Overhung Fans
Reliability Improvement
Opportunities

For
Vibration Institute
Piedmont Chapter
May 9, 2002

Ken Singleton
KSC Consulting, L.L.C.
OVERHUNG FANS

Introduction

Life Cycle of Fans

Mechanical Problems Common To Overhung Fans

Considerations When Purchasing Overhung Fans
Introduction

This presentation was in response to requests for information on overhung fans. The presentation was made initially at the Feb 22, 2002 Vibration Institute Piedmont Chapter Meeting. It contains information on overhung fans that was learned over several years.

Centrifugal fans, and in particular overhung centrifugal fans, are often more unreliable than other types of rotating machinery. The reasons are many and include the fact that fans are generally low margin machines due to the very stiff competition among manufacturers.

A maintenance cost analysis may show that fan maintenance costs in $/Hp/Yr are higher than more high-end machines such as compressors or turbines. Some of these costs are directly related to the quality of the mechanical design and construction of the fans.

This presentation will discuss some documented mechanical problems with overhung fans of the AMCA arrangement 1, 2, 3, 4, 8, 9 and 10 types. References are provided for further reading. Some discussion is provided about the use of vibration, experimental modal and operating deflection shape, and rotor modeling.

The battle to achieve a smooth running, low maintenance fan will likely be fought and won over the price tag. There are many competent fan manufacturers that have the knowledge and skills to build a quality fan but quality comes at a cost. The criticality of the fan in the production stream may be used to justify those additional upfront costs to purchase are more reliable machine.
### OVERHUNG FANS

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</table>
Life Cycle of Fans

Time Line
Life Cycle of Fans

Time Line

Project Definition
Life Cycle of Fans

Develop a thorough purchase specification that references the applicable industry standards such as AMCA, ARI, ISO 1940, ANSI, ASHRAE, etc.

It is better to work with a quality vendor than try to build a bulletproof specification.
Life Cycle of Fans

Vendor Selection

Evaluate fan manufacturers and select top three.

Selecting a quality vendor will make your life much easier. Many of the problems documented have resulted from selecting the wrong fan for an application and from a vendor that did not have the capabilities to build a quality machine.

There are many Fan Manufacturers that build quality products and have excellent engineering capabilities. But, the opposite is also true.
Life Cycle of Fans

- Vendor Selection
- Specifications Development
- Project Definition
- Machine Built Inspections Witnessed Testing
- Identify and correct machine design and performance problems at the shop.
  Much easier and less expensive to correct before it leaves the shop than at your plant site.
This is the time to catch installation oversights, check isolators, alignment, vibration, performance, inlet vane and dampers operate properly, bearing properly installed and properly lubed, oil levels correct & no leaks, belts adjusted properly, bolts are tight, etc.
Life Cycle of Fans

- Vendor Selection
- Specifications Development
- Project Definition
- Machine Built
- Inspections Witnessed
- Testing
- Installation & Startup
- Operating Life Cycle

Time Line
Life Cycle of Fans

- Vendor Selection
- Specifications Development
- Project Definition
- Machine Built Inspections Witnessed Testing
- Installation & Startup
- Operating Life Cycle
- Maintenance Life Cycle

Time Line
Life Cycle of Fans

Vendor Selection
Specifications Development
Project Definition
Machine Built Inspections Witnessed Testing
Installation & Startup

Reliability Cycle
Includes Specifications & Condition Monitoring
- Vibration Analysis
- Lube Analysis
- Brg Temp Etc.

Operating Life Cycle
Maintenance Life Cycle

Time Line
Life Cycle of Fans

Be Aware: That Problems can be introduced during assembly & repair from the initial build thru-out the machine’s life cycle.

- Time Line
  - Project Definition
  - Specifications Development
  - Vendor Selection
  - Machine Built Inspections Witnessed Testing
  - Installation & Startup
  - Operating Life Cycle
  - Maintenance Life Cycle
  - Condition Monitoring using Vibration Analysis, Lube Analysis, Brg Temp & Machine Protection Instruments, Etc., Life Cycle
OVERHUNG FANS

Introduction

Life Cycle of Fans

AMCA Overhung Fan Arrangements

Overview of Documented Fans Maintenance Costs

Mechanical Problems Common To Overhung Fans

Considerations When Purchasing Overhung Fans
AMCA Arrangements 1, 2 and 3

No. 1, 2 & 3 SWSI

For belt drive or direct connection. Wheel overhung. Two bearings on base.

Arrangement 2, bearing mounted on bracket supported by fan housing.

Fan base typically attached to steel frame either grouted to concrete or supported on spring isolators.
AMCA Arrangements 4, & 8

No. 4, SWSI
For direct drive. Wheel overhung on prime mover shaft. No bearings on fan. Prime mover base mounted on integrally direct connected.

No. 8, SWSI
For belt drive or direct connect. Arrangement No. 1 plus extended base for prime mover.

†Special drawings required when using outlet flange.
AMCA Arrangements 9 and 10

No. 9, and 10 SWSI

Arrangement No. 9, prime mover outside base.

Arrangement No. 10, prime mover inside base.
AMCA Fan Class

Three classes are specified by AMCA. The classes were developed to regulate actual structural limitations of the wheels, bearings, and housing of fans.

When using a fan rating table provided by fan manufacturer, the class of the fan changes if fan speeds and static pressures increase above certain conditions.
Documented Mechanical Problems Common to Overhung Fans
AMCA Arrangements 1, 3, 4, 7, 8, 9 and 10

- AMCA Overhung Arrangements
- Overview of Reported Problems & Cost
  - Fan Frame Mounting
  - Flexible Frame & Base Design
  - Rotor Critical Speeds Often Near Operating Speed Range
  - Overhung Fans Require Lower Residual Unbalance
  - Overhung Fan Bearings More Likely To See Reduced Life
  - Fan Wheel Resonance
  - Often designed to operate above 1800 rpm.
Most of these fans were centrifugal, SWSI, AMCA Arrangement 4, and 8.
### Fan Vendor Selection Process
Data from study at one plant site

Maintenance Cost/Hp/Yr for different fan manufacturer
fans range $12.83 to $128.43

<table>
<thead>
<tr>
<th>MANUFACTURER</th>
<th>TOTAL For 4 Years</th>
<th>TOTAL # Fans</th>
<th>AVG COST PER W.O.</th>
<th>AVG. HP</th>
<th>Low HP</th>
<th>High HP</th>
<th>AVG. $/HP/YR</th>
<th>AVG. RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manufacturer 1</td>
<td>$3,256,077</td>
<td>273</td>
<td>$2,501</td>
<td>143.45</td>
<td>0.2</td>
<td>1750</td>
<td>$18.48</td>
<td>2104</td>
</tr>
<tr>
<td>Manufacturer 2</td>
<td>$3,198,578</td>
<td>627</td>
<td>$3,425</td>
<td>21.22</td>
<td>0.04</td>
<td>800</td>
<td>$53.42</td>
<td>1618</td>
</tr>
<tr>
<td>Manufacturer 3</td>
<td>$306,001</td>
<td>71</td>
<td>$13,304</td>
<td>7.00</td>
<td>0.25</td>
<td>25</td>
<td>$128.43</td>
<td>2093</td>
</tr>
<tr>
<td>Manufacturer 4</td>
<td>$203,508</td>
<td>11</td>
<td>$2,423</td>
<td>196.39</td>
<td>7.5</td>
<td>600</td>
<td>$20.93</td>
<td>1777</td>
</tr>
<tr>
<td>Manufacturer 5</td>
<td>$197,496</td>
<td>13</td>
<td>$3,086</td>
<td>105.96</td>
<td>7.5</td>
<td>300</td>
<td>$31.86</td>
<td>1630</td>
</tr>
<tr>
<td>Manufacturer 6</td>
<td>$23,102</td>
<td>2</td>
<td>200.00</td>
<td>(---)</td>
<td>(---)</td>
<td>(---)</td>
<td>(---)</td>
<td>(---)</td>
</tr>
<tr>
<td>Manufacturer 7</td>
<td>$4,652</td>
<td>2</td>
<td>1.00</td>
<td>(---)</td>
<td>(---)</td>
<td>(---)</td>
<td>(---)</td>
<td>(---)</td>
</tr>
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</table>
Maintenance Costs 4 years
Fan Study for One Well Known Brand of Heavy Duty Fan

<table>
<thead>
<tr>
<th>Service</th>
<th>Costs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Insulation Removal</td>
<td></td>
</tr>
<tr>
<td>Fuses Replace</td>
<td></td>
</tr>
<tr>
<td>Expansion Joint Repair</td>
<td></td>
</tr>
<tr>
<td>Guard Replace/Repair</td>
<td></td>
</tr>
<tr>
<td>Belts Adjust/Replace</td>
<td></td>
</tr>
<tr>
<td>HVAC Test</td>
<td></td>
</tr>
<tr>
<td>Fan Shaft Repair/Replacement</td>
<td></td>
</tr>
<tr>
<td>Motor Replacement</td>
<td></td>
</tr>
<tr>
<td>Bearing Replacement</td>
<td></td>
</tr>
<tr>
<td>Coils</td>
<td></td>
</tr>
<tr>
<td>Damper Repair</td>
<td></td>
</tr>
<tr>
<td>Fan Base Repair/Replace</td>
<td>$100,000.00</td>
</tr>
</tbody>
</table>

The chart shows the maintenance costs for different services over a 4-year period, with Fan Base Repair/Replace being the most expensive.
Documented Mechanical Problems Common to Overhung Fans
AMCA Arrangements 1, 3, 4, 7, 8, 9 and 10

• AMCA Overhung Arrangements
• Overview of Reported Problems & Cost
• Fan Frame Mounting
  • Flexible Frame & Base Design
  • Rotor Critical Speeds Often Near Operating Speed Range
  • Overhung Fans Require Lower Residual Unbalance
• Overhung Fan Bearings More Likely To See Reduced Life
• Fan Wheel Resonance
• Often designed to operate above 1800 rpm.
If Spring Isolator Mounted,
Specify 95% Isolation
Better, Concrete Mount or Inertia Base

Arrangement 3 Overhung Fan
Suggested Purchasing Guidelines
<table>
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<tr>
<th>Fan base/frame support:</th>
</tr>
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<tbody>
<tr>
<td>• <em>Spring isolators</em></td>
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<td>• Center of gravity must be within the base. If located above the base, machine sway/rocking can be a problem.</td>
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<td>• Attachment method can result in relative movement between fan frame and concrete. Cementitious grout often used, epoxy is better but proper prep work must be done.</td>
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## Fan Support

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<td>• Other mounting methods may include mounting fan on bottle rubber stoppers, no hold downs.</td>
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<td>• These methods often result in high vibration transmission, fretting corrosion, and fatigue cracking.</td>
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</table>
Coil spring isolators available with up to 5” static deflection which is equivalent to 1.4 Hz natural frequency.
### Spring Isolation Problems Often Include:

**Isolator stiffness incorrect:**
- Stiffness **too high** causing resonance to be within operating range of fan or driver.
- Stiffness **too low**, causing bottoming or collapse of isolators.

Incorrectly locating isolators so that equal static deflection is not achieved.

Incorrectly adjusting isolators causing some isolators to be overloaded and some seeing no load.

Failure to account for fan thrust load in sizing isolators causing isolators to either collapse or unload.

Under designed isolator attachment structure (fan housing) causing fatigue failure.
Isolator Selection:

- Catalog Charts
- Slide Rule Type Calculator
- Calculations
- Hand off to isolator vendor
- Let Fan vendor do it


For 600 RPM Fan with Transmissibility Ratio of 10%, Static Deflection 1 inch & Natural Freq = 180 cpm
For 600 RPM Fan with Transmissibility Ratio of 5%, Static Deflection 2 inch & Natural Freq = 140 cpm
Slide Rule Type Calculator also include isolator mounting considerations which include:

- Basement & Below Grade
- Grade and 20' Span
- 25' Span
- 30' Span
- 40' Span
- 50' Span
Example using Slide Selection Guide:
Axial and Centrifugal Fans
Floor Mounted, Above 576 RPM
II-B Isolator
Basement and Below Grade
Grade and 20' Span
25' Span
30' Span
40' Span
50' Span

<table>
<thead>
<tr>
<th>Span</th>
<th>Static Deflection</th>
</tr>
</thead>
<tbody>
<tr>
<td>25’</td>
<td>1.75” Static Deflection</td>
</tr>
<tr>
<td>30’</td>
<td>1.75” Static Deflection</td>
</tr>
<tr>
<td>40’</td>
<td>2.00” Static Deflection</td>
</tr>
<tr>
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<td>2.50” Static Deflection</td>
</tr>
<tr>
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Isolator Selection Process:

- Specified Vibration Isolation efficiency = 90% or higher
- Machine: Wt. 800lb, 4 point support with equal load distribution
- Operation speed $F_d = 540$ RPM
- Load per isolator = $800/4 = 200$ lb
- From catalog, smallest isolator to support 200 lb has stiffness of 63 lb/in

- Static Deflection
  \[ W/K = \frac{200\text{lb}}{63\text{ lb/in}} = 3.2 \text{ in} \]

- Natural frequency of isolation system
  \[ F_n = 188 \times (1/3.2)^{0.5} = 106 \text{ CPM} \sim 1.77 \text{ Hz} \]

- Isolation efficiency
  \[ I_e = 100 - \frac{100}{(F_d/F_n)^2 - 1} = 100 - \frac{100}{((540/106)^2 - 1)} = 96\% \]

- Transmissibility Ratio
  \[ TR = \frac{1}{(F_d/F_n)^2 - 1} = \frac{1}{(540/106)^2 - 1} = 0.04 \sim 4\% \]

Reference: Per Buffalo Forge Fan Engineering
Transmissibility Ratio (TR) should be <0.20 and preferable <0.10
Notes on Isolators:

- Several isolators are usually required to support the fan.
- They should be installed according to the distribution of load to achieve equal static deflection. Specify minimum 90% isolation, better 95% each isolator.
- The isolator load includes the dead weight of the equipment, and thrust forces.
- The fan and drive motor should be mounted on a common base.
- Steel springs are typically used. Cork can be used above 1200 RPM. Rubber in-shear can be used above 700 RPM (Rubber should not be used in compression since it is considered incompressible)
- Vibratory short circuits must be prevented. Flexible connections must be used between the fan and duct work at the inlet and discharge. Steam, water, power, lubrication lines, must be flexibly connected.
- An inertial mass may be needed to restrict the amplitude of vibration regardless of whether the fan is mounted on isolators or not. Center of gravity must be below the top of the mass.
- For concrete base; rule of thumb, mass of concrete should be three times the mass of the fan and drive. Pay attention to location of center of gravity, too high can cause rocking (rigid body modes) of the base and fan.

Reference:
Fan Engineering 8th Edition, Buffalo Forge Company
WMC Korfund Steel Spring Vibration Isolators Bulletin No. K29/93
Documented Mechanical Design Issues Common to Overhung Fans
AMCA Arrangements 1, 2, 3, 4, 8, 9 and 10

- Fan Frame Mounting/Support
- Flexible Frame & Base Design (especially if isolator mounted)
- Rotor Critical Speeds (Often Near Operating Speed Range)
- Overhung Fans Require Lower Residual Unbalance
- Overhung Fan Bearings More Likely To See Reduced Life
- Fan Wheel Resonance
- Operating Speed Above 1800 (Reduced reliability)
ACMA Arrangement 1
Frame Supported On Isolators

Problems:
• Frame Natural Frequency in Bending (Long Axis of Frame)
• Frame Natural Frequency in Torsion

Frame bending mode, long axis:
Controlled by depth of side beams. May require boxing beams or adding gussets.

Fan shown was modified by welding stiffeners to the channel frame.

A second channel was welded to this side.
ACMA Arrangement 1
Frame W/Isolators

Problems:
- Frame Natural Frequency in Bending
- Frame Natural Frequency in Torsion

Frame bending mode, long axis:
Controlled by depth of side beams. May require modification by boxing beams or adding gussets.

Frame torsional mode:
Controlled by depth of beams. May require frame modification by boxing beams or adding bracing.
ACMA Arrangement 1
Frame W/Isolators

Problems:
- Frame Natural Frequency in Bending
- Frame Natural Frequency in Torsion

Frame bending mode, long axis:
Controlled by depth of side beams.
May require modification by boxing beams or adding gussets.

Frame torsional mode:
Controlled by depth of beams.
May require frame modification by boxing beams or adding bracing.

Frame bending mode, side-side:
Controlled by cross bracing.
**Undeformed Structure**

Mode 1: Fan 1X, 1230 CPM  
Freq: 20.50 Hz  Damp: 0.00%

Mode 3: FAN 2X, 2490 CPM  
Freq: 41.50 Hz  Damp: 0.00%

**Isolator Resonance**

- Fan 1X = 1200 cpm
- Fan 2X
Inertia Base, Concrete Filled
Reference: Vibration Mountings & Controls, Inc.
Wide Flange Beam Base W/Isolators

Reference: Vibration Mountings & Controls, Inc.
Specification Notes, Frames Supported on Isolators:

- Specify heavy duty frames. Square or rectangular tubing or wide flange beam is better than channel.
- Use inertial base if mounting fan on structural steel. Make sure center of gravity is below top of base.
- Specify that no global frame resonance is allowable within design operating frequency range.
- Witness test critical fans for conformance to specification.
Frame/base resonance can still be a problem when attachment is directly to concrete base. The fan base and the motor support plate can exhibit excessive flexure due to resonance. In example shown, modal analysis and operating deflection shape analysis were used to diagnose the problem of excessive vibration on test stand.
The fan vendor added stiffeners under the motor support plate to eliminate the resonance problem. Several stiffeners were added inside the base to increase stiffness, thus moving resonance away from fan running speed.
Specification Notes, Fan Base/Frame Attached to Concrete:

- Specify that no base resonance is allowable within specified operating frequency range.

- **Specify epoxy grouting. (Make sure proper process is used and vendor preps frame properly)** Reference: Perry C. Monroe, Pump Baseplate Installation And Epoxy Grouting Seminar, Aug 1991

- Provide option to fill base with grout but make sure vent holes are drilled to prevent voids.

- The stiffness of the bearing base plate, pedestal and foundation should be $> 1 \times 10^7$ lbs/in.
Bearing Pedestal Design

Common problems:
- Pedestal Material Too Thin, and Flexible
- Bearing Support Surface Not Flat
- Bearing Support Resonance
Bearing Pedestal Plate 5/16” Bearing Support Surface out of flat 1/8 “ caused housing distortion, short bearing life. Severe bearing failures damaged shafts which required replacement.

Concrete pad mass too low resulted in excessive vibration.

A major part of the costs for shafts, base repair & replacement, and bearing replacement was for these AMCA Arrangement 8 Fans.
Specification Notes, Bearing Pedestal:

- Specify minimum thickness bearing support plate of 1” Pl < 50Hp, 2” Pl > 50Hp.
- Machine bearing support plate to 0.002 to 0.003 in/ft flatness.
- The bearing supports shall not exhibit structural resonance within the operating speed range of the fan.
- Alignment jack bolts shall be provided to facilitate alignment of the bearing housings.
- Mounting holes shall be drilled (not burned)
Mechanical Problems Common to Overhung Fans
AMCA Arrangements 1, 4, 8 and 9

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- Often designed to operate above 1800 rpm.
Overhung Fan Criticals

- Modeling of overhung fan rotors can be difficult. Wheel gyroscopics forces transferred to the shaft affect the location of the 1st critical.

- Bearing dynamic loading is affected by location of 1st critical above or below running speed.

- Bearing loading is lower if operation super-critical.

- Not uncommon for fan manufacture to calculate 1st critical higher than it measures.
Overhung Fan Criticals

- Example rotor modeled using ARMD. Operating speed range 500 to 4,000 rpm.
- Bearing stiffness of 500,000 lbs/in, Estimated bearing damping was added.
Operating Speed 3,000 RPM
Bearing K .5E6 lb/in, damping 500 lb-sec/in.
Unbalance 1 Oz-in at fan wheel

1st Critical 3,750 RPM

Wheel End Brg Dynamic Load 105 lb
Design Operating Speed 3,000 RPM
Increasing fan speed very common when more gas flow is needed.

Increasing speed to 3550 RPM
Wheel End Brg Dynamic Load increases to 300lb
Fluid Film Bearings:
Fluid film bearings provide more damping than rolling element bearings. Usually provide longer life. Consider fluid film bearings for critical applications and where shaft size > 4.00 inches.

Dodge Sleevoil bearings have elliptical bore which creates oil wedge in bottom and top. Dip ring feed or re-circulation oil system required.

Practical experience: at lightly loaded drive end, high eccentricity ratio should be maintained to provide stable operation, i.e., oil whirl. May require reducing effective length of bearing pad.

Scrubber Fan,
3.937 Shaft Dia,
SWSI Fan Wheel,
Overhung
Dodge Sleevoil Fluid Film Bearings
Wheel end bearing, 1087 lb load down
Brg Length 7.500 inch. Oil pressure wedge develops to support shaft.

Drive end bearing, 125 lb load up
Brg Length 2.75 inch. Oil film bearing can become unstable unless adequate eccentricity ratio is maintained. Note that oil wedge develops in the top of the bearing since shaft load is up.
Notes, Overhung Fan Criticals

- Rotor criticals will likely be close to operating speed region due to overhung mass of the wheel. These will include 1st and 2nd rigid body modes controlled by bearings/bearing support stiffness and the 1st flexure mode primarily controlled by shaft stiffness and the overhanging wheel mass.
- Stiffness of the wheel & hub affect gyroscopic forces transferred to the shaft, which affect the 1st critical. The more flexible the wheel, the lower the critical.
- Not uncommon for fan manufacturers to calculate the critical higher than it actually measures. (Another item to confirm on test stand.)
- Flexible shaft design may lower bearing loading and provide longer bearing life. Shaft stresses must be evaluated.
- Bearing L10 life will likely be reduced when the rotor is operating too close the 1st critical due to increased dynamic loads.
- Bearing support stiffness is of paramount importance. Flexible, stitch welded bases will lower the bearing support stiffness.
- Require rigid bearing base plate, pedestal and foundation stiffness > 1 X 10^7 lbs/in.
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Overhung Fan Balance

- Overhung fans required lower residual unbalance tolerance.

- Shops often fail to balance per ISO 1940/1 Standard for overhung rotors.

- Shops may use arbors rather than actual shaft and balance fan centerhung using $U_{per}$ for centerhung rather than overhung wheel.

- Wheel hub may use loose fit to shaft with set screws instead of interference fit.

- Airfoil design wheels typically hollow. Moisture can condense causing unbalance. (Specify continuous weld. Some HVAC wheels will have vent holes drilled in trailing edge.)
Fan Wheel Potential Unbalance Problems Areas

- Wheel radial and axial runout Exceeding 0.003 X OD
- Correction weights thicker than base material (potential stress cracks)
- Sharp Corners (potential stress cracks). Should have continuous welds, and NDT
- Loose Shaft to hub fits
- Published recommendations call for wheel hub initial fit in excess of 0.002 inch for thermal expansion plus an allowance for hub expansion due to centrifugal force.
- Set screws may initially hold hub tightly in position on the shaft but setscrew tips may corrode or wear.
Fan Wheel Potential Unbalance Problems Areas

- Wheel radial and axial runout Exceeding 0.003 X OD
- Correction weights thicker than base material (potential stress cracks)
- Sharp Corners (potential stress cracks). Should have continuous welds, and NDT
- Loose Shaft to hub fits
- Loose fits of wheel capture plate if used
- Key Cut Off Center & loose
- Eccentricity of the solid hub
Fan Wheel Potential Unbalance Problems Areas

- Wheel radial and axial runout exceeding 0.003 X OD
- Correction weights thicker than base material (potential stress cracks)
- Sharp Corners (potential stress cracks). Should have continuous welds, and NDT
- Loose Shaft to hub fits

Note circular stiffener: Install a tapered section to help eliminate buildup of dirt or material in gas stream which could cause changes to wheel balance if large mass thrown off.
### Balance Quality Grades

**ISO 1940/1**

**Table 1 - Balance quality grades for various groups of representative rigid rotors**

(From ISO 1940/1)

<table>
<thead>
<tr>
<th>Balance quality grade</th>
<th>Product of the relationship ((\omega_p \times \omega_{f1}) \times 10^{-6})</th>
<th>Rotor types — General examples</th>
</tr>
</thead>
<tbody>
<tr>
<td>G4 000</td>
<td>4 000</td>
<td>Crankshafts/dies of rigidly mounted slow marine diesel engines with uneven number of cylinders</td>
</tr>
<tr>
<td>G1 600</td>
<td>1 600</td>
<td>Crankshafts/dies of rigidly mounted large two-cycle engines</td>
</tr>
<tr>
<td>G30</td>
<td>630</td>
<td>Crankshafts/dies of rigidly mounted large four-cycle engines</td>
</tr>
<tr>
<td>G250</td>
<td>250</td>
<td>Crankshafts/dies of rigidly mounted marine diesel engines</td>
</tr>
<tr>
<td>G100</td>
<td>100</td>
<td>Crankshafts/dies of fast diesel engines with six or more cylinders</td>
</tr>
<tr>
<td>G40</td>
<td>40</td>
<td>Complete engines (gasoline or diesel) for cars, trucks and locomotives</td>
</tr>
<tr>
<td>G16</td>
<td>16</td>
<td>Car wheels, wheel rims, wheel sets, drive shafts</td>
</tr>
<tr>
<td>G10.3</td>
<td>10.3</td>
<td>Crankshafts/dies of semi-automatically mounted fast four-cylinder engines (gasoline or diesel) with six or more cylinders</td>
</tr>
<tr>
<td>G2.5</td>
<td>2.5</td>
<td>Crankshafts/dies of engines with six or more cylinders under special requirements</td>
</tr>
<tr>
<td>G0.4</td>
<td>0.4</td>
<td>Drive shafts (propeller shafts, cardan shafts with special requirements)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Parts of crushing machines</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Parts of agricultural machinery</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Individual components of engines (gasoline or diesel) for cars, trucks and locomotives</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Electric turbines</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Paper machinery rolls, print rolls</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Fans</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Assembly of aircraft gas turbine rotors</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Flywheels</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Pump impellers</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Machine-tool and general machinery parts</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Medium and large electric armatures of electric motors having at least 80 mm shaft height without special requirements</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Small electric armatures, often mass-produced, in vibration insensitive applications and/or with vibration-isolating mountings</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Individual components of engines under special requirements</td>
</tr>
</tbody>
</table>

1) \(\omega_p = \frac{2 \pi f_1}{60} = \frac{\pi}{10}\), if \(\omega_p\) is measured in revolutions per minute and \(f_1\) in radians per second

2) For allocating the permissible residual unbalance to correction planes, refer to “Allocation of “per to correction planes”

3) A crankshaft/die is an assembly which includes a crankshaft, flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc.

4) For the purposes of this part of ISO 1940/1, slow diesel engines are those with a piston velocity of less than 9 m/s; fast diesel engines are those with a piston velocity of greater than 9 m/s

5) In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft/die described in note 3 above.

---

**Use G2.5 for fans Rather than G6.3 which is most frequently used**
Balance Quality Grade Tolerance for correction planes between bearings can be selected using charts.

Select Rotor Speed and Weight, connect with line and read the unbalance tolerance in each plane for the quality grade.

**Fig 5-8 BALANCE TOLERANCE NOMOGRAM FOR G 2.5 & G 6.3**

- **Based on ISO 1940, and ANSI G2.19**
- **MAX SERVICE SPEED OF ROTOR IN RPM**
- **RECOMMENDED TOLERANCE IN GRAM-INCHES**
- **ROTOR MASS IN LB**

<table>
<thead>
<tr>
<th>QUALITY GRADE</th>
<th>G-2.5</th>
<th>G-6.3</th>
</tr>
</thead>
<tbody>
<tr>
<td>100000</td>
<td>10</td>
<td>20</td>
</tr>
<tr>
<td>50000</td>
<td>5</td>
<td>10</td>
</tr>
<tr>
<td>20000</td>
<td>2</td>
<td>5</td>
</tr>
<tr>
<td>5000</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>2000</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

**NOTE 1:** For disc-shaped rotors, use full recommended value for one plane. For rigid rotors with 2 correction planes, use one-half the recommended value for each plane.

**NOTE 2:** For additional instructions, see ISO or ANSI documents.

**EXAMPLE:** 2000 lb rotor with service speed of 1800 rpm in quality grade G-2.5, has 2700 gram-inch tolerance, i.e. 1350 gram-inches in each of (two correction planes).

1 g-in = 0.053 oz-in
1 oz-in = 28.35 g-in
Balance Quality Grades ISO 1940/1

Symmetrical Rotor, Center Hung

\[ U_{\text{per \ Left}} = U_{\text{per \ Right}} = U_{\text{per}} / 2 \]
Where: \( U_{\text{per}} \) = Permissible Residual Unbalance

Center Hung Rotor
d = 16 in
G2.5
RPM 3000
b = 4 in
Shaft+Wheel Wt 187 lb
a = 6 in
\( U_{\text{per}} = 28 \text{ gr-in} \sim 1 \text{ oz-in} \)
or 14 gr-in/plane
Overhung Rotor
G2.5
RPM 3000
Shaft+Wheel Wt 187 lb

\[
U_{\text{per Static}} = \frac{U_{\text{per}}}{2} \times \frac{d}{2c}
= \frac{28 \text{ gr-in}}{2} \times \frac{16}{2 \times 24}
= 4.6 \text{ gr-in} \sim 0.16 \text{ oz-in}
\]

\[
U_{\text{per Couple}} = \frac{U_{\text{per}}}{2} \times \frac{3d}{4b}
= \frac{28 \text{ gr-in}}{2} \times \frac{3 \times 16}{4 \times 4}
= 42 \text{ gr-in} \sim 1.47 \text{ oz-in}
\]

In this example:
- Unbalance Static Component Centerhung wheel 1 oz-in
- Overhung wheel .16 oz-in
- Tolerance 1/6 in overhung

Reference:
IRD Mechanalysis, The Practical Application of ISO 1940/1, Technical Paper 14
Schenck Trebel, Fundamentals of Balancing April 1990
Specification Notes, Balancing Overhung Fans

- Require hub bore to be interference fit to shaft
- Wheel radial & axial run out no greater than 0.003 X OD
- Dynamically balance with correction in 2 or 3 planes
- Welded weights no thicker than base material, full radius corners, continuous weld, minimum 1 inch from wheel OD, NDT all welds
- All welding to be complete prior to balancing
- Balance on fan shaft, not arbor
- Balance to quality grade G2.5 per ISO 1940/1 for overhung rotors
- Specify largest access panel possible to provide access to wheel for cleaning and balancing.
- Verify balance on test stand with unbalance test run.
  - Take initial amplitude and phase during 1hr test run
  - Add test mass
  - Spin up to measure amplitude and phase
  - Calculate residual unbalance in rotor. If out of spec, then correct either in place or return to balance machine.
Specification Notes, Balancing Overhung Fans

- Request copy of balance machine calibration & date.
- Note general condition of the balance machine. You don’t want your shaft bearing fits damaged by grit or worn rollers.
- Request complete balance report showing initial unbalance, final unbalance, residual unbalance check, weight correction and location.
- AMCA 204-96 Balance Quality and Vibration Levels for Fans
  - Assigns Fan Application Category
  - Based on Category, assign allowable vibration and balance quality grade
  - Category BV-4 (>400 Hp, Industrial Process and Power Generation) calls for Balance G2.5
- Probably no need to witness balance. The run test will identify any balance problem.
### Field Balancing Overhung Fans

Balancing cannot correct:
- Flexible bearing pedestals
- Flexible or cracked frames
- Loose components
- Defective bearings
- Misaligned sheaves or sheave runout
- Misaligned shafts
- Airfoil blades partially filled with water
- Structural resonance, etc.

| Have you ever received a request about a vibrating fan and the customer wanted you to “balance it”?
| Everyone knows that the only reasons fans vibrate is unbalance, right? |

- Perform detailed vibration analysis and inspection prior to balancing. Clean fan.
- Overhung fans are more sensitive to unbalance forces than center hung. More tolerant of couple unbalance than static.
- Be very precise in weight placement, amount, and that data is stable. Adjacent fans with similar run speeds can cause phase instability as can seal rubs inducing thermal bowing.
Mechanical Problems Common to Overhung Fans
AMCA Arrangements 1, 4, 8 and 9

- AMCA Overhung Arrangements
- Overview of Reported Problems & Cost
- Fan Frame Mounting
- Flexible Frame & Base Design
- Rotor Critical Speeds Often Near Operating Speed Range
- Overhung Fans Require Lower Residual Unbalance
- Overhung Fan Bearings More Likely To See Reduced Life
- Fan Wheel Resonance
- Often designed to operate above 1800 rpm.
Overhung Fan Bearings

- Wheel end bearing sees highest loading due to overhung mass of wheel.

- Drive end bearing may be up loaded and see very low load. Can result in skidding of rolling elements and high temperatures, reduced life.

- Bearing life can be adversely affected by bearing support stiffness and flatness.

- Some high speed overhung fans may require special bearings to obtain satisfactory L10 life.

- Maintenance of bearings can be difficult since shaft removal, coupling removal, and motor movement may be required if non-spacer coupling used.
Shafts above 4.00 inch dia, consider fluid film bearings

Check d*N Index:
- \( d \) = bearing bore mm
- \( d^*\) = pitch diameter mm of bearing
- \( D \) = Bearing OD mm
- \( N \) = RPM

- 340,000 Grease lubed radial ball
- 300,000 Grease lubed cylindrical roller
- 145,000 Grease lubed spherical roller
- 140,000 Grease lubed thrust bearing ball & roller

Oil sump lubrication can be used up to a speed index of 300,000 for lubrication of rolling bearings (Reference: The Lubrication of Rolling Element Bearings, FAG Bearings Corp., Pub. No. WL81 115/2 ED).

For oil lubrication, consider dry sump rather than wet sump lubrication where high temperatures are a concern

Bearing L10 life is a function of bearing load/speed capability & operating speed, radial and axial loads
• On high speed fans & blowers, it is often necessary to use special bearings (tunnel bearings,) to achieve satisfactory L10 life and operating temp.
• Ball bearings have higher speed rating than spherical roller bearings

• Check the fixed bearing’s ratio of radial to axial loads. This often calls for fixing the wheel-end bearing and allowing the drive-end bearing to float (due to very low radial load on the drive end bearing). The shaft is allowed to grow toward the coupling. If this setup is used, verify that the coupling can handle the required growth.

• Use of taper fit increases size of spherical roll bearing and reduces the upper speed limit.
The minimum load for spherical roller bearings can be considered to be 2% of the bearing's dynamic capacity.

Reference: William Detweiler, Spherical Roller Bearings in Fans, SKF

Fluid film bearings can offer advantages over rolling element bearings for overhung fans.

- Split bearing and housing permits easier change-out. Rolling element bearing change-out is typically more difficult and may require pulling the shaft, coupling removal, and moving the motor when a non-spacer coupling is used.
- Longer running life, higher radial and axial load capacity.
- Sealing of oil must be addressed.
- Pay attention to journal surface speed for ring oiling, <2500 ft/min.
Mechanical Problems Common to Overhung Fans
AMCA Arrangements 1, 4, 8 and 9

- AMCA Overhung Arrangements
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- Fan Wheel Resonance
- Often designed to operate above 1800 rpm.
Fan Resonance

- Overhung fan wheels will have structural resonance which may be excited by 1X, harmonics and blade passing frequencies
- Fatigue failures of the backplate (web), blades and shrouds can occur
- Better fan companies are using FEA and modal testing
- Modal testing can identify potential problems if used during shop testing
- Modal testing in the field can also be used to identify resonant frequency problems of fan wheels and rotor
Fan Wheels Are Flexible Structures:

- Excitation of the Wheel 1\textsuperscript{st} Circular Mode Can cause fatigue failure of the back plate.

- The resonance may not present a reliability problem if wheel construction is robust.

- If 1\textsuperscript{st} diametrical mode is near running speed, found to cause unbalance sensitivity and can cause fatigue failure of the back plate.
60" Blower Wheel. Hub Failure due to cyclic fatigue.
60" Blower Wheel

Hub Failure due to cyclic fatigue

1st diametrical wheel mode near running speed and 1st rotor critical also near running speed
30” HVAC Wheel

Wheel backplate fatigue failure due to excitation of wheel resonance. Cracked about 270 degrees through bolt circle attachment to hub.

Very light weight construction, 1/8 material, aluminum.
FRF of fan wheel on test. 2\textsuperscript{nd} diametrical mode too close blade passing frequency. But response was below 1 mil/lbf.
Wheel, Shaft and Base
Modal Test Points

MODAL TEST IN FAN BEARINGS
Wheel and Shaft
Free-Free Modal Test Points
Specification Notes, Fan Wheel Resonance

- Modal test critical fan wheels. Wheel can be tested Free-Free mounted on shaft supported by nylon slings or installed which will also show shaft static critical and base modes.

- Modal test should identify natural frequencies of the wheel backplate (web), blades, shroud, and blade stiffeners.

- No natural frequencies should have response greater than 1.0 mil/lbf in frequency range 2 Hz to 800 Hz.

- No wheel resonance should occur with in 10% of fan 1X, 2X or blade passing frequency.

  - Shroud OD and ID midway between two blades
  - Backplate at the OD in axial and traverse direction
  - Backplate in the radial (torsional mode)
  - Blade tip midway between backplate and shroud (looking for first mode with highest stresses at blade attachment points)
  - Blade inlet midway between hub and shroud
Specification Notes, Fan Wheel Resonance

- Large diameter, high tip speed wheels with long shroud spans between blades can have resonant frequency response problems that result in vibration and fatigue failure unless adequate thickness or blade numbers are used.

- Radial blade wheels may experience blade fatigue failure due to bending or backplate (web) fatigue failure causing scalloping (also occurs with centrifugal compressors).
Mechanical Problems Common to Overhung Fans
AMCA Arrangements 1, 4, 8 and 9

- AMCA Overhung Arrangements
- Overview of Reported Problems & Cost
- Fan Hung Fan Frames Often Mounted on Spring Isolators
- Flexible Frame & Base Design
- Overhung Fans Require Lower Residual Unbalance
- Overhung Fan Bearings More Likely To See Reduced Life
- Rotor Critical Speeds Often Near Operating Speed Range
- Fan Wheel Resonance
- Often designed to operate above 1800 rpm.
Fans at 3560 rpm are often used for extremely high pressure/low volume applications (above 60 in H₂O static pressure. However, in many applications this desired operating point can be achieved with

* A single stage 1780 rpm fan or
* Two 1780 rpm fans in series or
* Two-stage unit operating at 1780 rpm

Wheel erosion/corrosion unbalance affects much greater at higher speeds

Example:

30 dia wheel, 3550 rpm
1.0 oz buildup lost at radius of 15 inch
\[ F = 1.77 \times 1 \times 15 \times (3550/1000)^2 = 335 \text{ Lbf} \]

60 dia wheel, 1780 rpm
1.0 oz buildup lost at radius of 30 inch
\[ F = 1.77 \times 1 \times 30 \times (1780/1000)^2 = 168 \text{ Lbf} \]
Higher speeds tend to cause reduced bearing life, higher vibration, lower reliability. The moment of inertia \( I = \frac{\pi d^4}{64} \) of a shaft. Shafts and bearing are proportionally smaller above 1800 rpm, i.e. L10 life usually lower for higher speed.
OVERHUNG FANS

Introduction
Life Cycle of Fans
Mechanical Problems Common To Overhung Fans
Considerations When Purchasing Overhung Fans
General Purchasing Guidelines
Mechanical Design Issues
For Overhung Fans

- **Determine the importance of the fan in the process or “Fan Criticality”**
  The criticality determines the quality of the machine, details of the specification, testing and condition monitoring requirements.

- **Qualify and Select a maximum of three Fan Vendors to quote.**
  Evaluate the vendors in the following areas:
  - **Engineering Capability** (Performance Testing, Performance Modeling, Rotor-Bearing Dynamics, Finite Element Analysis, Materials, etc.)
  - **Shop Capabilities** (Fabrication, CAD & CNC for cutting, shaft machining, welding, NDT, rotor balancing, etc.)
  - **Testing facilities** (Can they test your fan to full power?) What test equipment is available, condition, calibration? Does the facility have transient multi- channel vibration equipment if required?

- There can be advantages to establishing business relationships with rotating equipment vendors.
• Be aware of the practice of low balling bids, then expensive adders begin to show up for items such as run tests, balancing to G2.5, vibration limits, etc.

• Develop Specifications that reference applicable industry standards such as AMCA, ASME, ISO, NEMA, etc.
  • Rating of the fan should be established in accordance with ANSI/AMCA Standard 210 or ASME Power Test Code 11.

  • Clearly define vibration and bearing temperature limits.
  • AMCA 204-96 Balance Quality and Vibration Levels for Fans Assigns Fan Application Category
  • Based on Category, assigns allowable vibration and balance quality grade
  • Category BV-4 (>400 Hp, Industrial Process and Power Generation) calls for Balance G2.5 and filter in 0.10 in/sec. (Be careful of the filter in only vibration limit)
• Direct drive, 1800 rpm more reliable then higher speed belt drives.
  • If variable speed is required, consider VFD.
  • Variable inlet guide vanes can help reduce the need for variable speed.
  • Typically, bearing life and belt life are shorter on higher speed units.
  • If reliability is a primary concern, consider two separate fans in series, or a two stage fan.

### Sample vibration tolerance

<table>
<thead>
<tr>
<th>Bearing Type &amp; Mount</th>
<th>Freq Span</th>
<th>Max Vib Overall</th>
<th>Max Vib Discrete Freq</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rolling Element, Rigid Support</td>
<td>2Hz - 1KHz</td>
<td>0.12 in/sec pk</td>
<td>0.07 in/sec pk</td>
</tr>
<tr>
<td>Rolling Element, Frame Isolator Mount</td>
<td>2Hz - 1KHz</td>
<td>0.15 in/sec pk</td>
<td>0.07 in/sec pk</td>
</tr>
<tr>
<td>Fluid Film, Rigid Mount-Casing Vibration</td>
<td>2Hz - 1KHz</td>
<td>0.10 in/sec pk</td>
<td>0.05 in/sec pk</td>
</tr>
<tr>
<td>Fluid Film, Rigid Mount-Journal Vibration</td>
<td>DC - 1KHz</td>
<td>1.0 mils pk-pk</td>
<td>0.50 mils pk-pk</td>
</tr>
</tbody>
</table>
  Relative disp (compensated)                    |
  Relative disp (compensated)                    |
• **Witness Test to enforce the specification**
  • Run test with all contract components, 1 hr run after bearing temperatures stabilize
  • Monitor vibration, bearing temperatures
  • Follow with unbalance response test to qualify balance and sensitivity
  • Open bearings for inspection
  • If on isolators, don’t permit isolators to be blocked
  • If inertia base specified, test on inertia base

• **Better to select a quality manufacturer than try to force a lower quality manufacturer to meet a strict specification**
• Don’t ship until it passes

• Be careful of delay tactics to avoid running witnessed tests or attempt to ship and fix later at the site. Tests will be pre-run by the vendor before you are allowed to witness. If there are problems during the pre-run, the witnessed tests may be delayed without explanation.

There may be efforts to delay or avoid running the tests or attempt to ship and fix later at the site. (Project managers have been known to buy the excuse that manufacturing schedules have slipped. This can bring pressure on site reliability people if construction schedule will be impacted.)

• Identify installation problems at the site during startup and correct.
**Mechanical Design Considerations For Arrangement 8 Overhung Fan**

- Bearing Housing Prepped For Accelerometers & RTD/Thermocouples
- Horizontal & Axial Alignment Screws, At Center of Motor Feet VFD
- Stainless Shims
- Top Plate, <50 Hp 1", >50Hp 2", >250Hp 3" Machined Flat
- Access Holes To Bolts
- Internal Gusseting
- Hinged Access Door For Cleanout And Balancing, Large As Possible
- Fan Rotor Balance Per ISO 1940/1 To G2.5
- Heavy Duty Spindle Brg W/ Oil Lubrication or Fluid Film Bearing Design
- Non-lubricated Coupling, <50Hp Elastomeric >50Hp Disc-Pak
- Heavy Duty Linkage & Construction If Discharge Damper Used
- Fan Wheel Shrink Fit To Shaft, Radial & Axial Runout 0.003 X OD
- Flexible Connection to Inlet
- Inlet Guide Vanes
- Heavy Duty Construction External Linkage, W/Lubrication
- Drain, Accessible Without Crawling Under Fan
- Flexible Connection to Inlet
- Epoxy Grout Frame To Concrete if Grouted, Inertial Base if Installed On Structural Steel
- Concrete Mass Minimum 3 Times Mass of Fan
- Air Gap Between Fan Housing And Base for Hot Gases

Mechanical Design Considerations For Arrangement 3 Overhung Fan,

- Top Plate, 1”, Machined Flat 0.002-0.003”/ft
- Sheave Runout 0.002 Radial & Axial
- Minimum Overhang
- Tubing or Wide Flange Frame
- Spindle Type Bearing Housing Preferred
- Gusseting Between Housing and Base May Be Required
- Fan Rotor Balance Per ISO 1940/1 To G2.5
- Bearing Pedestal Continuous Weld Internal Stiffeners
- If Spring Isolator Mounted, Specify 95% Isolation Better, Concrete Mount or Inertia Base
- No Resonance of Adjustable Motor Mounts In Operating Range
And finally; if you don’t want the problems of an overhung fan:
And finally; if you don’t want the problems of an overhung fan:

• Buy an Arrangement 3, or
• Modify an existing problem overhung fan to an arrangement 3
References:

- Fan Performance and Design, Robinson Industries, Inc.
- Engineering Letters, The New York Blower Company
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- Ken Singleton, Analysis of Blower Vibration Using Experimental Modal and Operating Deflection Analysis, 1993
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The End