></tr>
<tr><td>2.48</td><td>45</td></tr>
<tr><td>3.94</td><td>71</td></tr>
</table>

A:Good
B:Allowable
C:Tolerable
D:Not Permissible
Suggested Classifications:

Class Ⅰ: Small (up to 15kW) machines and subassemblies of larger machines.
Class Ⅱ: Medium size (15kW to 75kW) machines without special foundations, or machines up to 300kW rigidly mounted on special foundations.
Class Ⅲ: Large rotating machines rigidly mounted on foundations which are stiff in the direction of vibration measurement.
Class Ⅳ: Large rotating machines mounted on foundations which are flexible in the direction of vibration measurement.

Vibration unit conversion table

1g = 32.174 ft/sec 2
1g = 9.807 m/sec 2
in/sec 2 = 0.0254 m/sec 2

DISPLACEMENT
1mil = 0.001 in
1mil = 0.0254 mm
1in = 25.4 mm
1cm = 10 mm

FREQUENCY
1Hz = 1 cps
1Hz = 0.159 rad/sec
1Hz = 60 rpm
1rpm = 0.0167 Hz
1rpm = 1 cpm

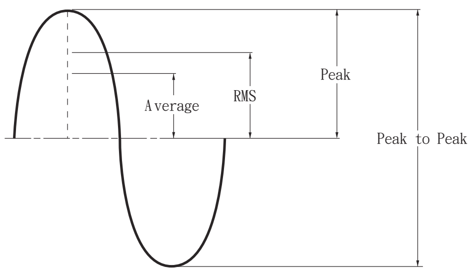
Unit:dB

dB	Gain
60	1000
40	100
20	10
10	3.16
6	2
3	1.41
1	1.12
0	1
-1	0.891
-3	0.708
-6	0.501
-10	0.316
-20	0.1
-40	0.01
-60	0.001

dB=20log (V/Vref)

Vref=10^{-9} m/s

Vibration parameter conversion	
SI UNITS g=2.013f^2D g=0.641Vf	INCH UNITS g=0.0511f^2D g=0.0162Vf
v=πfD D=m，Peak to Peak V=m/s，Peak f=Hz(cps) g=9.80665m/s^2	v=πfD D=inch，Peak to Peak V=inch/s，Peak f=Hz (cps) g=386.1 inch/s^2



Average Value
= 0.637 x Peak Value
RMS Value
= 0.707 x Peak Value
Peak Value
= 1.414 x RMS Value
Peak to Peak Value
= 2 x Peak Value
Peak to Peak Value
= 2.828 x RMS Value

- Peak Amplitude (Pk) is the maximum excursion of the wave from the zero or equilibrium point.
- Peak-to-Peak Amplitude (Pk-Pk) is the distance from a negative peak to a positive peak. In the case of the sine wave, the peak-to-peak value is exactly twice the peak value because the waveform is symmetrical, but this is not necessarily the case with all vibration waveforms, as we will see shortly.
- Root Mean Square Amplitude (RMS) is the square root of the averageof the squared values of the waveform. In the case of the sine wave, the RMS value is 0.707 times the peak value, but this is only true in the case of the sine wave. The RMS value is proportional to the area under the curve -- if the negative peaks are rectified, i.e., made positive, and the area under the resulting curve averaged to a constant level, that level would be proportional to the RMS value.

Vibration Amplitude Measurement

	Peak	Peak to Peak	RMS	Average
Peak	1	0.5	1.414	1.57
Peak to Peak	1	1	2.828	3.14
RMS	0.707	0.354	1	1.11
Average	0.637	0.319	0.901	1

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