Balance & Vibration

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- Leads development and publication of technical articles, white papers and educational materials.
- Editor-in-chief of the award-winning *AMCA inmotion* magazine.

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Rad Ganesh

Director, Product Applications,
AMCA Member Company

- Over 32 years of experience in the Movement (HVAC and Fan) industries.
- Currently has 6 patents and 5 pending patent applications.
- Very actively involved with AMCA and ASHRAE, on committees, contributing to publications and giving presentations.
- Earned Ph.D. in Mechanical Engineering and is a registered P.E. in Wisconsin.

Balance & Vibration

Purpose and Learning Objectives

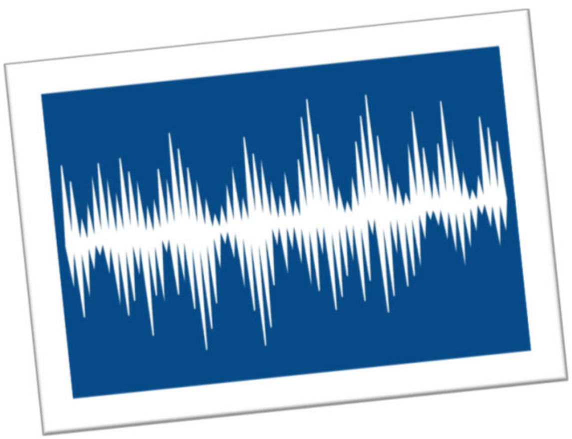
The purpose of this presentation is to inform industry professionals about Balance and Vibration topics as related to Fans for commercial and industrial applications.

At the end of this presentation you will be able to:

1. Explain balancing, and vibration, and how they differ.
2. Identify the standards used for balancing fans and the acceptable balancing grades.
3. Describe how vibration problems are diagnosed and fixed.
4. Compare the various vibration specifications related to fans.

Why Is This Topic Important?

- Balance and vibration characteristics are used as indicators of quality.
- Understanding the terminology and specifications is important to users and those selecting fans.



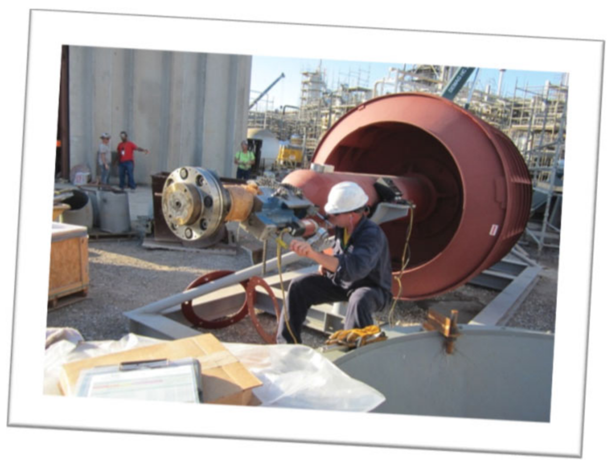
Topics to be Covered

- Balance
  - This is a property of the fan and stays the same at any location.
  - Imbalance is present even if the fan is not running.
- Vibration
  - Strongly influenced by the fans environment and operating speed.
- Do not mix these two concepts!

Balance Terminology

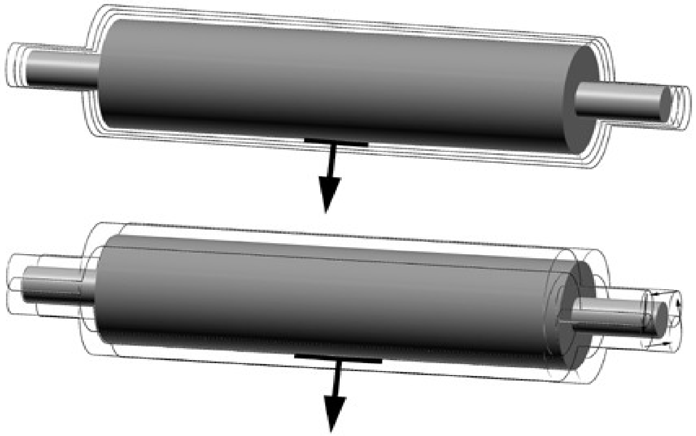
Balancing:

- The process of adding or removing weight on a rotor in order to move the center of gravity towards the axis of rotation.
- The purpose of balancing is to reduce the unbalance forces.



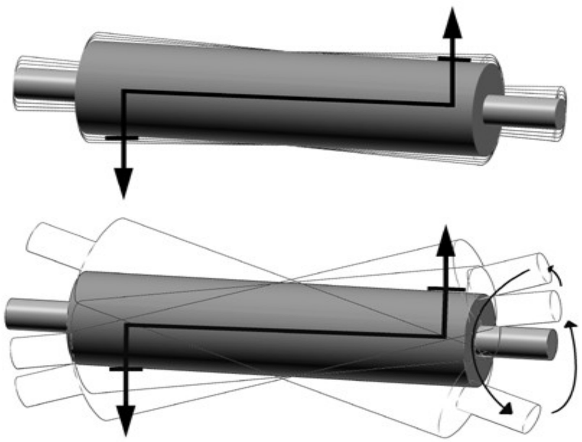
Static Unbalance

- Static unbalance can be corrected by adding a single weight.
- Heavy spot will “bottom out” when rotor is placed in bearings.



Dynamic Unbalance

- Dynamic (couple) unbalance requires correction in two or planes.
- Heavy spot will not “bottom out” when rotor is placed in bearings.

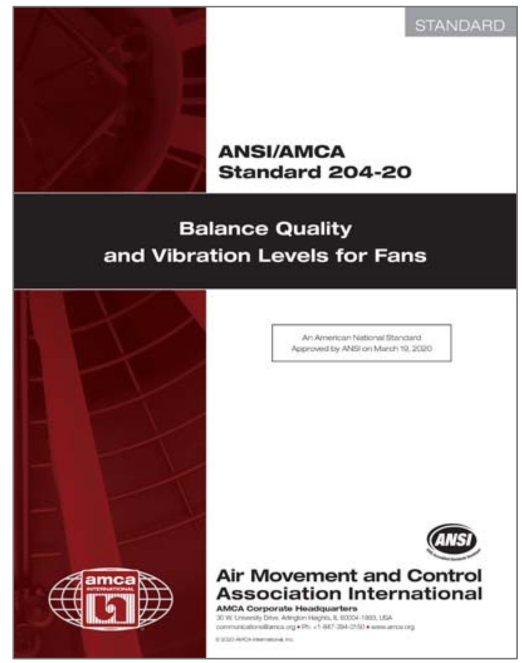


Balance Tolerances

- Defines the maximum amount of residual unbalance remaining after balancing.
- Originally defined in international standard ISO 1940/1.
- Adapted to fans in AMCA 204-20.

Recommended Tolerances Guide

- Use AMCA 204 - 20 to specify balance & vibration tolerances
  - All relevant terms are defined
  - Tolerances are realistic
  - Allows for different categories of fans
  - Provides for long life and reliable operation without excessive cost



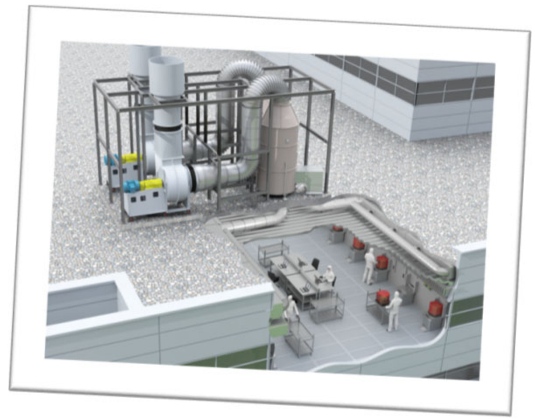
AMCA 204 Fan Categories

Table 6.1—Fan Application Categories for Balance and Vibration

Application	Examples	Driver Power Limits, kW (hp)	Fan Application Category, BV
Residential	Ceiling fans, attic fans, window air-conditioning unit	≤ 0.15 (0.2) > 0.15 (0.2)	BV-1 BV-2
HVAC and agricultural	Building ventilation and air-conditioning systems; commercial systems	≤ 3.7 (5.0) > 3.7 (5.0)	BV-2 BV-3
Industrial process and power generation etc.	Baghouse, scrubber, mine, conveying, boilers, combustion air, pollution control, wind tunnels	≤ 298 (400) > 298 (400)	BV-3 BV-4
Transportation and marine	Locomotives, trucks, automobiles	≤ 15 (20) > 15 (20)	BV-3 BV-4
Transit and tunnel	Subway emergency ventilation, tunnel fans, garage ventilation	≤ 75 (100) > 75 (100)	BV-3 BV-4
	Tunnel jet fans	ALL	BV-4
Petrochemical process	Hazardous gases, process fans	≤ 37 (50) > 37 (50)	BV-3 BV-4
Computer chip manufacturer	Clean room	ALL	BV-5

AMCA 204 Balance Tolerance

- Fans Categorized by application category
  - BV-3 covers all but the smallest and largest fans that cover commercial & industrial applications.
  - BV-4 used for larger industrial fans (over 400 HP) and certain special applications.
  - BV-5 applies to computer chip manufacturing.



AMCA 204 Balance Tolerance

Table 7.1—Fan Application Categories and Balance Quality Grades

Fan Application Category, BV	Balance Quality Grade for Rigid Rotors/Impeller
BV-1*	G 16
BV-2	G 16
BV-3	G 6.3
BV-4	G 2.5
BV-5	G 1.0
Note: *Category BV-1 may include some extremely small fan rotors weighing less than 227g (8 oz). In such cases, residual unbalance may be difficult to determine accurately. The fabrication process must ensure reasonably equal weight distribution about the axis of rotation.	

What is “G6.3”

- “G” values refer to the maximum speed that the shaft center would rotate about the center of gravity if suspended in free space
- **G = e x ω (mm/second)**
  - e = Specific Unbalance or distance between center of gravity and axis of rotation (mm)
  - ω = angular velocity at max design speed (rad/sec)
- “G” has units of velocity but is not appropriate for specifying vibration tolerances.
- Residual unbalance calculations clearly shown in AMCA 204

AMCA 204 Example: Permissible Residual Unbalance, Max Permissible Weight

- G = 6.3 mm/s; N=4000rpm
- Rotor mass, m = 10lbs = 4.5 Kg; Rotor Radius, r = 6 in
- From Fig C.1, e_{per} = 15 gm mm / Kg
- Max Residual unbalance U_{per} = 15 * 4.5 ~ 67.5 g mm ~ 67.5/25.4 ~ 2.68 gm in.
- Max permissible mass at rotor OD for 2 plane balance = 2.68/ (2* 6) ~ 0.22 gms per plane
  - A steel cube 1/8” per side ~ 0.25 gms!
- Thus small, light weight rotors running at high speeds are the most difficult to balance with the added uncertainties of material homogeneity and fabrication.
- Balancing machines are used today to direct the balancer on the magnitude and location (phase) of balance weight placement - a vector quantity.

Annex C

Maximum Permissible Residual Imbalance (Informative

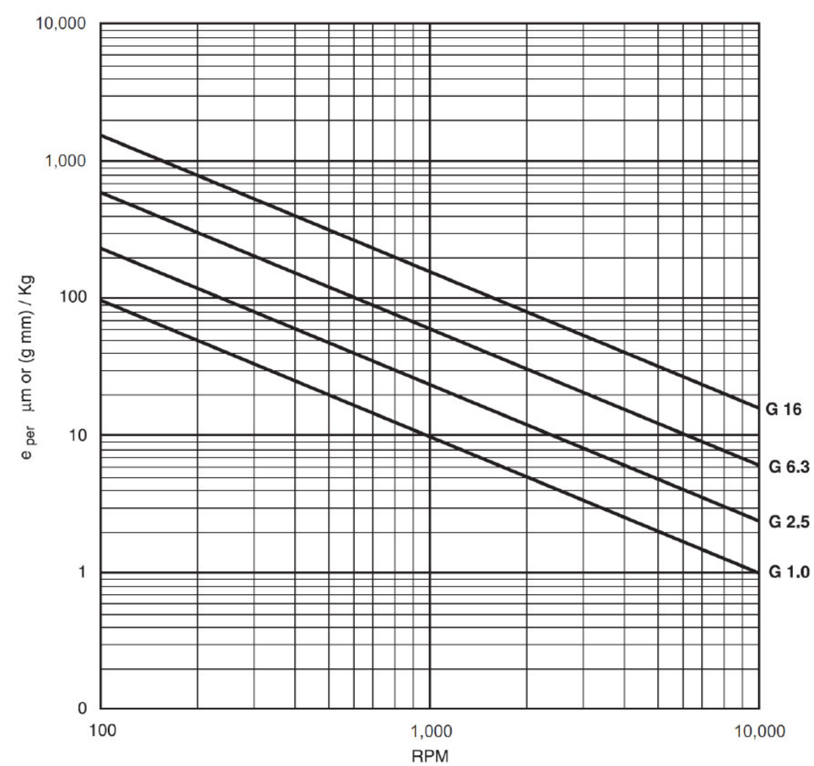
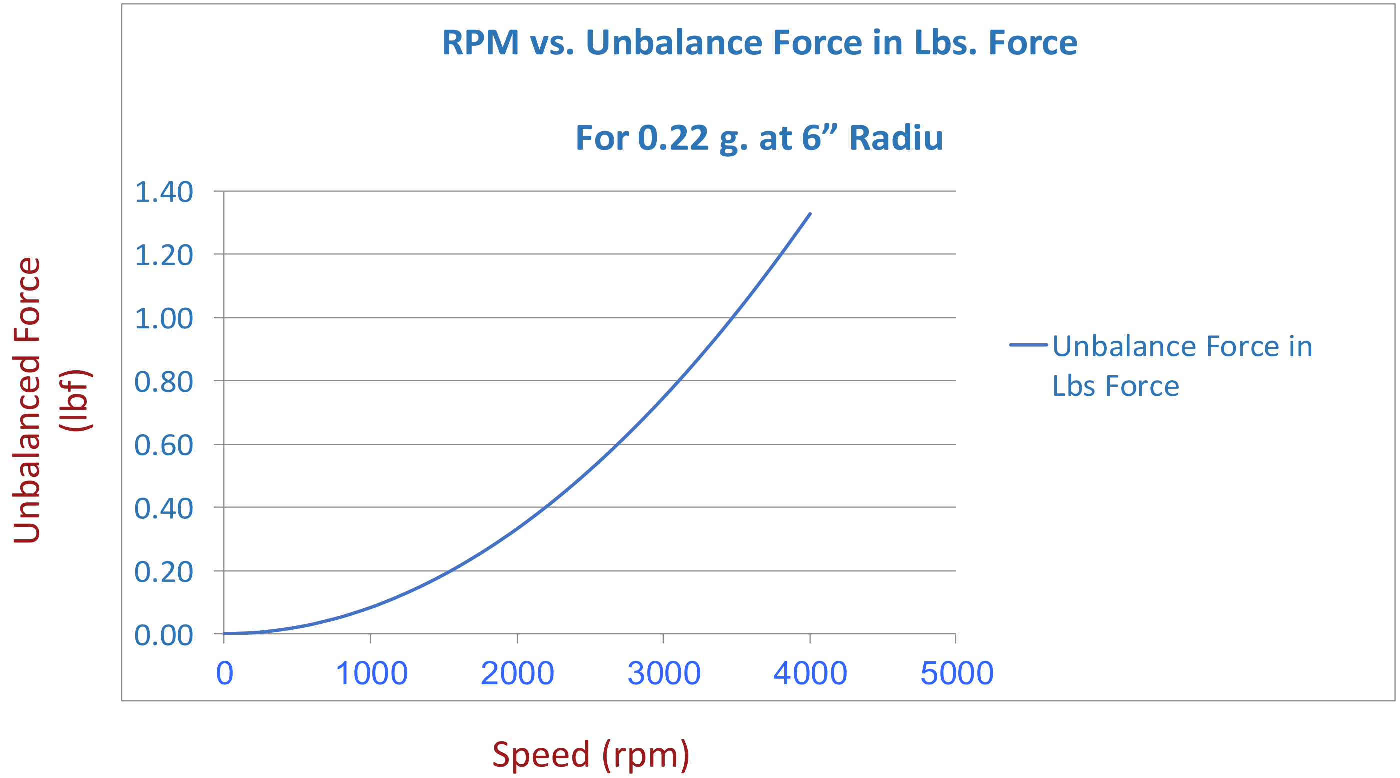


Figure C.1—Maximum Permissible Residual Unbalance (SI)

Centrifugal Force From Unbalance

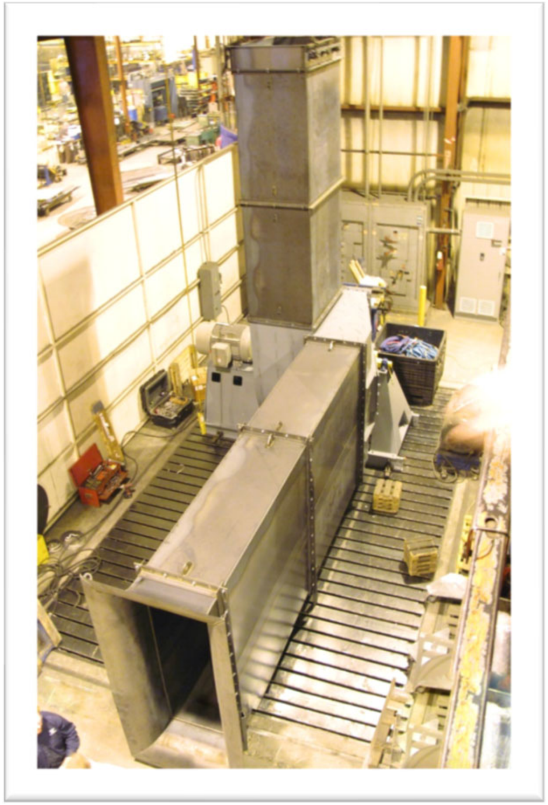


Balancing Machines

- Sized to accommodate weight of rotor
- Fan wheels are usually balanced using a precision arbor to match bore of impeller.

Trim Balancing

- After fan assembly, fans are run tested and checked for vibration.
- If vibration is high, trim balancing is performed.
  - This corrects for fit-up of assembled parts.
  - Most fans use welded on trim weights.
- Two plane balancing is normally not required for trim balancing



Vibration

- What is vibration?
  - Forces that vary in amplitude or direction over time cause repetitive motions called vibration.
- When is vibration bad?
  - All fans must generate some vibration.
  - Only vibration that exceeds certain amplitudes is bad.

Types of Vibration

- Bending vibration refers to fan parts such as blades bending in a cyclic manner.
  - Advanced topic. Not covered here.

- Torsional vibration refers to a periodic twisting of the fan shaft due to a multiple components with high inertia in the drive train.
  - Advanced topic. Not covered here.

- Linear or lateral vibration refers to motion of an assembly in a radial or axial direction.
  - This is our topic for this session.

Vibration Terminology

Frequency:

- The number of times that a vibrating object repeats its motion per unit of time.
- The fan rotational frequency in Hertz equals the RPM / 60.
- The rotational frequency is often called 1X.
- Other frequencies are often identified as multiples of 1X, such as 2X, 3X, 3.6X, 19.2X, etc. These are called ‘Harmonics’ or ‘Orders’

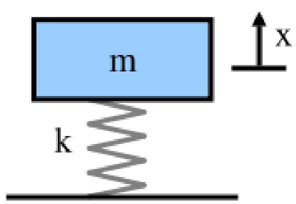
Vibration Terminology

Amplitude:

- The measure of cyclic energy or movement in a vibrating object.
- Three common ways of expressing:
  - Displacement in mils peak to peak
  - Velocity in inches/second peak
  - Acceleration in g's rms
- Amplitude measurements can be converted between the different units if the frequency is known.

What is Natural Frequency?

- Every object has its own Natural Frequency of vibration in its supporting structure.
- For fans there are typically two supporting structures: Rigid Mount and Flexible Mount .
- Fundamentals of vibration: Wikipedia



K= stiffness, lbf/in
M= mass, lbm.]

undamped natural frequency.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}.$$

Fan Engineering by R. Jorgensen

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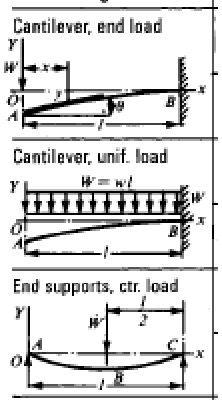


Support and Loading Diagrams	Deflections, Max
Cantilever, end load	$y = -\frac{1}{6}\frac{W}{EI}(x^3 - 3l^2x + 2l^3)$
Cantilever, unif. load	$Max\ y = -\frac{1}{3}\frac{Wl^3}{EI}\ (at\ A)$
End supports, ctr. load	$y = -\frac{1}{24}\frac{W}{EIl}(x^4 - 4l^3x + 3l^4)$

Complex computer programs are available for calculating critical speeds. These include the effects of the bearings, the supports, the foundation, and the soil. Also, gyroscopic and impeller flexibility effects can be included when appropriate. Detailed response calculations are also possible. That is, the motions at various locations can be calculated for a given excitation force.

Approximate values of critical speed can be predicted from some fairly simple equations. For a simple system, the maximum static deflection y as determined from the equations in Table 17.2, can be used in

$$N_{cr} = \frac{187.7}{\sqrt{y}} \qquad (17.31)$$

to determine the critical speed N_{cr}. The 187.7 factor corresponds to speeds in rpm and deflections in inches. For SI units, substitute 0.4984 instead.

The amount of force transmitted between members of a vibrating system depends on the masses, stiffnesses, and damping present. The ratio of transmitted force to impressed force is called transmissibility TR and is related to the natural frequency f_n of the system as well as to the disturbing frequency f. Ignoring damping,

$$TR = \frac{1}{\left(\frac{f}{f_n}\right)^2 - 1}. \qquad (17.36)$$

The natural frequency of a fan mounted on isolators can be determined from the static deflection y using

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{y}}. \qquad (17.37)$$

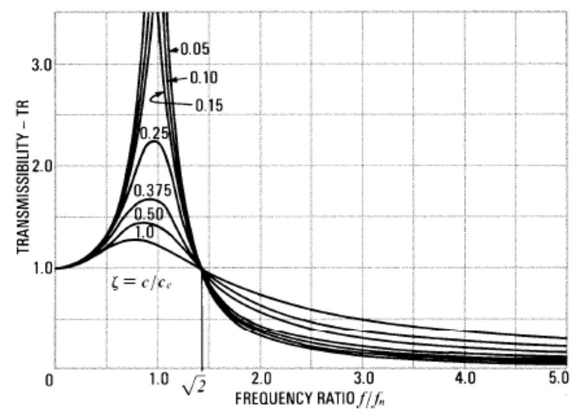


Figure 17.13 Transmissibility

Vibration Conversions

- Velocity (in/s)

  V = D(2pf)

  Where:

  D = peak displacement, (in)

  F = frequency (Hz)

  p = 3.14

- Acceleration (in/s2)

  A = V(2pf)

  Where:

  A = Acceleration, (in/s2)

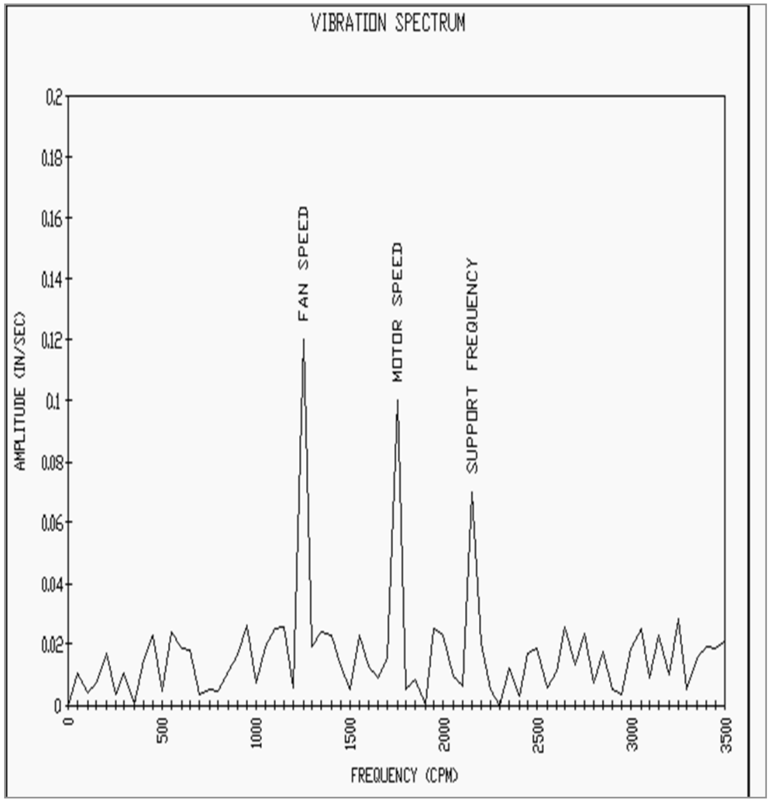
  1g = 386.1 in/s2)

- Peak = 1,414 * rms

Vibration Terminology

- Vibration Spectrum
  - A display of frequency vs. amplitude.

- The horizontal axis is linear, not logarithmic or in octave bands such as used with sound.



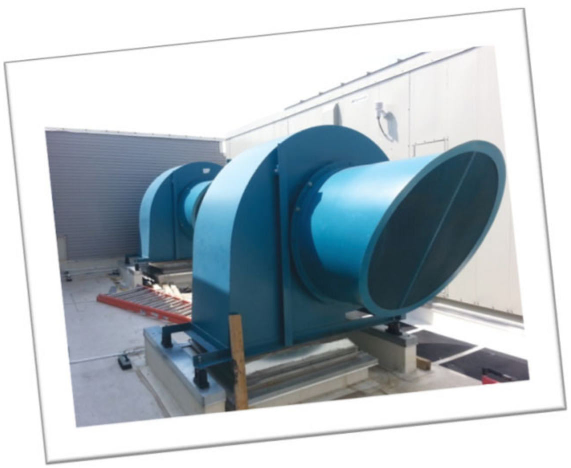
Vibration Terminology

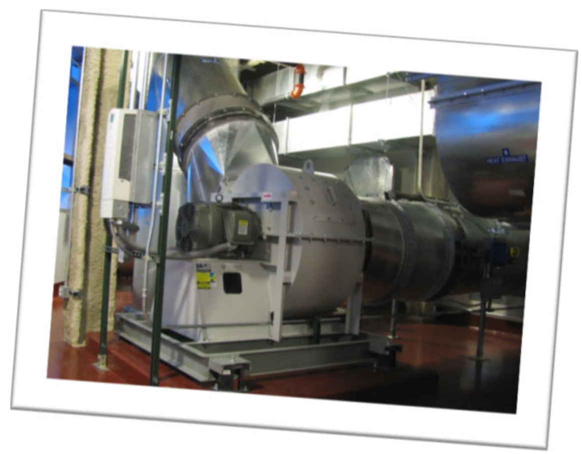
Rigidly Mounted:

- Generally applies to a fan mounted to a concrete slab.
- A very rigid steel structure may qualify.

Vibration Terminology

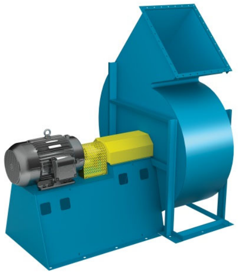
- Flexibly Mounted
  - Fan is mounted on springs or a spring isolation base.
  - An inertia base is filled with concrete then the whole assembly is mounted on springs.
    - If isolation is required, this performs best for vibration.

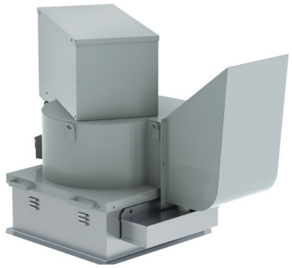




Vibration Measurement

- Vibration can be measured at any location.
  - Readings include Displacement (mils), Velocity (in/s) or acceleration (g, in/s^2).
- Normally, only the vibration on the fan or motor bearings are of concern.
- Normally, horizontal, vertical and axial readings are taken on each bearing.
  - This is also called a tri-axial set of readings.





Overall vs. Filtered Amplitudes

- Overall (filter-out) amplitude is a measure of the vibration energy over a wide frequency range.

- Filtered (filter-in) amplitude usually refers to the vibration at 1X (the operating speed).

- Overall amplitude includes the effects of all sources — even those external to the fan.

Mechanical Resonance

- Every piece of mechanical equipment has components that have their own natural frequencies.
  - For fans these include Blades, wheels, structures, housings, rotating component assemblies etc.
- When a fan's rotating frequency or its harmonics (1X, 2X…exciting frequency) coincides with the natural frequency of a fan or its components, they are excited and high amplitude vibration can result. This is phenomena called resonance.
- There are detrimental and non-detrimental amplitudes in resonance.
- Only amplitudes that exceed certain limits are considered detrimental.

Causes of Vibration

- Unbalance (the most common cause, typically 1X)
  - Fan impeller, sheaves, coupling or motor

- V-belt drive induced vibration (typically 2X)
  - Sheaves not concentric
  - Belts which are not uniform in length, worn or that have hard spots.
  - Misaligned sheaves
  - Adjustable pitch sheaves often have unequal groove spacing.

Causes of Vibration

- Shaft induced (typically 2X)
  - Shafts not straight
  - Keyways create local weak spots (center hung rotors)
  - Out-of-round shaft
  - Undersize shafting
- Bearing Faults induced (~High: 10-20 X)
  - Bearing flaws and non-symmetries
  - Bearing contamination
  - Distorted bearings (pinched)
  - Bearings which do not freely self-align
  - Distortions caused by set-screws
  - Grease or oil churning

Causes of Vibration

- Fan construction induced
  - Imperfect fan wheel components (run-out)
  - Rubbing
  - Misalignment (bearing and coupling)

- Aerodynamic (Random X)
  - Turbulent and eccentric flow through the fan
  - Fan in stall
  - Rotating stall ~ 2/3X, 4/3X…

Causes of Vibration

- Installation and environmental
  - Inadequate support (resonance)
  - Loose or missing anchor bolts
  - Loose access doors, rattling panels, etc.
  - Solids which impact the fan wheel
  - Fans with stacks vibrate due to vortex shedding

- Motor eccentricities (electrical and mechanical)

Causes of Vibration Increasing with Time

- Abrasive wear or corrosion
- Build-up of water in airfoil blades
- Material or dirt build-up
- Deformed fan components due to impact
- Damage due to over temperature
- Bearing wear or deterioration

Causes of Vibration Increasing with Time

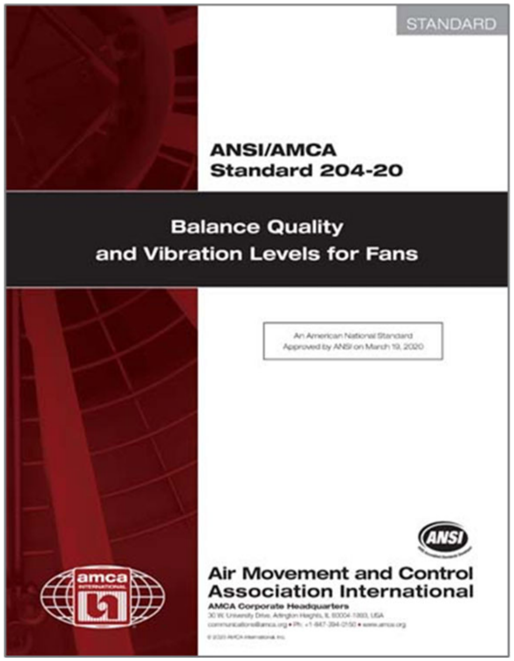
- V-belt drive wear
- Fatigue failure due to abnormal loads
- Loosening of hardware
- Change in RPM
- Lost balance weights

Consequences of Excessive Vibration

- Shortened bearing life
- Shortened lubrication life
- Structural damage to fan
- Increased noise
- Damage to adjacent processes
- Fasteners come loose

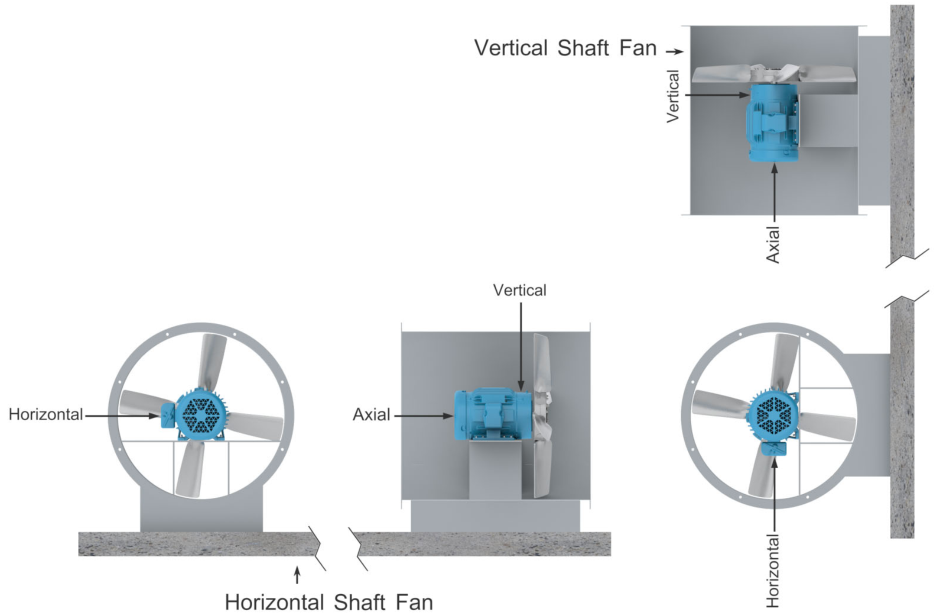
Recommended Vibration Tolerances

- Use AMCA 204 to specify vibration tolerances.
- All relevant terms are defined.
- Tolerances are realistic.
- Allows for different categories of fans.
- Provide for long life and reliable operation without excessive cost.



AMCA 204 – Vibration Measurements

- The standard says where to take vibration readings.



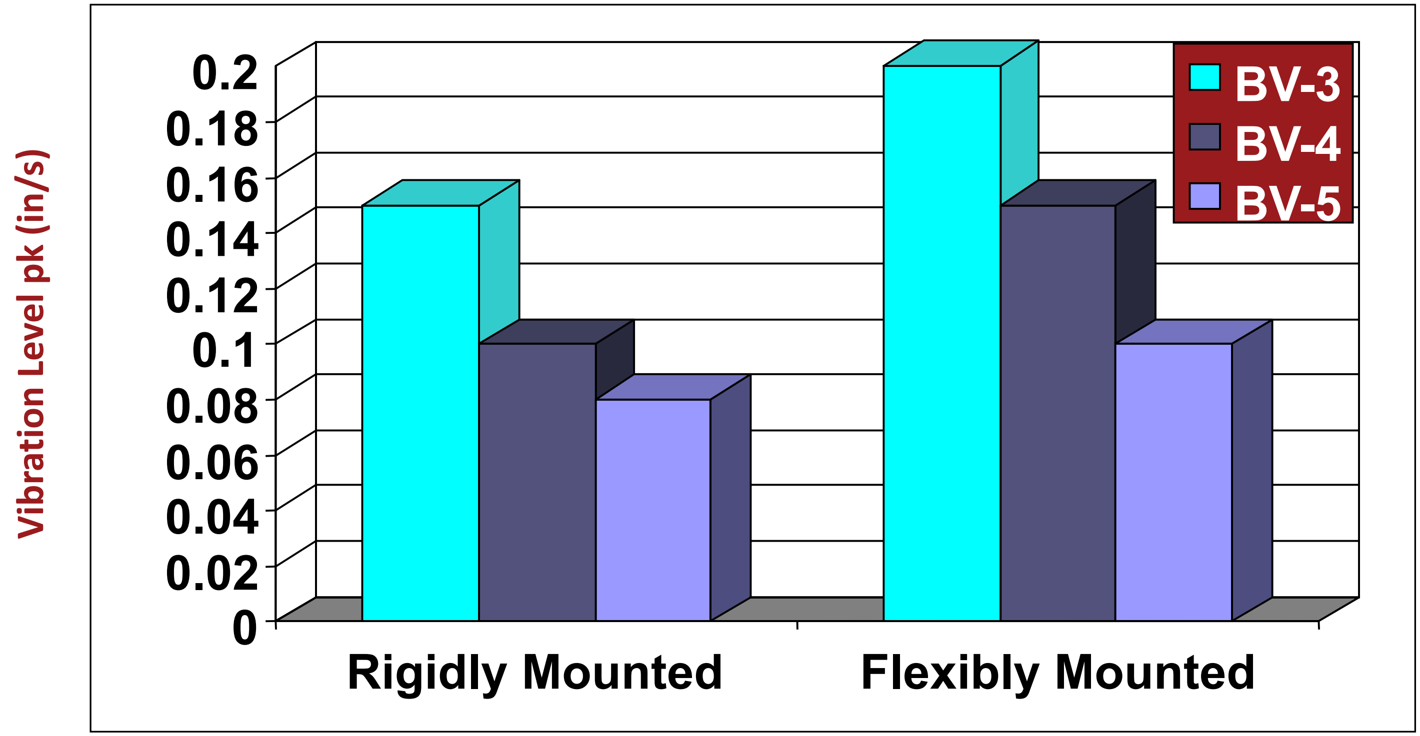
AMCA 204 Fan Categories

- The same category as for balance tolerance.

Table 6.1—Fan Application Categories for Balance and Vibration

Application	Examples	Driver Power Limits, kW (hp)	Fan Application Category, BV
Residential	Ceiling fans, attic fans, window air-conditioning unit	≤ 0.15 (0.2) > 0.15 (0.2)	BV-1 BV-2
HVAC and agricultural	Building ventilation and air-conditioning systems; commercial systems	≤ 3.7 (5.0) > 3.7 (5.0)	BV-2 BV-3
Industrial process and power generation etc.	Baghouse, scrubber, mine, conveying, boilers, combustion air, pollution control, wind tunnels	≤ 298 (400) > 298 (400)	BV-3 BV-4
Transportation and marine	Locomotives, trucks, automobiles	≤ 15 (20) > 15 (20)	BV-3 BV-4
Transit and tunnel	Subway emergency ventilation, tunnel fans, garage ventilation	≤ 75 (100) > 75 (100)	BV-3 BV-4
	Tunnel jet fans	ALL	BV-4
Petrochemical process	Hazardous gases, process fans	≤ 37 (50) > 37 (50)	BV-3 BV-4
Computer chip manufacturer	Clean room	ALL	BV-5

AMCA 204 Vibration Limits – Factory Test



AMCA Standard 204-20

Table 8.2—Vibration Limits for Factory Tests

Fan Application Category, BV	Rigidly Mounted mm/s (in./s)		Flexibly Mounted mm/s (in./s)	
	Peak	RMS	Peak	RMS
BV-1	12.7 (0.50)	9.0 (0.35)	15.2 (0.60)	11.2 (0.44)
BV-2	5.1 (0.20)	3.5 (0.14)	7.6 (0.30)	5.6 (0.22)
BV-3	3.8 (0.15)	2.8 (0.11)	5.1 (0.20)	3.5 (0.14)
BV-4	2.5 (0.10)	1.8 (0.07)	3.8 (0.15)	2.8 (0.11)
BV-5	2.0 (0.08)	1.4 (0.06)	2.5 (0.10)	1.8 (0.07)

Note: Peak values are widely used in North America and are made up of several sinusoidal waveforms that do not necessarily have an exact match with RMS values. They also may depend to some extent on the instrument used.

AMCA 204 Tests Conducted On-Site

- **Note: Do not use for fan specifications.**

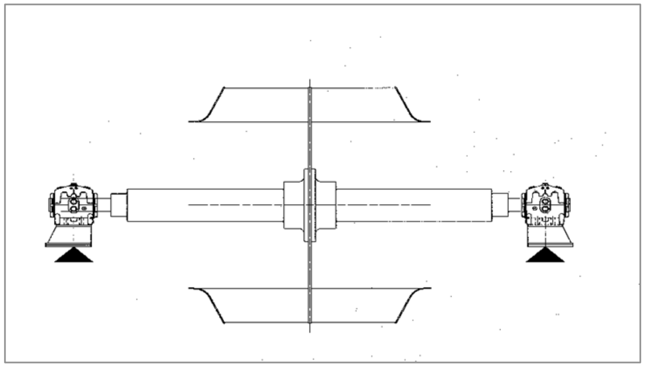
Table 8.3—Vibration Limits for In-situ Tests
The values in this table represent the overall vibration value with a minimum frequency range of 10-1,000 H
calculated FFT and analog measurements.

Condition	Fan Application Category	Rigidly Mounted mm/s (in./s)		Flexibly Mounted mm/s (in./s)	
		Peak	RMS	Peak	RMS
Startup	BV-1	14.0 (0.55)	10 (0.39)	15.2 (0.60)	11.2 (0.44)
	BV-2	7.6 (0.30)	5.6 (0.22)	12.7 (0.50)	9.0 (0.35)
	BV-3	6.4 (0.25)	4.5 (0.18)	8.8 (0.35)	6.3 (0.25)
	BV-4	4.1 (0.16)	2.8 (0.11)	6.4 (0.25)	4.5 (0.18)
	BV-5	2.5 (0.10)	1.8 (0.07)	4.1 (0.16)	2.8 (0.11)
Alarm	BV-1	15.2 (0.60)	10.6 (0.42)	19.1 (0.75)	14.0 (0.55)
	BV-2	12.7 (0.50)	9.0 (0.35)	19.1 (0.75)	14.0 (0.5)
	BV-3	10.2 (0.40)	7.1 (0.28)	16.5 (0.65)	11.8 (0.28)
	BV-4	6.4 (0.25)	4.5 (0.18)	10.2 (0.40)	7.1 (0.28)
	BV-5	5.7 (0.20)	4.0 (0.16)	7.6 (0.30)	5.6 (0.22)
Shutdown	BV-1	NOTE 1	NOTE 1	NOTE 1	NOTE 1
	BV-2	NOTE 1	NOTE 1	NOTE 1	NOTE 1
	BV-3	12.7 (0.50)	9.0 (0.35)	17.8 (0.70)	12.5 (0.49)
	BV-4	10.2 (0.40)	7.1 (0.28)	15.2 (0.60)	11.2 (0.44)
	BV-5	7.6 (0.30)	5.6 (0.22)	10.2 (0.40)	7.1 (0.28)

Notes:
1. Shutdown levels for fans in Fan Application Grades BV-1 and BV-2 must be established based on historical data.
2. Peak values are widely used in North America and are made up of a number of sinusoidal waveforms. The peak values do not necessarily have an exact match with RMS values and also depend on the instrument used to some extent.

Design Resonant Speed (Frequency)

- Natural Frequency of the Combined spring-mass of the Rotating Element, Oil Film or Rolling element, Bearing Housing and Bearing Support (of complete fan) but excluding the Foundation.
- Typical range of Design Resonant Speed (or critical speed) is ~1.3 of Max operating speed depending on the application.



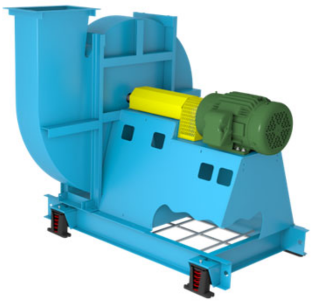
$$\frac{1}{S_{Combined}} = \frac{1}{S_{Shaft}} + \frac{1}{S_{Bearing}} + \frac{1}{S_{Housing}} + \frac{1}{S_{Pedestal}}$$

Rigid vs. Flexible Mounting

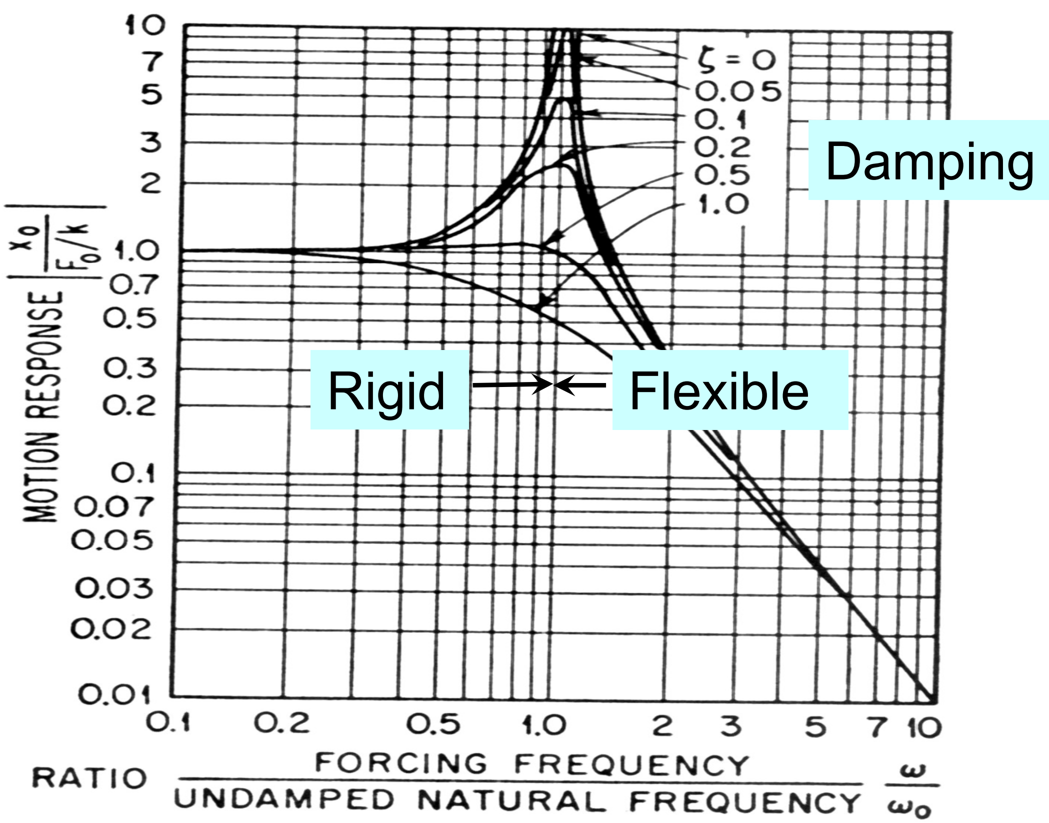
- Rigid mounting operates below the first fundamental frequency of the support system.
- Flexible mounting operates above the first fundamental frequency of the support system.

Avoid Operation at the first Fundamental Frequency


Rigid Mount


Flexible Mount

Rigid vs. Flexible Mounting



Fan Foundations

- Design for static and dynamic loads.
- Good foundations help keep the vibration levels low.

Rigid Foundation

- Employs massive concrete base.
- Grout between bottom of fan and concrete.

Flexible Mounting With Steel Base

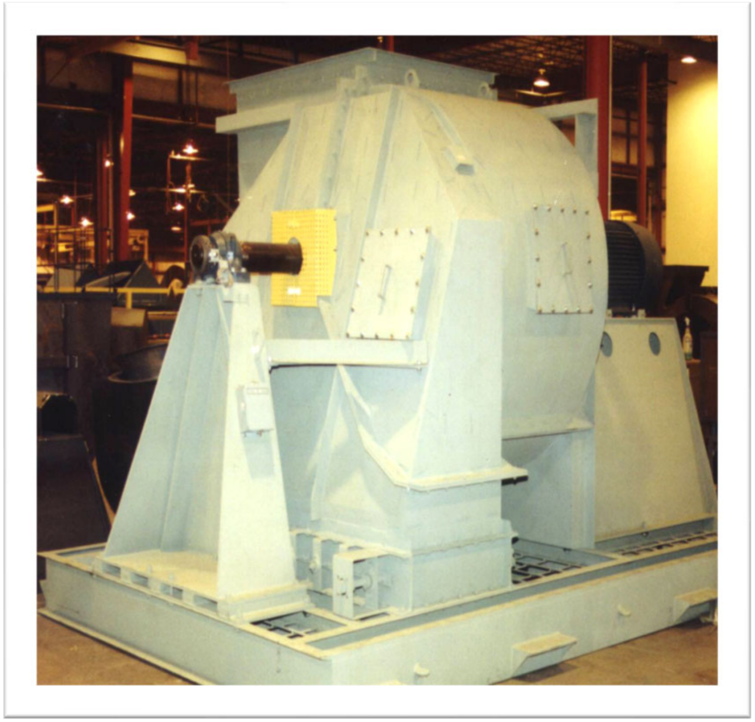
- Often uses structural C-channel or I-Beam.
- Shim between bottom of fan and steel base.
- Isolators are shipped loose, not mounted.

Flexible – Inertia Base

- Inertia base: Similar to structural steel base, except the base is filled with concrete.
- Isolators are shipped loose, not mounted.

Flexible – Inertia Base – In Factory

- Fan and Motor can be shipped mounted when specified.
- Concrete filled in field
- Isolators are shipped loose, not mounted.



Questionable Rigid Foundation Design

- Fans mounted to long span beams.
- Minimal diagonal support
- Low mass under fan
- Dynamic analysis of this structure is required to ensure problem free operation.

"Small" Fan Foundation

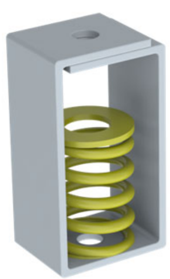
- Many small fans are mounted directly on isolators or have isolation built-in.
- Bearing vibration may be quite high.
- Low total vibration energy makes failure rates are low.
- Lower installation costs make this construction common.

Vibration Isolators

- Used to prevent fan vibrations from being transmitted to the support structure.
  - Rubber in shear isolators are used for small fans with speeds of 1,000 rpm or higher.
  - For good isolation, spring isolators are used for large fans, with lower operating speeds.

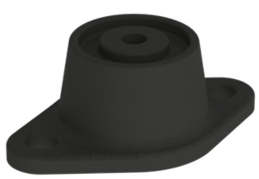
Spring Type

Floor Mount



Ceiling Mount

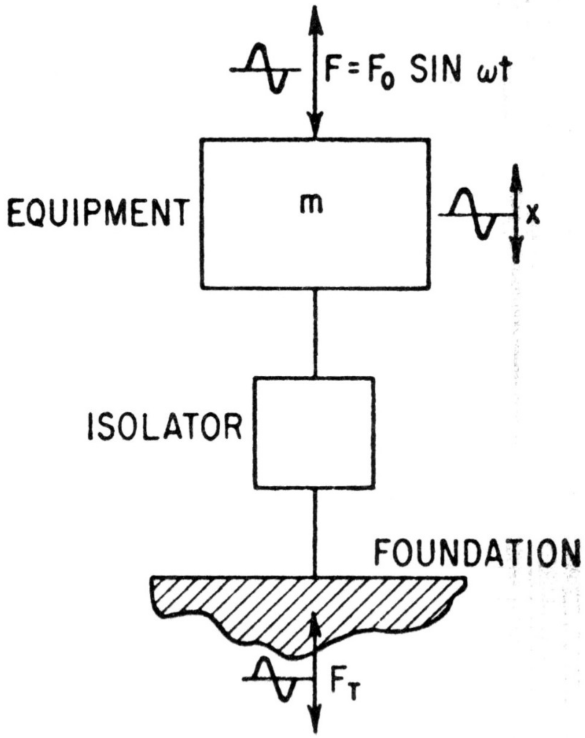
Rubber-in-Shear



Floor Mount

Ceiling Mount

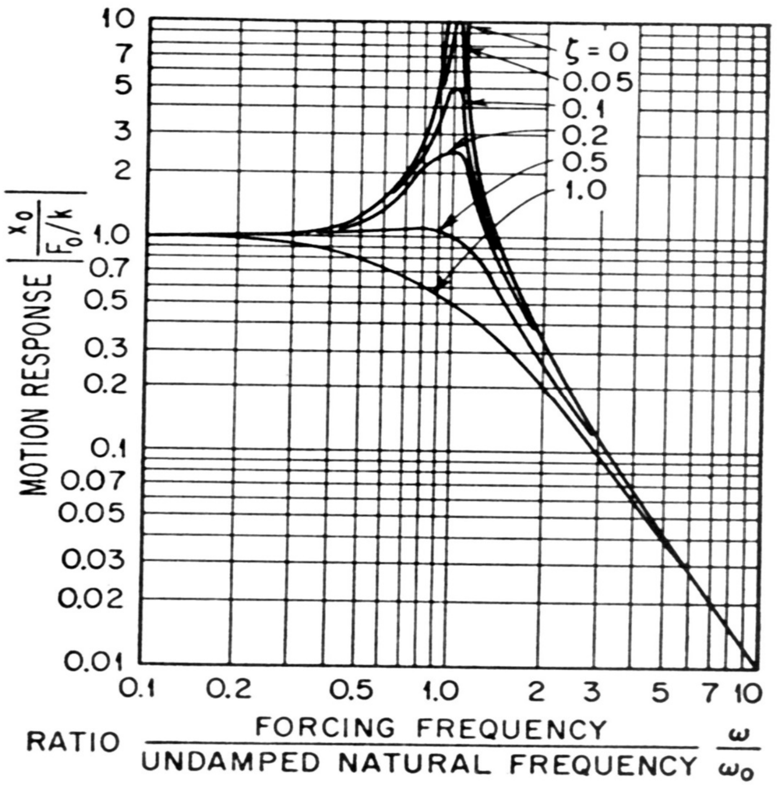
Model Used for Flexible Foundation



Single Degree of Freedom System

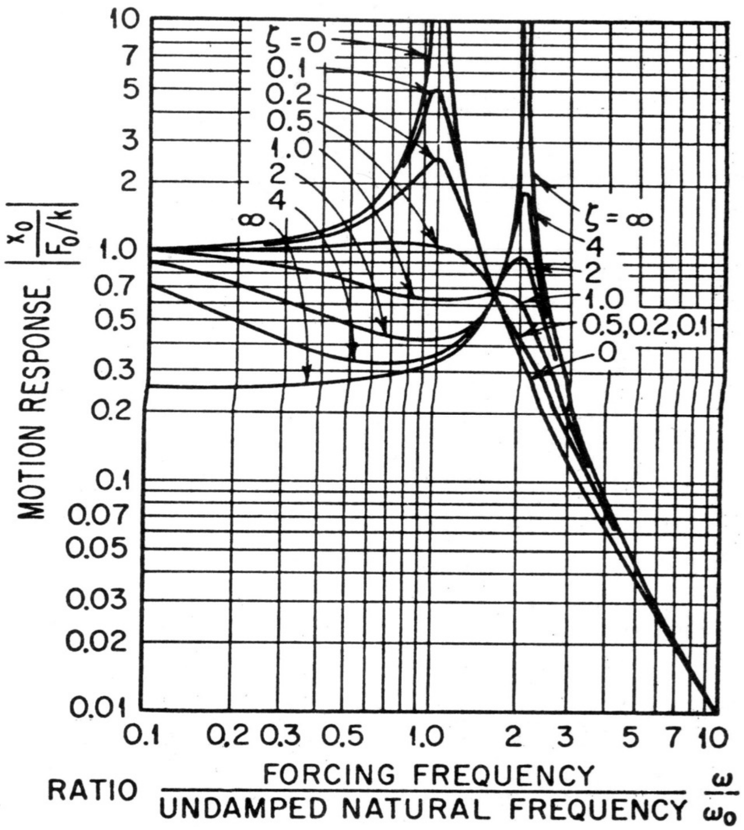
- Fan, Motor, and Base
- Springs or Rubber Pads
- Infinitely Rigid Structure

Ideal System Response

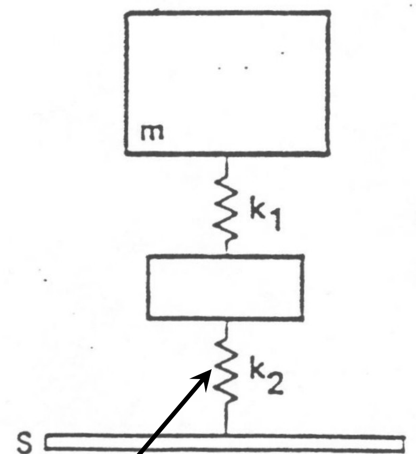


Single Degree of Freedom System

Model with Flexible Foundation

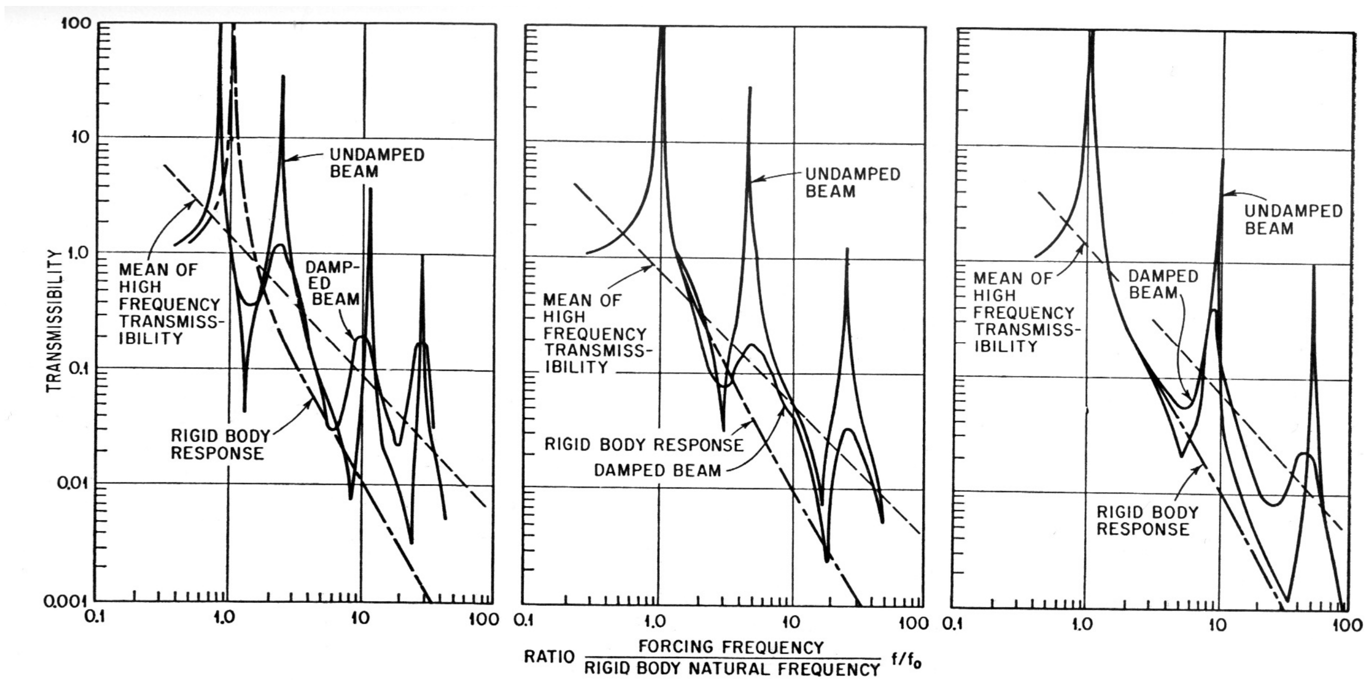


Two Degrees of Freedom System



“Springy” support (like roof) under fan springs

Multi-Degree of Freedom System



Real Systems Have Many Frequencies Where Vibration Amplification Can Occur

Characteristics of Flexibly Mount Fans

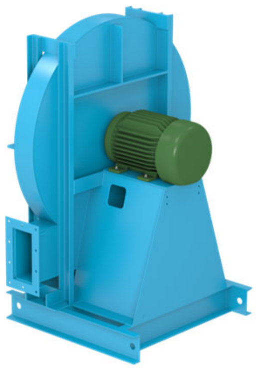
- High reliability requires sophisticated analysis.
- Most operate with higher transmitted forces than calculated.
- Almost all will “work” in spite of the potential for problems.
- Bearing vibration will be higher (most specifications allow higher tolerances).

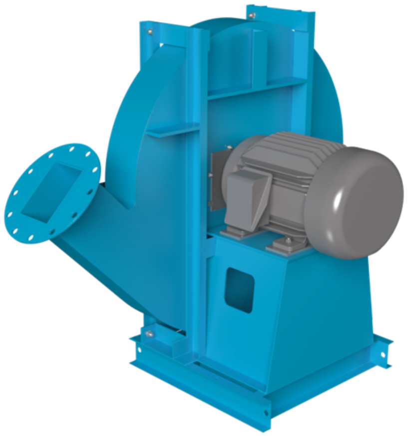
Roof Mounted Fans

- Roof mounted fans are normally rigid mounted.
  - Fans with stacks must be rigid mounted.
- Low stiffness in roofs can cause vibration problems as they act like springs.

Preferred Foundation Design

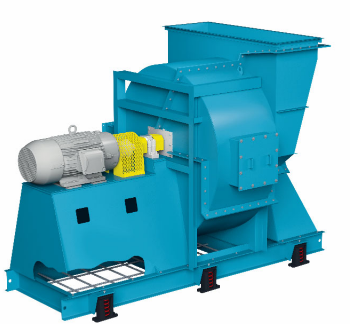
- Rigid mounting is the most reliable.
- Inertia type flexible mounting rarely has problems.
- Non-inertia type steel “spring” bases may result in problems.
- Use springs directly mounted to fans (or internal isolation) only on low power, non-critical applications.





Other Vibration Tolerances

- Come from many different sources in many different forms using varying terminology.
  - Difficult to interpret
  - May be difficult or impossible to comply due to ultra low amplitude requirements.
  - Can add significant cost to the fan.
- Fan manufacturers recommend using AMCA 204.



Lower than Standard Vibration Tolerances

- Do not specify ultra-low vibration limits unless absolutely necessary.
- Adds cost with minimal extension of fan life.

Achieving Lower Vibration Levels

- Involves better precision of many components.
  - Better balance of all rotating components.
    - Fan wheel (balance with fan shaft), sheaves, couplings and motor
  - Straighter shafts with precision diameter tolerance and roundness.
  - Premium bearings with special construction.
  - Bearing mounting surfaces machined flat.
  - Premium quality v-belt drives with precise installation.
  - Less hop and wobble in fan wheel.
- All special modifications add cost.

Dealing with Ultra-Low Vibration Specifications

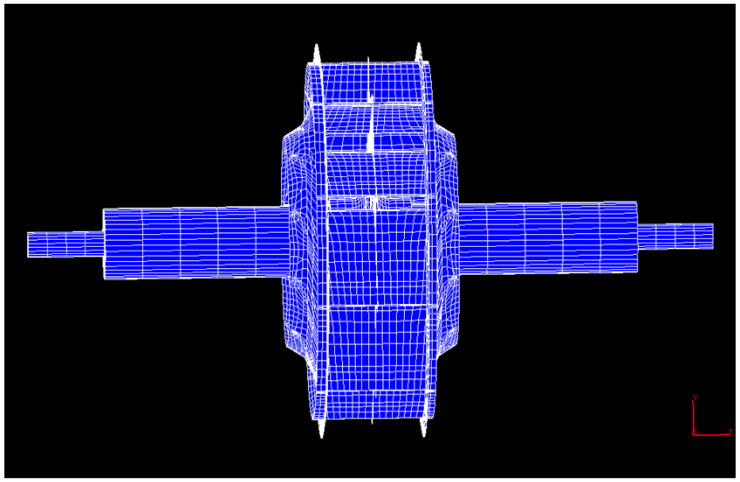
- Lower than standard filtered (1X) vibration can often be achieved with minimal difficulty.
- Overall levels or “banded” frequency ranges may be very difficult to achieve.
- Fan manufacturers normally only commit to vibration levels measured in their factories.
  - On-site conditions may have major effects.
  - Request start-up service to be part of the contract for commitment.

Some Advanced Engineering Tools

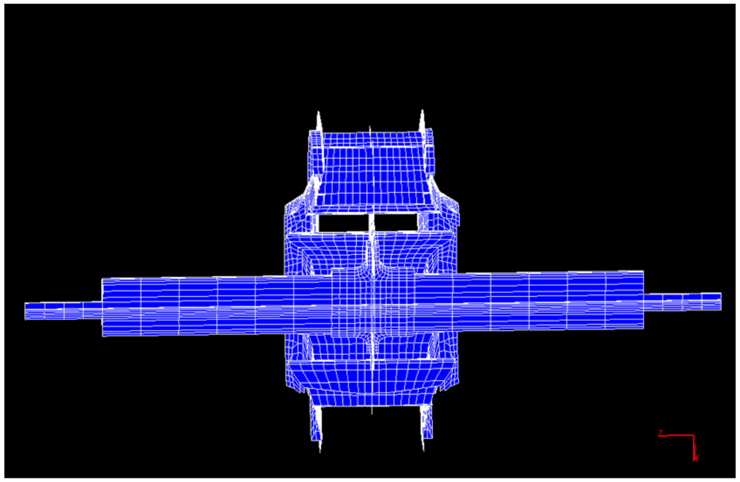
- Determination of Natural Frequencies
  - FEA Modeling
  - Bump or Impact Tests
- Coast down/ Ramp up Tests
- Waterfall Plots and ORDER analysis
- Modal Analysis, mode shapes
  - FEA modeling and animation
  - Experimental – ODS (Operational Deflection Shapes)
  - High speed Camera

Finite Element Analysis (FEA)- Modes of Vibration

Disc wobble or Diametral Mode

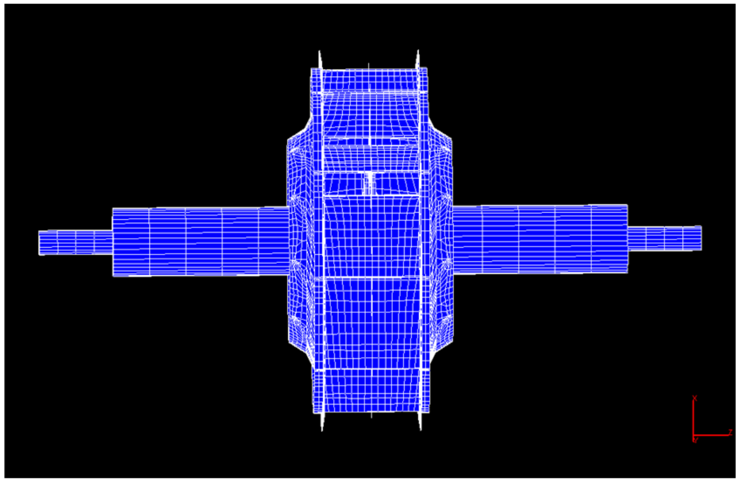


Umbrella Mode

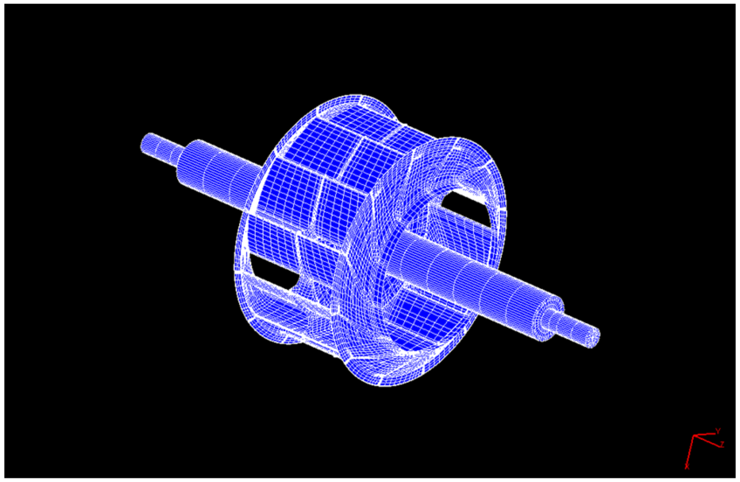


Finite Element Analysis (FEA)- Modes of Vibration

2nd Order Diametral Mode



Torsional Mode



Bump or Impact Tests to Validate FEA Models or Determine Natural Frequencies

Fan Tip Dia, in	Hub Size, in	Bld Len, in	Bld No.	Bld Angl, deg	Max. RPM	Max Shaft Fq, Hz	Max Bpass Fq, Hz	Tip Speed, fpm	Impeller Weight, lbs	Gross Weight, lbs
59.9375	16	21.9688	5	35	796	13.3	66	12491	98	<input>

Test 0	Hit laterally somewhere on frame [0] -- measure at front bearing [X]	frame/structure mount only
Test 1	Hit axially at mid-blade [1] -- measure at tip [A]	
Test 2	Hit axially at hub [2] -- measure at tip [A]	
Test 3	Hit axially at mid-blade [1] -- measure at hub [B]	
Test 4	Hit axially at hub [2] -- measure at hub [B]	
Test 5	Hit torsionally at vertical blade [3] -- measure at tip of horiz blade [C]	free-hanging torsional
Test 6	Hit axially at shaft end [5] -- measure at tip [A]	free-hanging umbrella
Test 7	Hit -- measure (custom)	
Test 8	Hit -- measure (custom)	
Test 9	Hit -- measure (custom)	

BUMP TEST Natural Frequencies, Hz

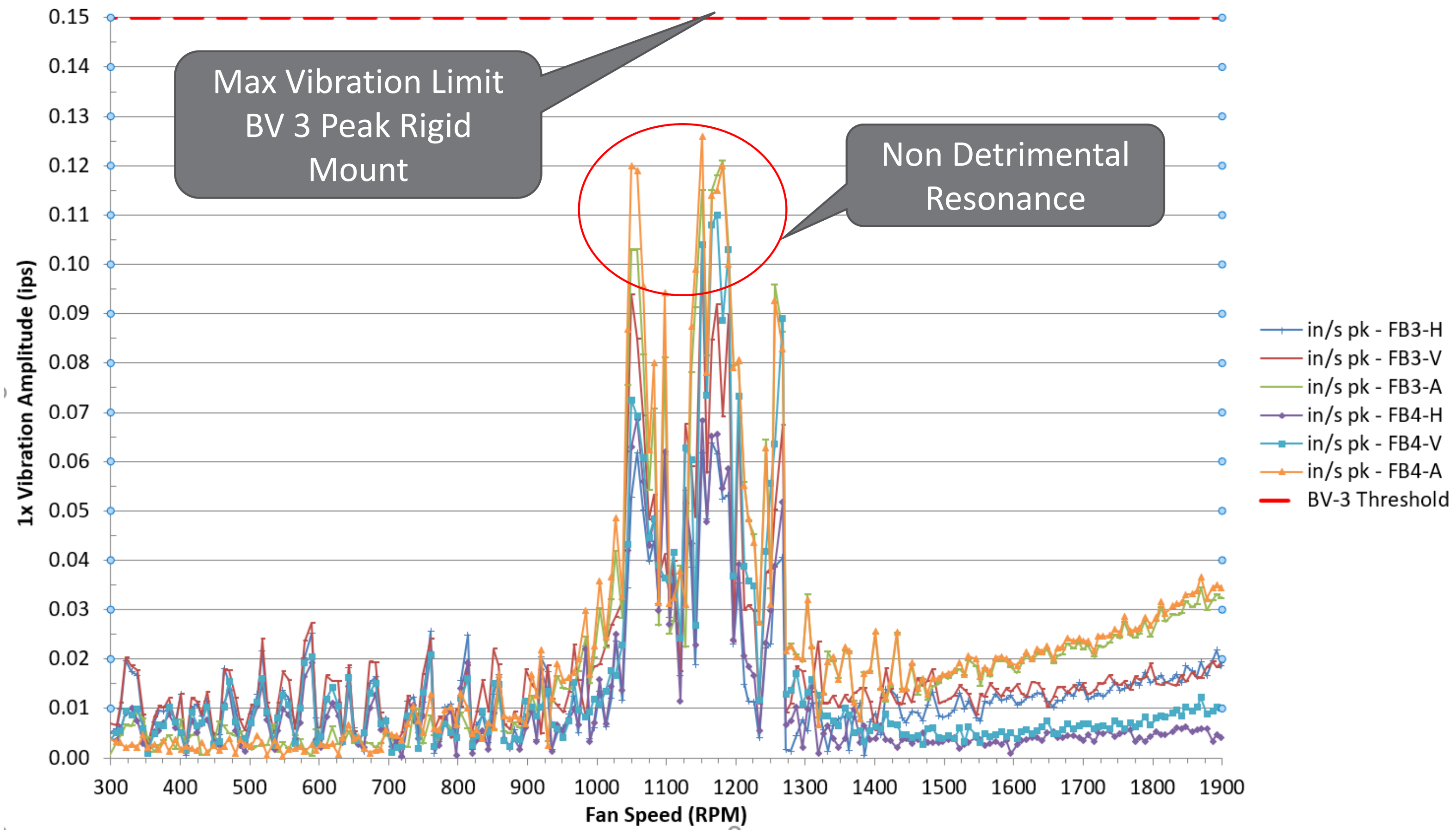
	Rigid Mount				Custom			Free Hanging					
Test 0	Test 1	Test 2	Test 3	Test 4	Test 7	Test 8	Test 9	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6
				16.21									
	16.98	16.98	16.98	16.98									
	18.88	18.88	18.88	18.88									
			22.13										
		34.71	34.71	34.71									
	49.02	49.02						49.59				49.59	
									49.78				49.78
	50.16	50.16						50.16					

Structural Natural Frequencies

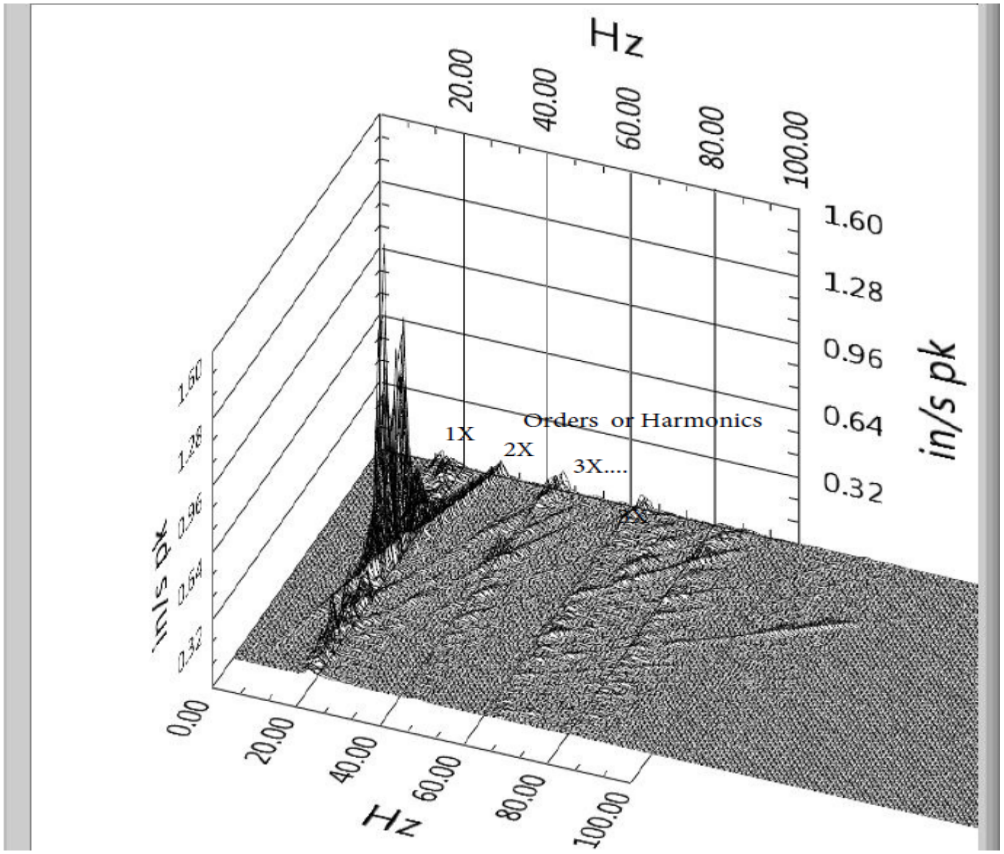
Blade Natural Frequencies

amca INTERNATIONAL

Coast Down Plot



Waterfall (order) Plots



High Speed Camera: Motion Visualization

- Basic Technology performs motion Amplification and allows measurement of displacement, Velocity and acceleration.
- This is useful in assessing root causes in vibration giving ideas on solution fixes on where / how to add mass or stiffness in a structure.

Other Reference Materials

- Twin City Fan FE-1900 - Vibration Isolation of Fans
- Twin City Fan FE-200 - Fan & Fan System Vibration
- Fan Engineering – By Robert Jorgensen, 1999 edition

Resources

- **AMCA International**: *www.amca.org*
- **ANSI/AMCA Standards:** *www.amca.org/store (Available for purchase)*
  - **> 204-20**: Balance Quality and Vibration Levels for Fans
- **AMCA Publications:** *www.amca.org/store (Available for purchase)*
  - **> 202-17**: Troubleshooting

Thank you for your time!

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If you viewed the webinar as a group and only one person registered for the webinar link, please email Lisa Cherney (lcherney@amca.org) for a group sign-in sheet today. Completed sheets must be returned to Lisa by tomorrow, June 4.

PDH credits and participation certificates will be issued electronically within 30 days, once all attendance records are checked and online evaluations are received.

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Questions?

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Join us for our next AMCA *insite* Pop-Up Webinar:

- Thursday, June 9
- 12:00-1:00pm CDT
- ***TOPIC: Louvers 101***
- Presenter: James Smardo, National Sales Manager-Louvers and Architectural Sales, AMCA Member Company

>> For additional webinar dates go to: *www.amca.org/webinar*