BALANCING AND VIBRATION

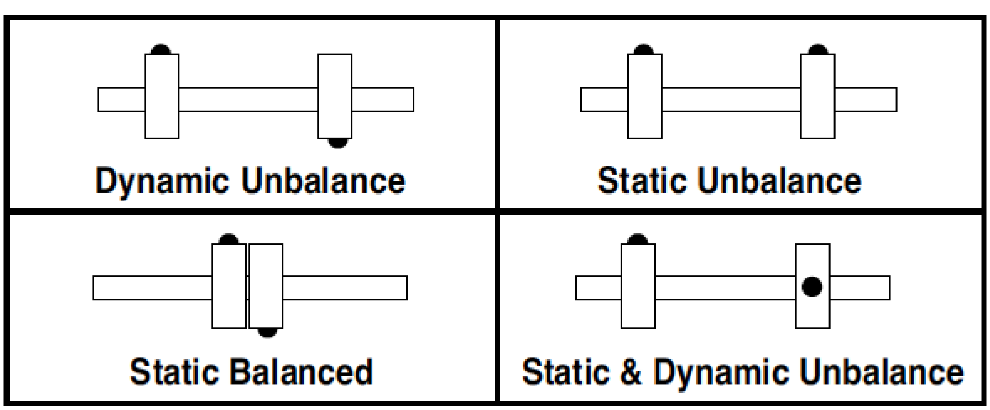
SINGLE-PLANE VERSUS TWO-PLANE BALANCING

Disk-shaped rotating parts usually can be balanced in one correction plane only, whereas parts that have appreciable width require two-plane balancing. As precision tolerances become more meaningful in better performance, dynamic balancing becomes more important, even on relatively narrow parts. Some guides indicate that the proportion of large diameter to relatively narrow face width suggests single-plane balancing. However, the distance between the two planes is more important than the width-to-diameter ratio.

For example, a rotor with a face width of 5” (125 mm) will usually require dynamic balancing whether its diameter is 4” (100 mm) or 40’ (12 m). Unbalance in two separate planes 5” (125 mm) apart is the reason it requires balancing in two planes regardless of its so-called “disk shape.”

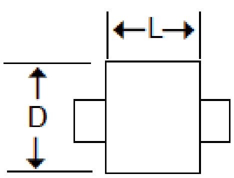
Experience also suggests that all parts that rotate at speeds high enough to require balancing of any kind should be dynamically (or force and couple) balanced on the rotor’s main body length.

Separating the disks but placing the unbalance weights on the same side of the rotor as shown below causes static unbalance that can only be corrected by adding weight at each disk or plane. Shifting one weight 90°, as also shown, produces a combination of static and dynamic unbalance. This condition can only be corrected by adding weights in each of the two planes.



SINGLE-PLANE VERSUS TWO-PLANE BALANCING—CONTD.

The type of correction or number of balance correction planes should be based on the length-to-diameter ratio—i.e., the length of the rotor (L) divided by the diameter (D). The L/D ratio is calculated using the dimensions of the rotor exclusive of the supporting shaft. For L/D ratios less than 0.5, single-plane balancing is sufficient for operating speeds up to 1000 rpm. For operating speeds above 1000 rpm, two-plane balancing is often required. For L/D ratios greater than 0.5, two-plane balancing is required for operating speeds greater than 150 rpm.

	L/D RATIO	BALANCE CORRECTION SINGLE PLANE	BALANCE CORRECTION TWO PLANE
	Less Than 0.5	rpm to 1000	Above 1000 rpm
	More Than 0.5	rpm to 150	Above 150 rpm

Select single-plane versus two-plane balancing based on the length-to-diameter (L/D) ratio and rpm of the rotor.

VIBRATION TESTS

The vibration tests should be in accordance with NEMA Stds. MG 1-1998, 7 for standard machines, as arranged with the customer, or as necessary to check the operating characteristics of the machine. When there are special requirements, i.e., lower than standard levels of vibration for a machine, NEMA Stds. MG 1-1998, 7 for special machines and IEEE 841 are recommended.

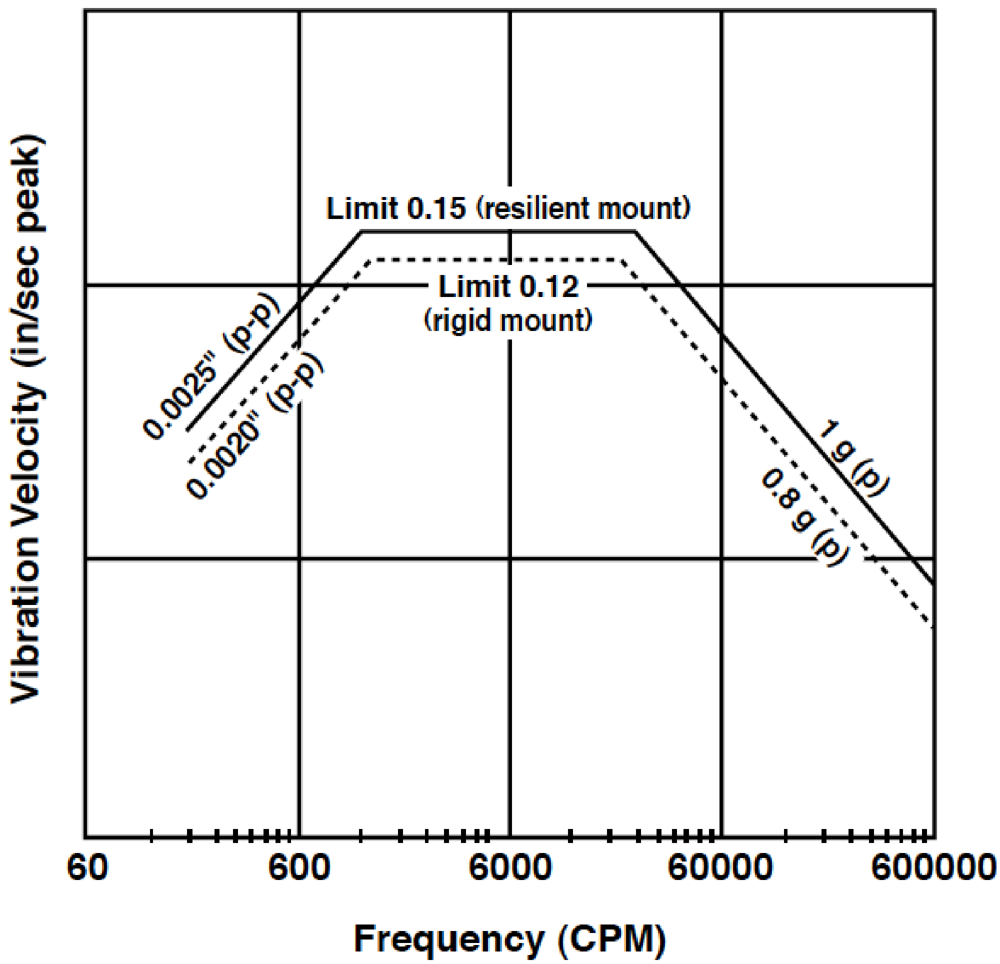
The unfiltered vibration limits for resiliently mounted standard machines (having no special vibration requirements), based on rotational speed, are shown in the table on Page 5 (“Unfiltered Vibration Limits”). Vibration levels for speeds above 1200 rpm are based on the peak velocity of 0.15 inch per second

(3.8 mm/s). Vibration levels for speeds below about 1200 rpm are based on the peak velocity equivalent of 0.0025 inch (0.0635 mm) peak-to-peak displacement. For machines with rigid mounting, multiply the limiting values by 0.8, as shown in the lower curve.

Note: International standards specify vibration velocity as rms in mm/s. To obtain an approximate metric rms equivalent, multiply the peak vibration in in/s by 18. (Reference: NEMA Stds. MG 1-1998, 7.8.)

UNFILTERED VIBRATION LIMITS

MECHANICAL VIBRATION



Note: The intersection of constant displacement lines with constant velocity lines occurs at approximately 1200 CPM. The intersection of constant velocity lines with constant acceleration lines occurs at approximately 24000 CPM.

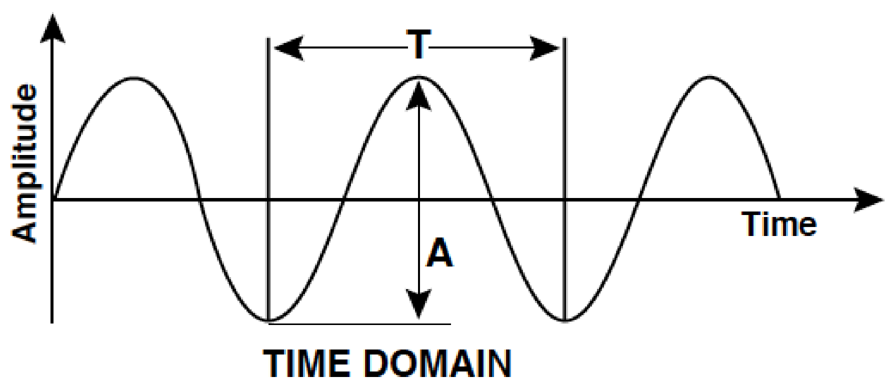
(Reference: NEMA Stds. MG 1-1998, Figure 7-5, Pg. 7.)

FFT VIBRATION ANALYSIS

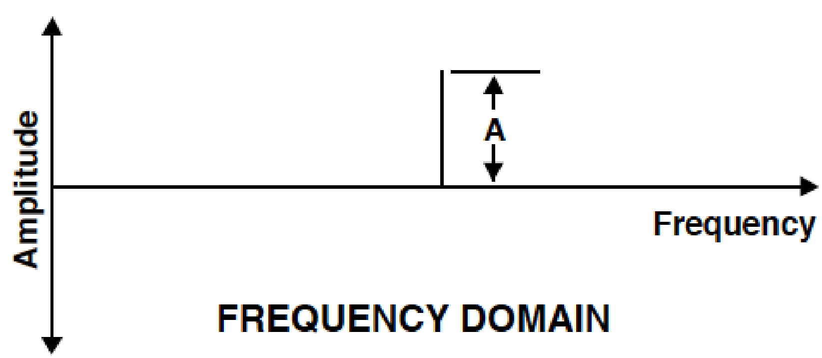
FFT (Fast Fourier Transform) vibration analyzers rely solely on digital techniques to acquire the spectral data. The signal is sampled and a FFT algorithm (mathematical operation) performed on the sampled data to obtain the signature.

A system response can be represented by displacement, velocity and acceleration amplitudes in both the time and frequency domains. The time domain consists of an amplitude that varies with time. When the amplitudes are represented in the frequency domain, they are shown as a series sum of sines and cosines which have a magnitude and phase that varies with the frequency.

The drawing below shows an example of time domain and frequency domain representation. Because measurements are made in the analog world (time domain), they must be "transformed" to the frequency domain. This is the purpose of the FFT (Fast Fourier Transform).



Where: A = peak-to-peak amplitude
T = period of vibration cycle



VIBRATION CONVERSION FACTORS

The relationships between displacement, velocity and acceleration are shown in the following formulas. The formulas are based on vibration waves due to harmonic motion (sine waves) and the frequency of vibration. Most machine vibration wave forms are close to sine waves and good accuracy will be obtained using these formulas. Accurate frequency values are required for these conversions. It is recommended that only filter-in readings of vibration and frequency be used to insure accuracy.

SYMBOLS	ENGLISH UNITS	METRIC UNITS
Displacement - D	in peak-to-peak	mm peak-to-peak
Velocity - V	in/s peak	mm/s peak
Acceleration - A	G's peak	G's peak
Force of gravity - G	1G = 386 in/s^2	1G = 9.81 m/s^2
Frequency - Hz	cycles/s	cycles/s

FORMULAS

$$D = \frac{0.318 \times V}{Hz} = \frac{19.607 \times A}{(Hz)^2}$$

$$V = 3.1416 \times D \times Hz = \frac{61.44 \times A}{Hz}$$

$$A = 0.051 \times D \times (Hz)^2 = 0.016 \times V \times Hz$$

EXAMPLE

SYMBOLS	ENGLISH UNITS	METRIC UNITS
Displacement - D	0.002 in p-p	0.05 mm p-p
Frequency - Hz	50 Hz	50 Hz
Velocity - V	3.1416 x 0.002 x 50 = 0.314 in/s peak	3.1416 x 0.05 x 50 = 7.85 mm/s peak
Acceleration - A	0.051 x 0.002 x 50^2 = 0.255 G's	0.051 x 0.05 x 50^2 = 6.38 G's

VIBRATION IDENTIFICATION GUIDE FOR ASSEMBLED UNIT

CAUSE	FREQUENCY RELATIVE TO MACHINE RPM	PHASE-STROBE PICTURE	AMPLITUDE	NOTES
Unbalance	1 x rpm	Single steady reference mark	Radial - steady proportional to unbalance	Common cause of vibration.
Defective anti-friction bearing	10 to 100 x rpm	Unstable	Measure velocity 0.2 to 1.0 in/s (5 to 25 mm/s) radial	Velocity largest at defective bearing. As failure approaches velocity signal increases, frequency decreases.
Sleeve bearing	1 x rpm	Single reference mark	Not large	Shaft and bearing amplitudes about the same.
Misalignment of coupling or bearing	2 x rpm. Sometimes 1 or 3 rpm.	Usually 2 steady reference marks. Sometimes 1 or 3.	High axial	Axial vibration can be twice race. Use dial indicator as check.
Bent shaft	1 or 2 x rpm	1 or 2	High axial	— — —
Defective gears	High rpm x gear teeth	— — —	Radial	Use velocity measurement.
Mechanical looseness	1 or 2 x rpm	1 or 2	Proportional to looseness	Radial vibration largest in direction of looseness
Defective belt	Belt rpm x 1 or 2	— — —	Erratic	Strobe light will freeze belt.
Electrical	Power line frequency x 1 or 2 (3600 or 7200 rpm)	1 or 2 rotating marks	Usually low	Vibration stops instantly when power is turned off.
Oil whip	Less than rpm	Unstable	Radial - unsteady	Frequency may be as low as half rpm.
Aerodynamic	1 x rpm or number of blades on fan x rpm	— — —	— — —	May cause trouble in case of resonance.
Beat frequency	1 x rpm	Rotates at beat rate	Variable at beat rate	Caused by two machines running at close rpm.
Resonance	Specific criticals	Single reference mark	High	Phase changes with speed. Amplitude decreases above and below resonant speed. Resonance can be removed from operative range by stiffening.

VIBRATION CONSTANTS

CONSTANT FOR TRUE SINE WAVES ONLY

rms value	=	0.707	x	peak value
rms value	=	1.11	x	average value
peak value	=	1.414	x	rms value
peak value	=	1.57	x	average value
average value	=	0.637	x	peak value
average value	=	0.90	x	rms value
peak-to-peak	=	2.0	x	peak value