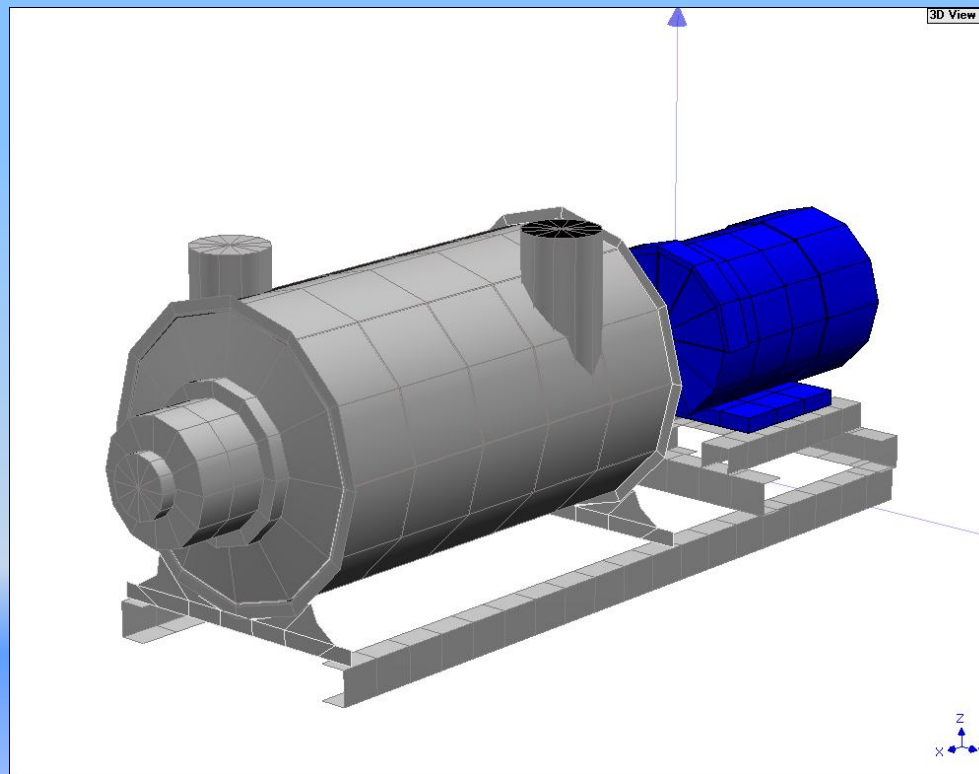


# Reliability Improvement Project Multi-Stage Centrifugal Blowers

For  
Piedmont Chapter  
Vibration Institute  
Charleston, SC Sept 9, 2011



**Steve Quillen**  
Eastman Chemical Company  
squillen@eastman.com

**Ken Singleton**  
KSC Consulting LLC  
ksingleton@vibrationconsulting.com

## Background

- Project to improve reliability of six Multi-Stage Centrifugal Blowers.
- Blowers typically operated 12 to 18 months between failures.
- Expected 5 year Mean Time Between Failure (MTBF).
- **Reported problems**
  - **Bearing failures**
  - **High vibration**
  - **Could not maintain alignment**
- **Blowers were fabricated, centrifugal type.**
  - 1500 ICFM, 150 Deg F Inlet Temp
  - Inlet Pressure 14.05 psia, Differential Pressure 10.0 psig
  - Discharge Pressure 24.5 psig, Specific Gravity 1.0
  - 125 HP, 3575 RPM
  - Direct Drive, Altra-Flex Couplings.
- **Motor:** 125 HP, 460 V, 143 Amp, 3563 RPM, Frame 444TS, Wt 1650 lb

## Background

- In-depth analysis of the six blowers included the following:
  - Vibration Analysis (Spectra, Time Waveform, PeakVue)
  - Transient Vibration Data Analysis (Runup/Coastdown).
  - Operating Deflection Shape Analysis (Three Blowers).
  - Experimental Modal Analysis.
  - Rotor Dynamic Modeling.
  - Piping Analysis (by the Plant's Engineering) .
  - Continuous Laser Alignment Measurements.
  - Witnessed Shop Disassembly, Balancing and Reassembly, Run Test.

## Background

Process gas was corrosive, required stainless steel construction of shaft, impellers, hubs, housings, piping.



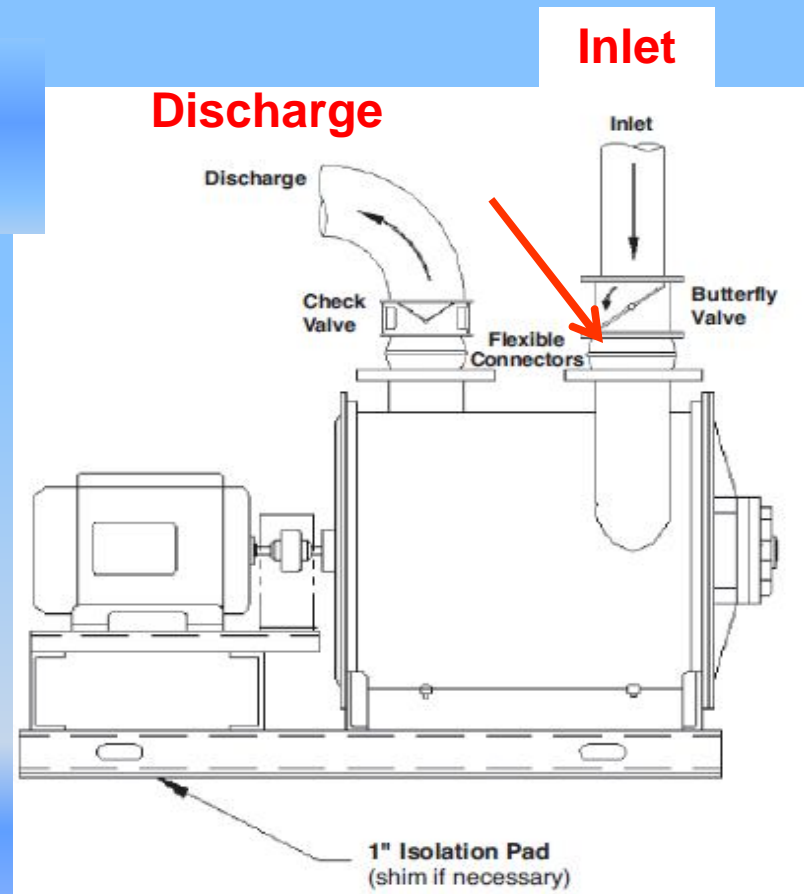
**Figure 1. Photo of Multi-Stage Centrifugal Blowers.**

# Background

**Stainless Steel Bellows  
Connected to Piping at Inlet and  
Discharge.**



**Figure 1. Photo of Multi-Stage Centrifugal Blowers.**



**Figure 2. Drawing From OEM Manual Showing Piping Connection and 1" Isolation Pad Under Skid.**

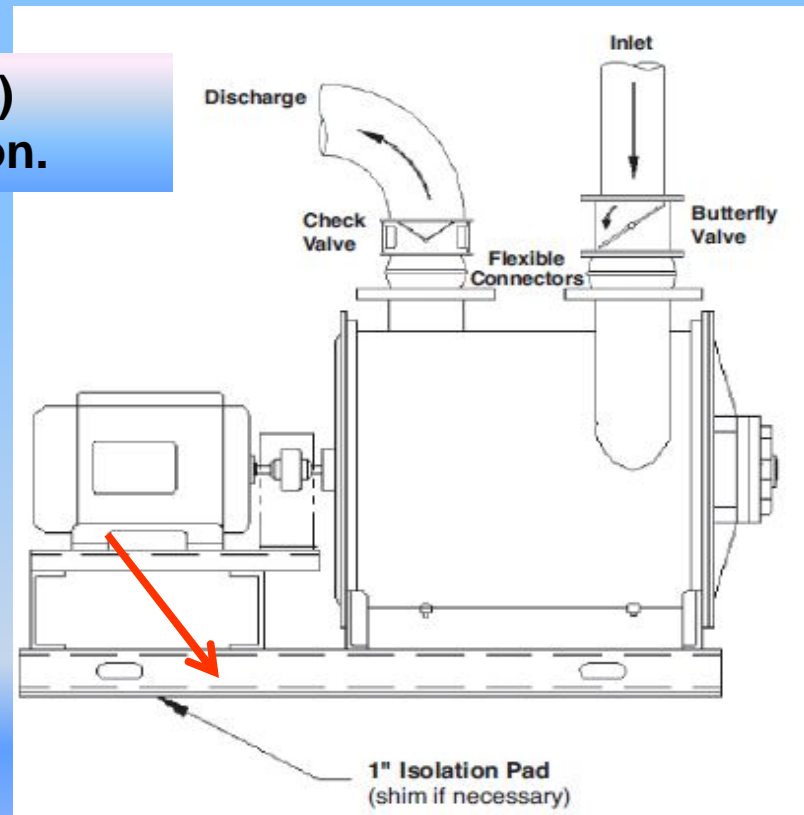


# Background

**Stainless Steel Skid (Channel)  
Supported on 1" Cork Isolation.**



**Figure 1. Photo of Multi-Stage Centrifugal Blowers.**

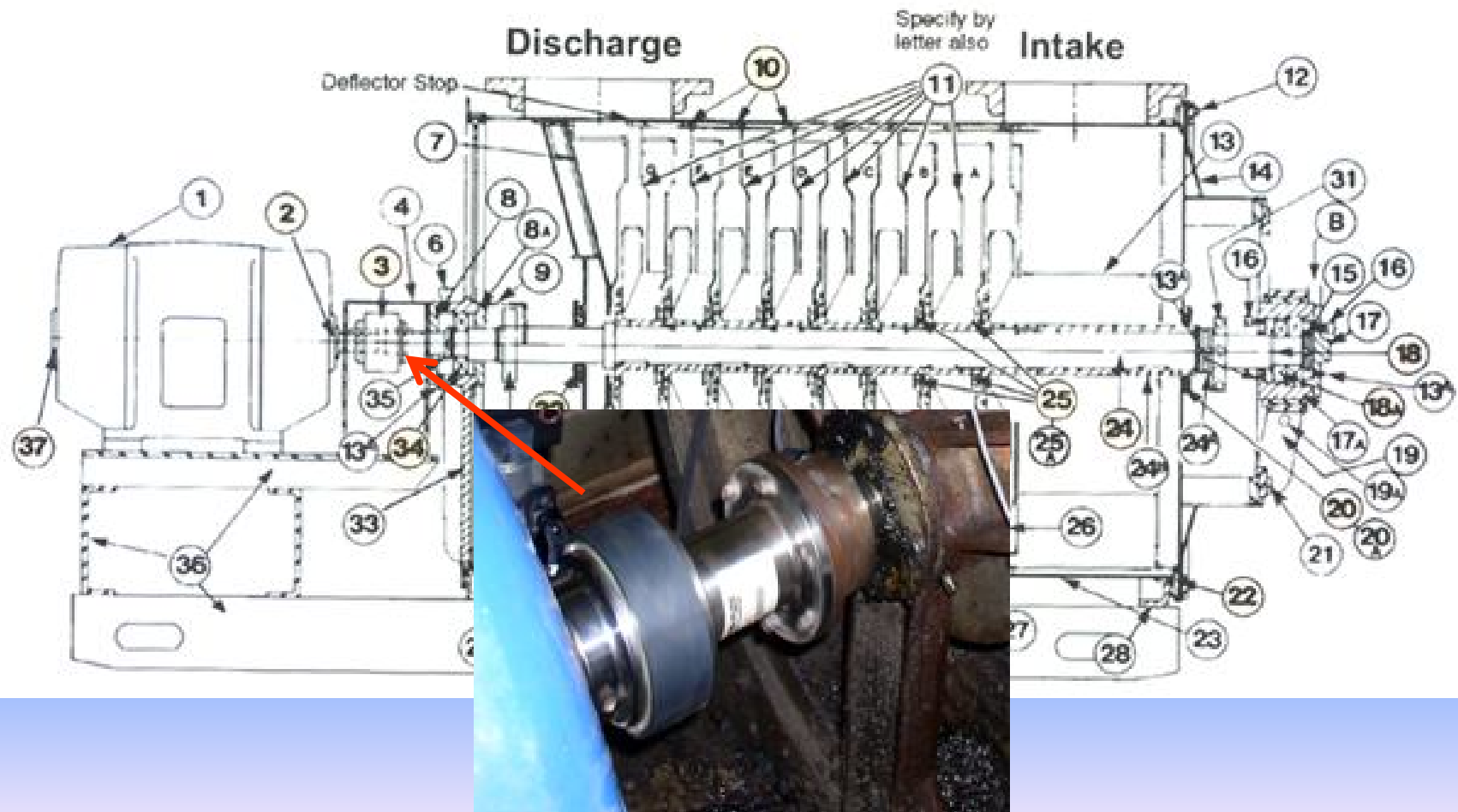


**Figure 2. Drawing From OEM Manual Showing Piping Connection and 1" Isolation Pad Under Skid.**

## Background

10 Stage Blower, Direct Coupled Using Alta-Flex Coupling.

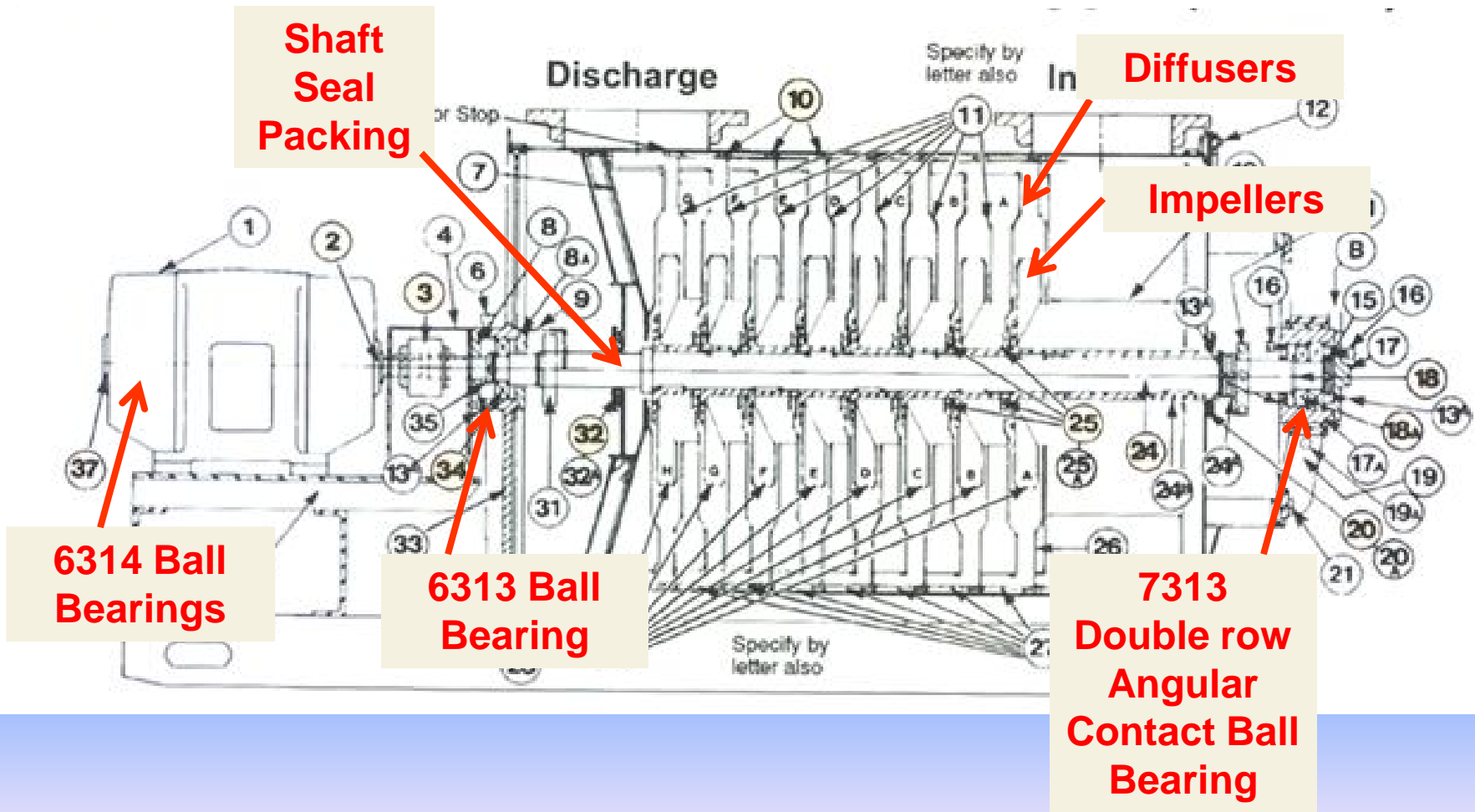
## Typical Multi-Stage Centrifugal Blower, Direct Drive.



# Background

## 10 Stage Blower

### Typical Multi-Stage Centrifugal Blower, Direct Drive.

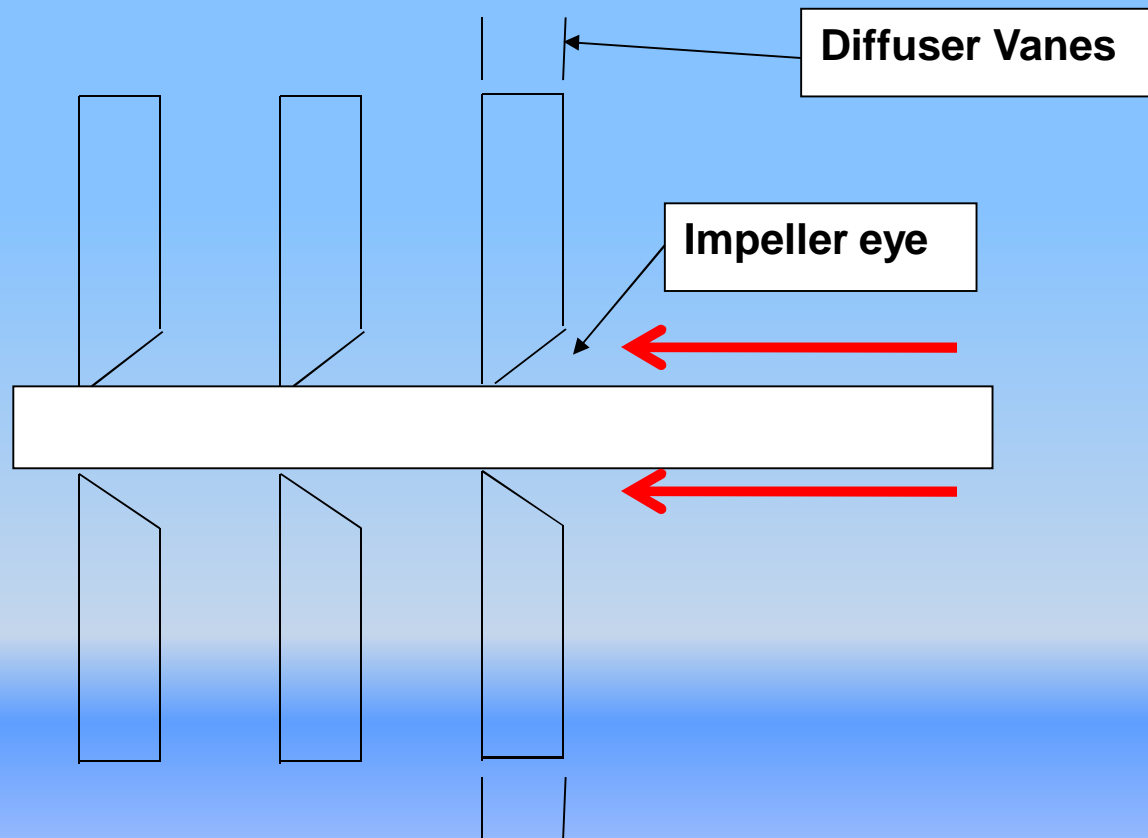




## Theory of Operation

Typical multi-stage compressor shown.

Flow approaches the impeller through the blower inlet duct in an axial inward direction.

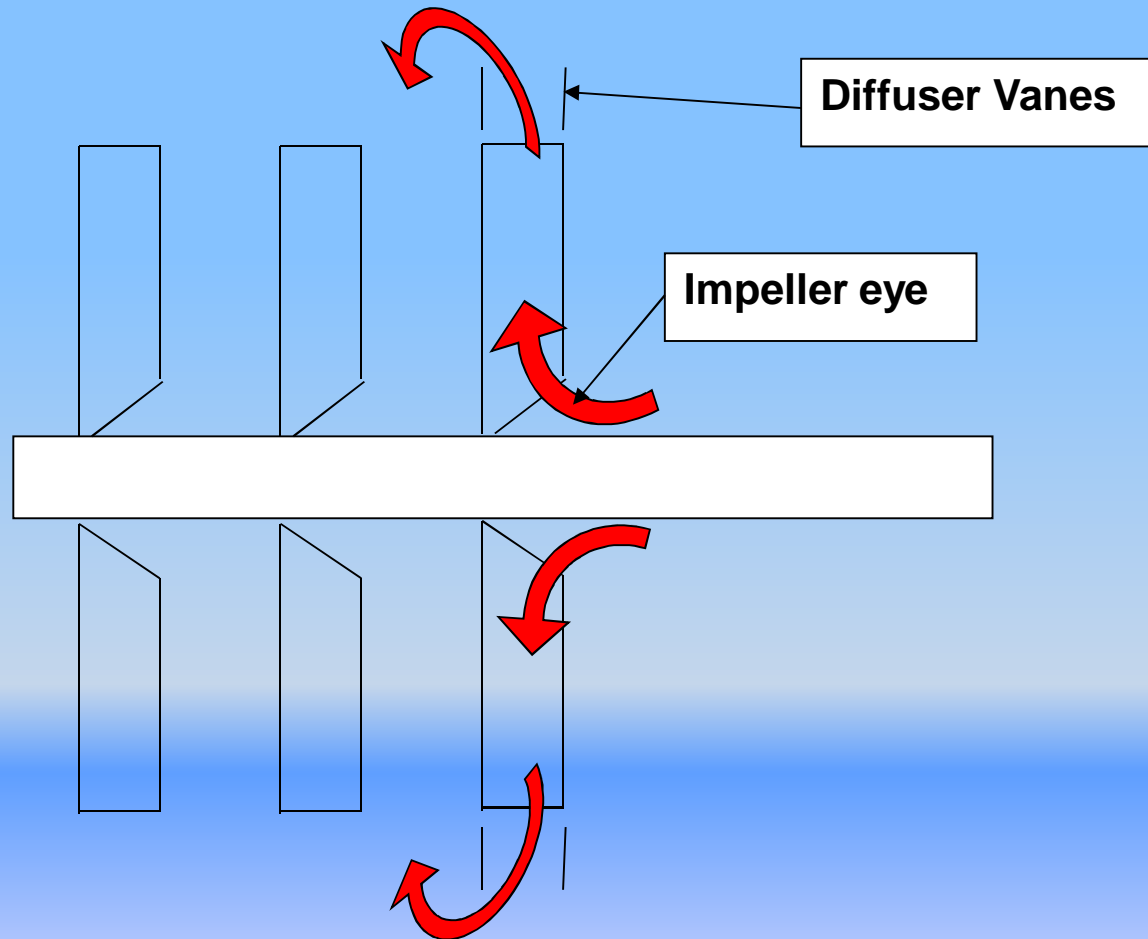


**Figure 3. Schematic of Rotor With Impellers Showing Gas Flow.**

## Theory of Operation

Flow then enters the rotating impeller.

The flow is then propelled through the impeller with work being continuously transferred to the flow as it transits through the impeller passages.

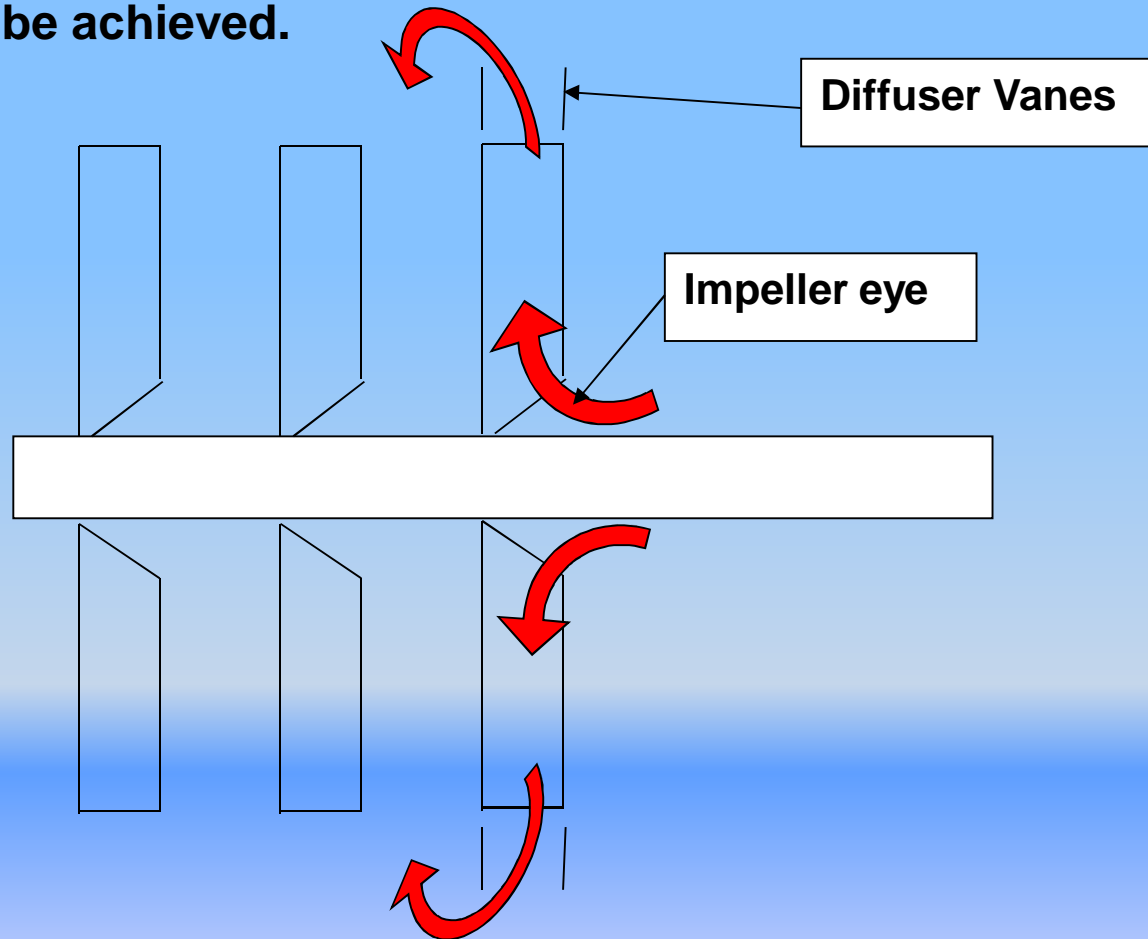


**Figure 3. Schematic of Rotor With Impellers Showing Gas Flow.**

## Theory of Operation

As flow exits the impeller it moves in a highly tangential direction, not a radial direction.

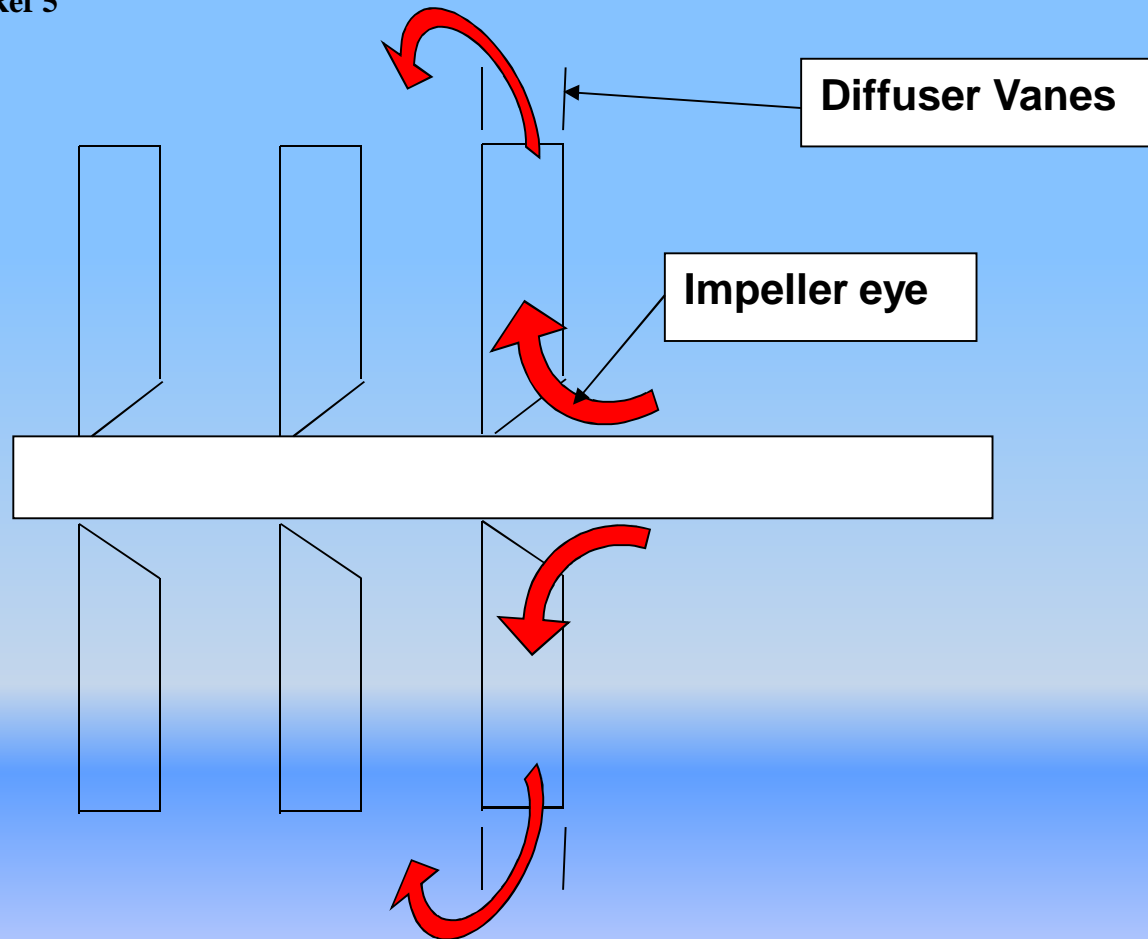
Kinetic energy level is very high which is required if any reasonable pressure rise is to be achieved.



**Figure 3. Schematic of Rotor With Impellers Showing Gas Flow.**

## Theory of Operation

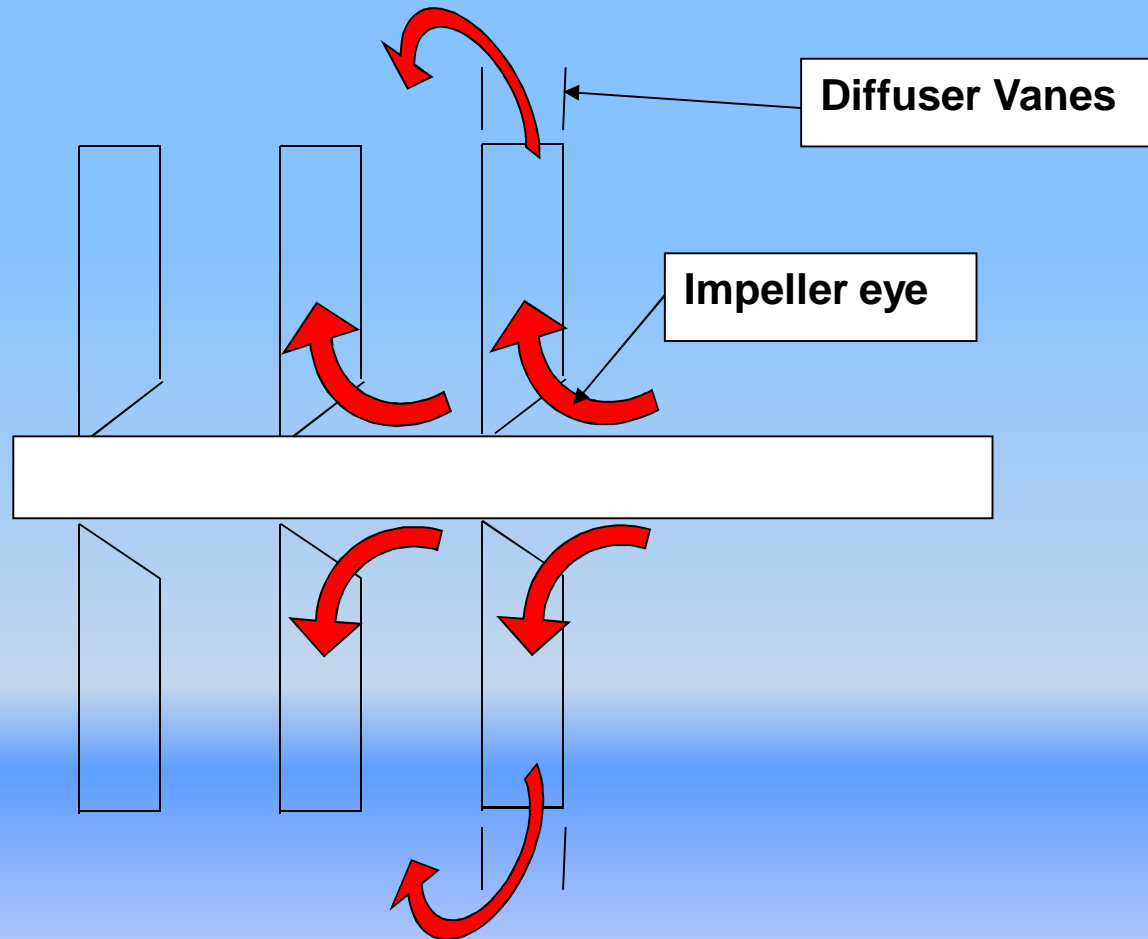
Approximately 2/3's of the pressure rise occurs in the impeller and 1/3 in the diffuser. Depending on design anywhere from 50% to 70% of the kinetic energy leaving the impeller may be recovered as a static pressure rise in the diffuser. <sup>Ref 5</sup>



**Figure 3. Schematic of Rotor With Impellers Showing Gas Flow.**

## Theory of Operation

Flow is directed into the next stage where more work is performed on the gas resulting in an additional pressure increase, etc.



**Figure 3. Schematic of Rotor With Impellers Showing Gas Flow.**

# Theory of Operation

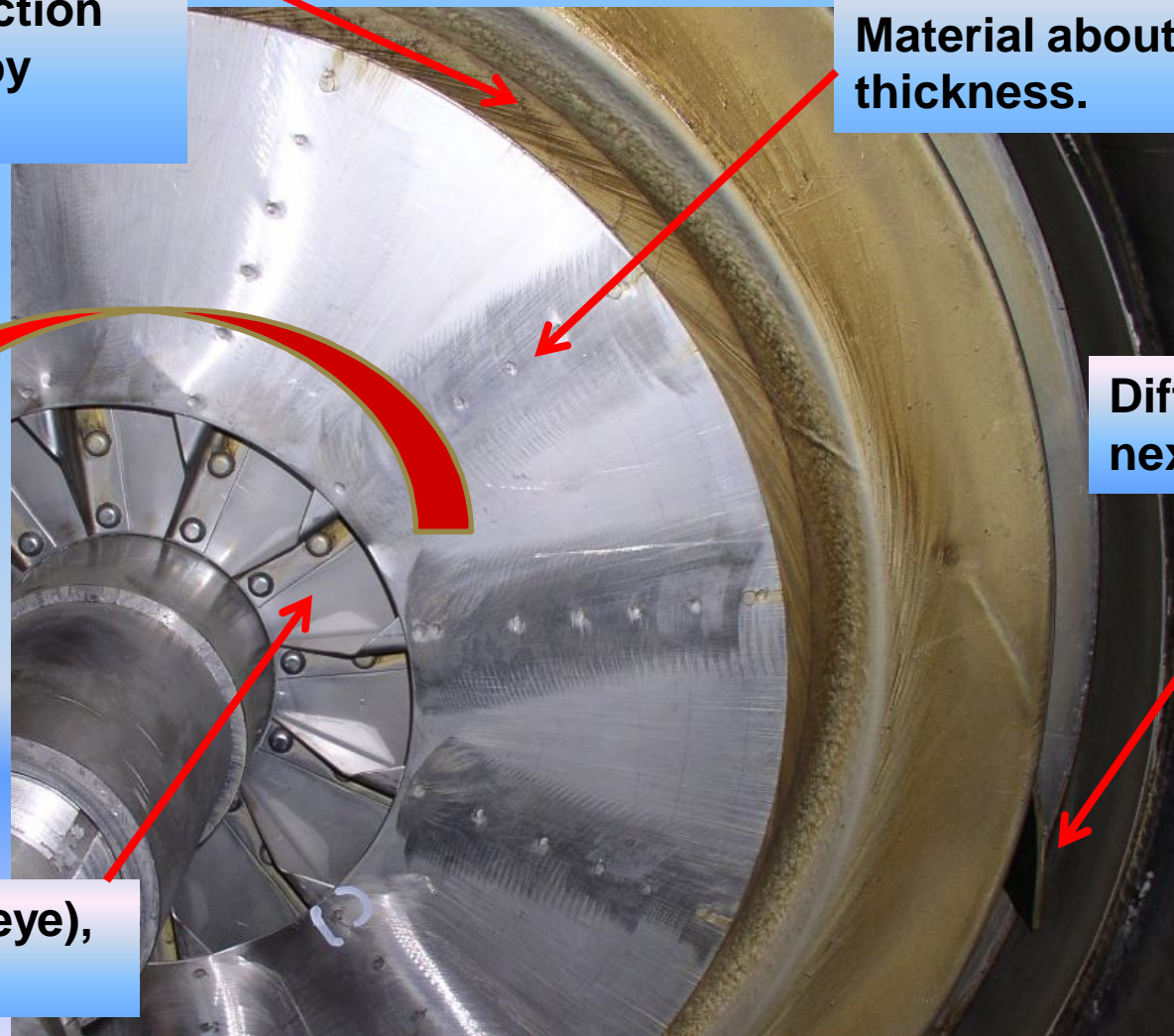
1<sup>st</sup> Stage Impeller, Deflector removed.

Flow exits impeller tangential direction as evidenced by residue.

Impeller with Riveted Construction. Shroud Material about 1/16" thickness.

Diffuser Blade, next stage.

Impeller inlet (eye), 12 Blades.





## Theory of Operation

Comparison of flow levels through rotary positive displacement, centrifugal and axial compressors.

Very simplified chart which only serves to illustrate a facet of machinery efficiency characteristics.

Axial flow compressors are more efficient than Centrifugal.

Centrifugal compressors are more efficient than rotary positive displacement.

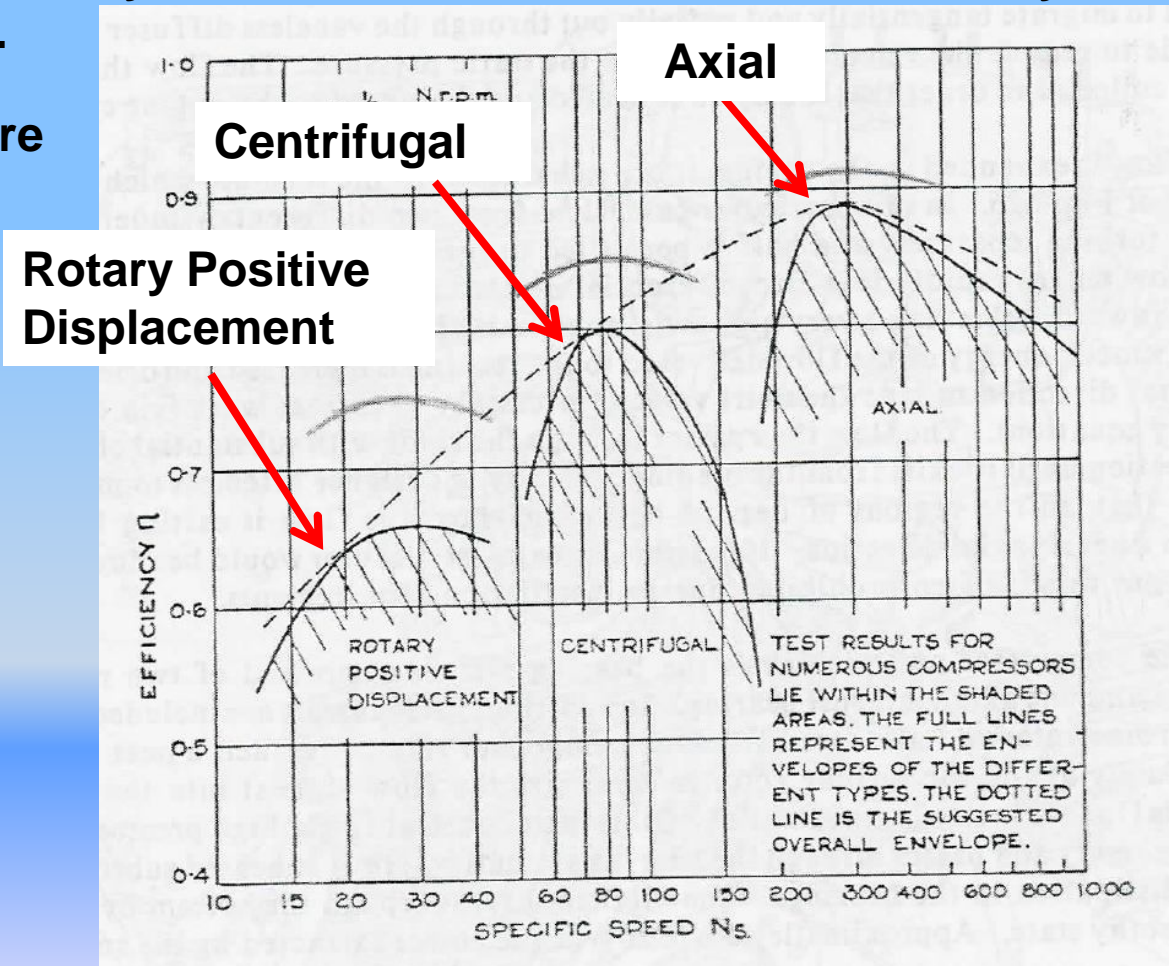


Figure 4. Variation of Efficiency With Specific Speed For Three Types of Compressors. <sup>Ref 5</sup>

## **Periodic Vibration Analysis:**

**At beginning of project, vibration data that had been acquired over several years by the plant's personnel was reviewed on the six motor-blowers.**

**The vibration data had been measured using a portable CSI Spectrum Analyzer.**

**Vibration data showed high amplitude vibration primarily at 1X and 2X blower/motor run speed.**

**For the analysis, additional data points were measured in the Vertical and Axial directions on the bearing housings.**

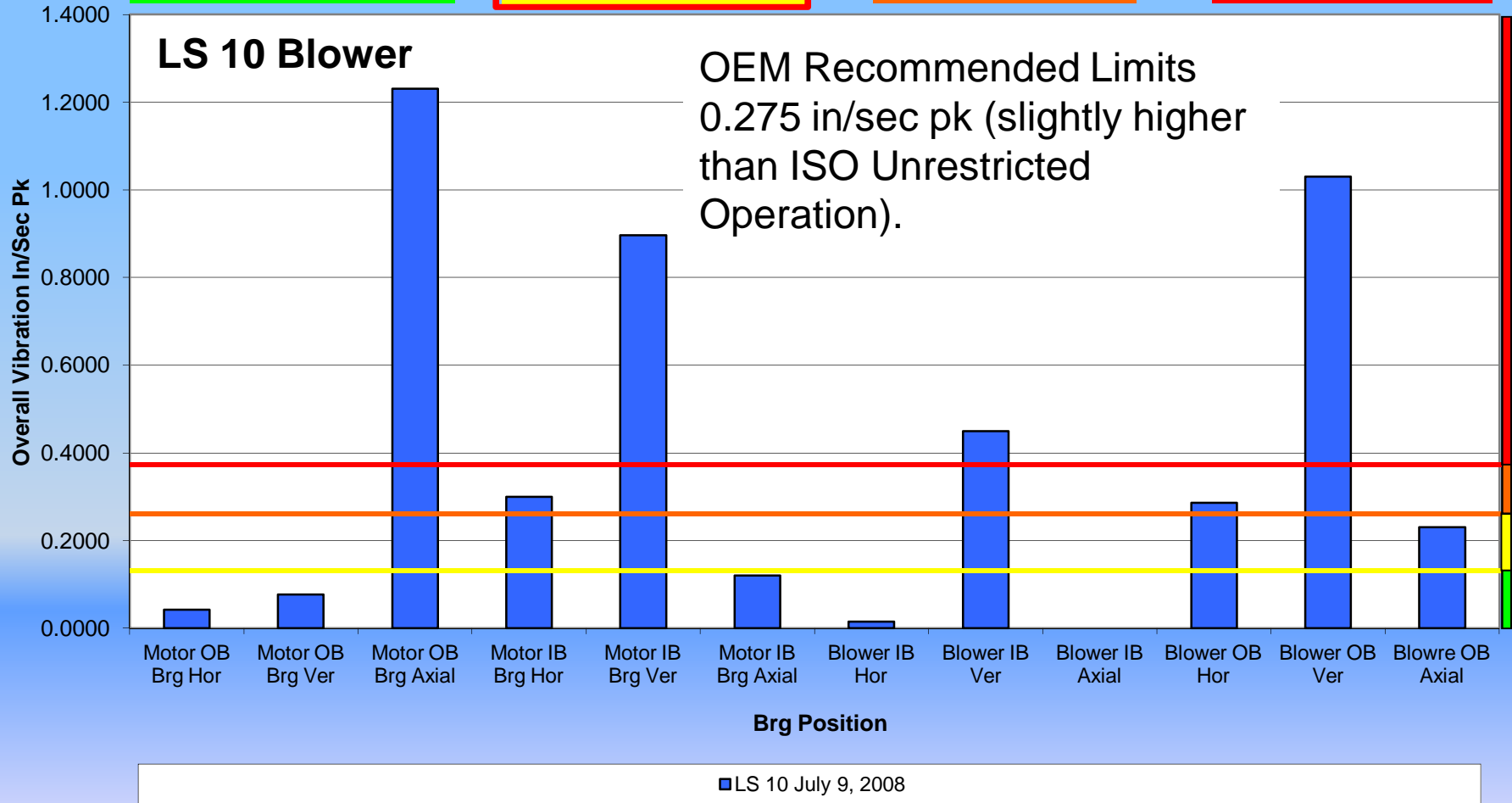
**LS 10 Multi-Stage Centrifugal Blowers**  
**ISO 10816-3 Vibration Standard for Overall Vibration Machinery Group 2 & 4, Flexible Mount**

**Newly Commissioned Machinery 0 to 0.13 in/sec pk Overall**

**Unrestricted Operation 0.13 to 0.25 in/sec pk Overall**

**Restricted Operation 0.25 to 0.39 in/sec pk**

**Damage Occurs > 0.39 in/sec pk**



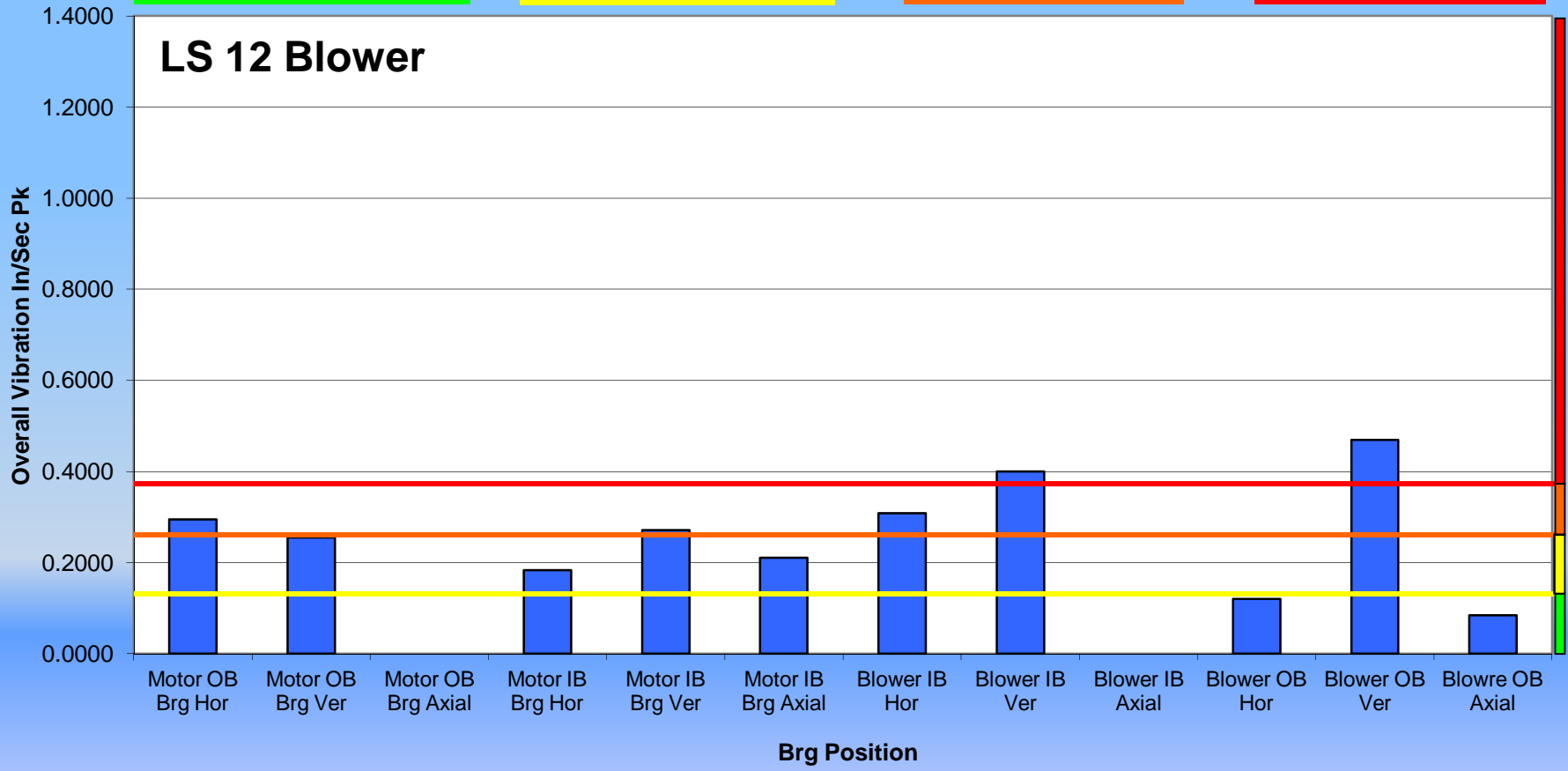
**LS 12 Multi-Stage Centrifugal Blowers**  
**ISO 10816-3 Vibration Standard for Overall Vibration Machinery Group 2 & 4, Flexible Mount**

**Newly Commissioned Machinery 0 to 0.13 in/sec pk Overall**

**Unrestricted Operation 0.13 to 0.25 in/sec pk Overall**

**Restricted Operation 0.25 to 0.39 in/sec pk Overall**

**Damage Occurs > 0.39 in/sec pk**



■ LS 12 July 9, 2008

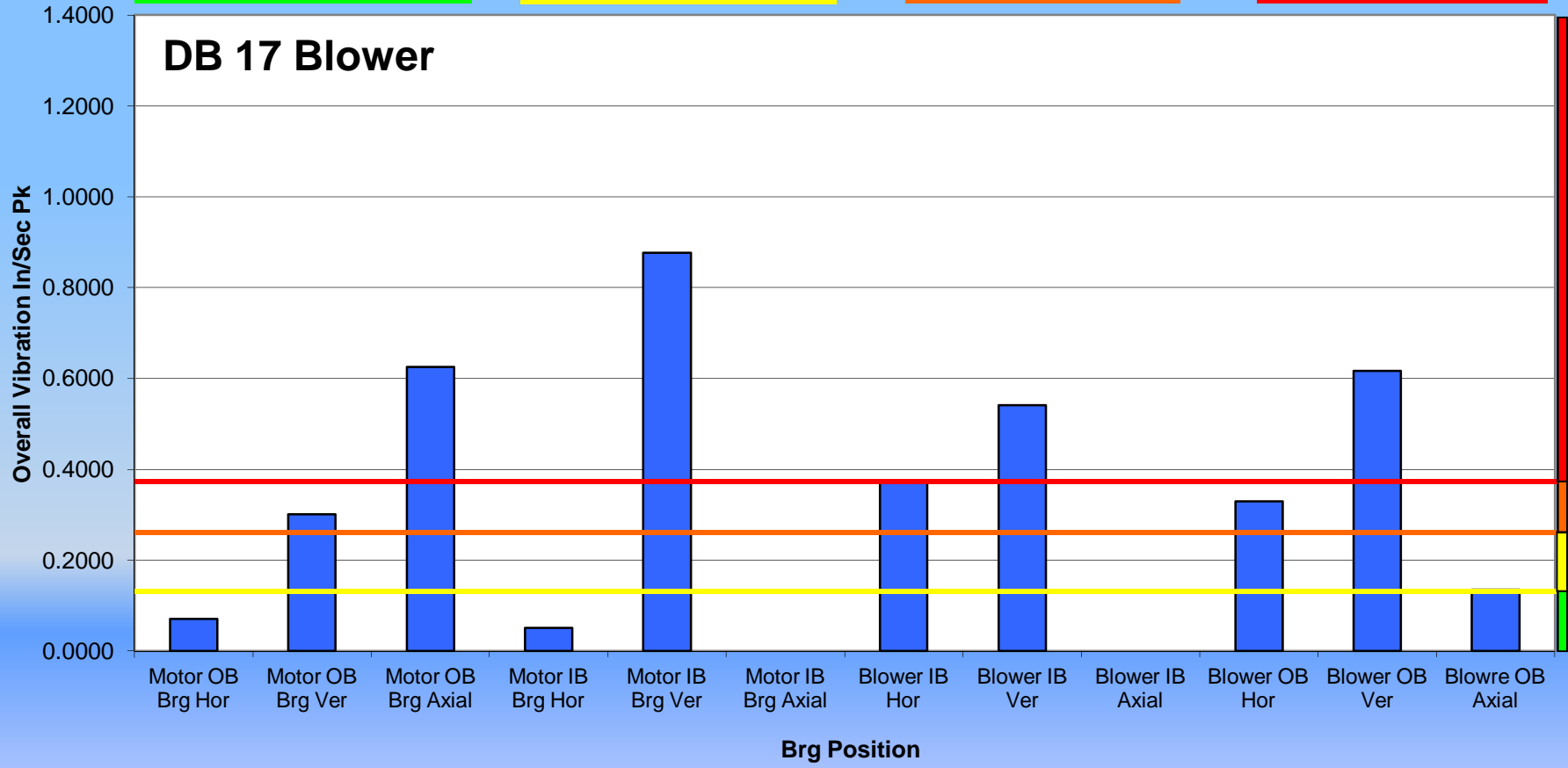
**DB 17 Multi-Stage Centrifugal Blowers**  
**ISO 10816-3 Vibration Standard for Overall Vibration Machinery Group 2 & 4, Flexible Mount**

**Newly Commissioned Machinery 0 to 0.13 in/sec pk Overall**

**Unrestricted Operation 0.13 to 0.25 in/sec pk Overall**

**Restricted Operation 0.25 to 0.39 in/sec pk Overall**

**Damage Occurs > 0.39 in/sec pk**



■ DB 17 July 9, 2008

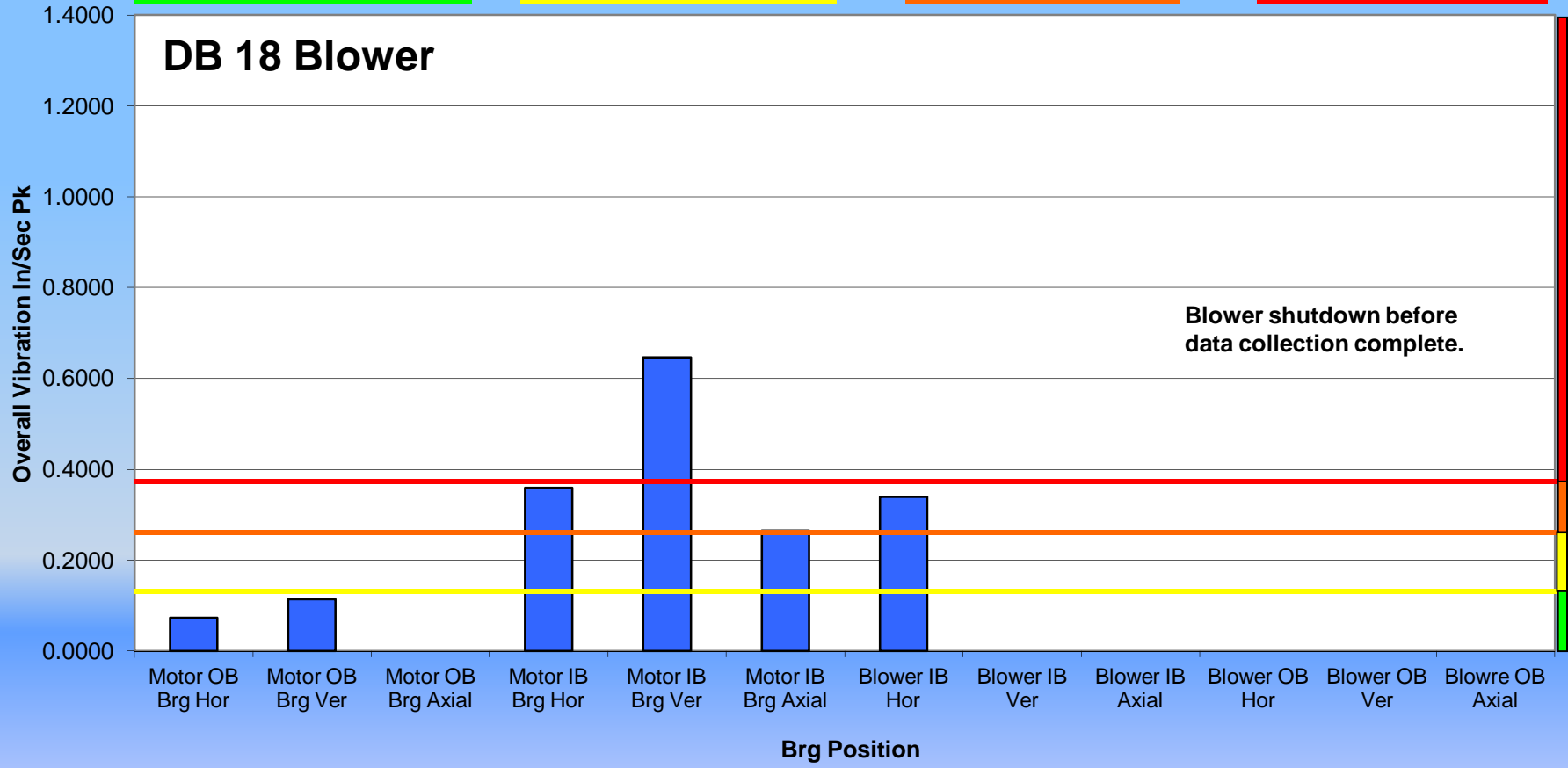
**DB 18 Multi-Stage Centrifugal Blowers**  
**ISO 10816-3 Vibration Standard for Overall Vibration Machinery Group 2 & 4, Flexible Mount**

**Newly Commissioned Machinery 0 to 0.13 in/sec pk Overall**

**Unrestricted Operation 0.13 to 0.25 in/sec pk Overall**

**Restricted Operation 0.25 to 0.39 in/sec pk Overall**

**Damage Occurs > 0.39 in/sec pk**



■ DB 18 July 9, 2008



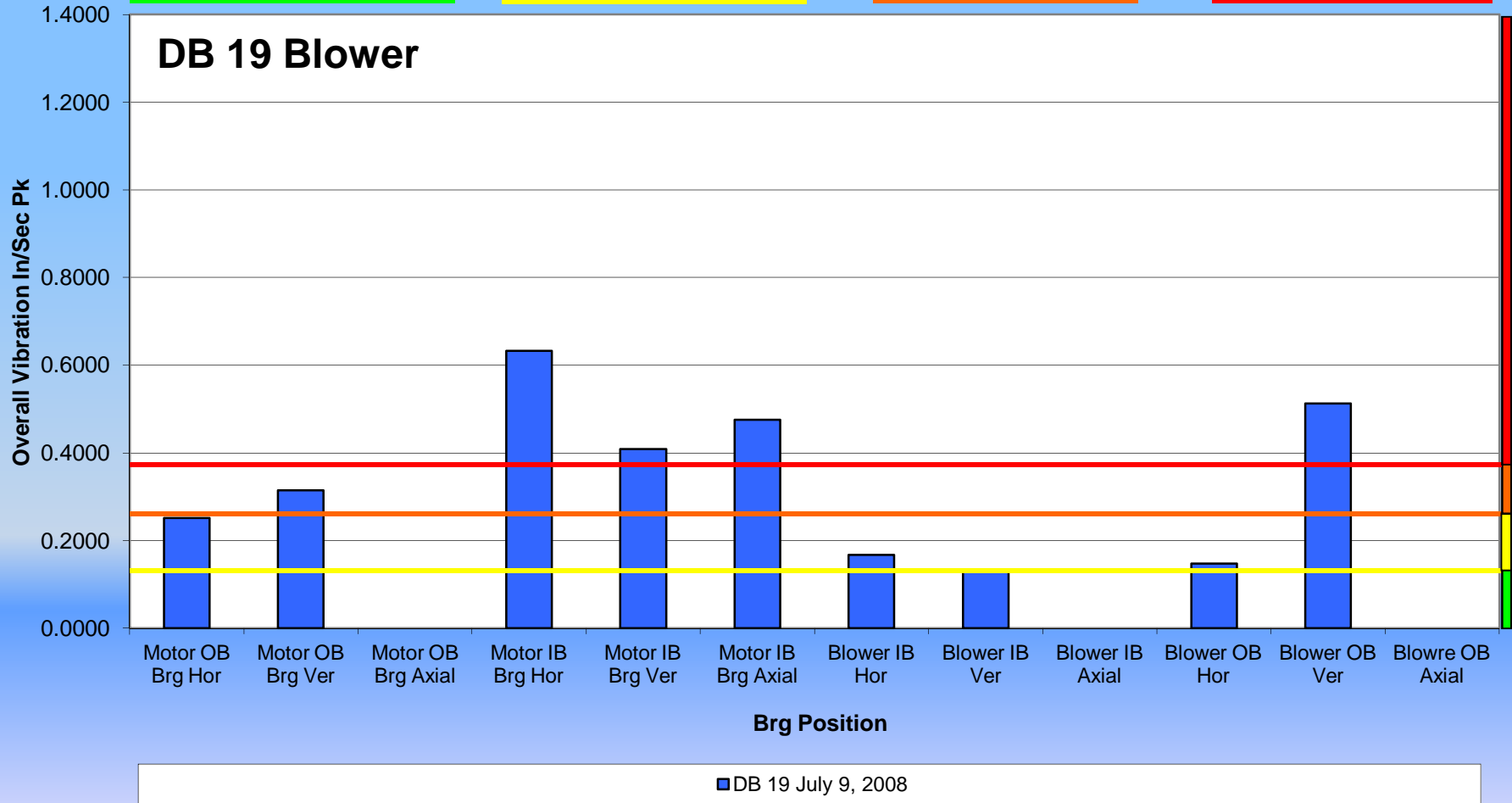
**DB 19 Multi-Stage Centrifugal Blowers**  
**ISO 10816-3 Vibration Standard for Overall Vibration Machinery Group 2 & 4, Flexible Mount**

**Newly Commissioned Machinery 0 to 0.13 in/sec pk Overall**

**Unrestricted Operation 0.13 to 0.25 in/sec pk Overall**

**Restricted Operation 0.25 to 0.39 in/sec pk Overall**

**Damage Occurs > 0.39 in/sec pk**



## Periodic Vibration Analysis:

Frequency Spectrum and Time Waveform for DB 19 Blower Motor IB Axial shown. Data at other point similar. Most vibration at 1X.

Could indicate unbalance, misalignment, resonance, bowed rotor, worn bearings, etc.

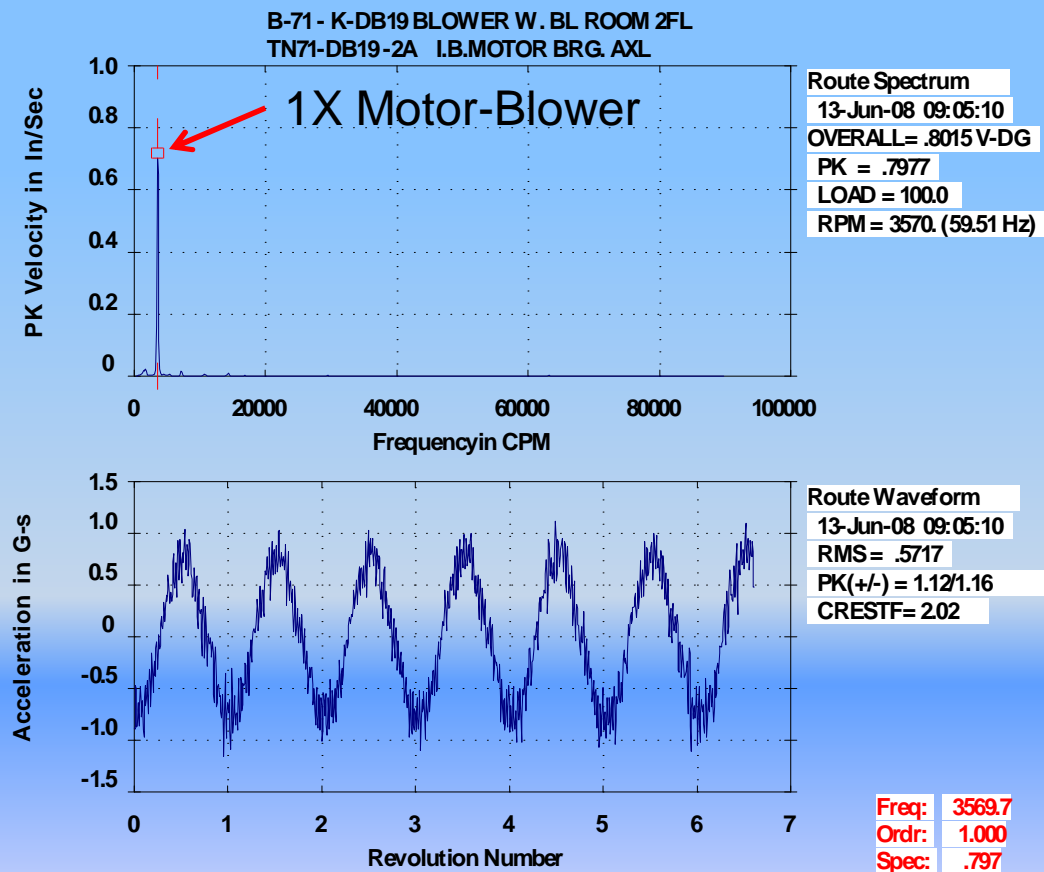


Figure 6. Frequency Spectrum and Time Waveform, DB 19, Motor IB Brg Axial.

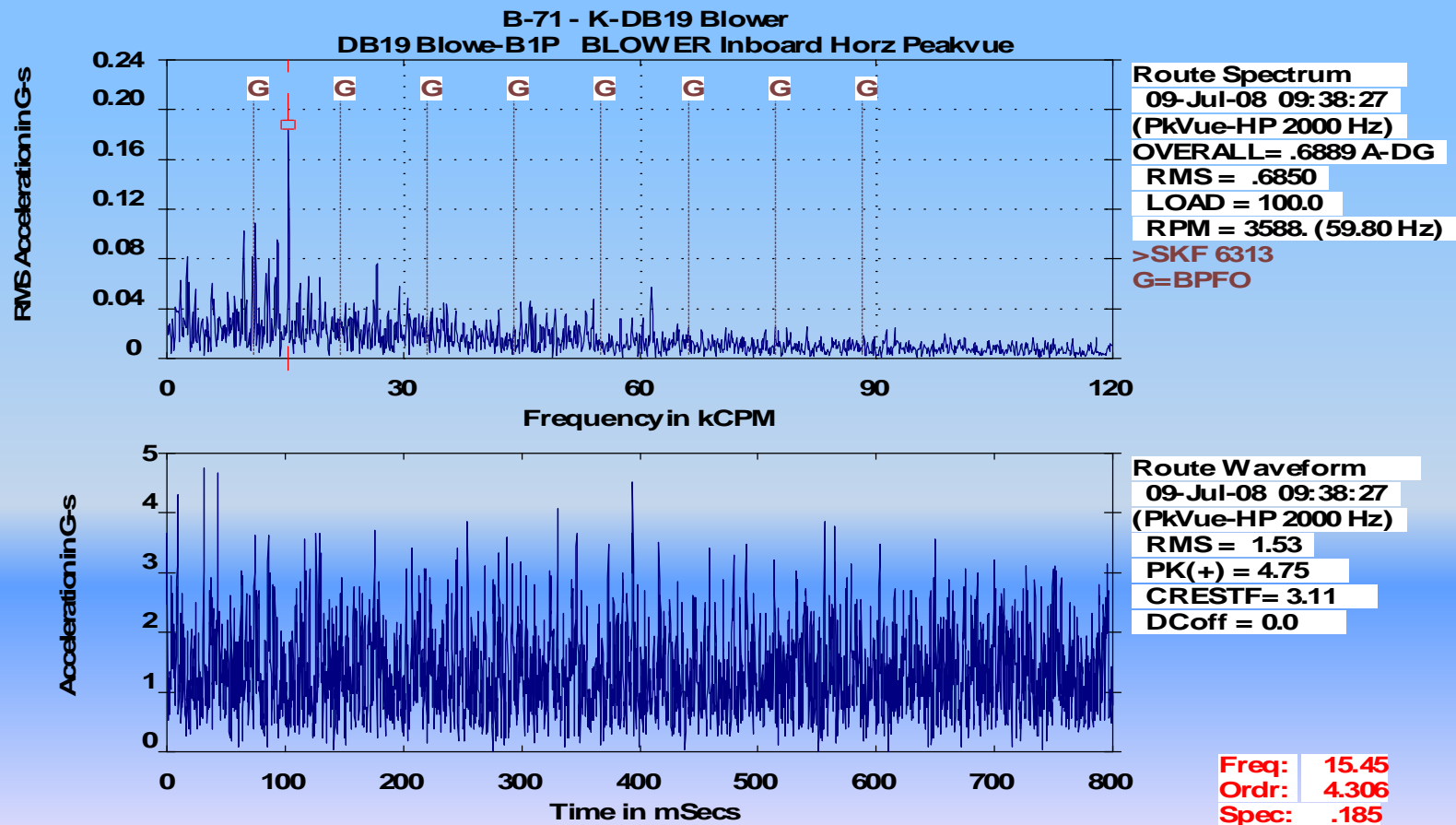
# Periodic Vibration Analysis:

Bearings in the Motors and Blowers:

Motor – 6314 Ball

Blower Inboard – 6313 Ball

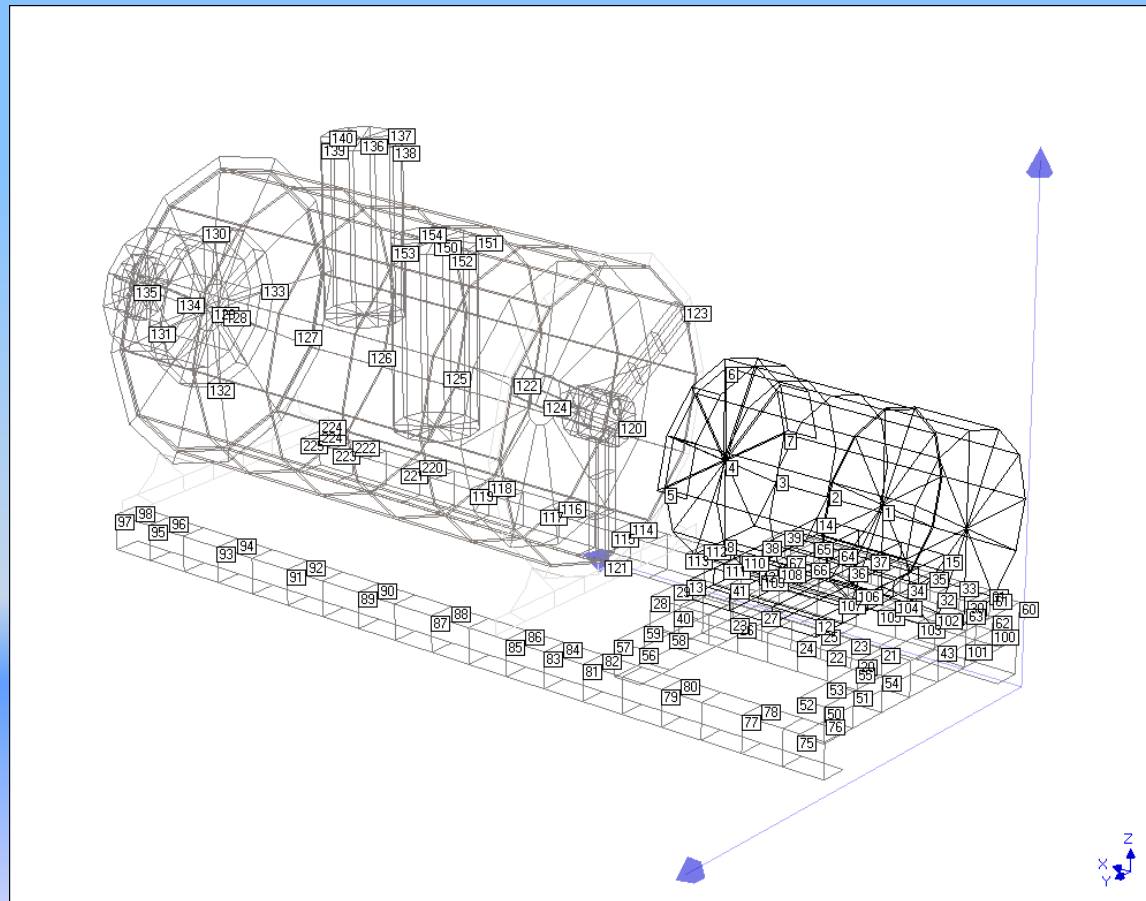
Blower Outboard – 7313 Double Row Angular Contact Ball



## Operating Deflection Shape Analysis (ODS):

ODS provides a 3D Computer model of a machine or structure that can be animated at the frequencies that vibration is occurring.

The vibration shape or pattern can be studied at any of the frequencies measured by the cross channel transmissibility data. Two channel analyzer required.

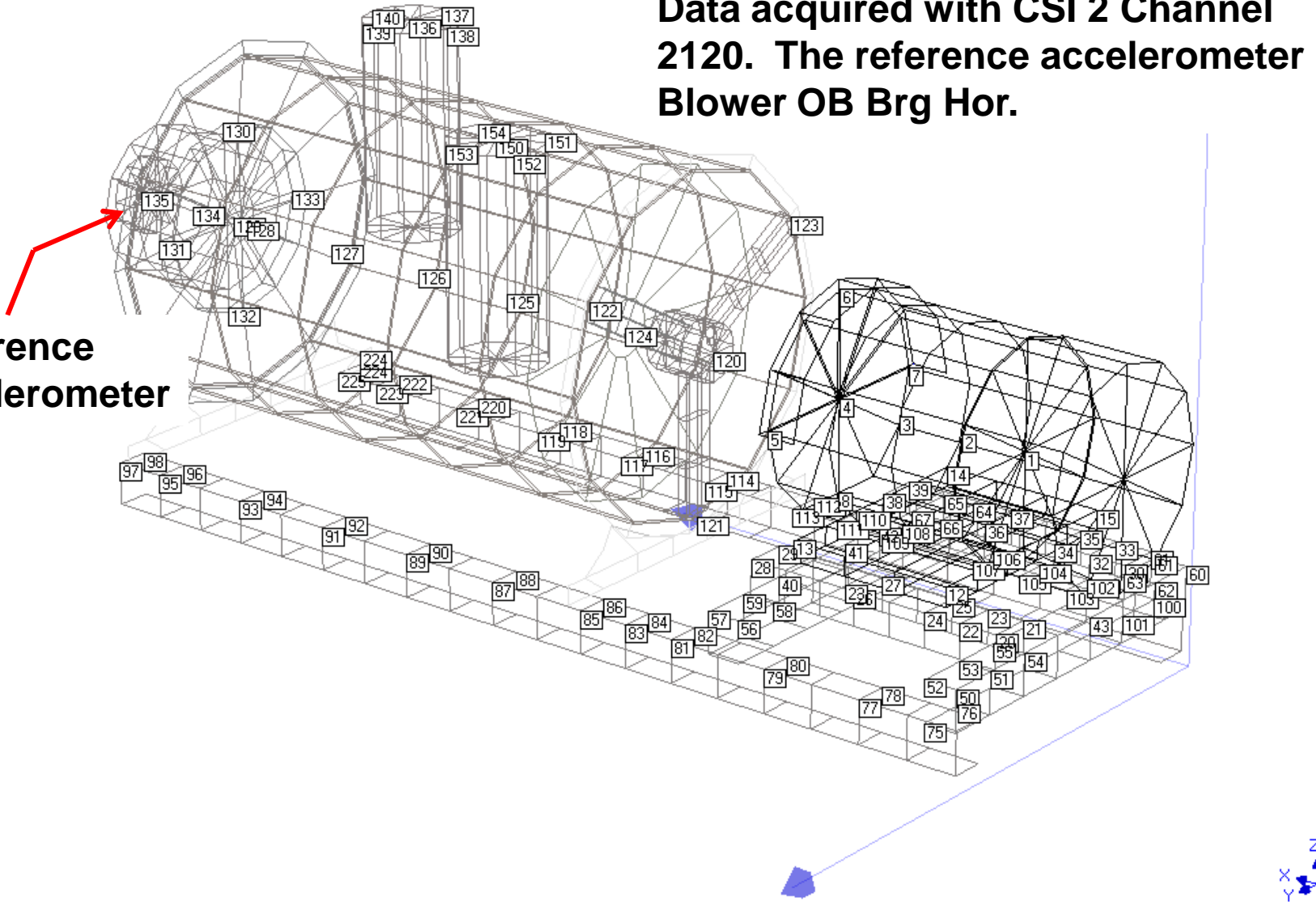


**Figure7. Wire Frame Model of a Motor, Blower and Skid With Measurement Point Locations Labeled.**

**ODS Models developed in ME'scopeVES V5.0 for Blowers DB 17, DB 19, and LS 10.**

**Data acquired with CSI 2 Channel 2120. The reference accelerometer Blower OB Brg Hor.**

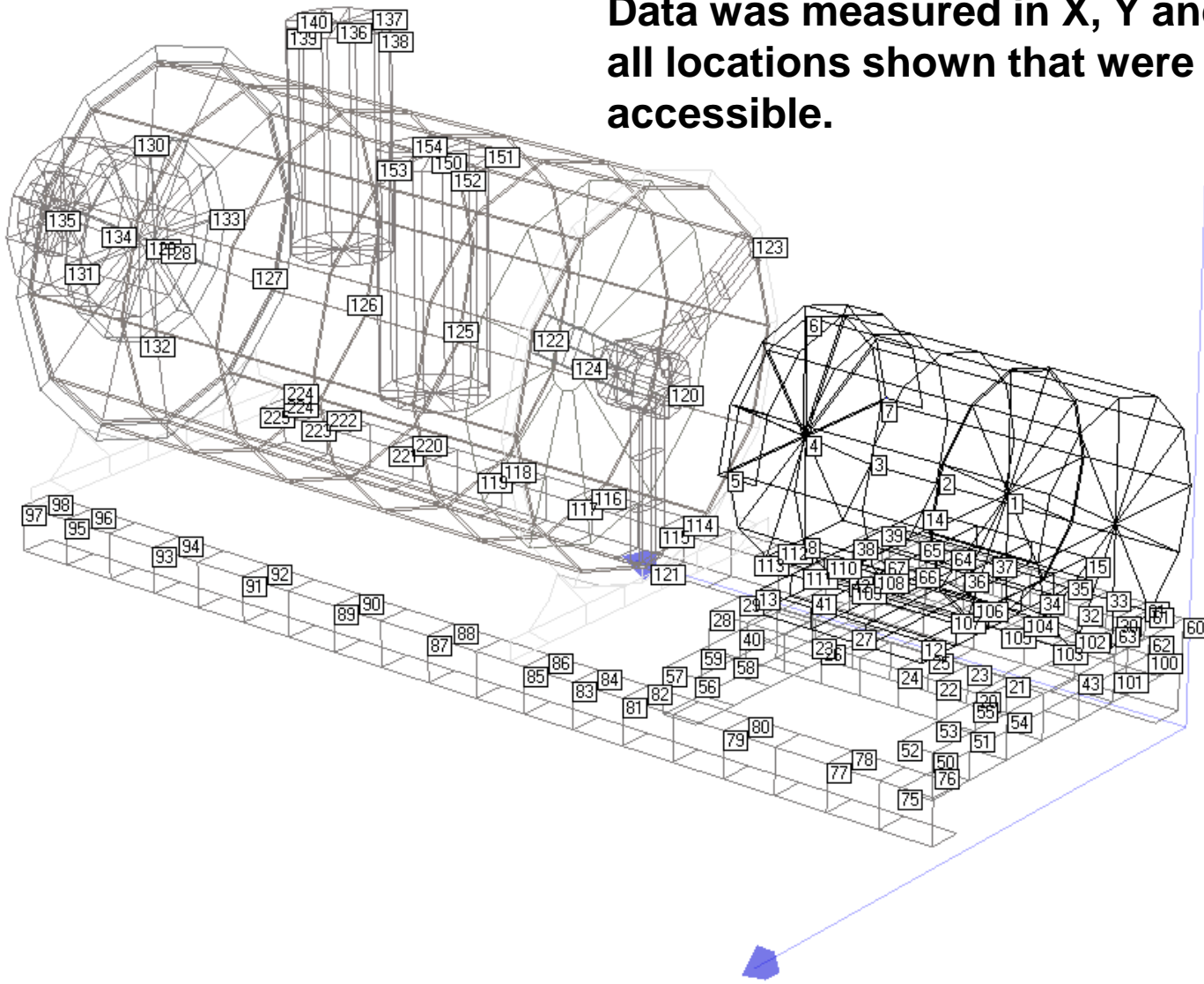
**Reference accelerometer**



**Figure7. Wire Frame Model of a Motor, Blower and Skid With Measurement Point Locations Labeled.**

**ODS Models developed in ME'scopeVES V5.0 for Blowers DB 17, DB 19, and LS 10.**

**Data was measured in X, Y and Z at all locations shown that were accessible.**

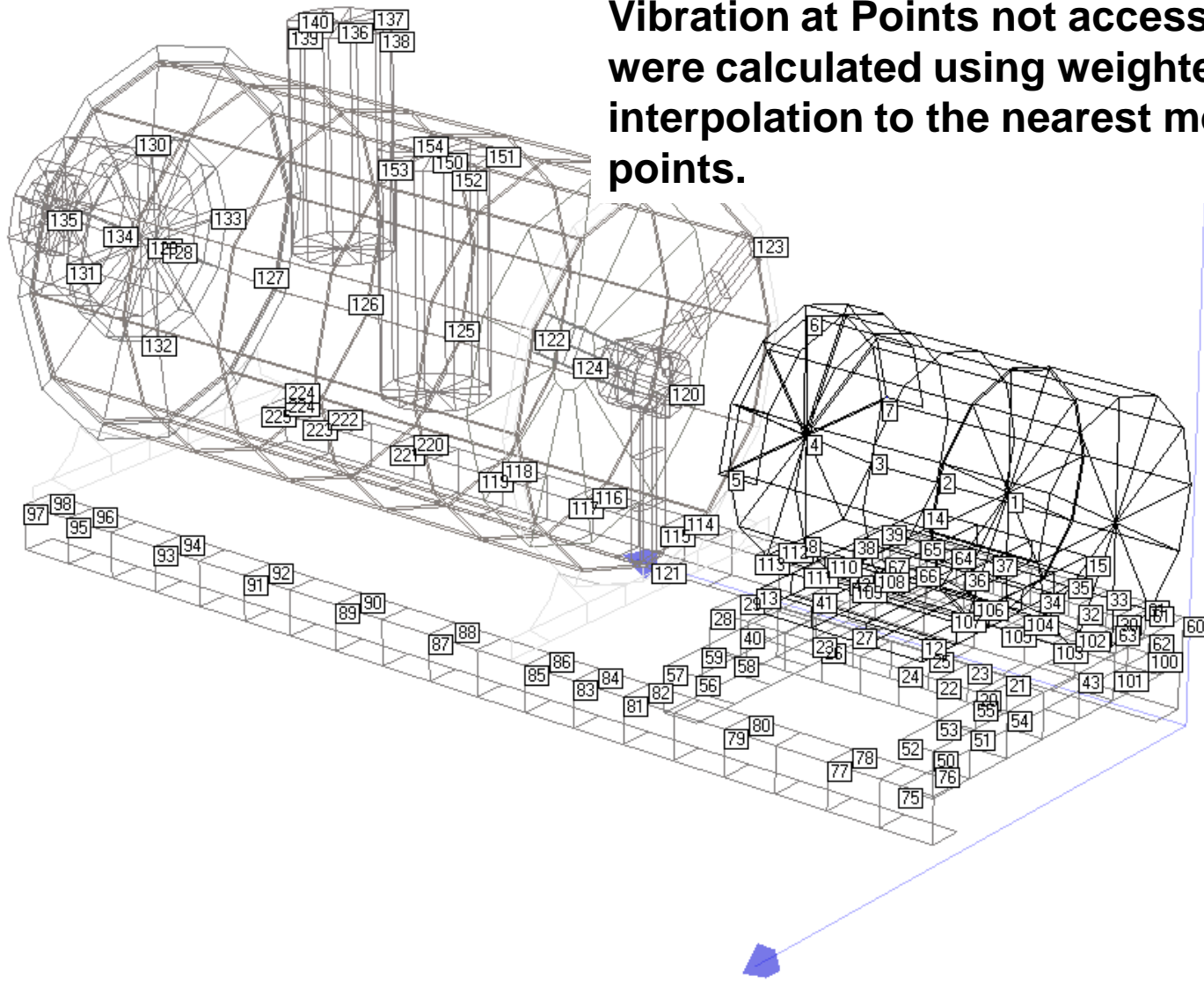


**Figure7. Wire Frame Model of a Motor, Blower and Skid With Measurement Point Locations Labeled.**



**ODS Models developed in ME'scopeVES V5.0 for Blowers DB 17, DB 19, and LS 10.**

**Vibration at Points not accessible were calculated using weighted interpolation to the nearest measured points.**



**Figure7. Wire Frame Model of a Motor, Blower and Skid With Measurement Point Locations Labeled.**

# ODS DB 19 Blower. Vibration at 1X.

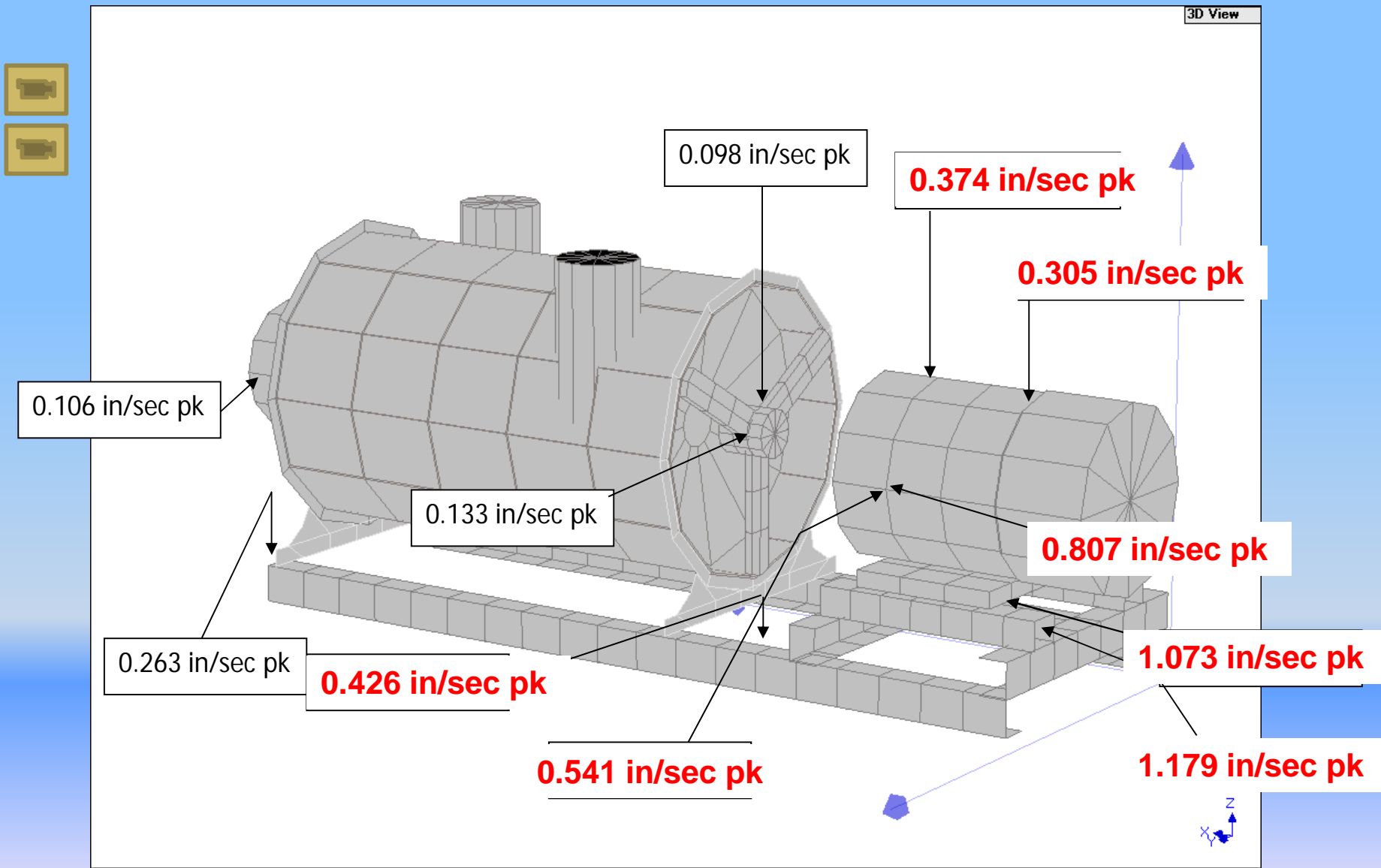


Figure 8. DB 19 ODS Vibration Amplitudes at 1X Run Speed Frequency.

# ODS DB 19 Blower. Vibration at 2X.

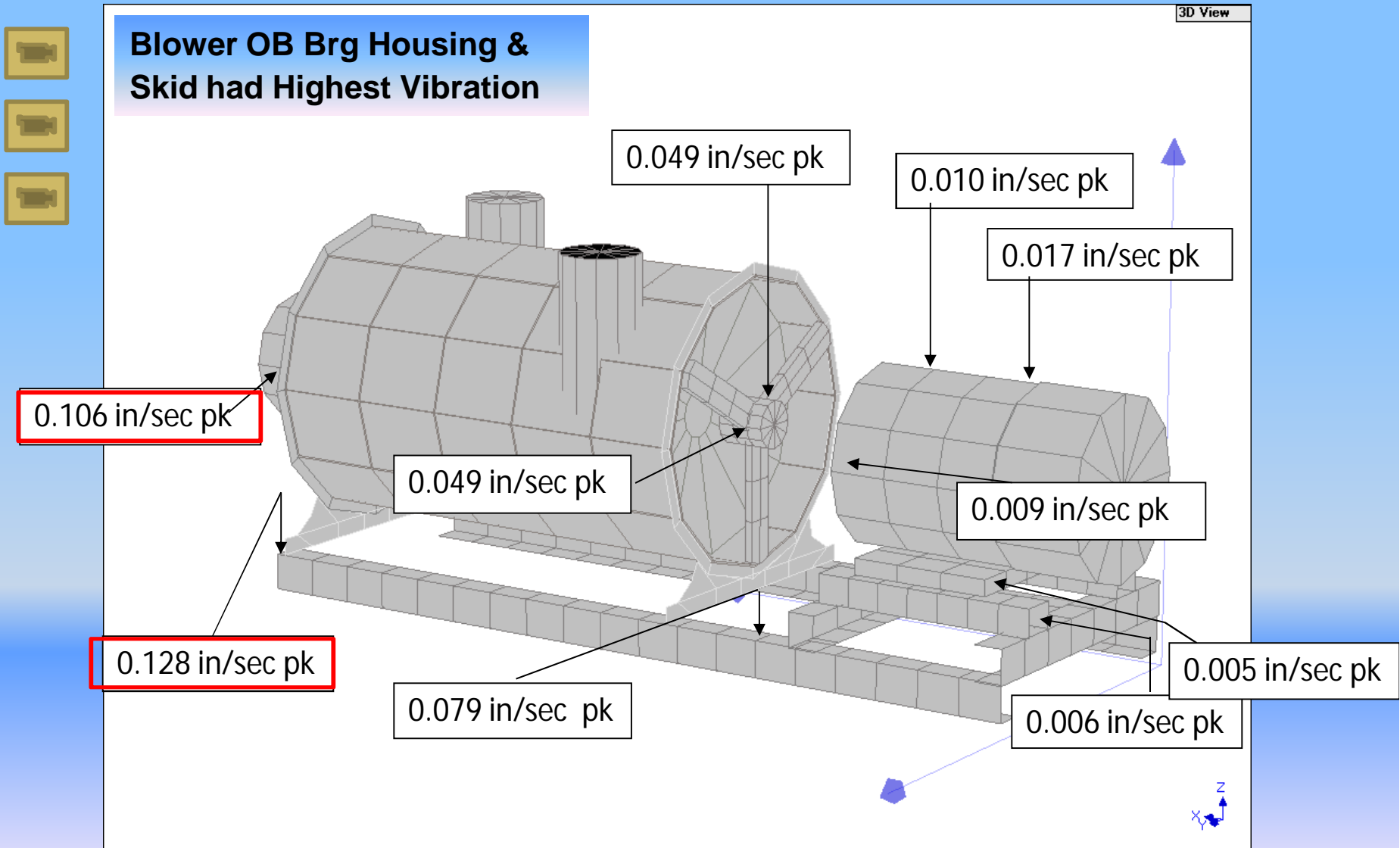
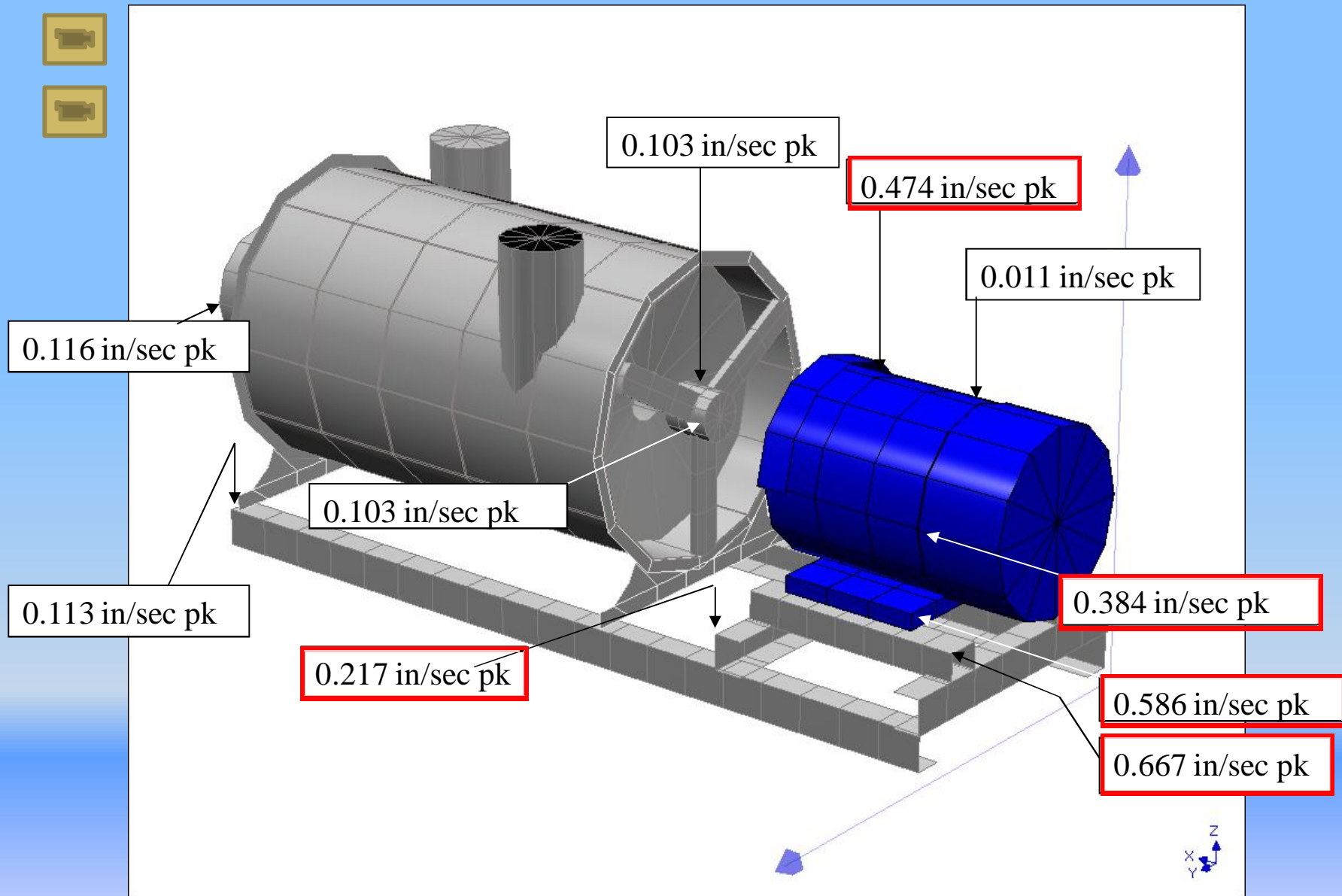


Figure 10. DB19 ODS Vibration Amplitudes at 2X Run Speed Frequency.

# ODS LS 10 Blower. Vibration at 1X.

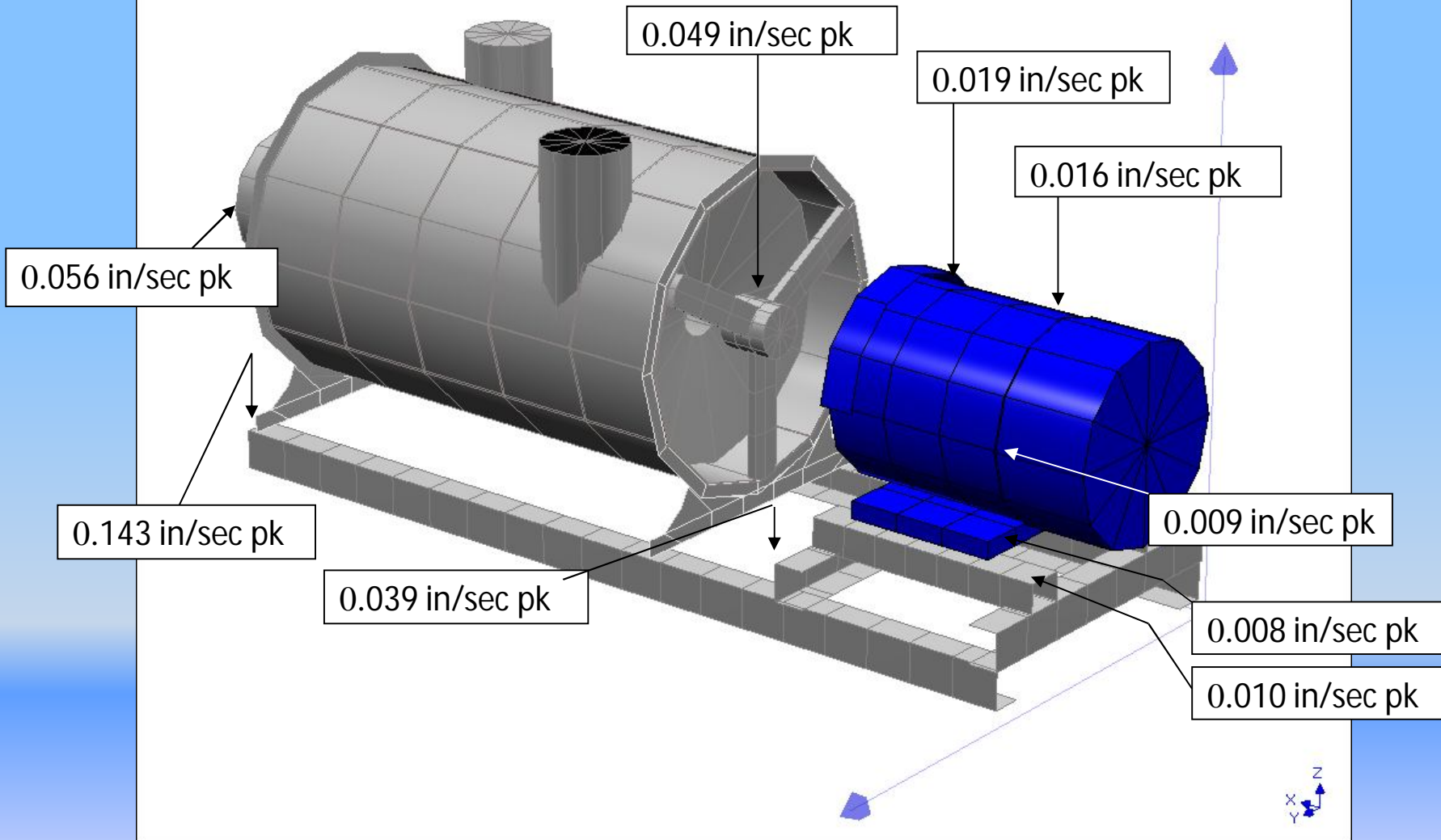


**LS 10 ODS Vibration Amplitudes at 1X Run Speed Frequency.**

# ODS LS 10 Blower. Vibration at 2X.

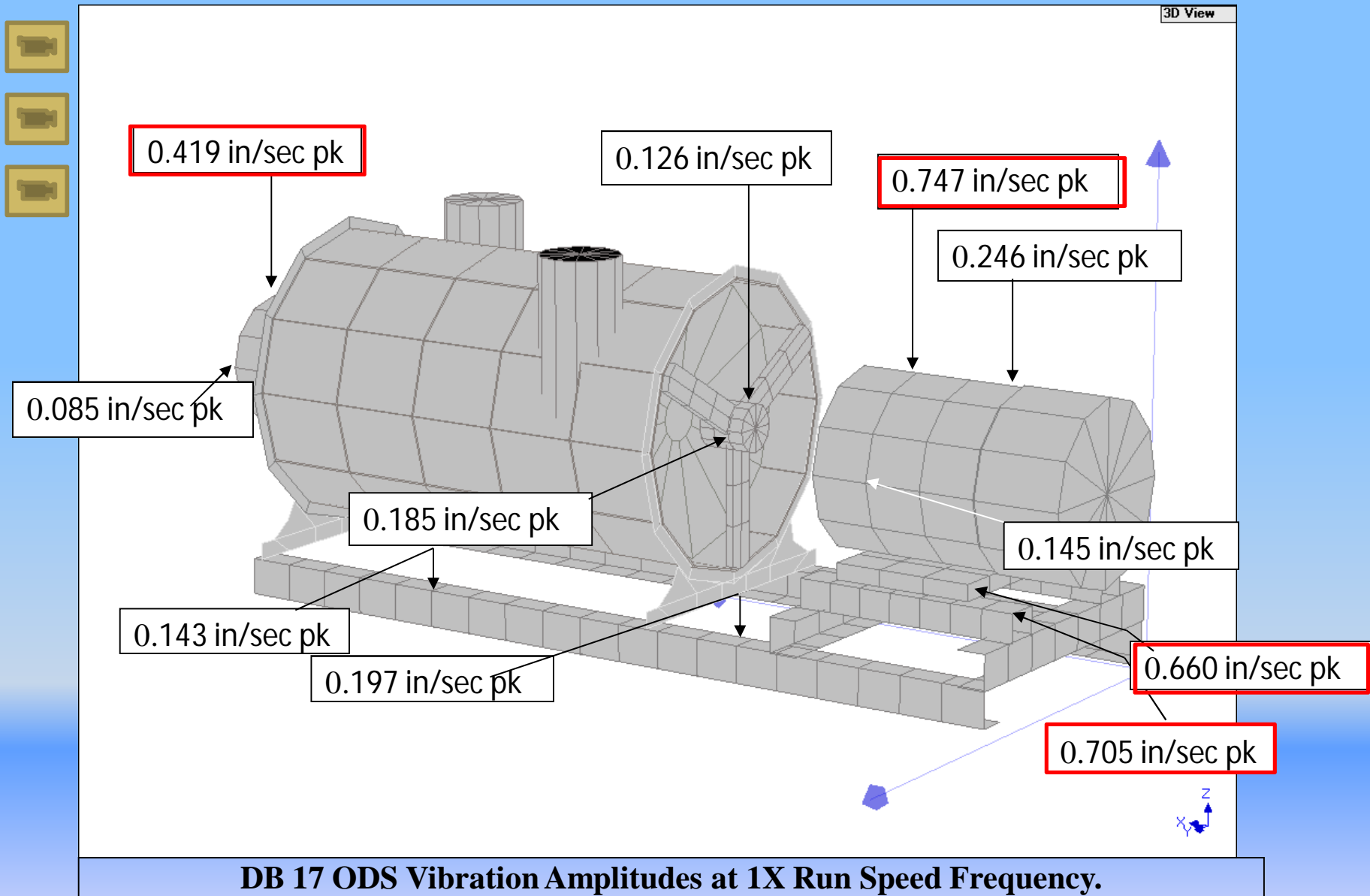


Vibration low amplitude, skid flexure shown by ODS.



LS 10 ODS Vibration Amplitudes at 2X Run Speed Frequency.

# ODS DB 17 Blower. Vibration at 1X.

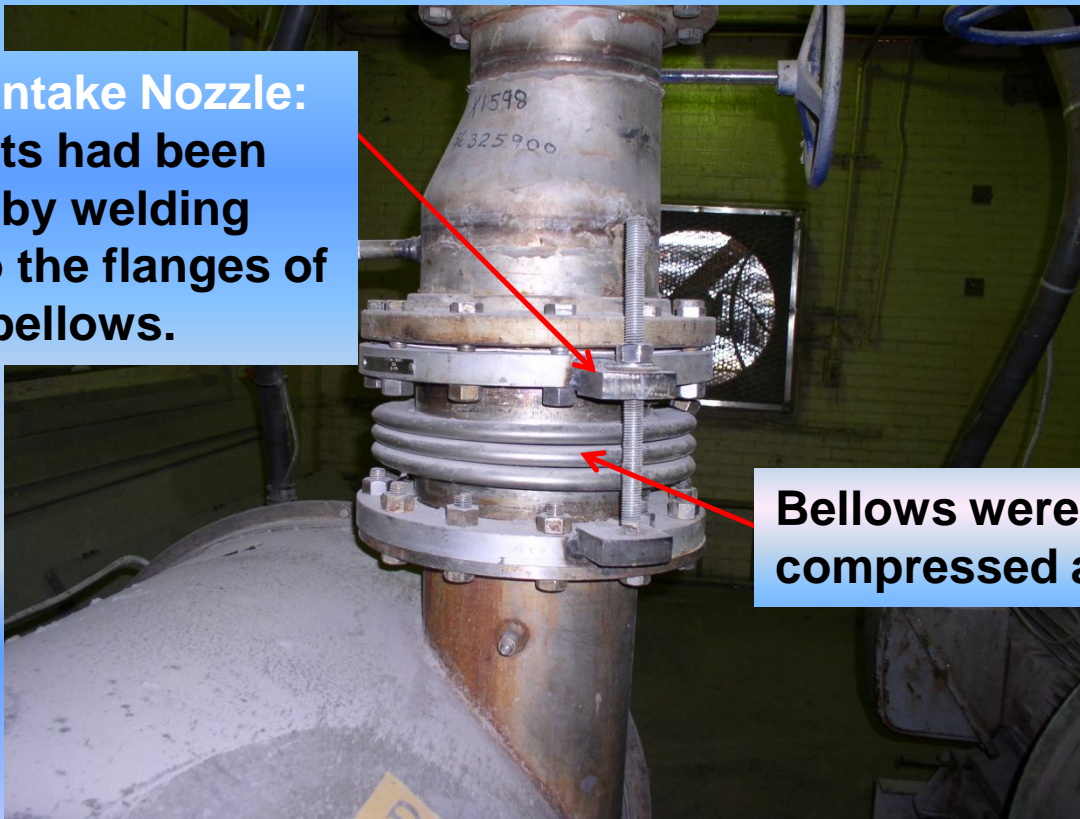




## Pipe Strain:

Inlet and discharge piping were connected to the blower nozzles with flexible stainless steel bellows.

**LS 12 Intake Nozzle:**  
Tie bolts had been added by welding lugs to the flanges of some bellows.



**Bellows were completely compressed axially.**

**Figure 11. LS 12 Blower, Compressed Bellows at Intake Nozzle.**

## Pipe Strain:

Inlet and discharge piping were connected to the blower nozzles with flexible stainless steel bellows.

**LS 10 Discharge Nozzle:**  
Tie bolts added using  
lugs welded to the  
flanges of bellows.

**LS 10 Blower Intake Nozzle:**  
Bellows compressed axially.



**Figure 12. Blower LS 10 Blower Discharge Connection, Bellows Collapsed.**

## Pipe Strain:

Flexible steel bellows can accommodate small amounts of axial and lateral movement.

They are not designed to compensate for piping misalignment errors.

LS 10 Discharge Nozzle:  
Tie bolts added using  
lugs welded to the  
flanges of bellows.

LS 10 Blower Intake Nozzle:  
Bellows compressed axially.



Figure 12. Blower LS 10 Blower Discharge Connection, Bellows Collapsed.

## Pipe Strain:

Flexible steel bellows can accommodate small amounts of axial and lateral movement.

They are not designed to compensate for piping misalignment errors.

Bellows compressed on one side.





## Pipe Strain:

Pipe guides should be installed within four diameters of the joint and then again after fourteen pipe diameters. Ref 6

The inlet and discharge piping to the blowers was not restrained so the bellows were transferring pipe movement forces to the blower nozzle flanges.

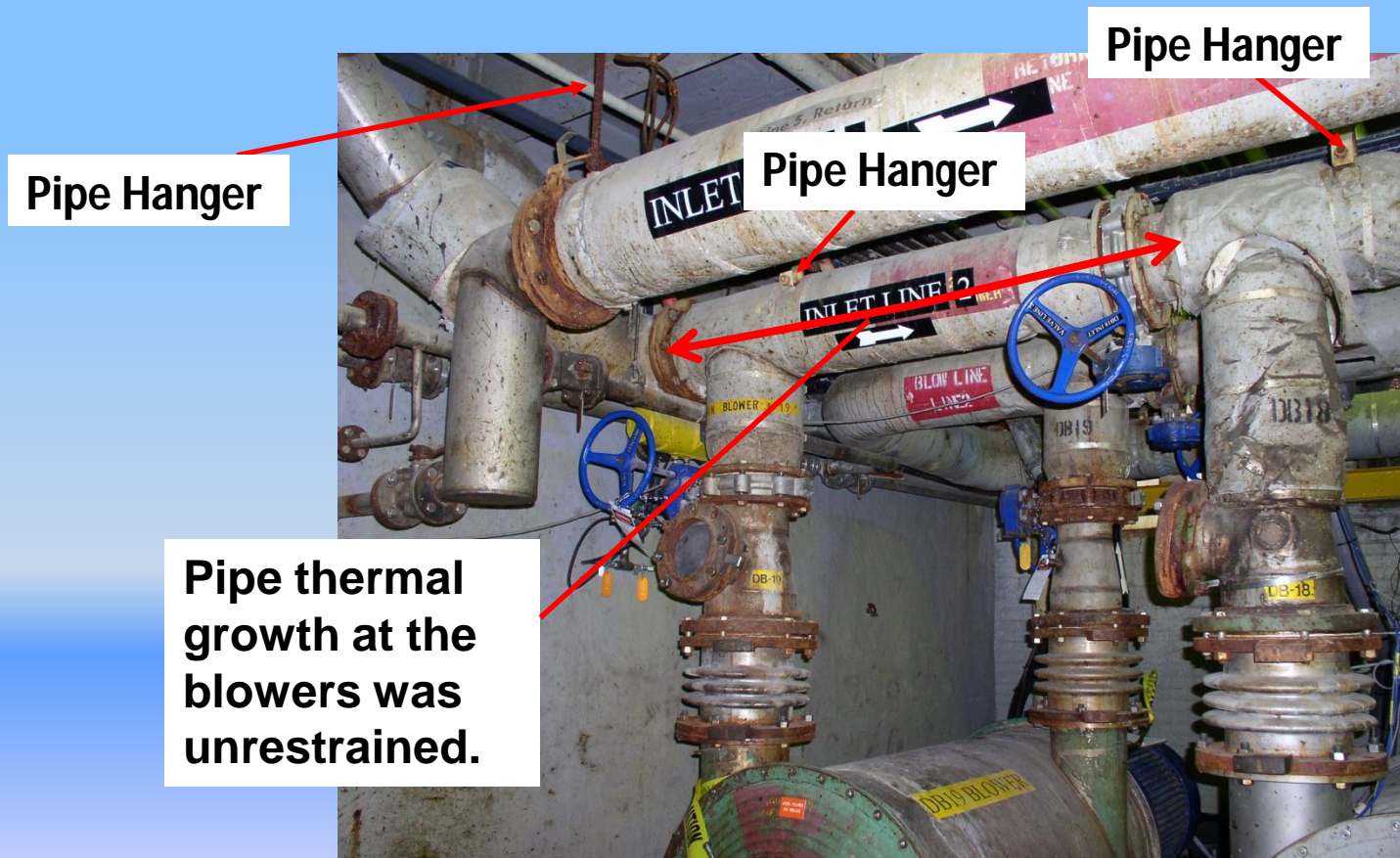


Figure 13. Piping Connected to Blowers, Blower DB 19 Shown.

## DB 19 Blower Continuous Vibration Acquisition:

A multi-channel spectrum analyzer, IOtech 618E & Tomas software was used to measure vibration data from accelerometers magnetically mounted on the motor and blower bearing housings.

Photo shows Permaalign, an accelerometer, Optical Tachometer, and the Alta-Flex coupling of DB19 Blower.



**Figure 14. DB 19 With Laser Alignment and Vibration Sensors Attached.**



## Continuous Vibration Acquisition:

Accelerometers were mounted at following locations:

Motor OB Hor 90 Deg Left

Motor OB Ver TDC

Motor IB Hor 90 Deg Left

Motor IB Ver TDC

Motor IB Axial 0 Deg

Blower IB Hor 90 Deg Left

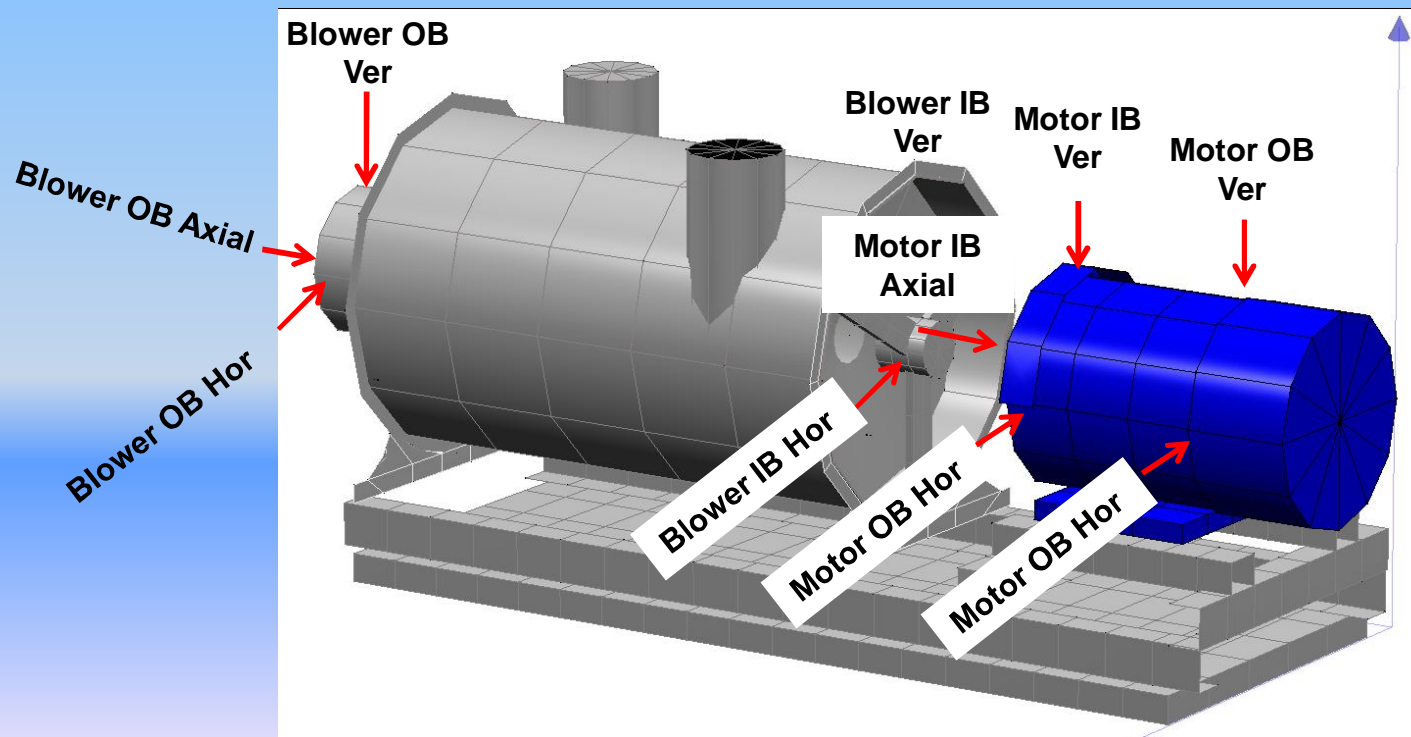
Blower IB Ver TDC

Blower OB Hor 90 Deg Left

Blower OB Ver TDC

Blower OB Axial 0 Deg

Optical Tachometer sensing reflection tape at the coupling.



# DB 19 Blower Continuous Vibration Acquisition:

Trend Plots show that the Blower was Started 3 times.

Vibration was about 6.0 in/sec pk Blower IB Horizontal each startup.

StripChart 09-Jul-2008 05:32:57 0 RPM  
09-Jul-2008 05:32:57 to 09-Jul-2008 11:12:27  
Your Company Name Your Plant Name Your Job Number Your Machine Name

Motor OB H: 90.R 100.0 mV/EU	Overall: .01120 ~ ips (pk)
Motor OB V: 0.L 100.0 mV/EU	Overall: .01476 ~ ips (pk)
Motor IB H: 90.R 100.0 mV/EU	Overall: .01515 ~ ips (pk)
Motor IB V: 0.R 100.0 mV/EU	Overall: .01281 ~ ips (pk)
Motor IB A: 90.R 100.0 mV/EU	Overall: .01868 ~ ips (pk)
Blower IB Hor: 0.R 100.0 mV/EU	Overall: .03567 ~ ips (pk)
Blower IB Ver: 90.R 100.0 mV/EU	Overall: .04865 ~ ips (pk)
Blower OB Hor: 0.R 100.0 mV/EU	Overall: .05792 ~ ips (pk)

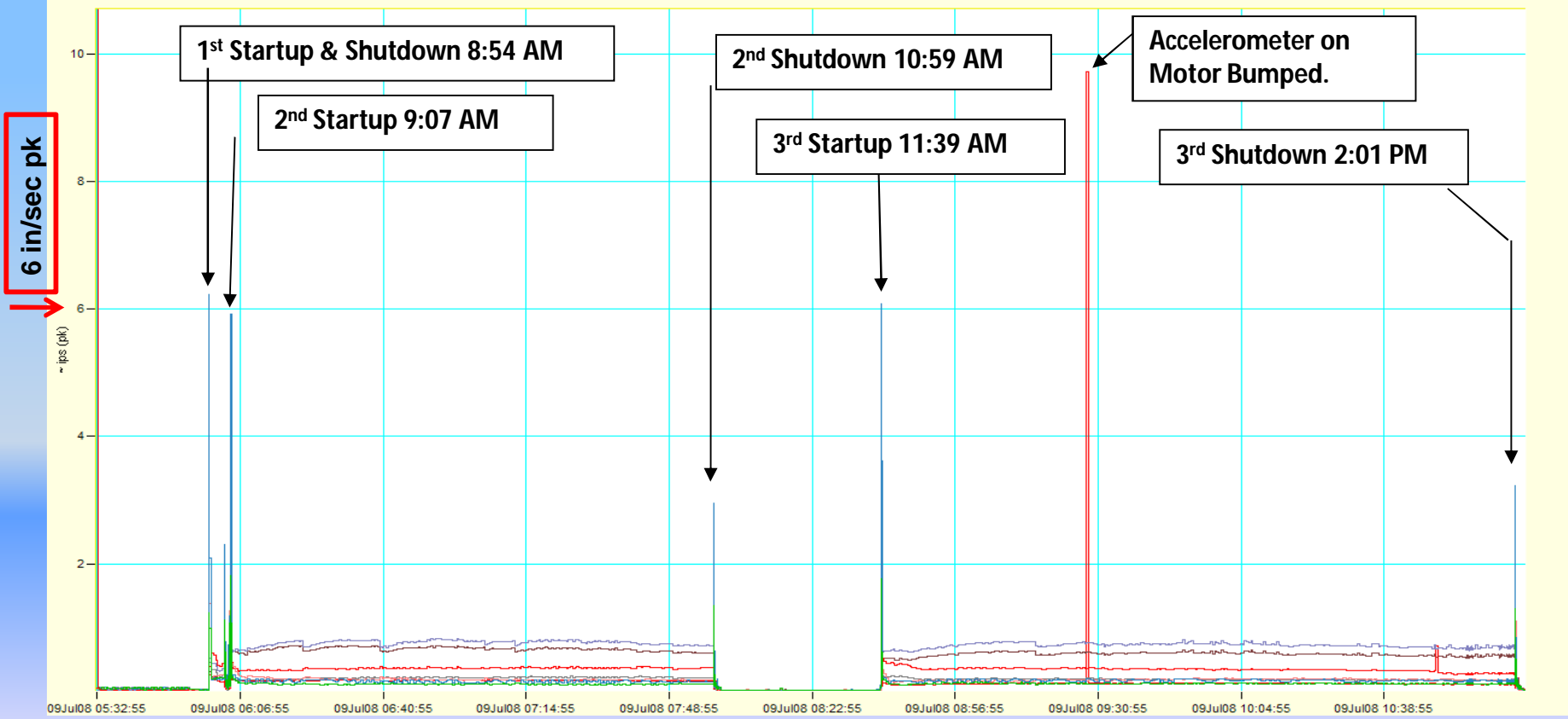


Figure 15. Overall Trend Vibration in/sec pk On DB 19 Blower, Channels 1 - 8.



## DB 19 Blower Continuous Vibration Acquisition:

Bode' plot of blower vibration indicated a resonance near 2400 RPM. The peak shifted from 2400 (spin up) to 2161 (coastdown).

Data indication of a rotor/bearing critical speed.

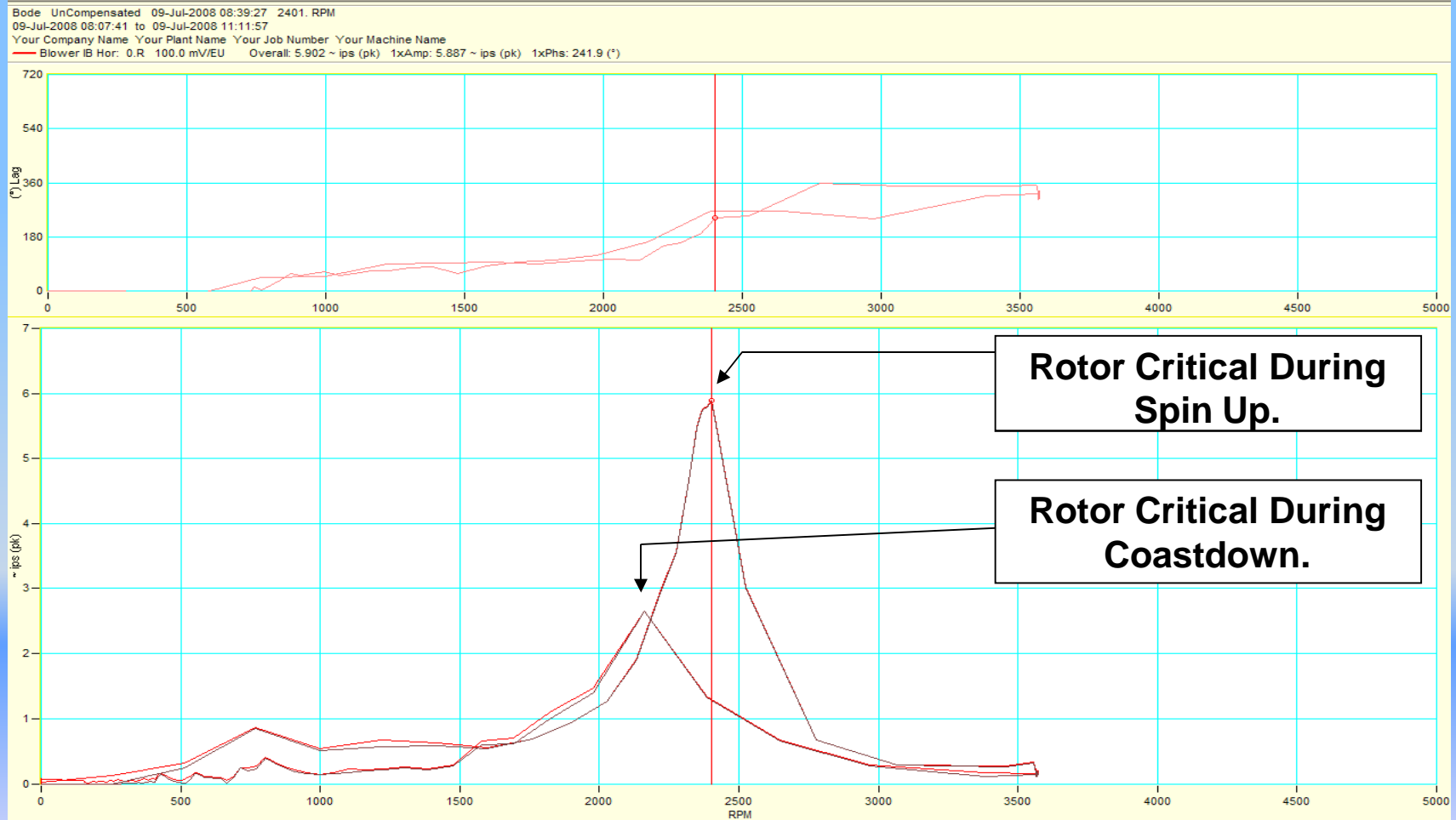
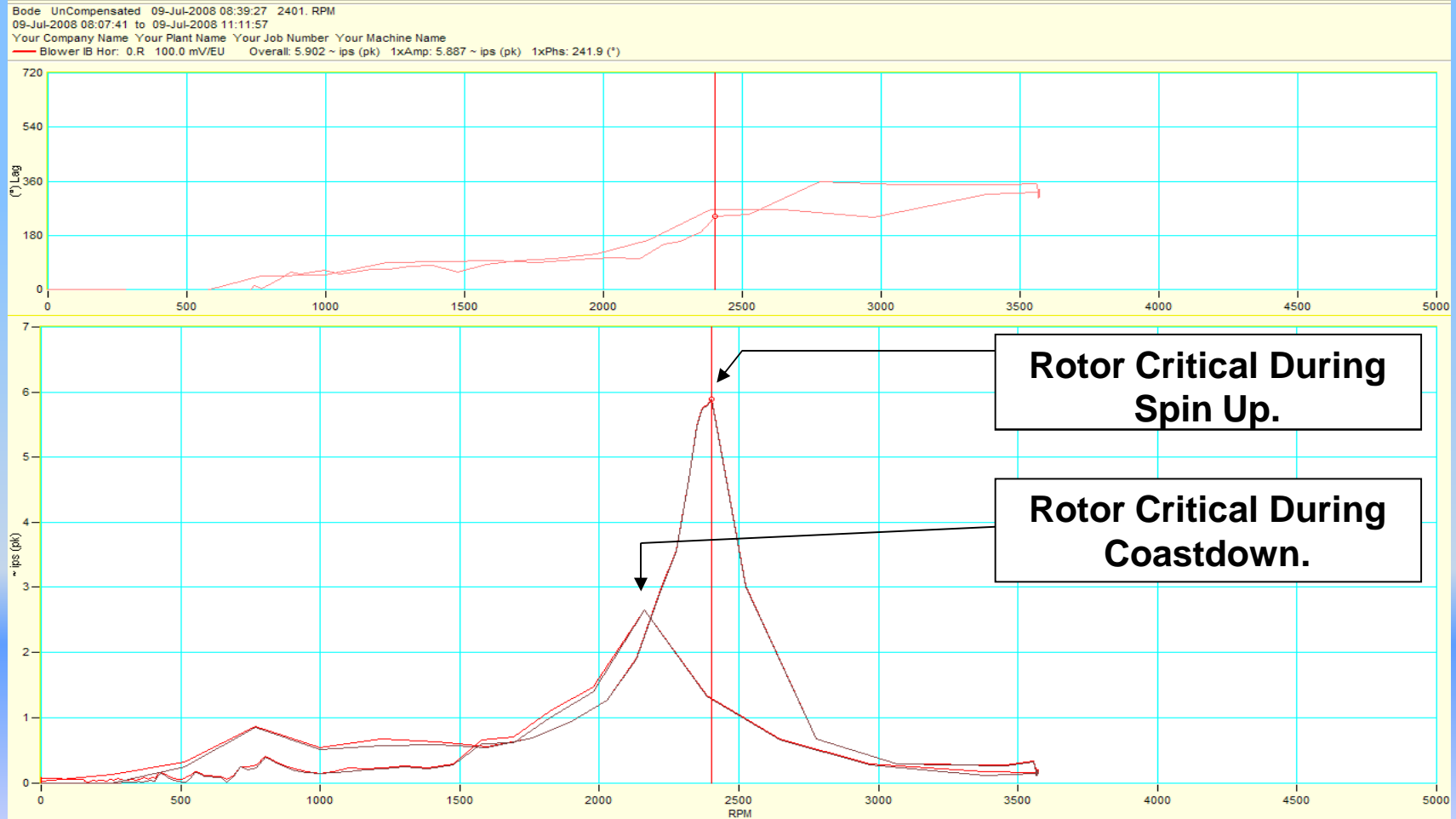


Figure 16: Bode' Plot of Blower IB Hor During 3<sup>rd</sup> Startup and Coastdown. Vibration Amplitude was Very High at About 5.9 in/sec pk.

## DB 19 Blower Continuous Vibration Acquisition:

A call to the Blower OEM about the critical speed location – 1600 RPM.

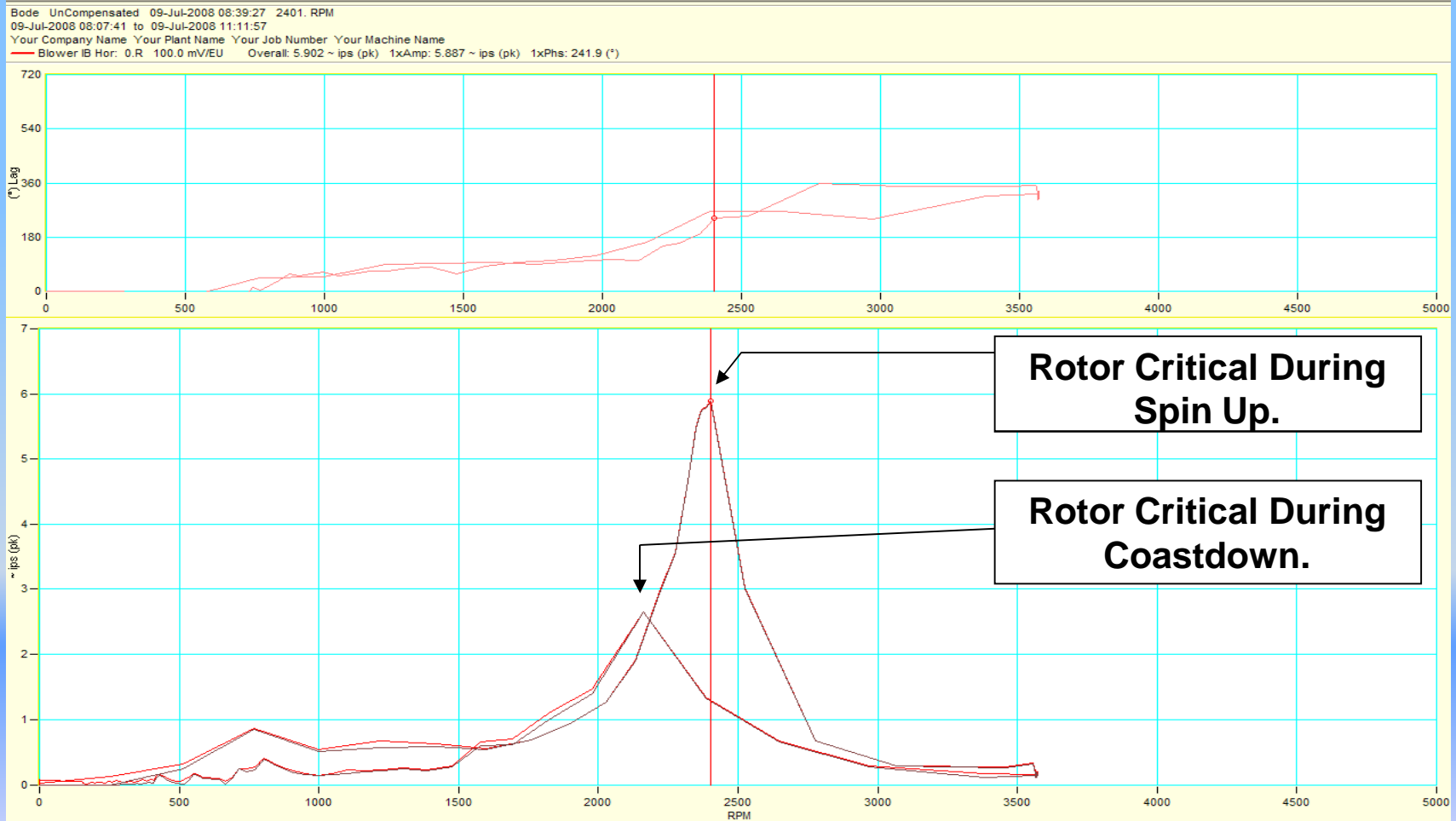
The indication of a critical speed much higher than calculated would not become clear until a rotor dynamic analysis was performed.



**Figure 16: Bode' Plot of Blower IB Hor During 3<sup>rd</sup> Startup and Coastdown. Vibration Amplitude was Very High at About 5.9 in/sec pk.**

## DB 19 Blower Continuous Vibration Acquisition:

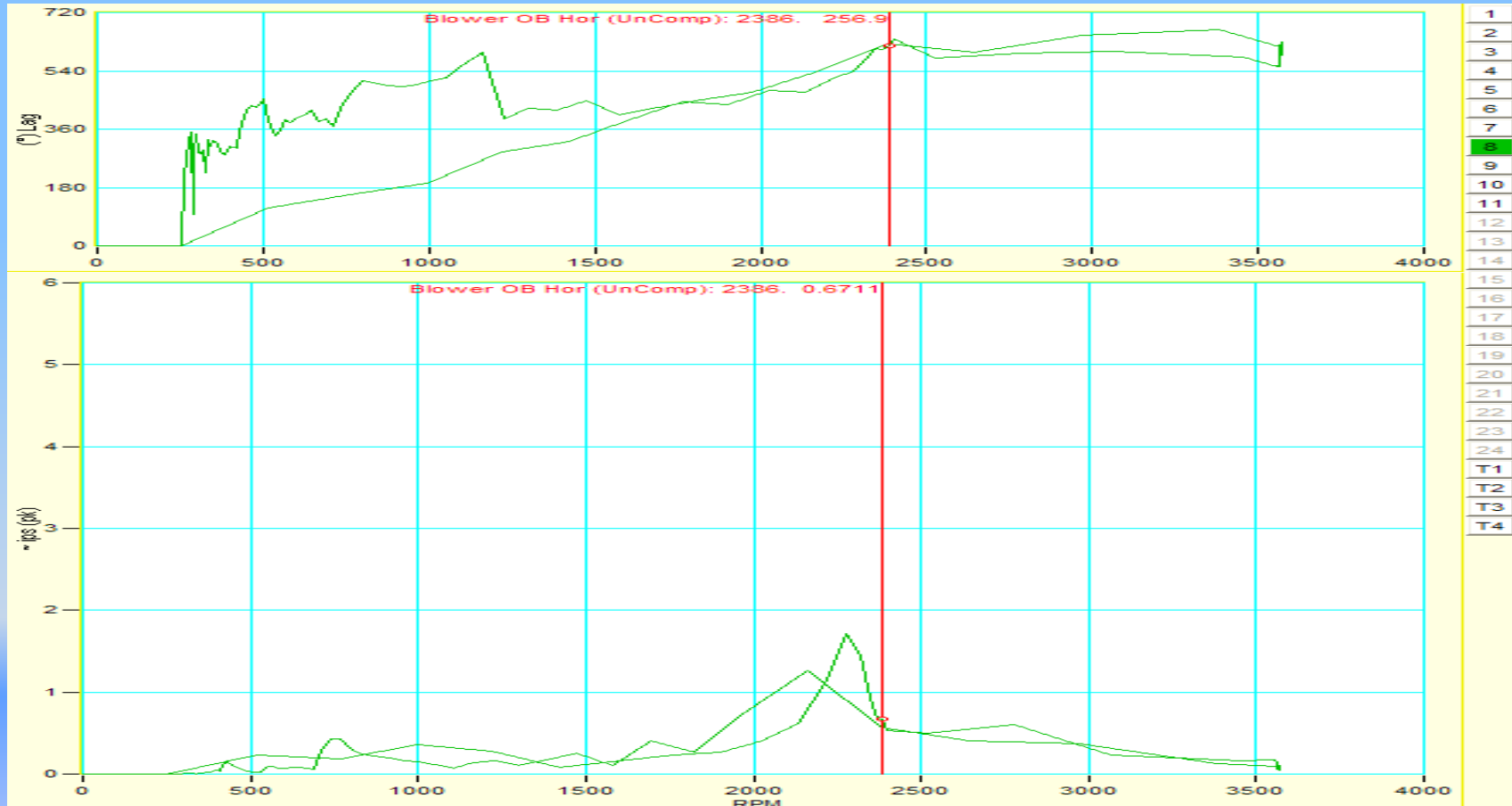
A rotor rub mid-span (between the bearings) due to high rotor unbalance would raise the critical speed.



**Figure 16: Bode' Plot of Blower IB Hor During 3<sup>rd</sup> Startup and Coastdown. Vibration Amplitude was Very High at About 5.9 in/sec pk.**

## DB 19 Blower Continuous Vibration Acquisition:

Bode' plot of Blower OB Bearing Housing Horizontal Response During 3<sup>rd</sup> Startup (2271 RPM) & Coastdown (2161 RPM).



**Figure 17. Bode' Plot of Blower OB Hor During 3<sup>rd</sup> Startup and Coastdown. Vibration Amplitudes Were Much Lower at the OB Bearing Housing and the Critical Speeds Were Also Lower.**

## Modal Test of Blower & Motor Mounted on Skid:

Motor & Blower Bolted to Skid for Impact Experimental Modal Test.  
Analyzer – CSI 2120 Two Channel, Medium Sledge Modal Hammer.

Coupling Removed.

Skid Supported on 1”  
Cork.

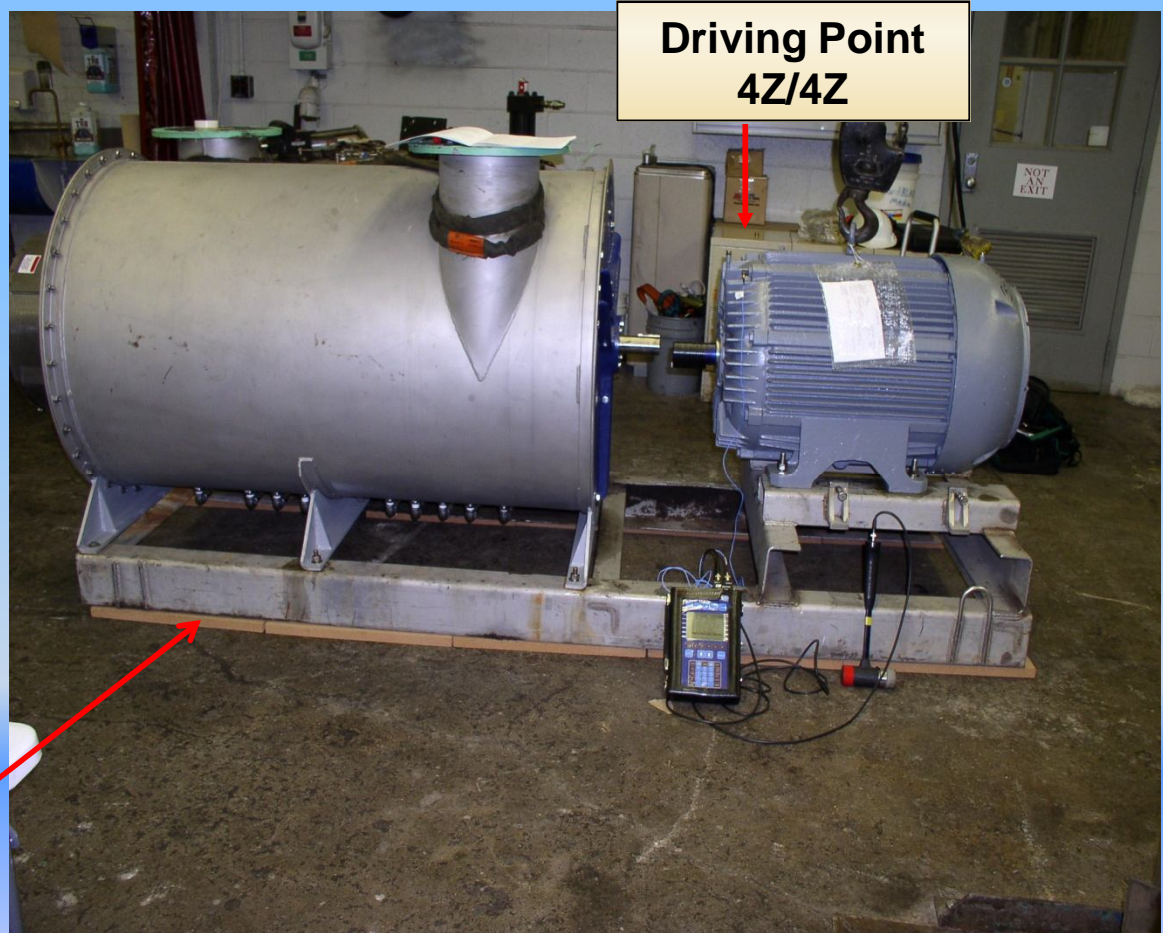


Figure 18: Blower & Skid Supported on Cork in the Maintenance Shop During Modal Test.

## Modal Test of Blower & Motor Mounted on Skid:

Driving Point Frequency Response Function (FRF).

The FRF Displays the Damped Natural Frequencies Excited by Impacting the Motor with Modal Hammer.



Rigid Body Modes of Motor-Blower & Skid on 1" Cork.

Rigid Body Modes are Bouncing/Yaw & Translation Modes of the skid on the Cork Isolation Material.

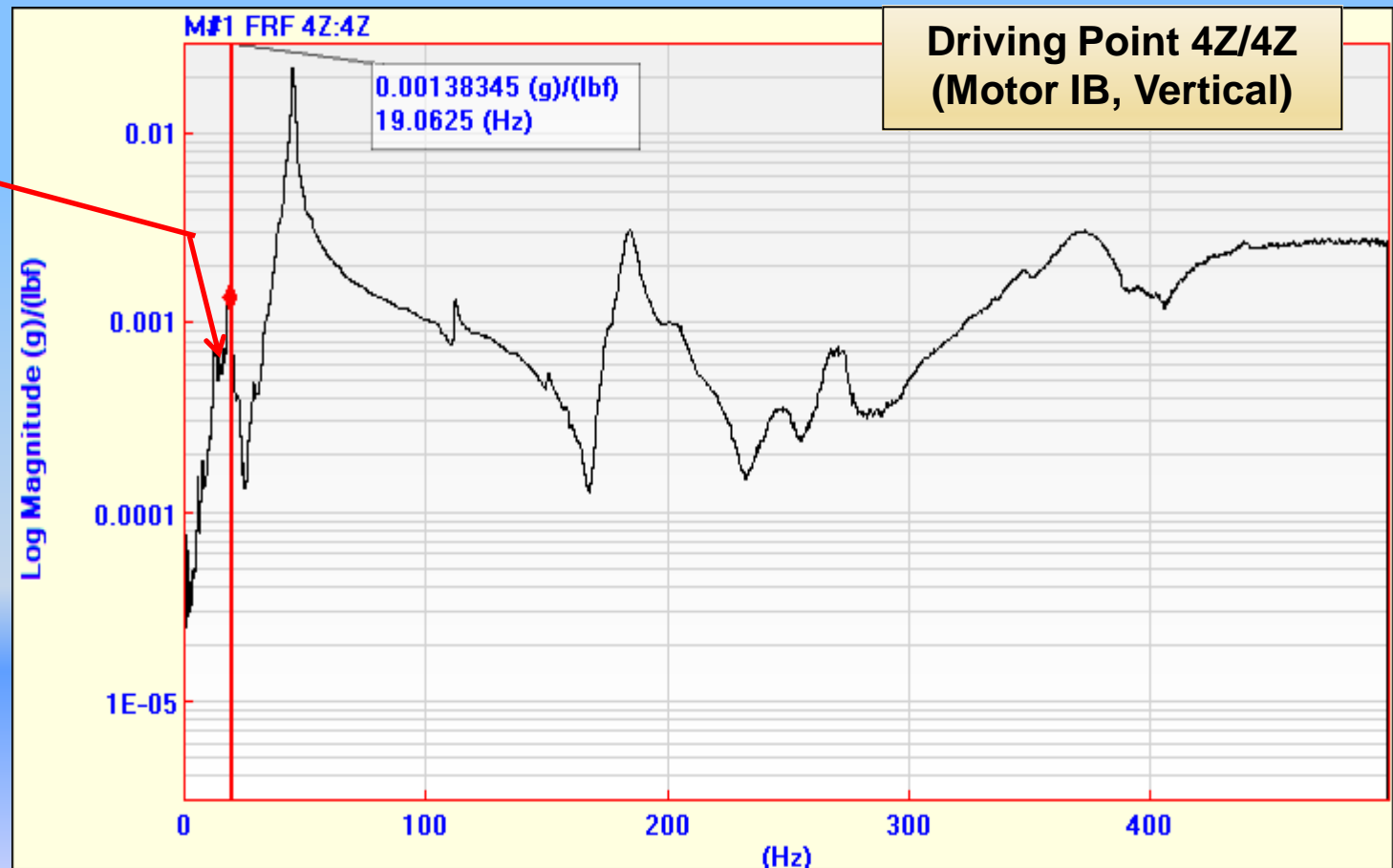


Figure 20: FRF on Motor IB Bearing Housing Showing Natural Frequency Near Running Speed.

## Modal Test of Blower & Motor Mounted on Skid:

Driving Point Frequency Response Function (FRF).

The FRF Displays the Damped Natural Frequencies Excited by Impacting the Motor with Modal Hammer.

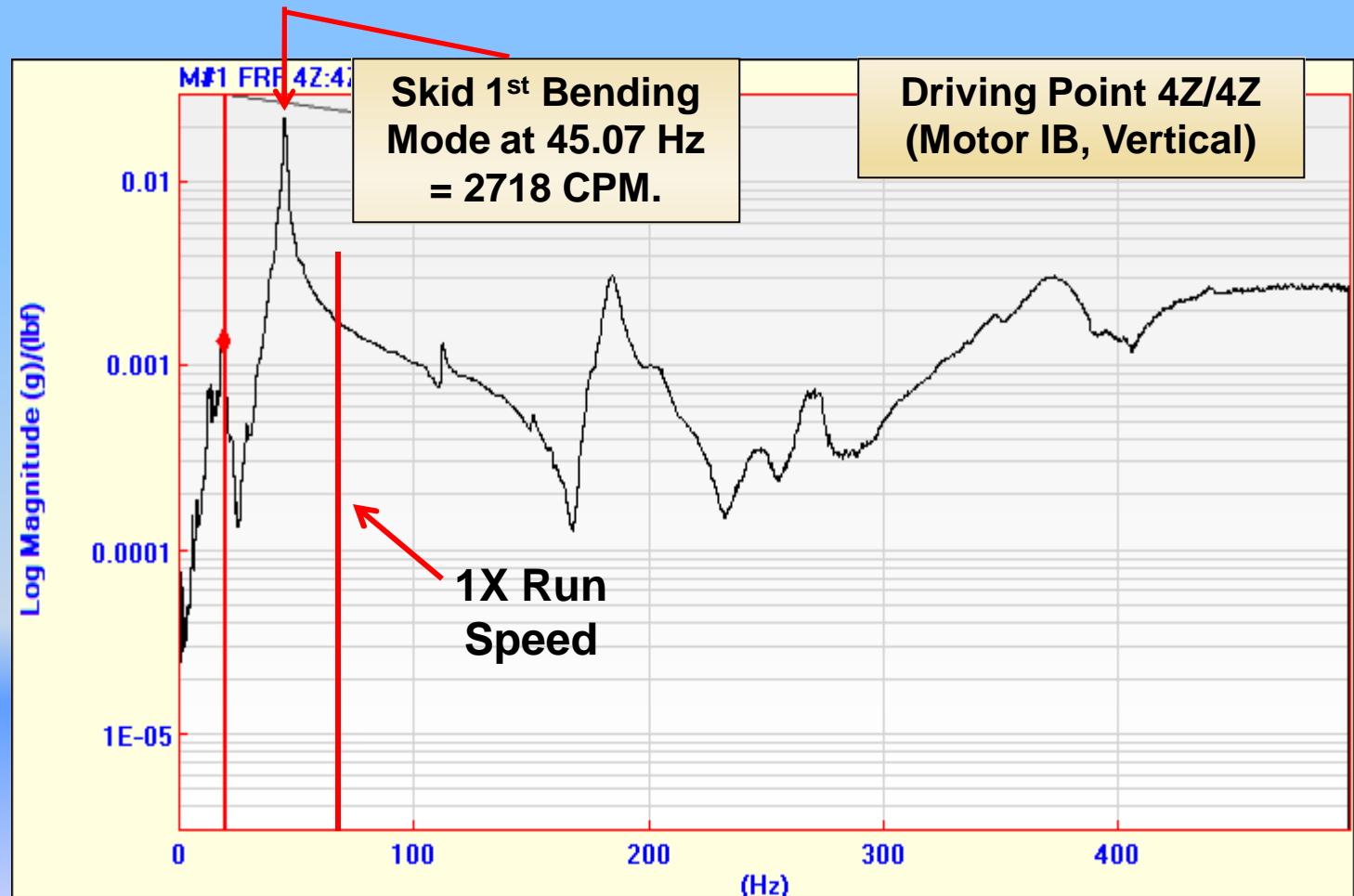


Figure 20: FRF on Motor IB Bearing Housing Showing Natural Frequency Near Running Speed.

## Modal Test of Blower & Motor Mounted on Skid:

The mode shape of the 2718 CPM natural frequency was bending of the skid rails in the vertical direction in the section between the motor support rails and the blower inboard feet mounting position.

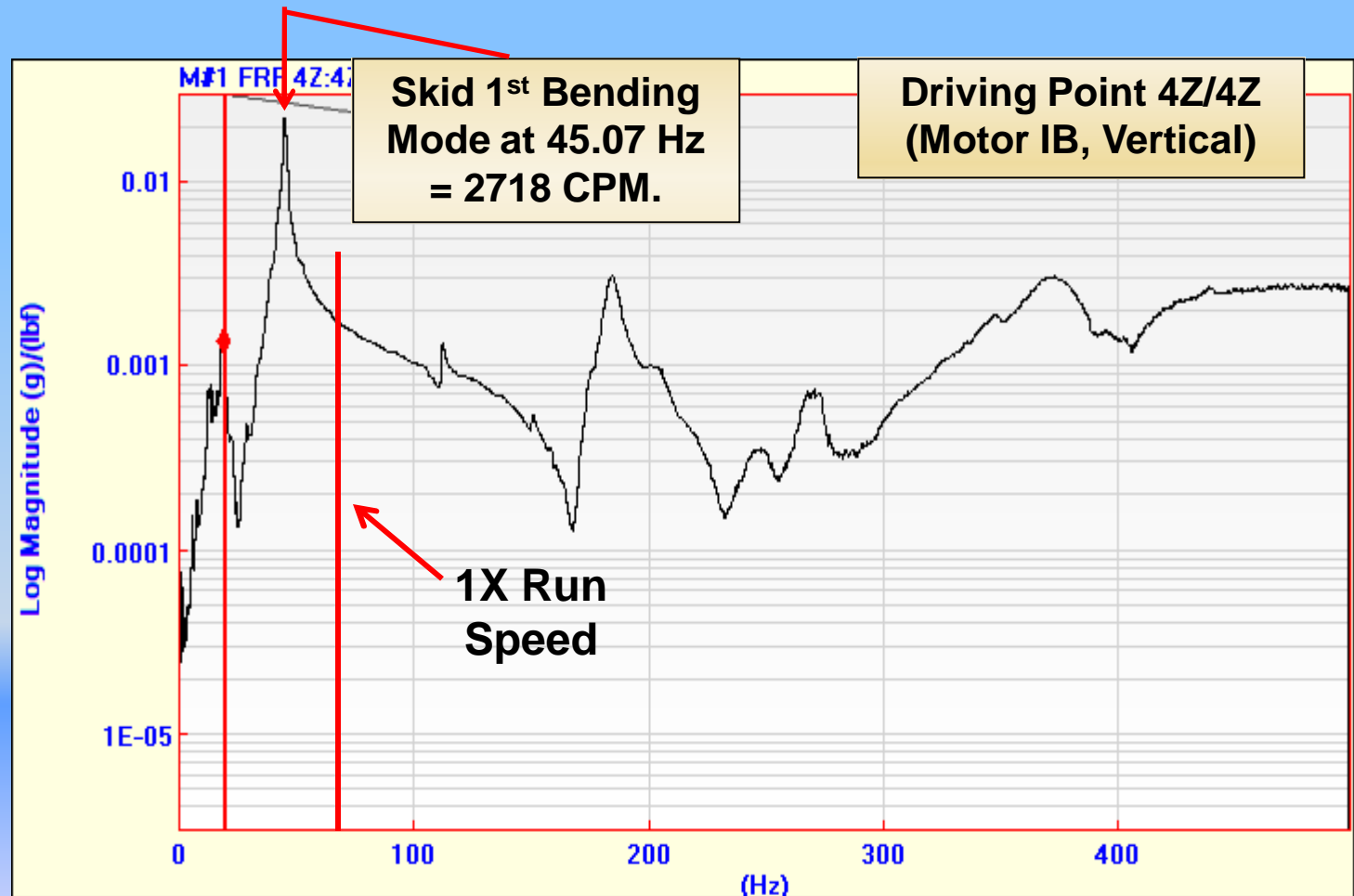


Figure 20: FRF on Motor IB Bearing Housing Showing Natural Frequency Near Running Speed.



## Modal Test of Blower & Motor Mounted on Skid:

Stiffening the Existing Skid Was Recommended by Welding Plate to the Channel (making the channel a rectangular tube) and also Adding plate Stiffeners.

Intent was to Move the Motor-Support Skid Frame Bending Mode Well Above The Run Speed Frequency.

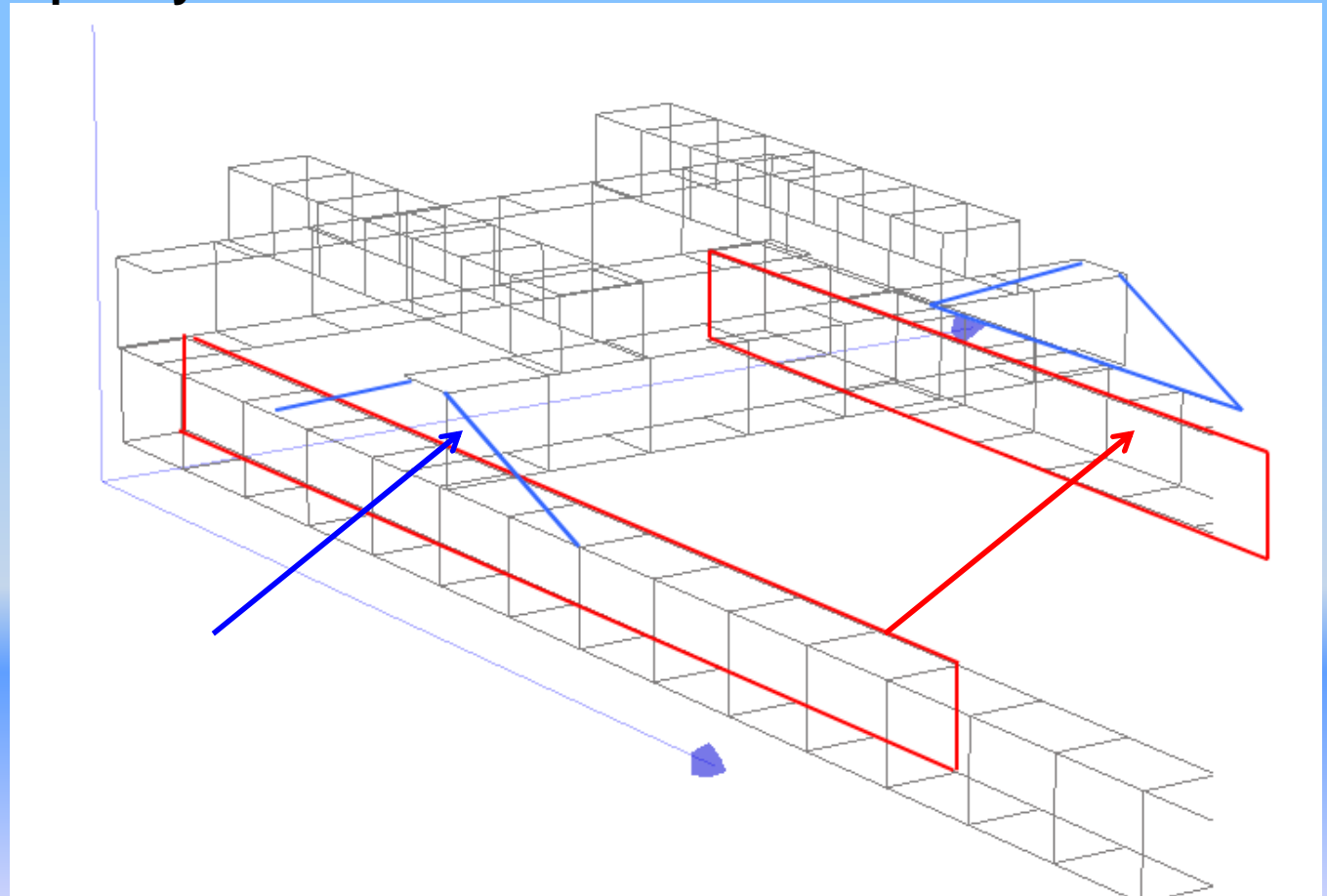
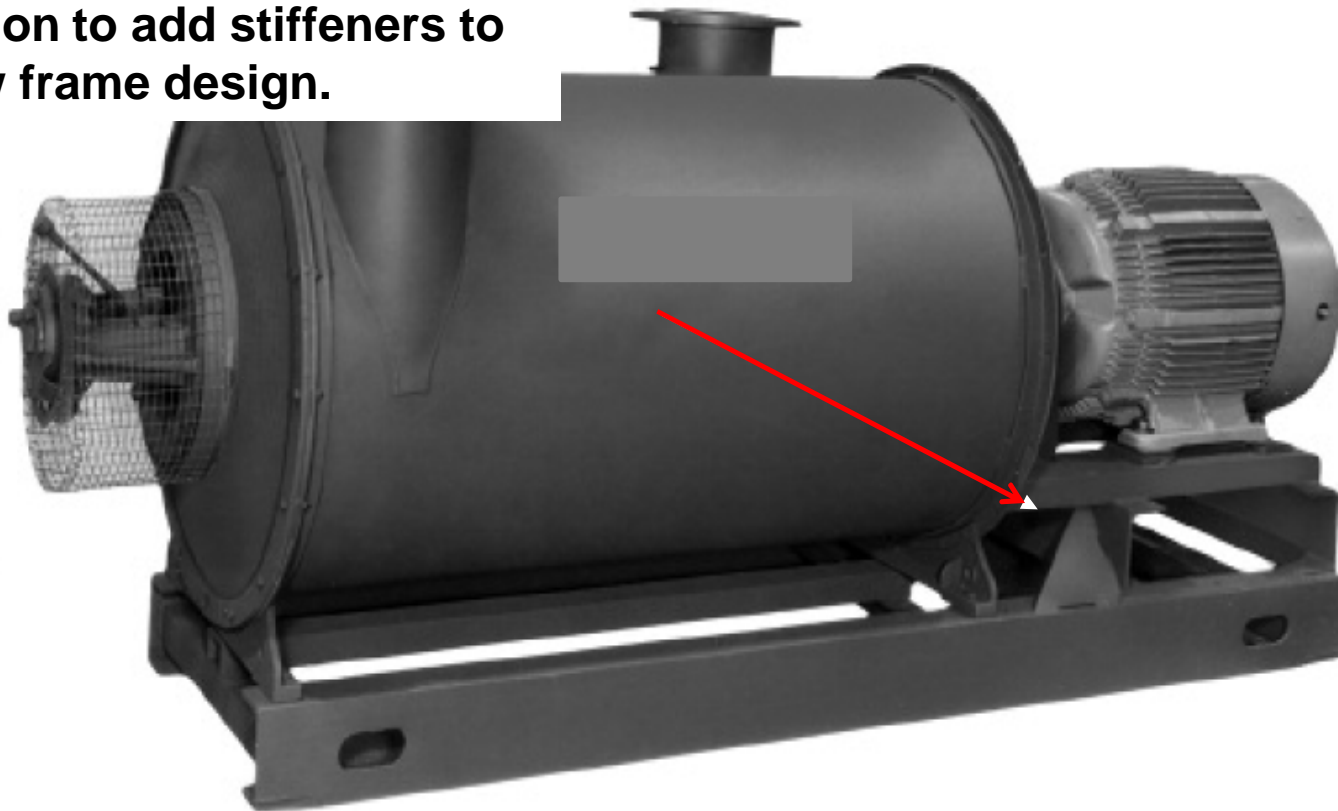


Figure 21: Suggested Stiffen Of the Skid Frame Rails Based on the Modal Test Results.

## **Modal Test of Blower & Motor Mounted on Skid:**

**It Turned Out that the Blower OEM Design Incorporated Stiffeners in this Area. During maintenance, the OEM Stiffeners had been Removed to Gain Access to Bolts but they had not been replaced.**

**The equipment owner used this information to add stiffeners to their new frame design.**



**Figure 22. Stiffeners Originally Included in the Blower Design.**

## Rotor Bearing Dynamic Analysis:

A Model of the Blower Rotor was Developed in DyRoBeS Finite Element Based Software. Ref 3 & 4 (Rodyne & DyRoBeS)

Shaft was Modeled using Dimensions from a Drawing.

The Model Included the Dynamic Mass and Stiffness for the Blower Bearing Housings Which was Measured Using Driving Point Modal Analysis Measurements.

Bearing Housing (Support Stiffness & Damping)  
 $K = 150,000 \text{ lb}_f/\text{in}$   
Damping Factor = 0.35 ( $Q=4.5$ )

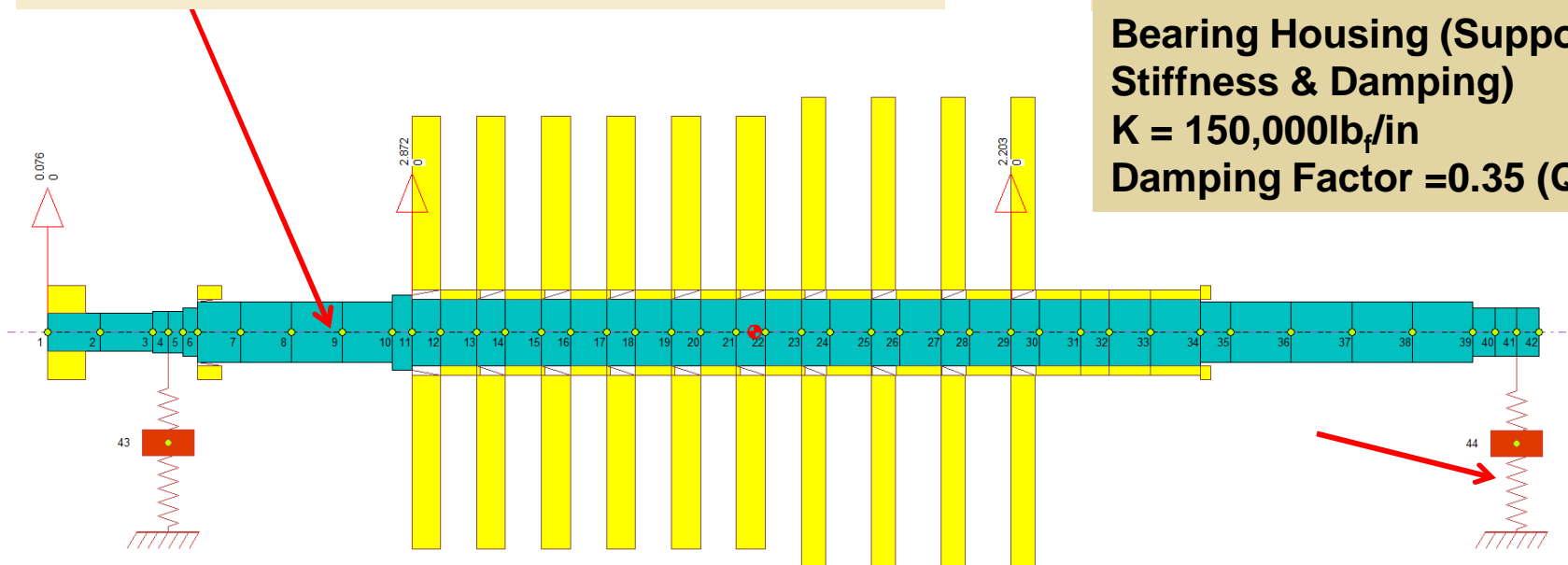


Figure 23. Blower Rotor-Bearing-Support Model.

# Rotor Bearing Dynamic Analysis:

A Model of the Blower Rotor was Developed in DyRoBeS <sup>Ref 1</sup> Finite Element Based Software.

The Ball Bearing Stiffness was Calculated using DyRoBeS.

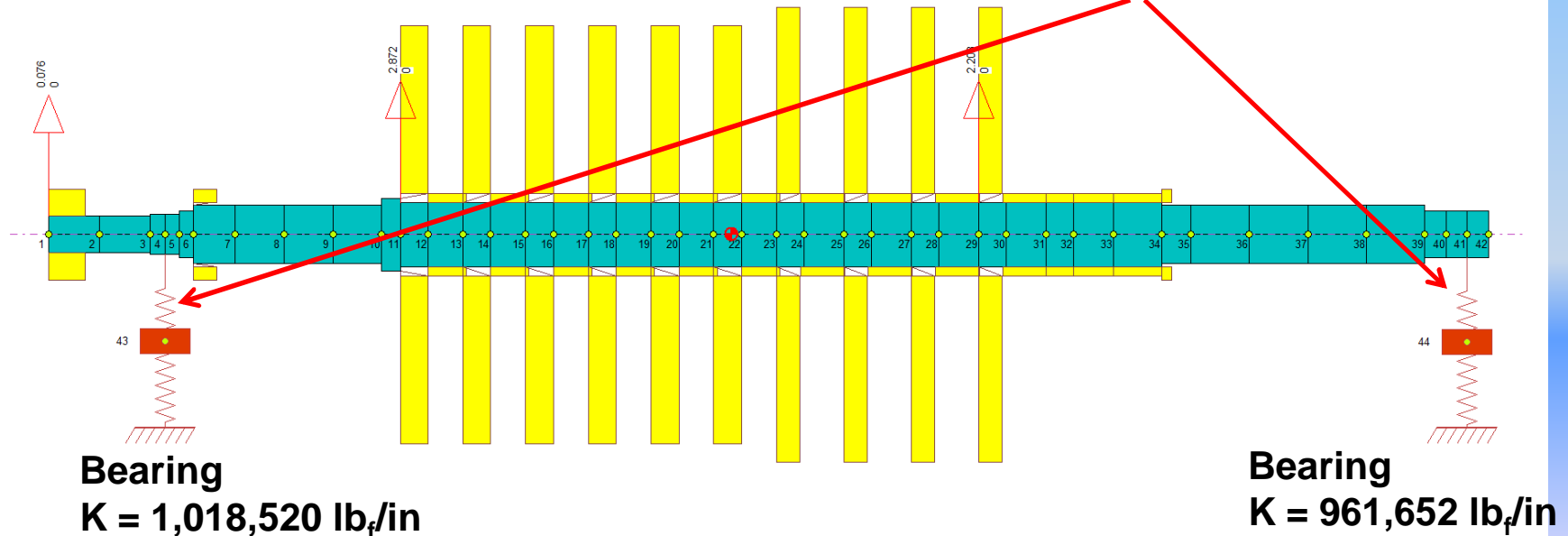


Figure 23. Blower Rotor-Bearing-Support Model.

## Rotor Bearing Dynamic Analysis:

A Model of the Blower Rotor was Developed in DyRoBeS <sup>Ref 1</sup> Finite Element Based Software.

The Impeller Weight Was Provided.

The Transverse & Polar Moment of Inertia For Each Impeller Was Calculated using Tools in DyRoBeS.

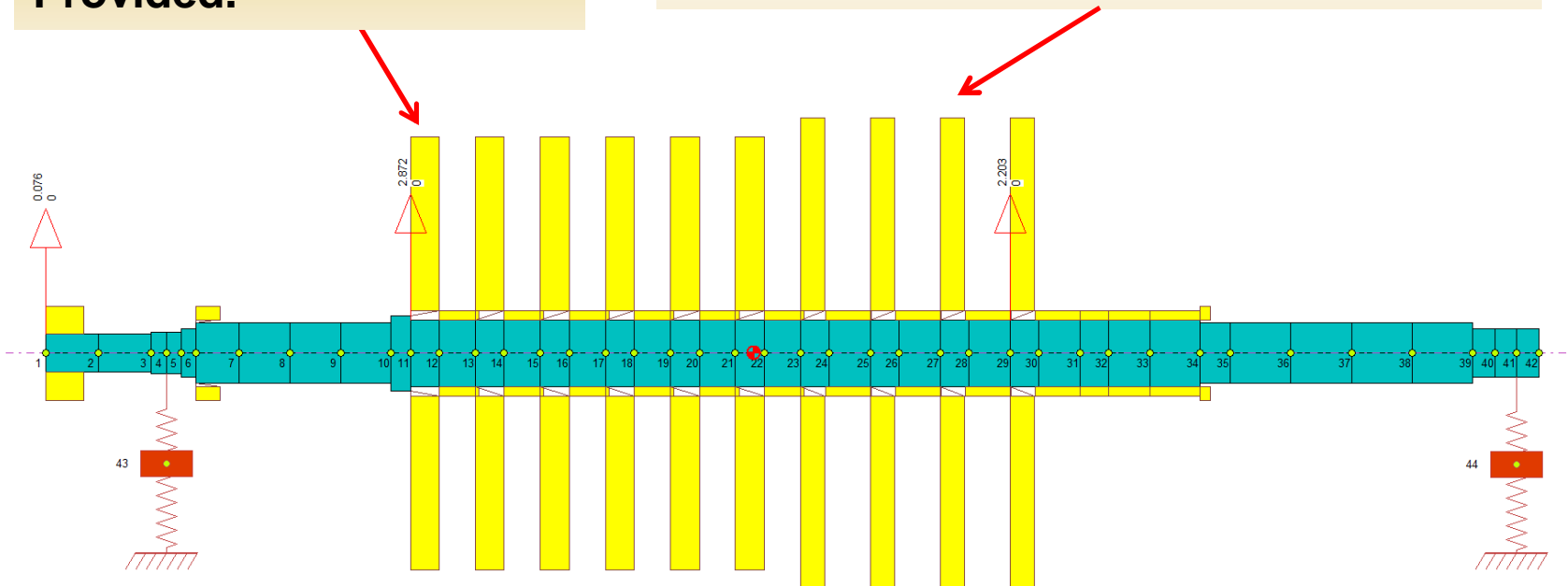


Figure 23. Blower Rotor-Bearing-Support Model.

## Rotor Bearing Dynamic Analysis:

A Model of the Blower Rotor was Developed in DyRoBeS <sup>Ref 1</sup> Finite Element Based Software.

Note that the 1<sup>st</sup> Four Impellers are Larger Diameter.

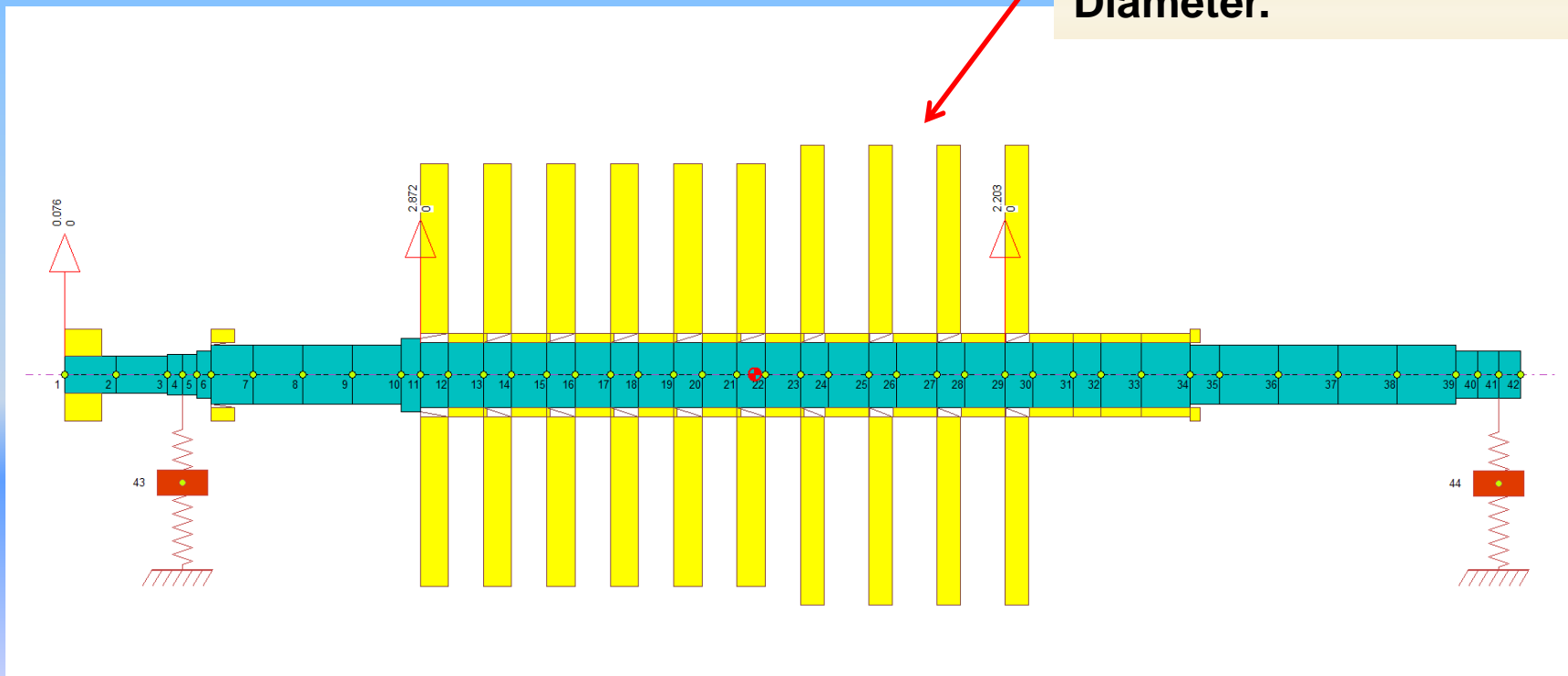


Figure 23. Blower Rotor-Bearing-Support Model.



## Rotor Bearing Dynamic Analysis:

The Rotor-Bearing 1<sup>st</sup> Critical Speed Calculated to 1660 RPM

The Nodal Points were Outboard of Each Bearing About 4 Inches.

Very important that the nodal points of a critical are not at the bearings. If this happens, the bearings provide very little control of the rotor vibration at the critical speed.

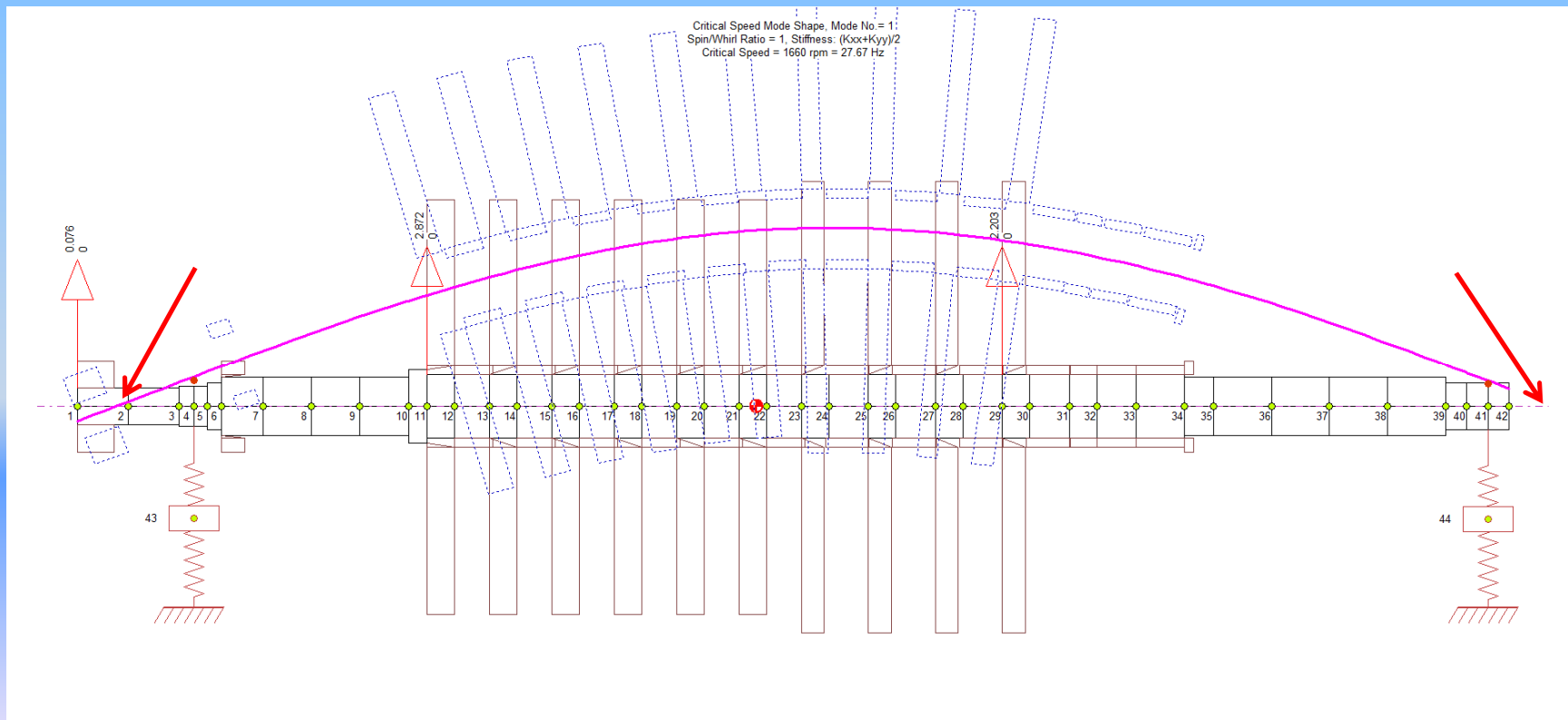


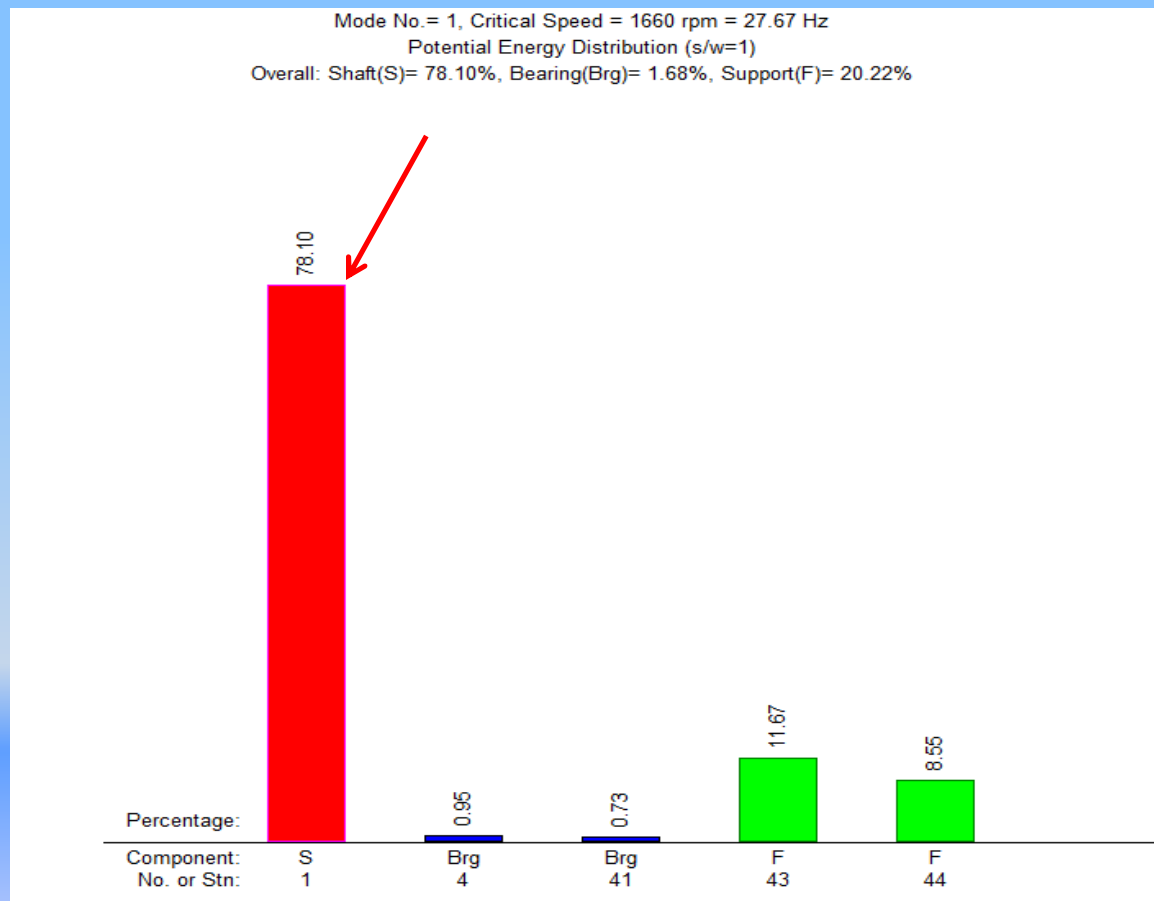
Figure 24. Rotor 1<sup>st</sup> Mode. Nodal Points are About 4" Outside Each Bearing.



## Rotor Bearing Dynamic Analysis:

Potential Energy is Calculated by DyRoBeS Which is Useful to Show if the Shaft, Bearings and Support are Properly Participating in the Critical Speed.

The Chart Shows that Most Potential Energy was in the Shaft.



**Figure 25. Potential Energy Distribution Showing Most Energy in the Shaft but the Bearings Have Very Low Participation in the Mode.**

## Rotor Bearing Dynamic Analysis:

Dr Gunter Recommends the bearings have at Least 20% Strain Energy .  
The Bearings Calculated to Have Less than 1% Strain Energy Which Means the bearings will not contribute to overall system damping. The rotor will be Very Sensitive to Unbalance and Expected to Have High Vibration at the 1<sup>st</sup> Critical.

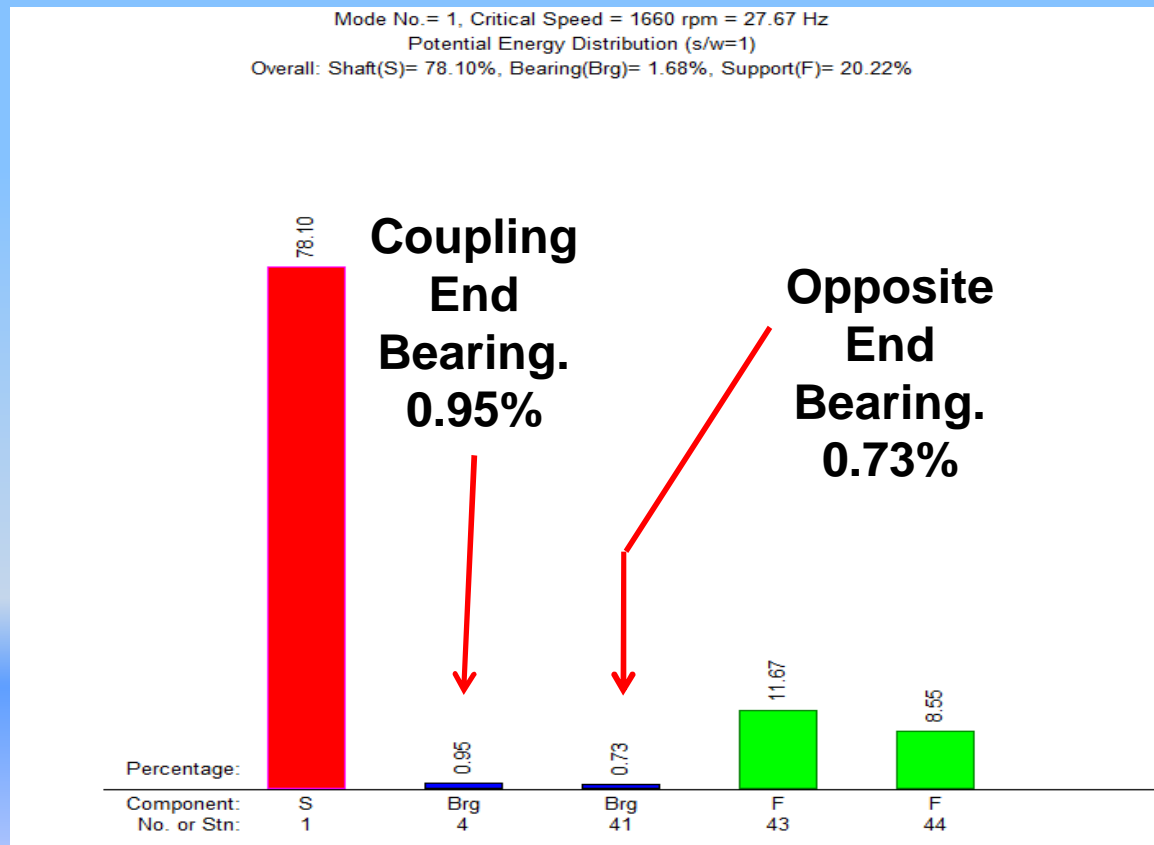
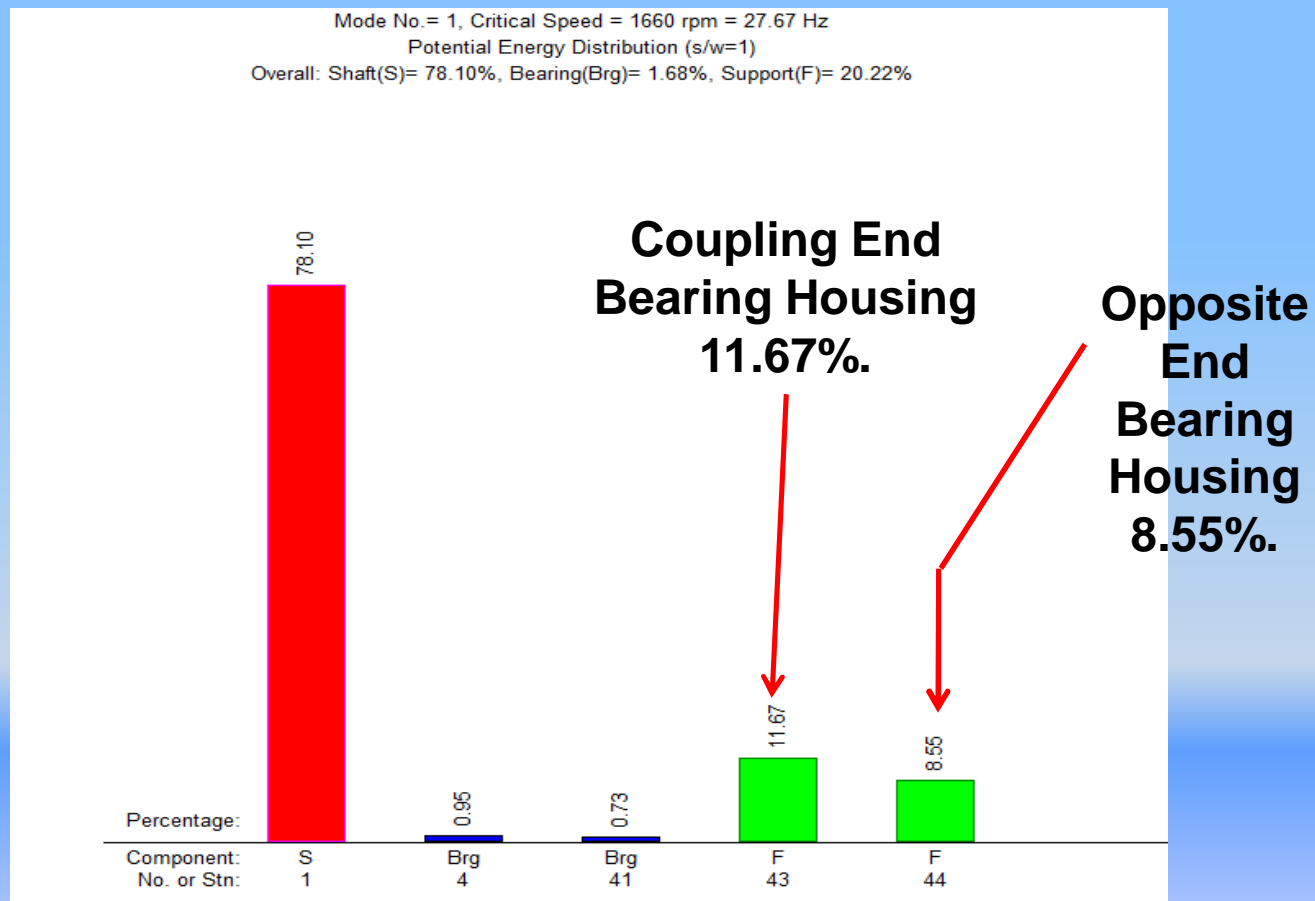


Figure 25. Potential Energy Distribution Showing Most Energy in the Shaft but the Bearings Have Very Low Participation in the Mode.

## Rotor Bearing Dynamic Analysis:

The Bearing Housing Calculated to about 12% and 9% Potential Energy.

The Shaft is the Primary Component Controlling Critical Speed Response.



**Figure 25. Potential Energy Distribution Showing Most Energy in the Shaft but the Bearings Have Very Low Participation in the Mode.**

## Rotor Bearing Dynamic Analysis:

The 1<sup>st</sup> Critical is Plotted to Show how the Rotor Bows Out During Passage Through the Critical Speed.

Unbalance = to ISO G 6.3 Was Placed in the Model.

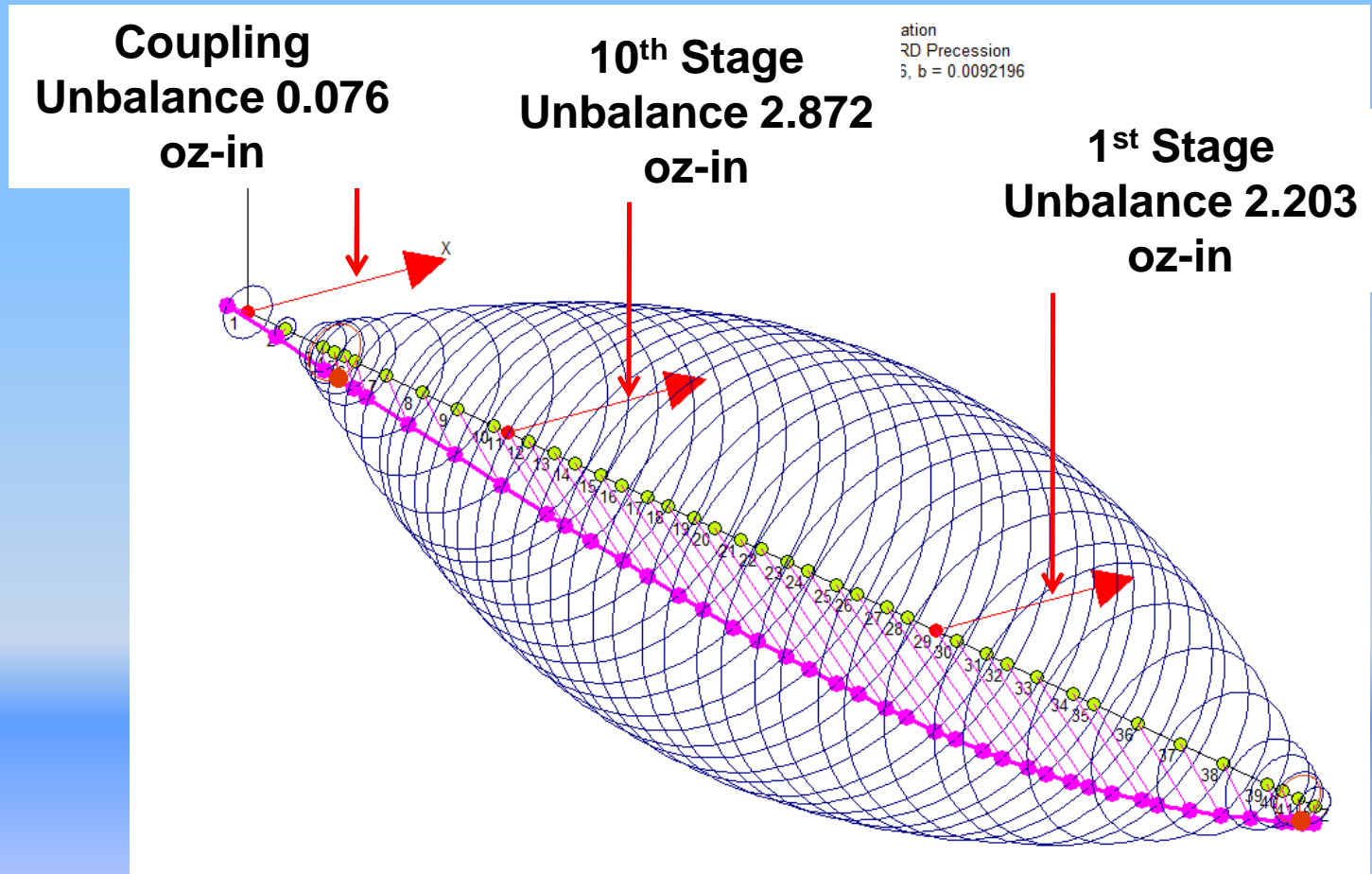


Figure 27: Rotor Response at 1<sup>st</sup> Critical to G 6.3 Unbalance in the 1<sup>st</sup> and 10<sup>th</sup> Stage Impellers. Maximum Orbit Mid-Span Calculated to About 11 mils pp.

## Rotor Bearing Dynamic Analysis:

The 1<sup>st</sup> Critical is Plotted to Show how the Rotor Bows Out During Passage Through the Critical Speed.

Unbalance = to ISO G 6.3 Was Placed in the Model.

The Rotor Response Lags the Unbalance Location about 90 Deg at the Critical And About 180 Deg Well Above the Critical.

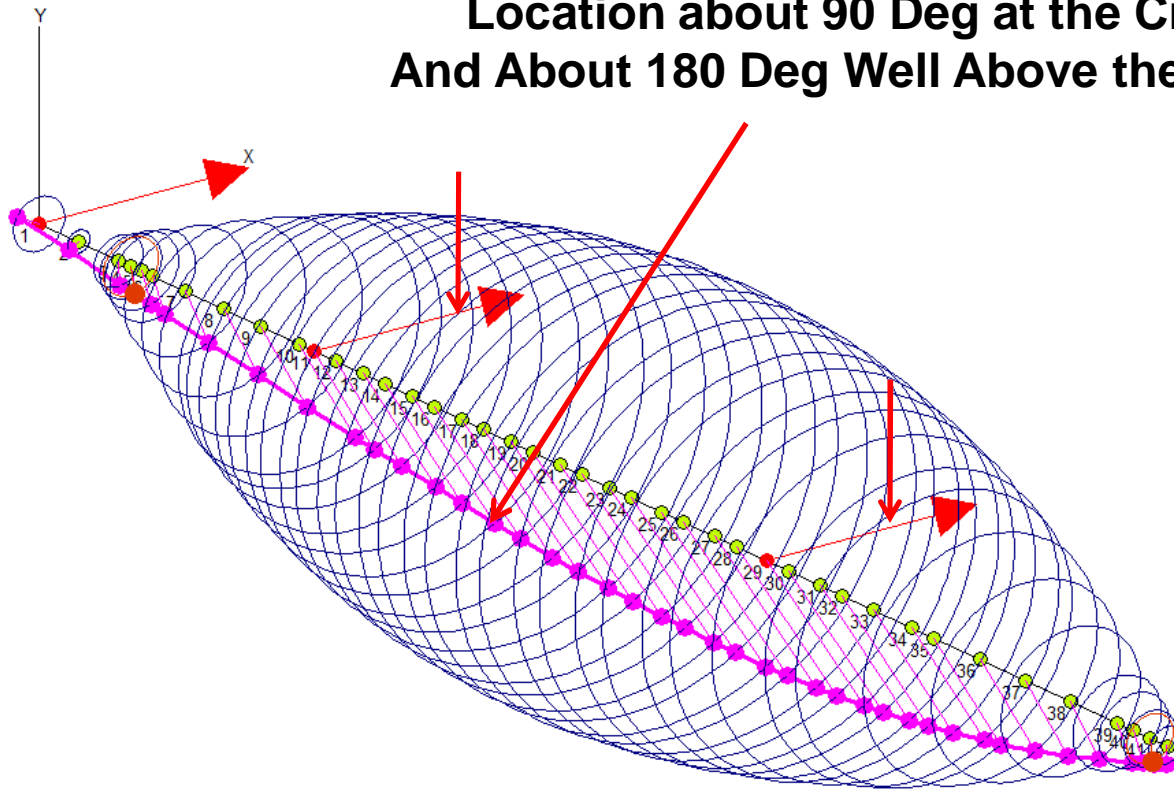


Figure 27: Rotor Response at 1<sup>st</sup> Critical to G 6.3 Unbalance in the 1<sup>st</sup> and 10<sup>th</sup> Stage Impellers. Maximum Orbit Mid-Span Calculated to About 11 mils pp.

## Rotor Bearing Dynamic Analysis:

The 1<sup>st</sup> Critical is Plotted to Show how the Rotor Bows Out During Passage Through the Critical Speed.

Unbalance = to ISO G 6.3 Was Placed in the Model.

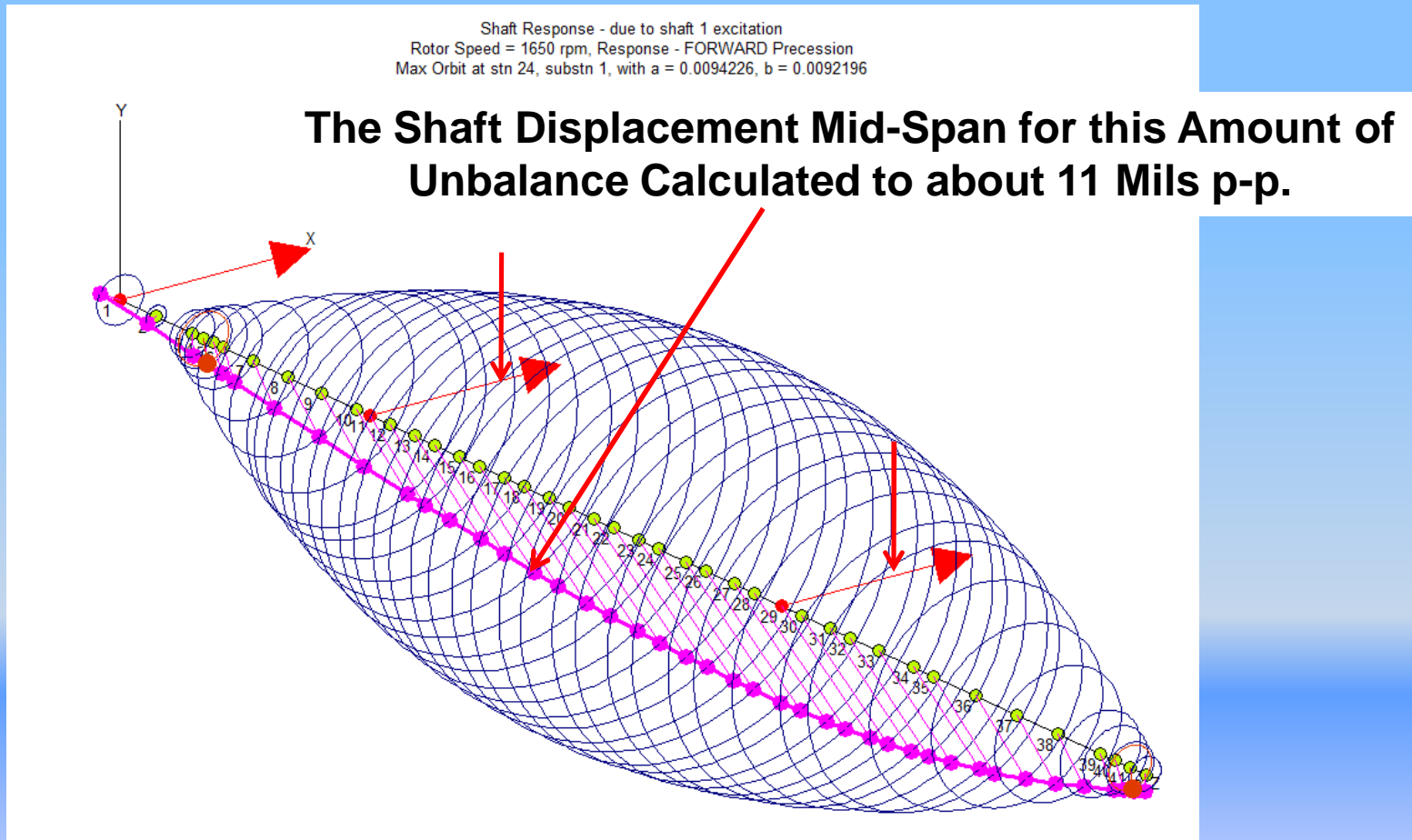


Figure 27: Rotor Response at 1<sup>st</sup> Critical to G 6.3 Unbalance in the 1<sup>st</sup> and 10<sup>th</sup> Stage Impellers. Maximum Orbit Mid-Span Calculated to About 11 mils pp.

## Rotor Bearing Dynamic Analysis:

The 1<sup>st</sup> Critical is Plotted to Show how the Rotor Bows Out During Passage Through the Critical Speed.

Unbalance = to ISO G 6.3 Was Placed in the Model.

The Unbalance Force at the Bearings was Calculated.

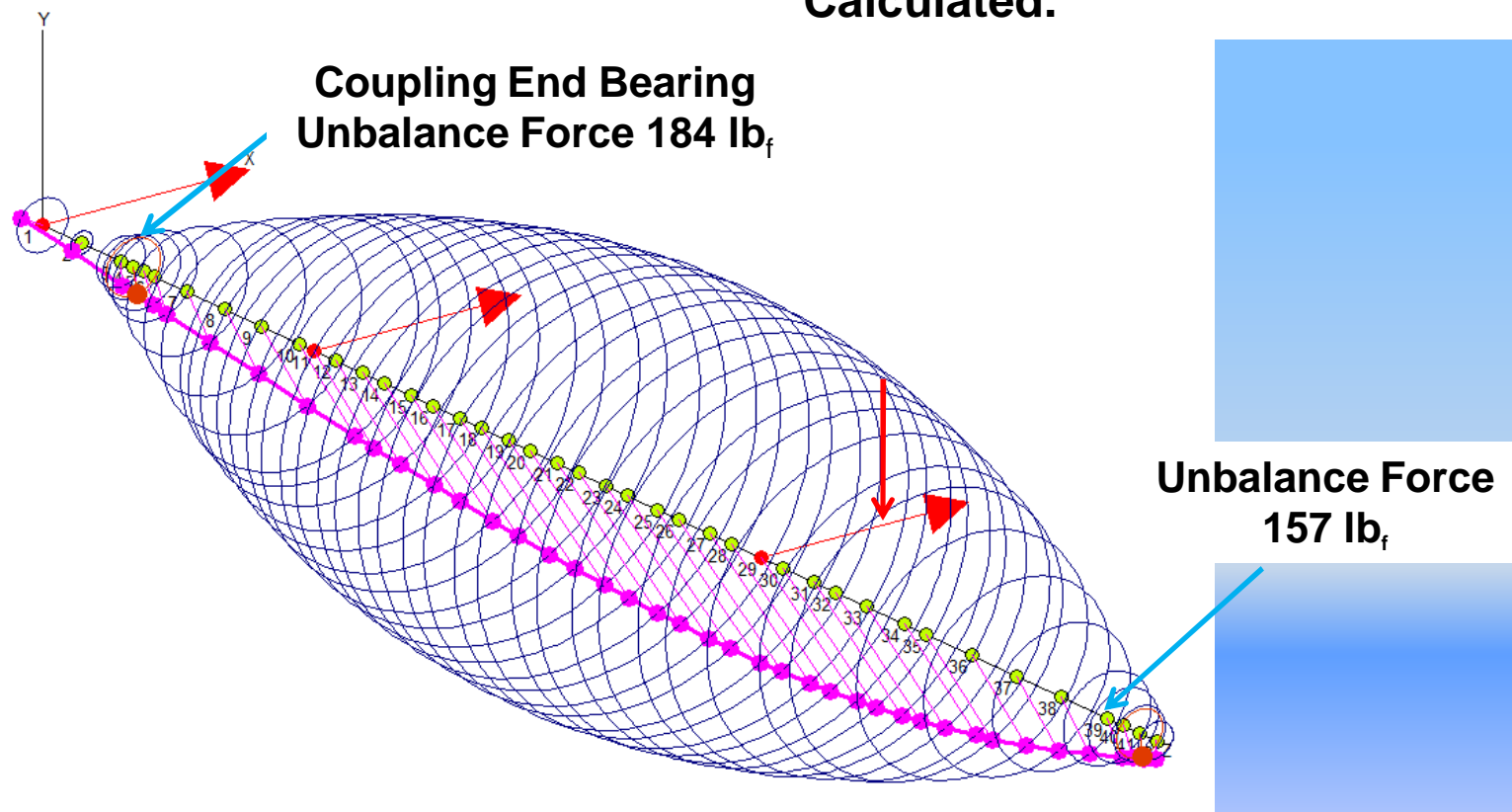


Figure 27: Rotor Response at 1<sup>st</sup> Critical to G 6.3 Unbalance in the 1<sup>st</sup> and 10<sup>th</sup> Stage Impellers. Maximum Orbit Mid-Span Calculated to About 11 mils pp.

## Rotor Bearing Dynamic Analysis:

The Damped Unbalanced Response for the Drive End Bearing is Shown in Figure 28 in Bode' Plot.

Response from X Probe 0 Deg and Y Probe 90 Deg is Plotted for Response to ISO G 6.3 Unbalance.

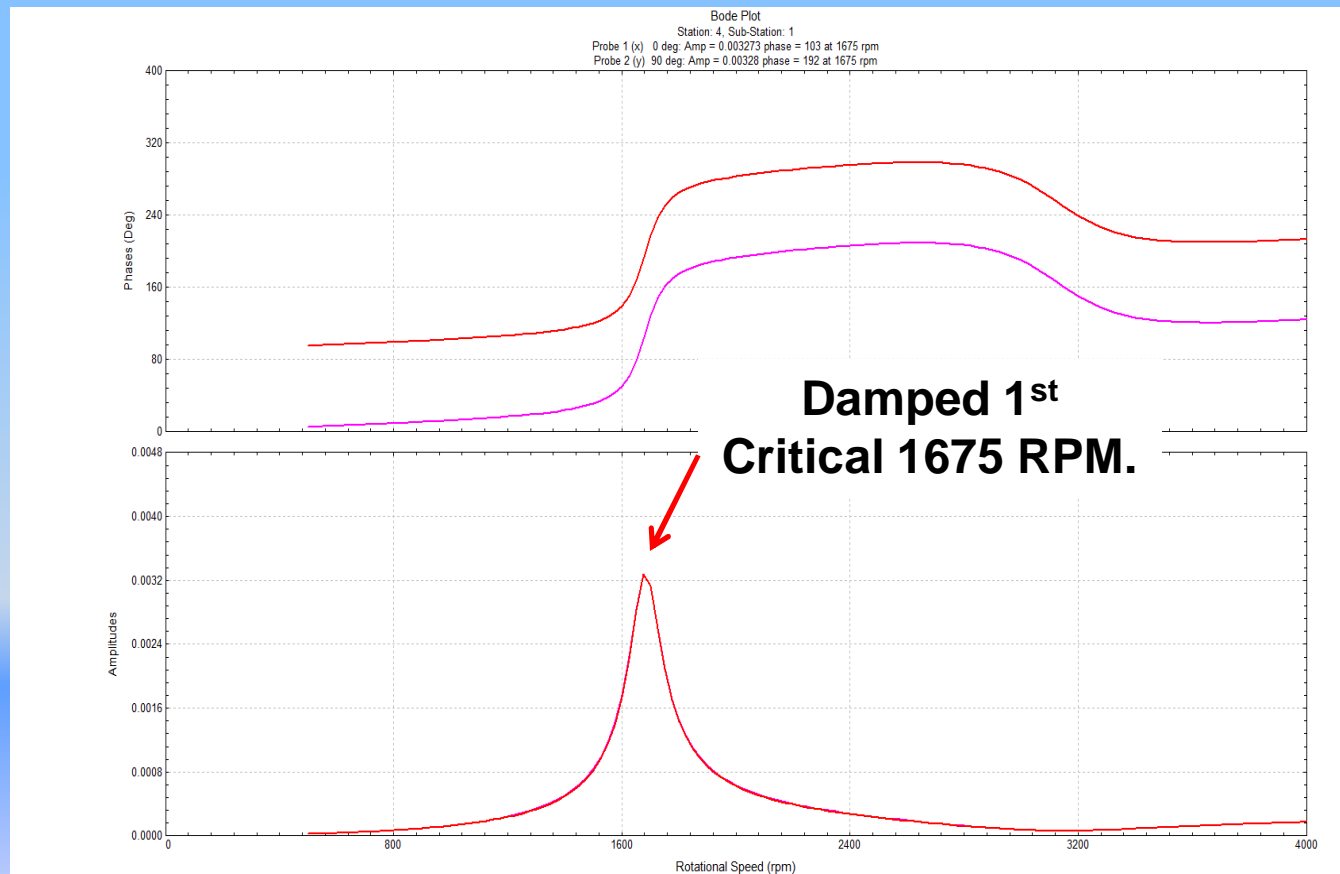


Figure 28. Bode' Plot of Damped Unbalanced Response at the Drive End Bearing to G6.3 Unbalance in 1<sup>st</sup> and 10<sup>th</sup> Stage Impellers.



## Rotor Bearing Dynamic Analysis:

A Rub at the Packing Was Simulated by Added a Third Bearings.

Different Stiffness Values were Tested to Find that 200,000 lb<sub>f</sub> Moved the 1<sup>st</sup> Critical to About 2400 RPM – Agreeing with the Field Vibration Data.

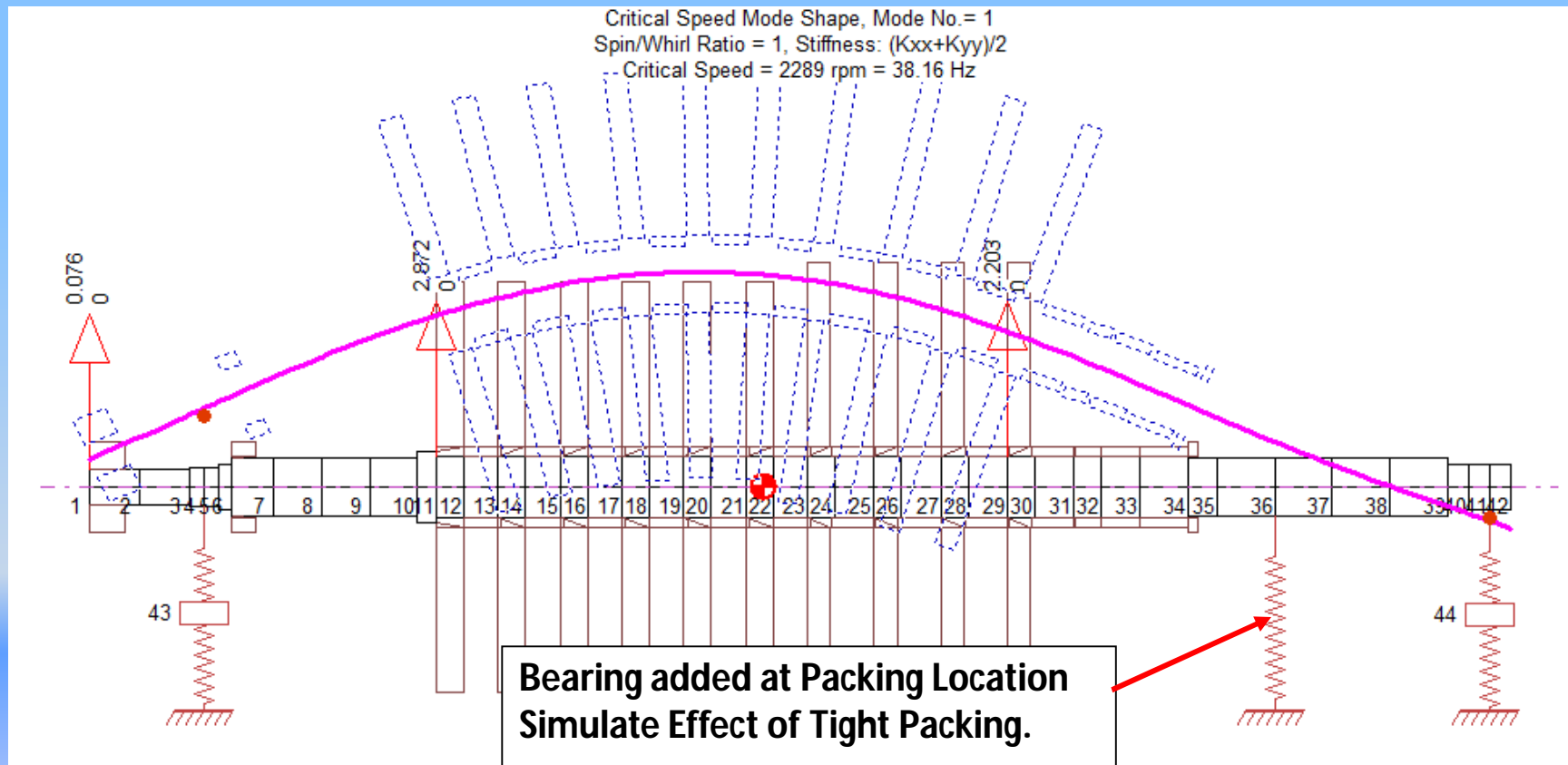


Figure 30. Blower Rotor Model with Bearing Added at the Packing Location to Simulate the Stiffening Effect of the Packing.

# Rotor Bearing Dynamic Analysis:

Based on the Vibration Analysis and Rotor Models:

Tight Packing Acting as a Bearing was predicted to shift the rotor 1<sup>st</sup> Critical to about 2400 RPM during spin up.



Animation of Damped 1<sup>st</sup> Forward Mode 2377 RPM

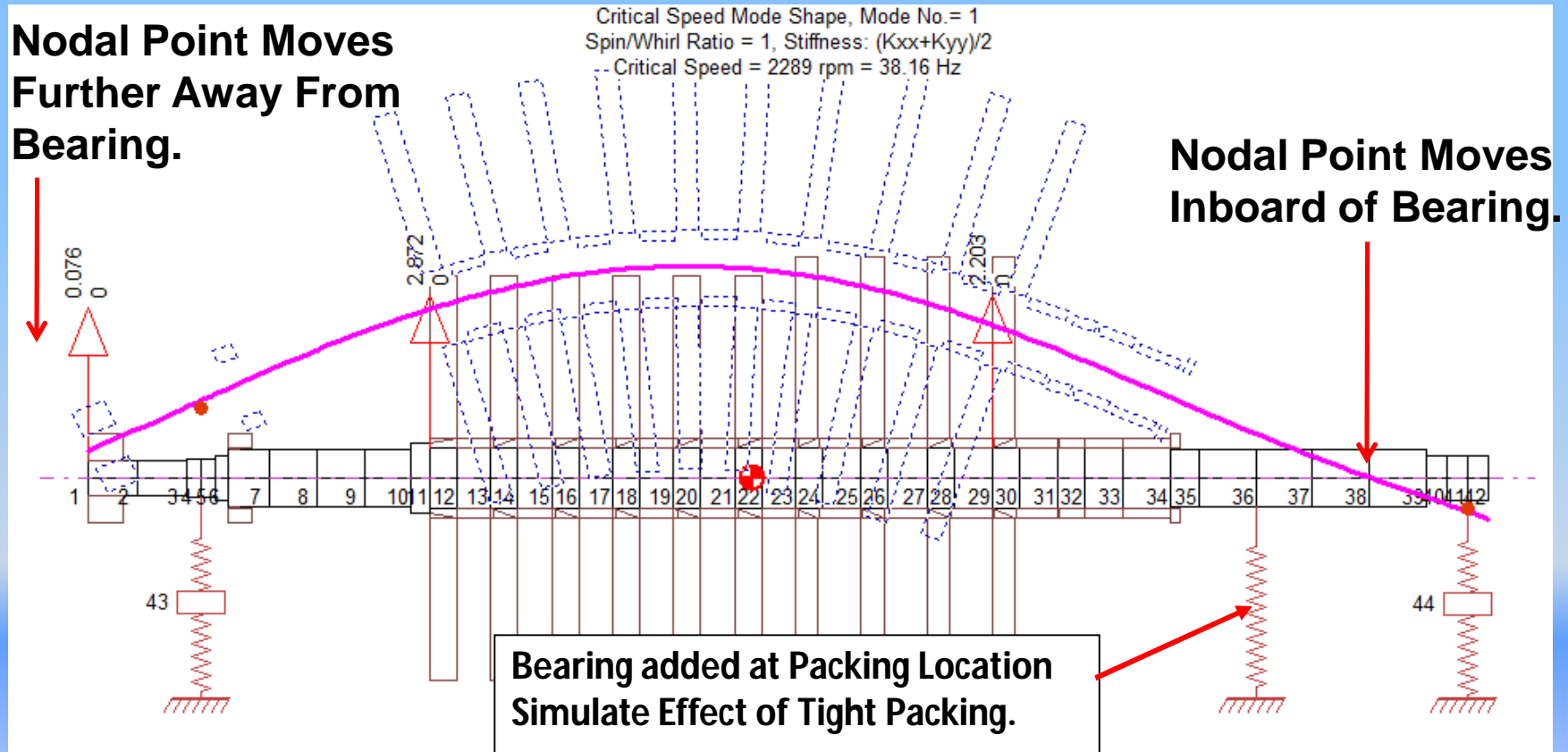


Figure 30. Blower Rotor Model with Bearing Added at the Packing Location to Simulate the Stiffening Effect of the Packing.

# Rotor Bearing Dynamic Analysis:

Based on the Vibration Analysis and Rotor Models:

Tight Packing Acting as a Bearing was predicted to shift the rotor 1<sup>st</sup> Critical to about 2400 RPM during spin up.



Animation of Damped 1<sup>st</sup> Forward Mode 2377 RPM

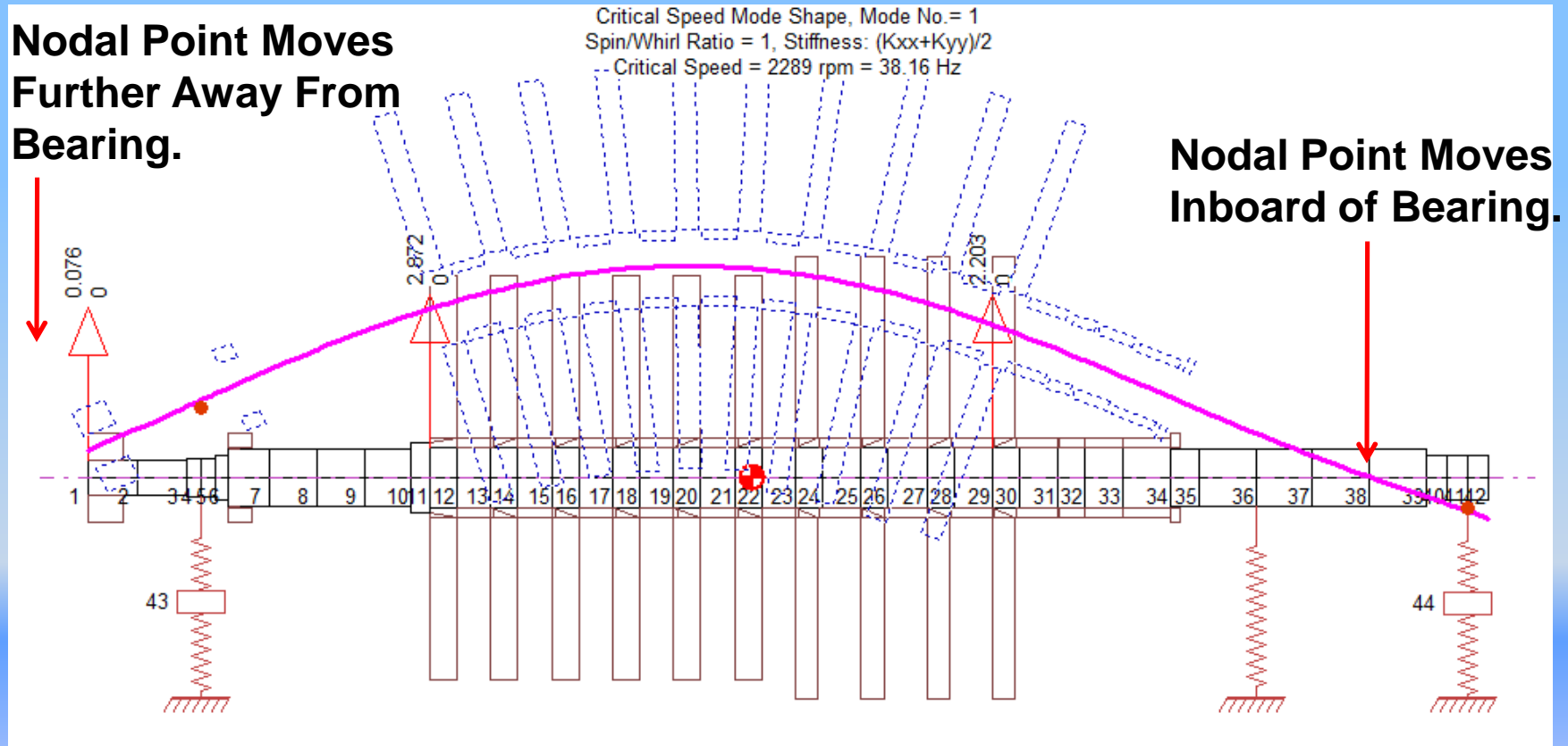
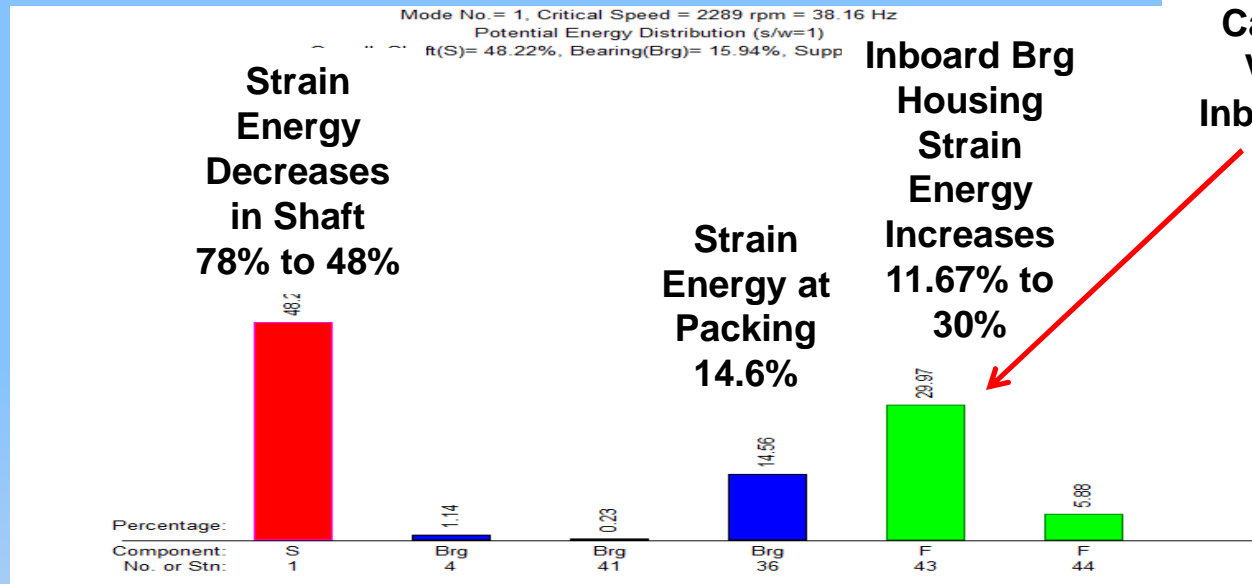


Figure 30. Blower Rotor Model with Bearing Added at the Packing Location to Simulate the Stiffening Effect of the Packing.

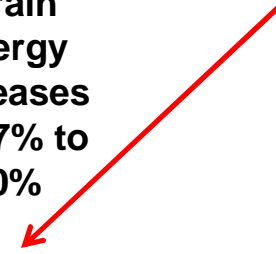
# Rotor Bearing Dynamic Analysis:

## Packing Rub Affect on Strain Energy:

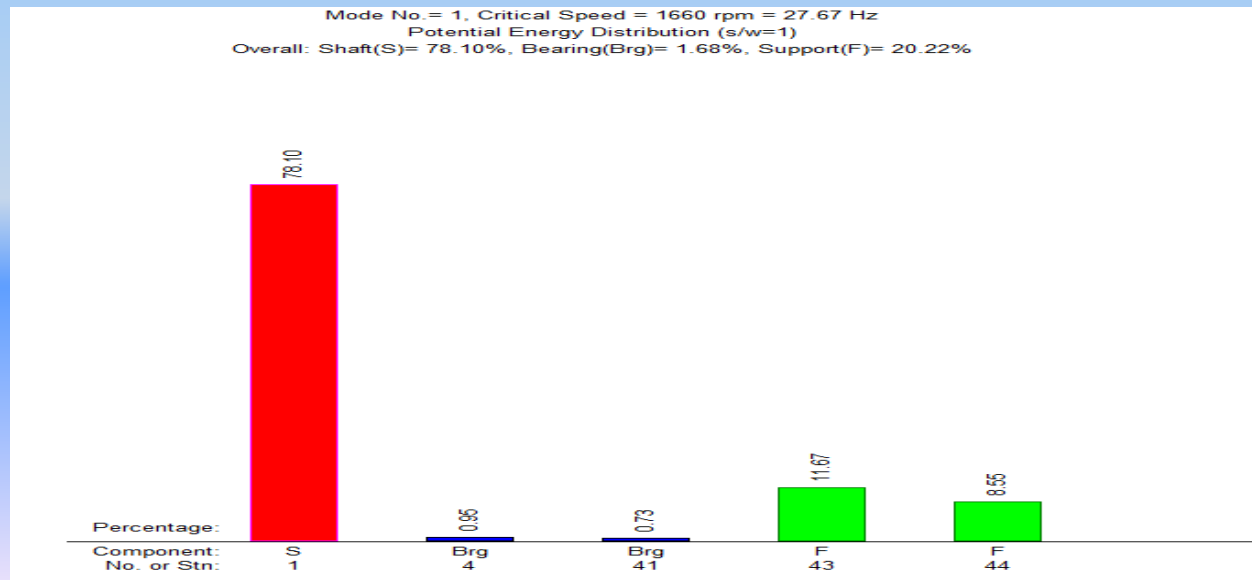
Model With Packing Acting as a 200,000 lb<sub>f</sub>/in Bearing.



Cause of High Vibration at Inboard Bearing.



Model With Packing Not Acting as a Bearing.



## Blower DB 19 3<sup>rd</sup> Startup – Frequency Spectra

This sequence of spectra at the blower inboard bearing housing horizontal show the rub-affect 1<sup>st</sup> critical speed moving higher in frequency.

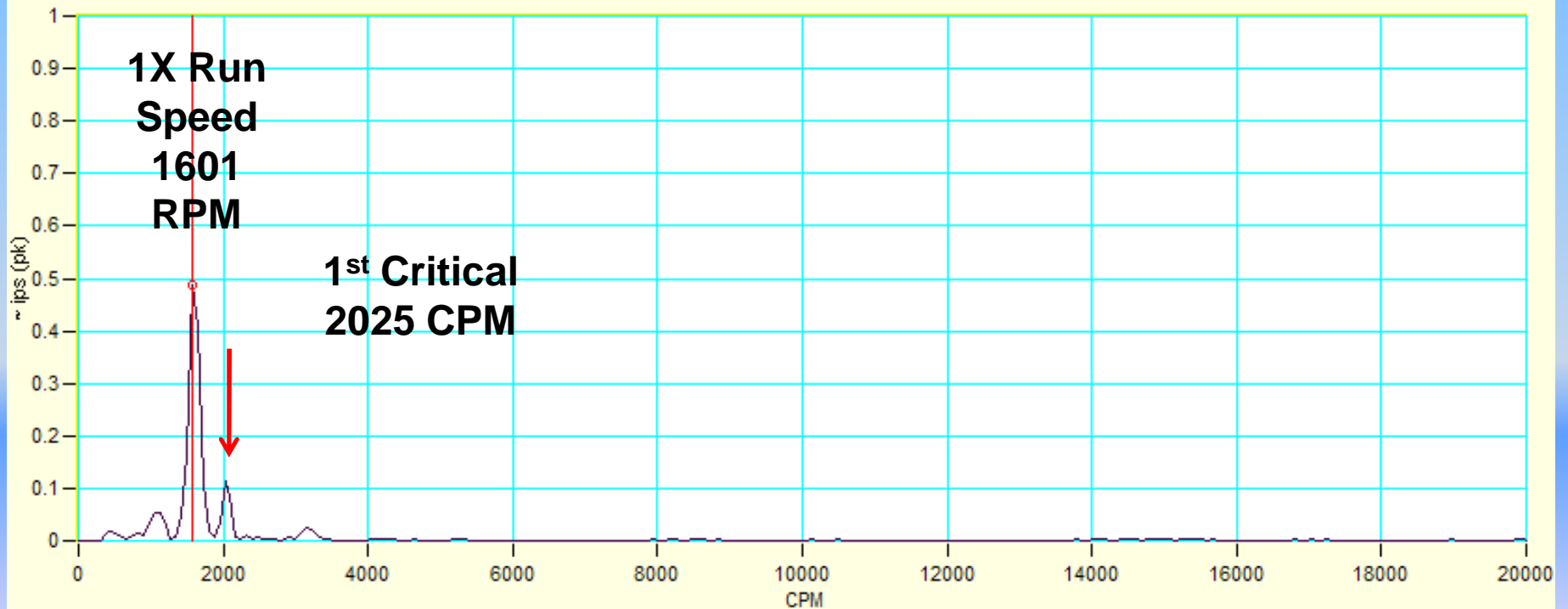
At rotor speed of 1601 RPM, the rotor rub affected 1<sup>st</sup> critical has shifted up to 2025 CPM.

Spectrum 09-Jul-2008 08:39:18 1601. RPM

60000. CPM, 800. Lines, Hanning

Your Company Name Your Plant Name Your Job Number Your Machine Name

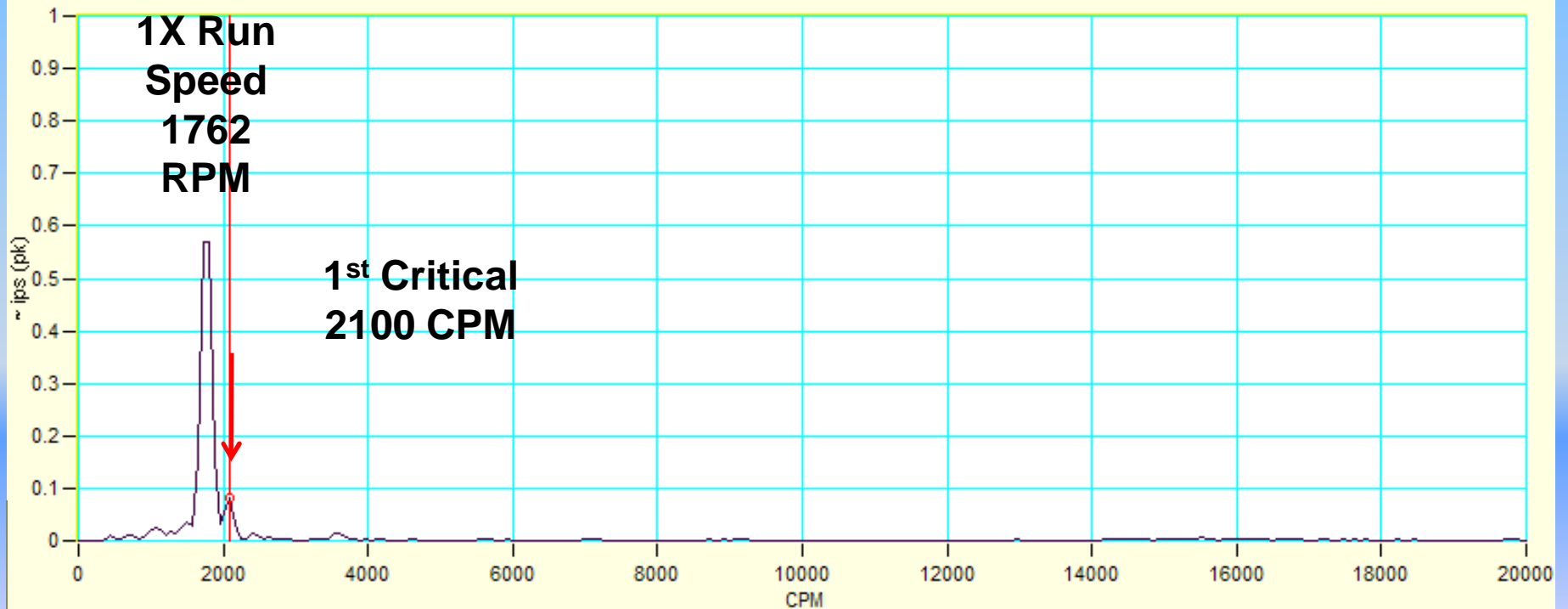
— Blower IB Hor Csr: 1575. CPM, 0.4878 ~ ips (pk) (OA: 0.5667)



# Blower DB 19 3<sup>rd</sup> Startup – Frequency Spectra

At rotor speed of 1762 RPM, the rotor rub affected 1<sup>st</sup> critical has shifted up to 2100 CPM.

Spectrum 09-Jul-2008 08:39:19 1762. RPM  
60000. CPM, 800. Lines, Hanning  
Your Company Name Your Plant Name Your Job Number Your Machine Name  
— Blower IB Hor Csr: 2100. CPM, .08275 ~ ips (pk) (OA: 0.6921)



# Blower DB 19 3<sup>rd</sup> Startup – Frequency Spectra

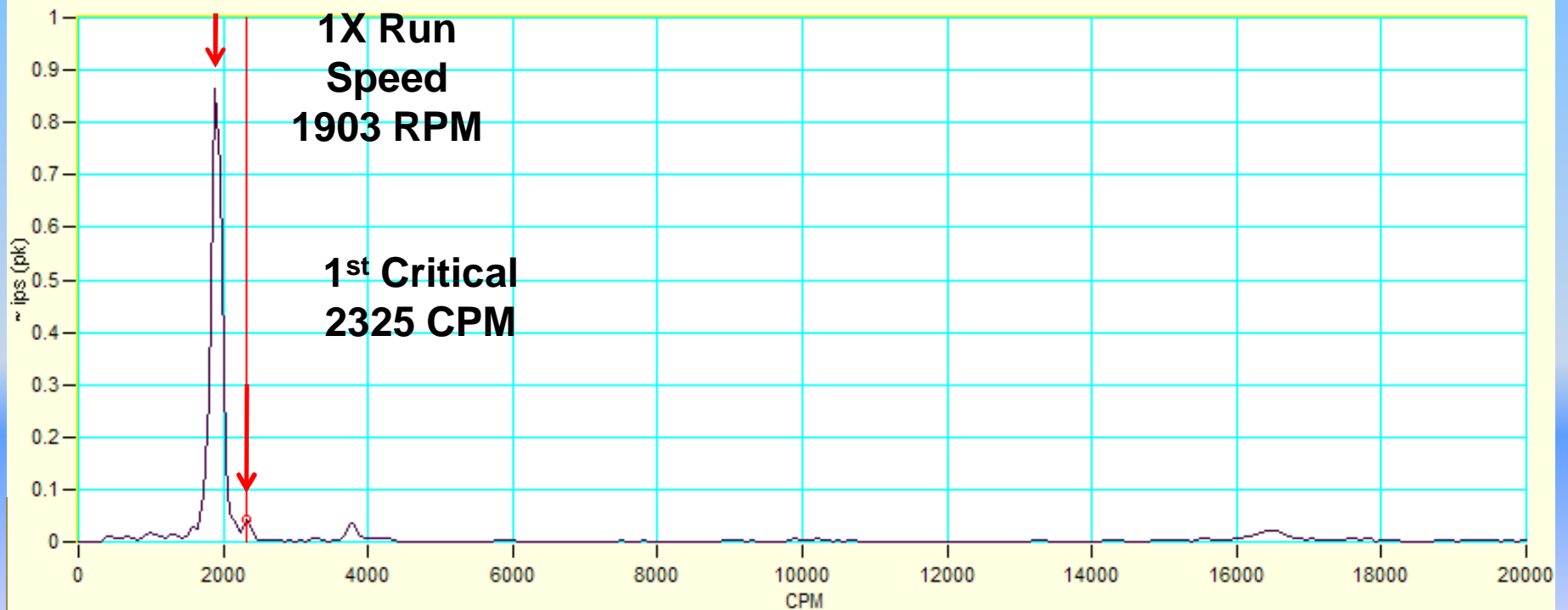
At rotor speed of 1903, the rotor 1<sup>st</sup> critical has shifted up to 2325 CPM.

Spectrum 09-Jul-2008 08:39:20 1903. RPM

60000. CPM, 800. Lines, Hanning

Your Company Name Your Plant Name Your Job Number Your Machine Name

— Blower IB Hor Csr: 2325. CPM, .04169 ~ ips (pk) (OA: 0.9606)



## Blower DB 19 3<sup>rd</sup> Startup – Frequency Spectra

Maximum amplitude of vibration occurred at rotor speed of 2400 RPM when the rotor 1X and rub affected 1<sup>st</sup> critical coincide.

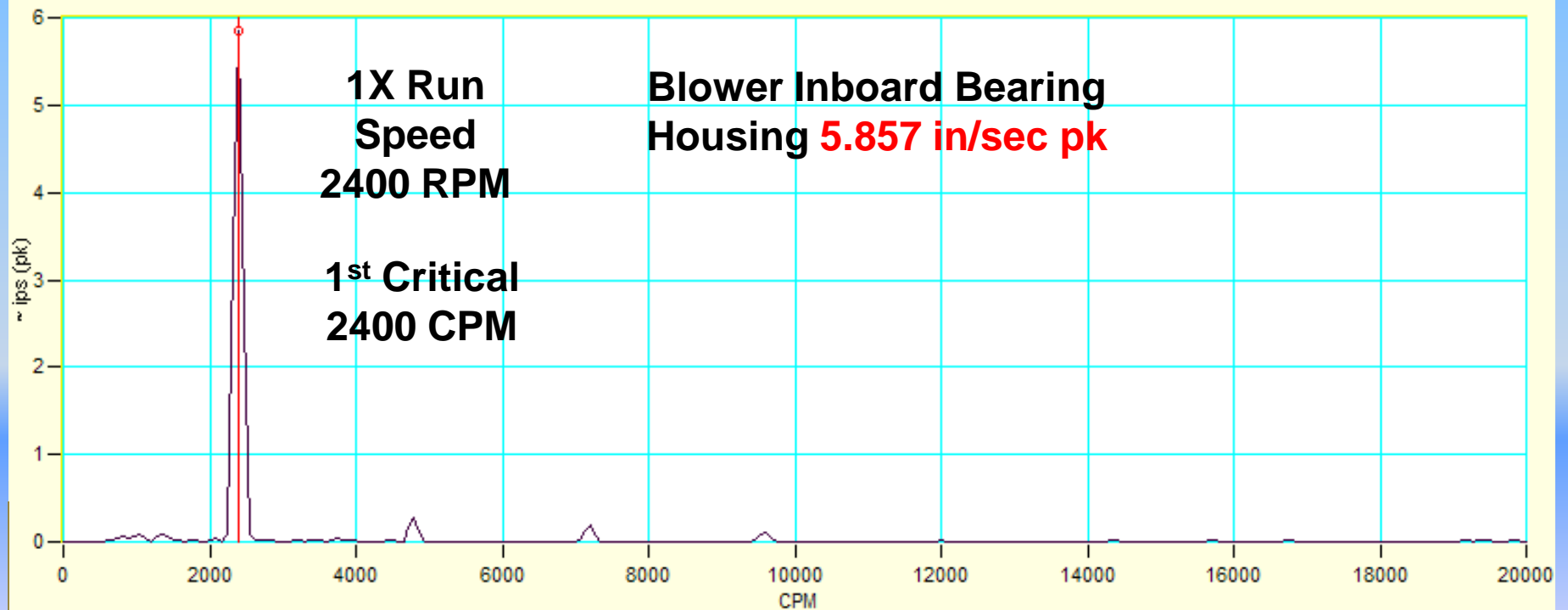
The data indicated that 2400 was the maximum frequency of the rub affected 1<sup>st</sup> critical.

Spectrum 09-Jul-2008 08:39:27 2401. RPM

60000. CPM, 800. Lines, Hanning

Your Company Name Your Plant Name Your Job Number Your Machine Name

— Blower IB Hor Csr: 2400. CPM, 5.857 ~ ips (pk) (OA: 5.902)





## Blower DB 19 3<sup>rd</sup> Startup – Frequency Spectra

At rotor speed of 2778 RPM, the rotor 1<sup>st</sup> critical speed drops slightly to 2325 CPM.

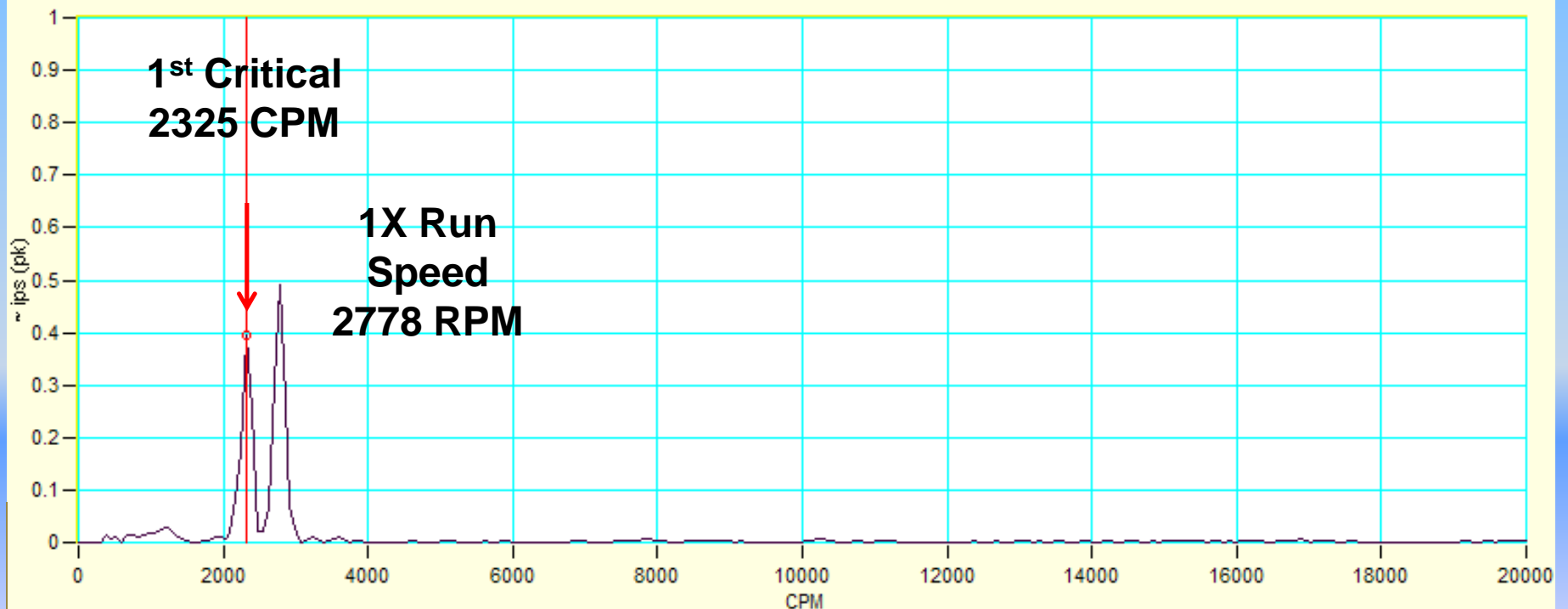
The bearing housing amplitude at 1X run speed has dropped from 5.857 in/sec pk to 0.50 in/sec pk.

Spectrum 09-Jul-2008 08:39:29 2778. RPM

60000. CPM, 800. Lines, Hanning

Your Company Name Your Plant Name Your Job Number Your Machine Name

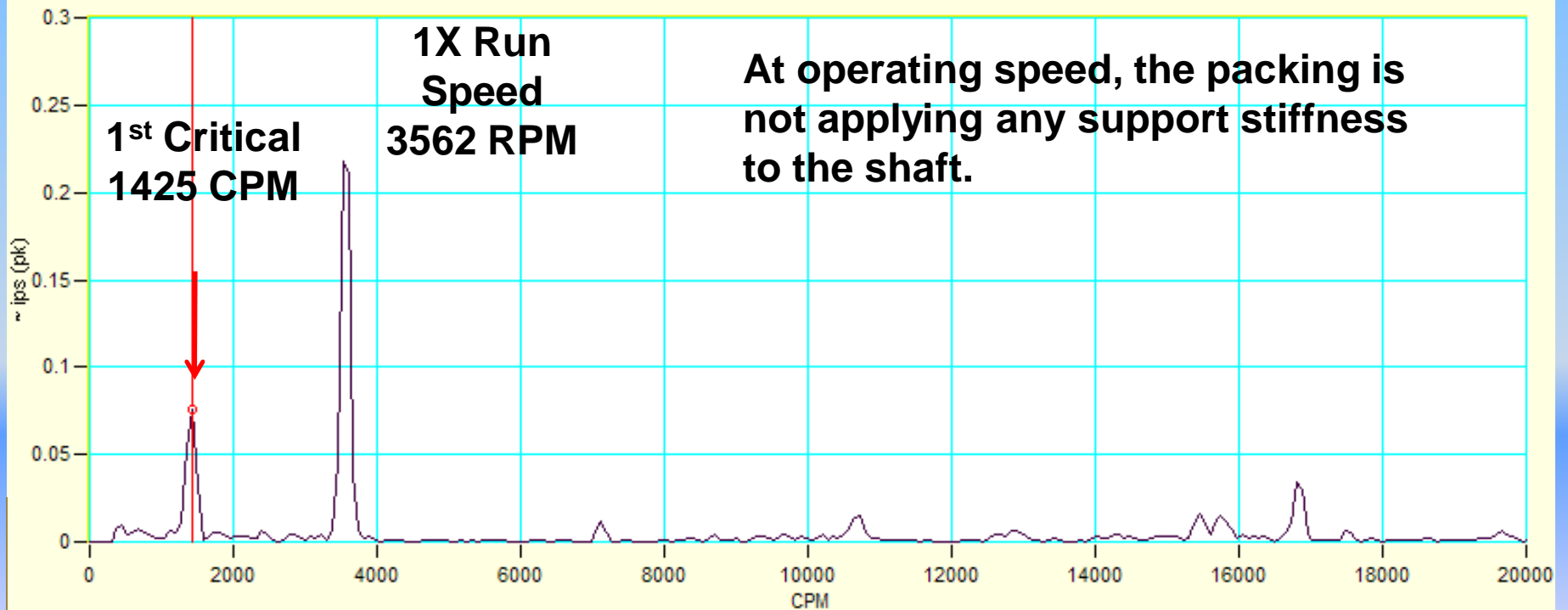
— Blower IB Hor Csr: 2325. CPM, 0.3965 ~ ips (pk) (OA: 0.6765)



# Blower DB 19 3<sup>rd</sup> Startup – Frequency Spectra

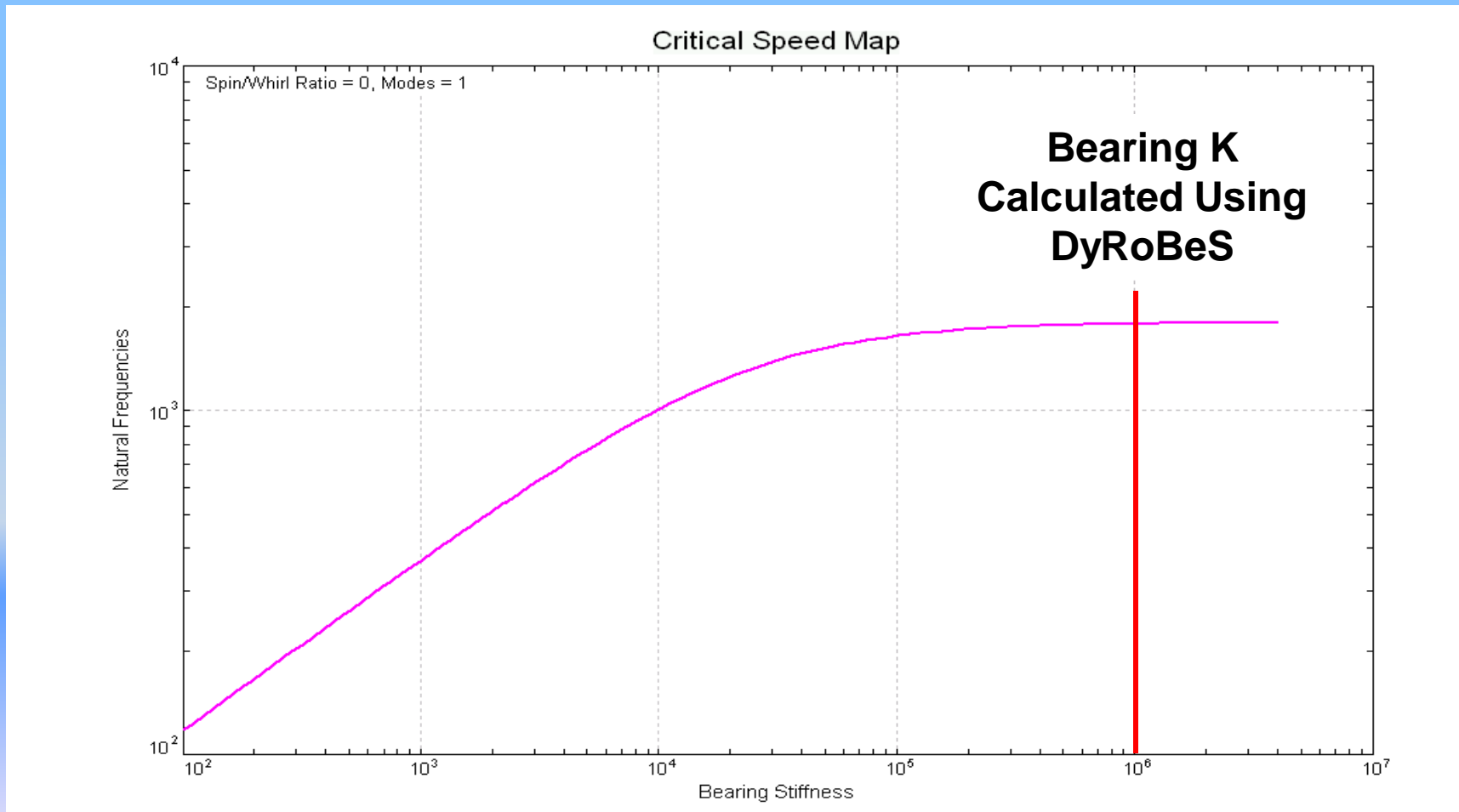
At running speed, the 1<sup>st</sup> critical measured about 1425 CPM.

Spectrum 09-Jul-2008 08:40:01 3562. RPM  
60000. CPM, 800. Lines, Hanning  
Your Company Name Your Plant Name Your Job Number Your Machine Name  
— Blower IB Hor Csr: 1425. CPM, .07630 ~ ips (pk) (OA: 0.2753)



## Blower DB 19 3<sup>rd</sup> Startup – Critical Speed Map

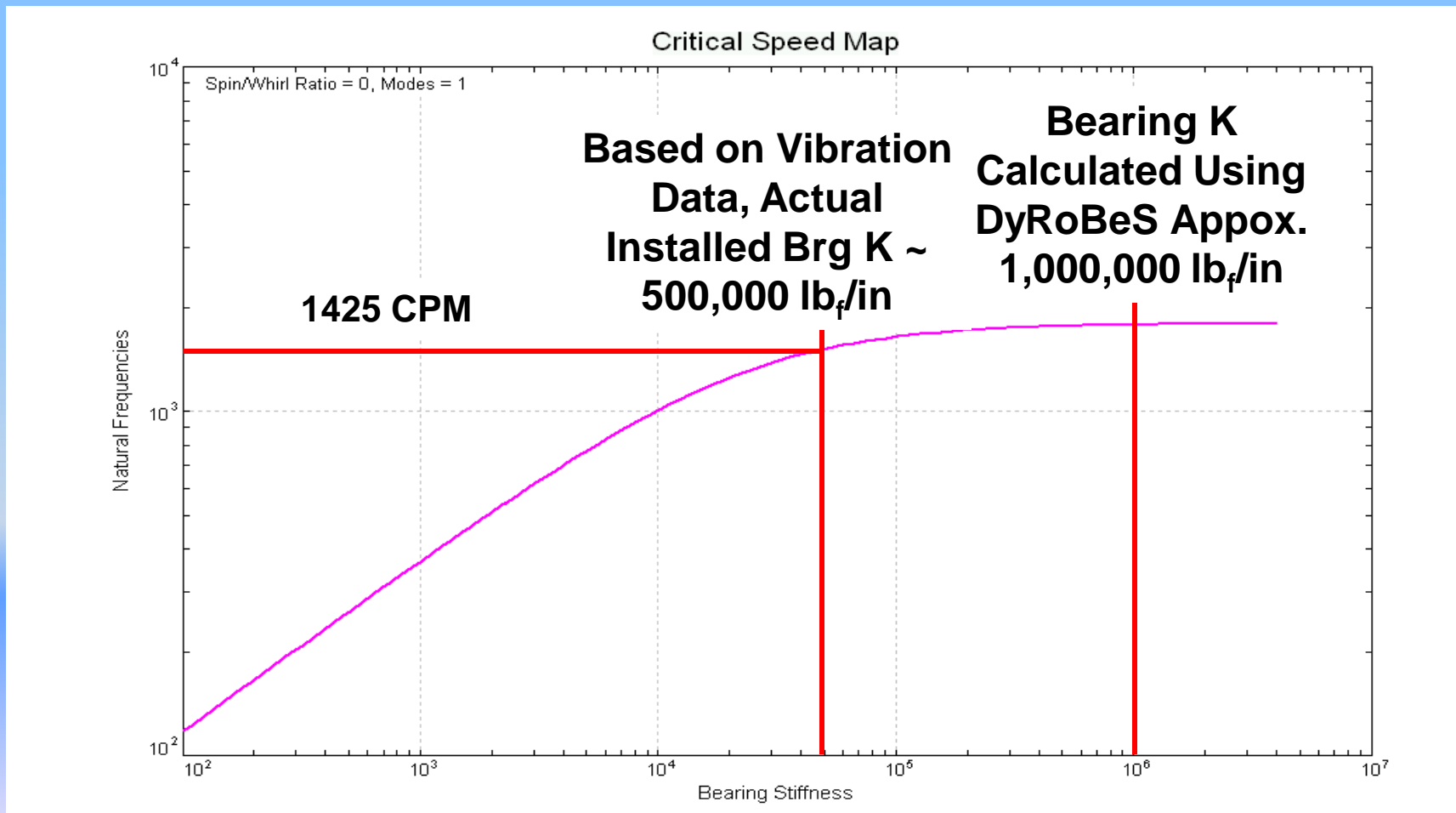
The Blower 1<sup>st</sup> Critical Speed Calculated to 1660 (Packing adding no support stiffness). The bearing stiffness calculated using DyRoBeS was approx 1,000,000 lb<sub>f</sub>/in.



## Blower DB 19 3<sup>rd</sup> Startup – Critical Speed Map

It would be found at disassembly that the shaft bearing fits were undersize and the bearing housing bores were oversize thus providing lower support stiffness than calculated (~500,000 lb<sub>f</sub>/in).

The model represented a new condition blower.



## Shaft Sealing:

Blower OEM Offered Three Options for Shaft Sealing.

**Packing:** Lowest Cost but tight packing rubbing the shaft causes heat, thermal bowing of the shaft and as calculations showed can act as a bearing raising the 1<sup>st</sup> critical speed.

Packing Box.

Carbon Ring

Mechanical Seal

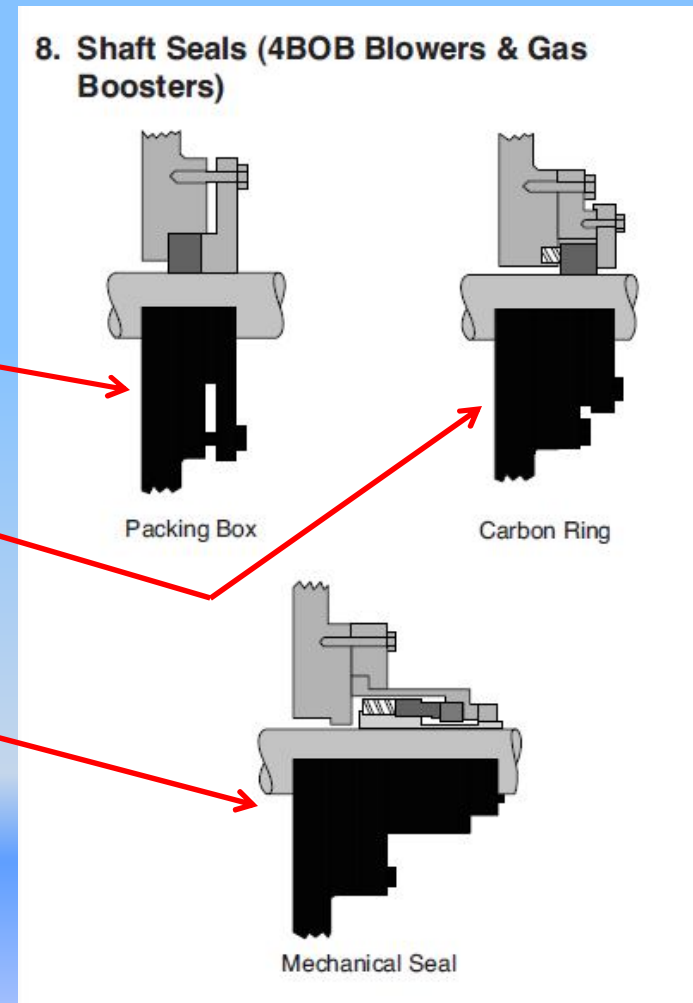


Figure 2. Seal Options Offered by Blower OEM.

## Thermal Growth Measurements:

Ludeca Permalign Laser System<sup>Ref 7</sup> was Installed on Blowers DB 18 and DB 19 To Measure Thermal Growth.

Blower DB 19 was Measured first.

The Blower was Shutdown and Allowed to Cool Overnight.

Permalign Was Installed After Shutdown.



Figure 33. Laser Alignment Equipment, Ludeca Permalign,<sup>Ref 7</sup> Installed on the Motor and Blower.



## Thermal Growth Measurements:

Ludeca Permalign Laser System <sup>Ref 7</sup> was Installed on Blowers DB 18 and DB 19 To Measure Thermal Growth.

Blower DB 19 was Measured first.

The Blower was Shutdown and Allowed to Cool Overnight.

Permalign Was Installed After Shutdown.

At Startup, Permalign Laser and Accelerometers Moved due to High Vibration.



Figure 34. Permalign Setup on Blower and Motor. <sup>Ref 7</sup>

Blower was Shutdown, Adjustments made & Blower restarted.

## Thermal Growth Measurements:

The Blower Was Started Two More times then Shutdown and Cooled Overnight Back to Ambient.

Cool Down Data Indicated Severe Pipe Stain in the Horizontal Direction.

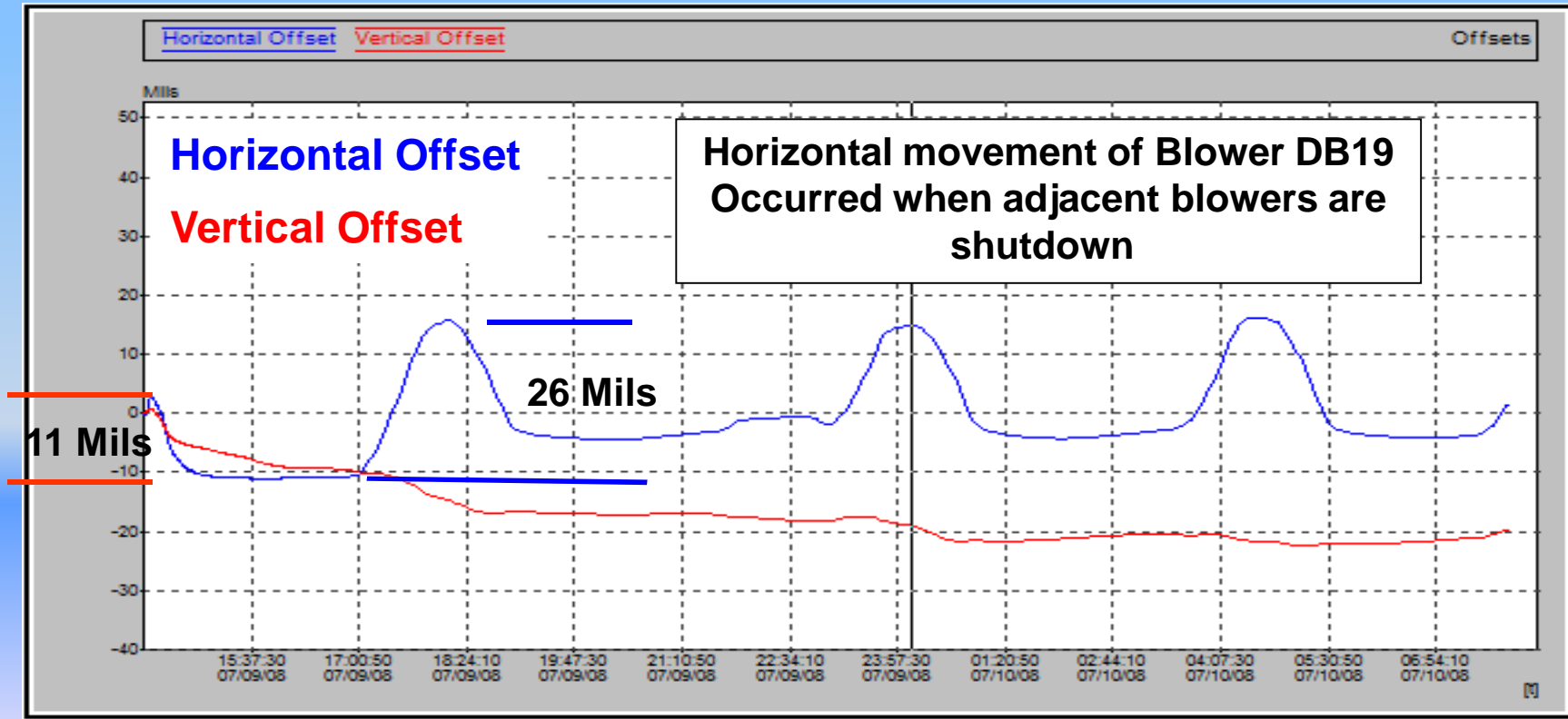


Figure 35. Blower DB19 Cool Down Alignment Data.



## Thermal Growth Measurements:

The Blower Was Being Pushed/Pulled by Pipe Thermal Growth.

The Data in Figure 35 Began Just Minutes Before DB 19 Shutdown. Blue Line Jumps When the Other Two Blowers Shutdown.

The Pipe Strain Forces the Blowers Out of Alignment as Much as 20 mils during Each Blower Startup.

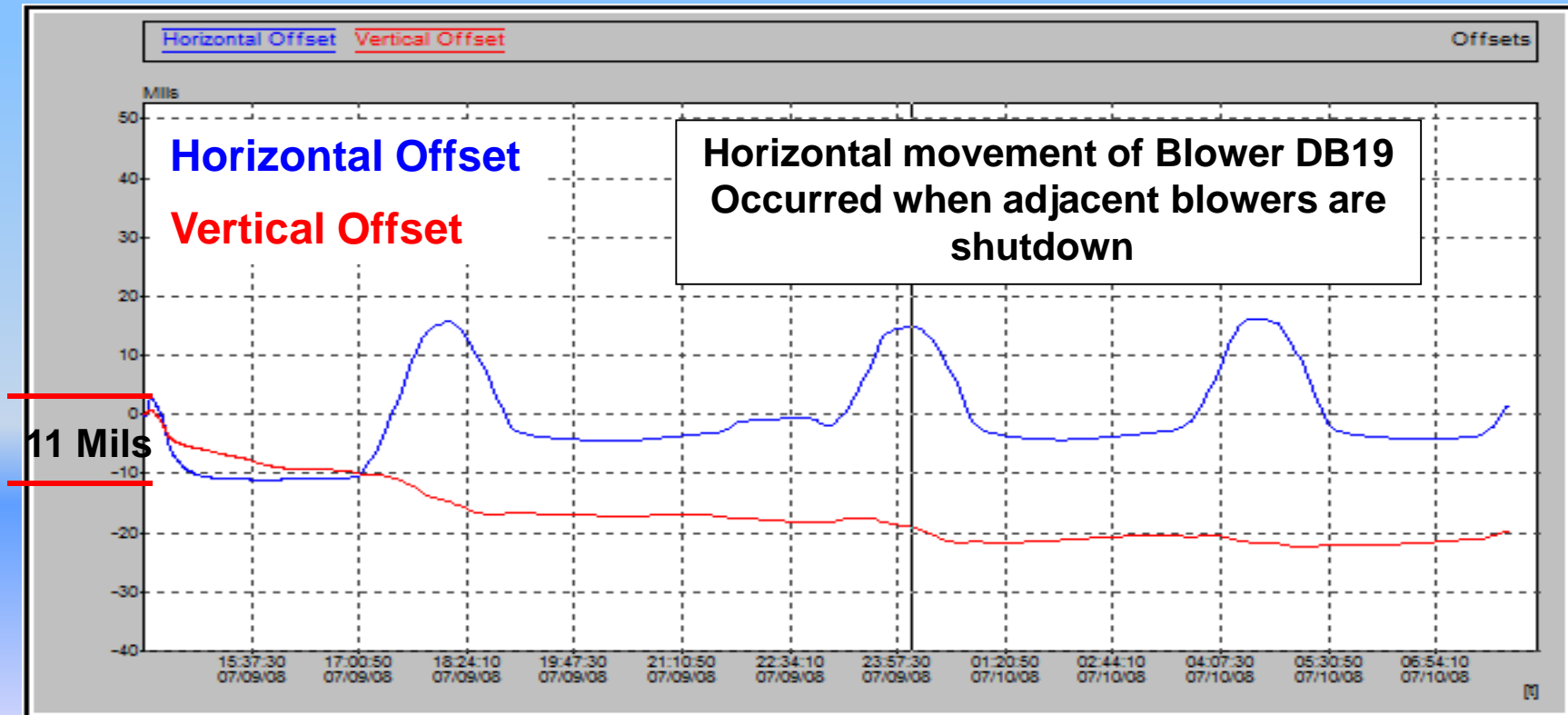


Figure 35. Blower DB19 Cool Down Alignment Data.

## Thermal Growth Measurements:

DB 18 Blower:

Permalign Was Installed on DB18 After DB19 Was Re-Aligned and Started. DB18 Was Started and Allowed to Reach Operating Temperature for About 5 Hours, Then Shutdown and Allowed to Cool Overnight.

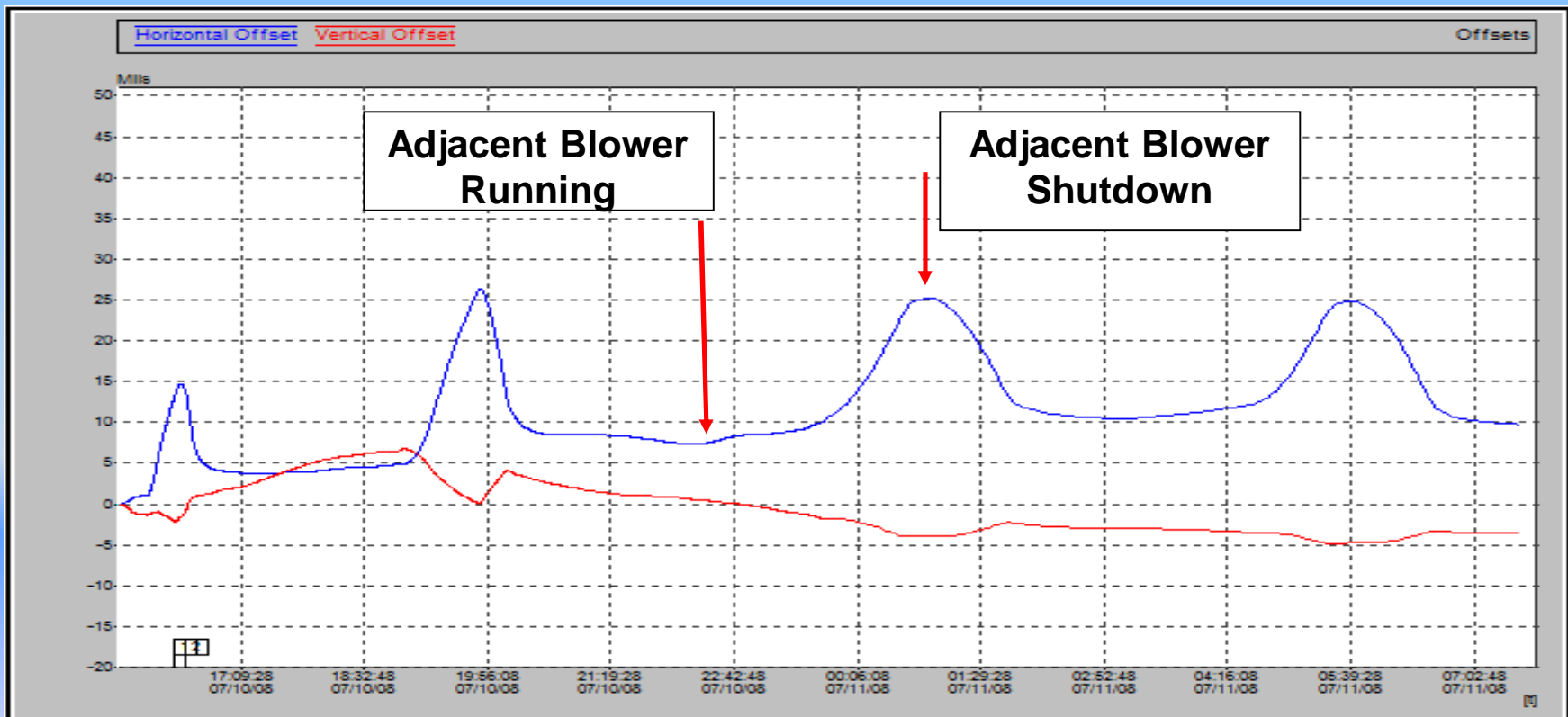


Figure 3. Permalign Data File for DB18 Blower. Horizontal Movement Occurs When Adjacent Blower Shutdown.

## Thermal Growth Measurements:

DB 18 Blower:

Results were Similar to DB19. Blower Was Pushed/Pulled When Other Blowers Shutdown.

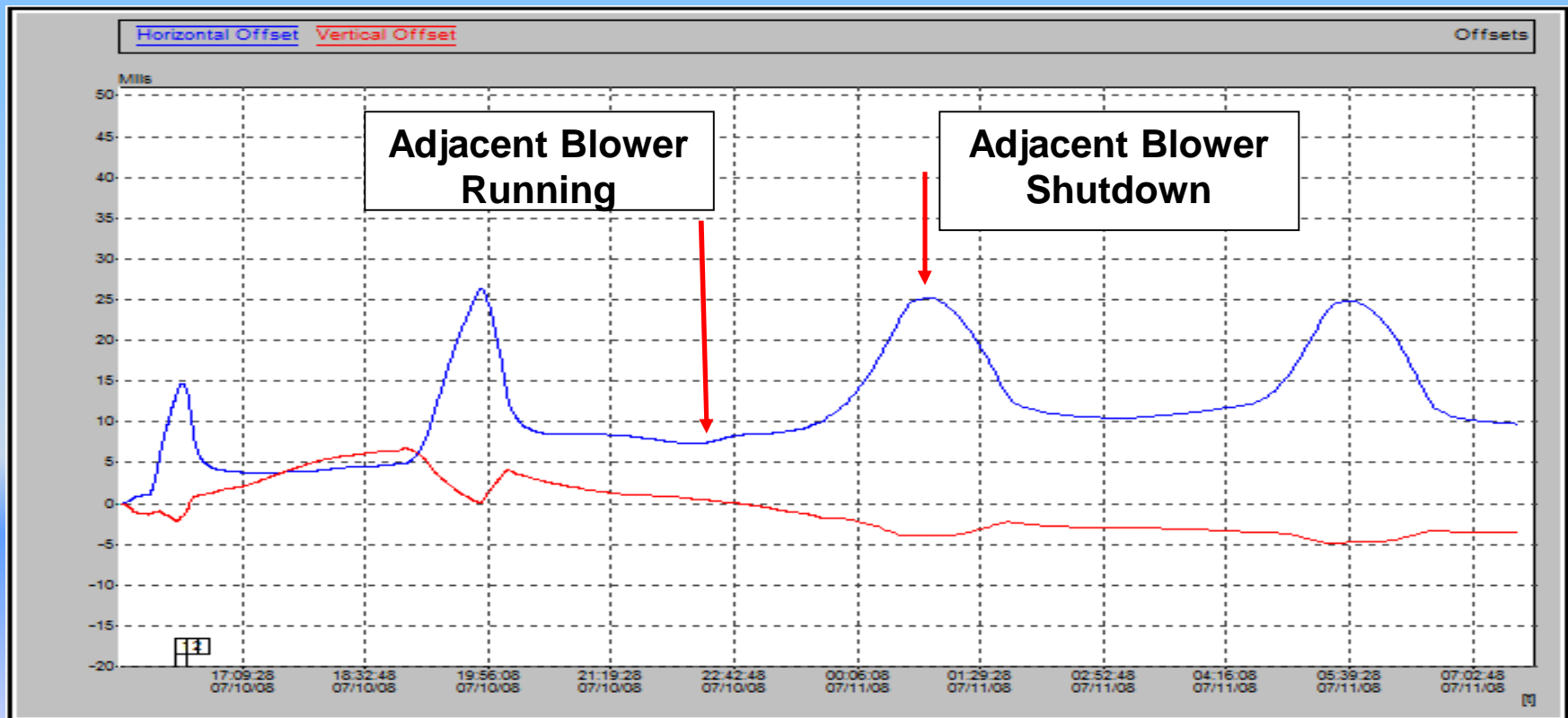


Figure 3. Permalign Data File for DB18 Blower. Horizontal Movement Occurs When Adjacent Blower Shutdown.

## **Blower Repair:**

**DB19 Sent to Blower Authorized Repair Facility.**

**Blower to be:**

- **Disassembled**
- **Inspected**
- **Repaired**
- **Balanced**
- **Re-assembled**
- **Run Tested.**



**Figure 37. Blower DB 19 As Received at Factory  
Authorized Repair Facility.**

## Blower Disassembly:

The shaft packing area had over 30 mils wear. Blower OEM allows up to 30 mils shaft wear.

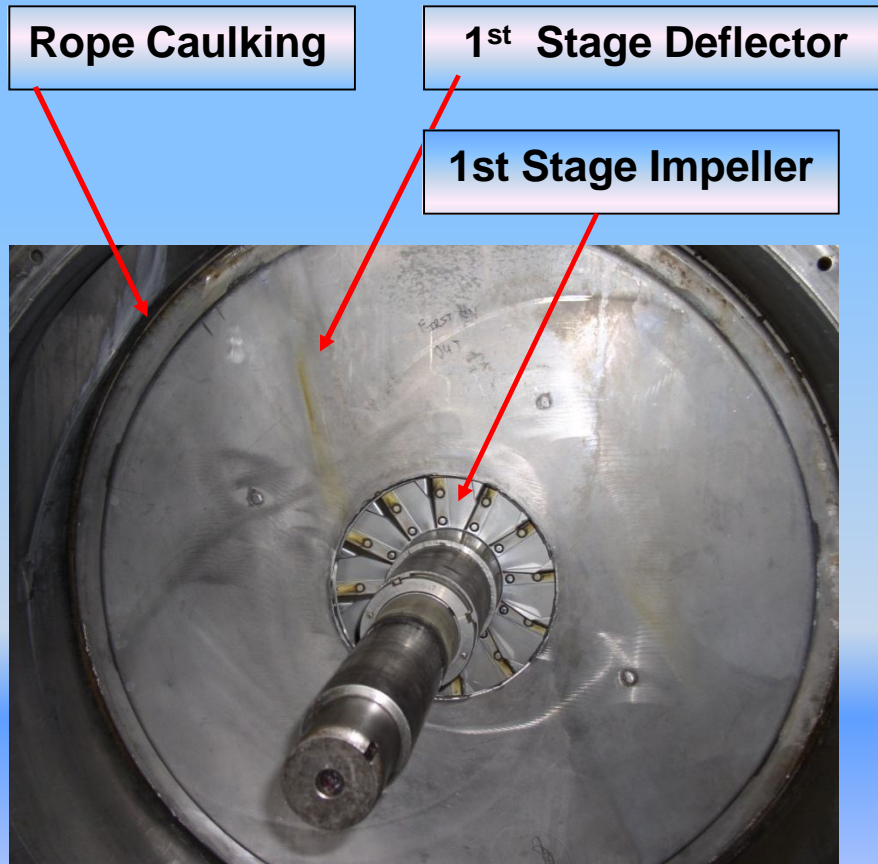


Figure 38. 1<sup>st</sup> Stage Impeller With Deflector and Rope Caulking at the OD.

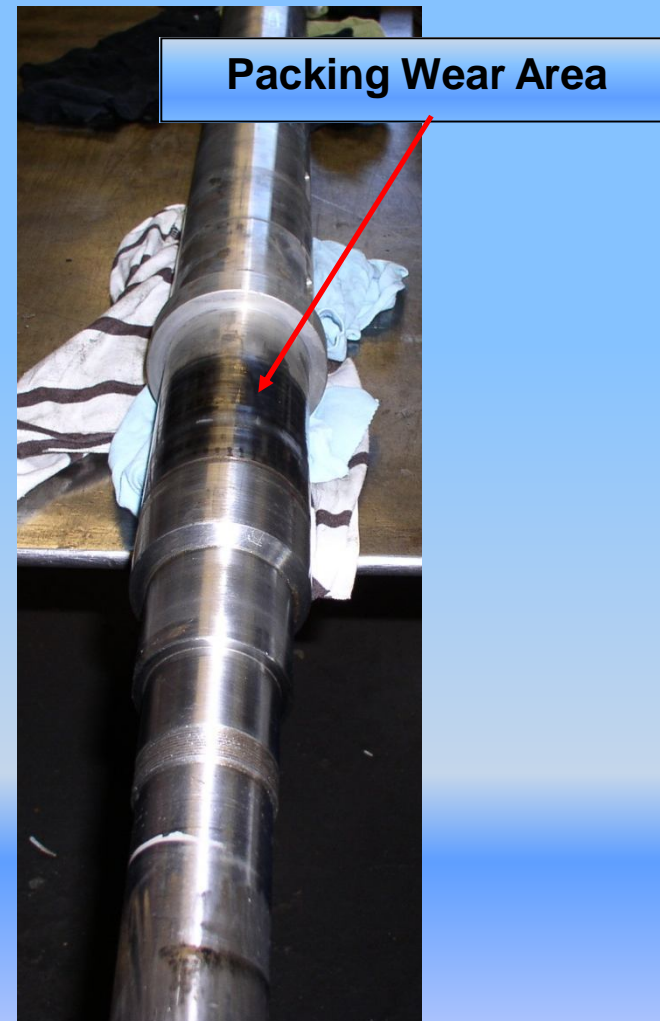


Figure 39. Shaft Packing Area Discolored by Packing Rubbing



## Blower Disassembly:

The impeller hub fit to shaft to have metal-to-metal fit (line-to-line).

Fretting in Bore –  
Indication Hub Loose  
Shaft Fit.

Heating of the Hub  
During Previous  
Assembly.

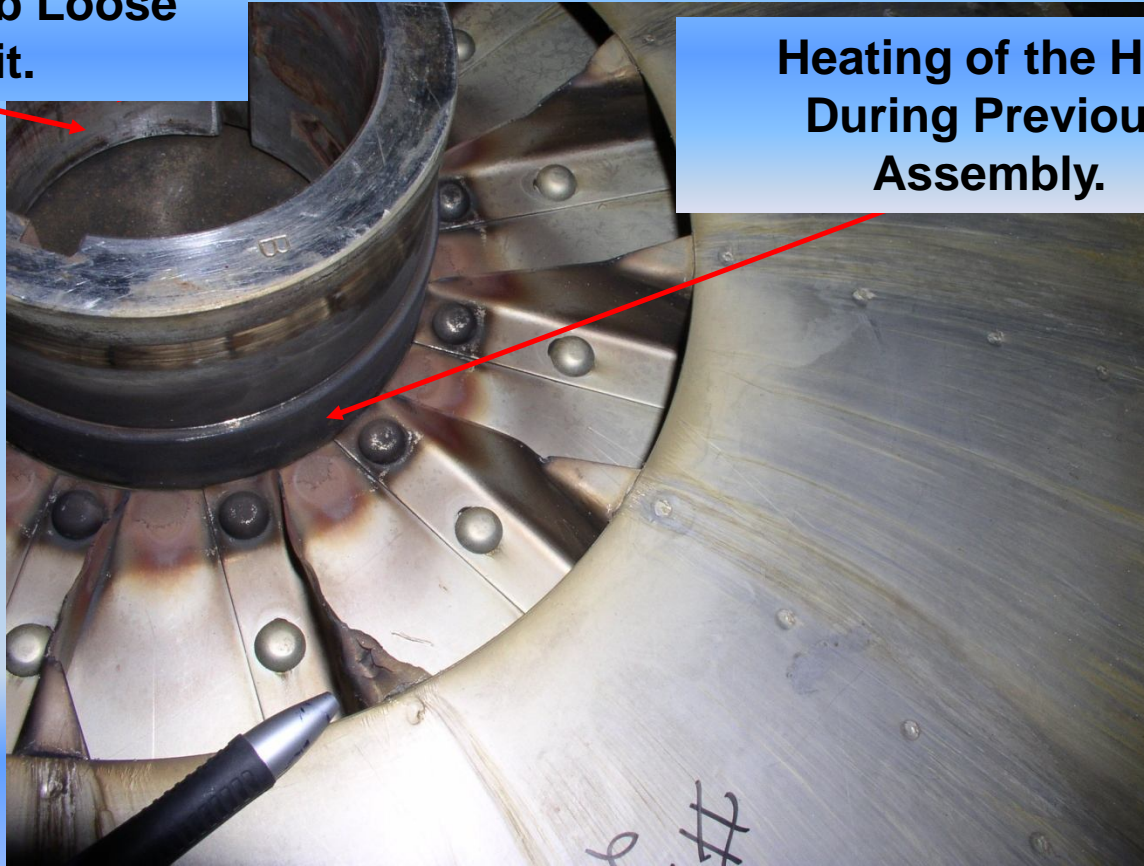


Figure 40. Impeller #2 Blade Damage. Note Heating of the Hub Which Occurred During Previous Assembly.

## Blower Disassembly:

Shaft keyseats are machined in 180 degree alternating pattern.

Keys Transfer Torque to Each Impeller Hub.

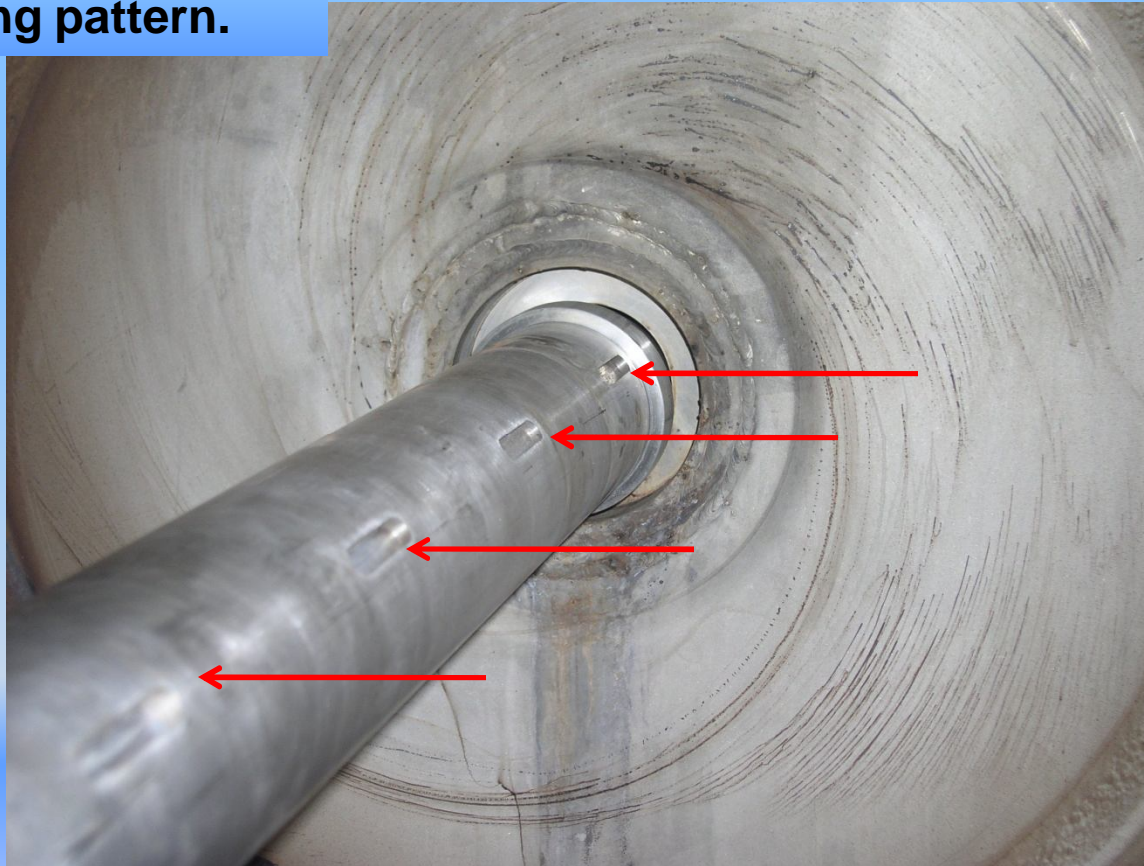
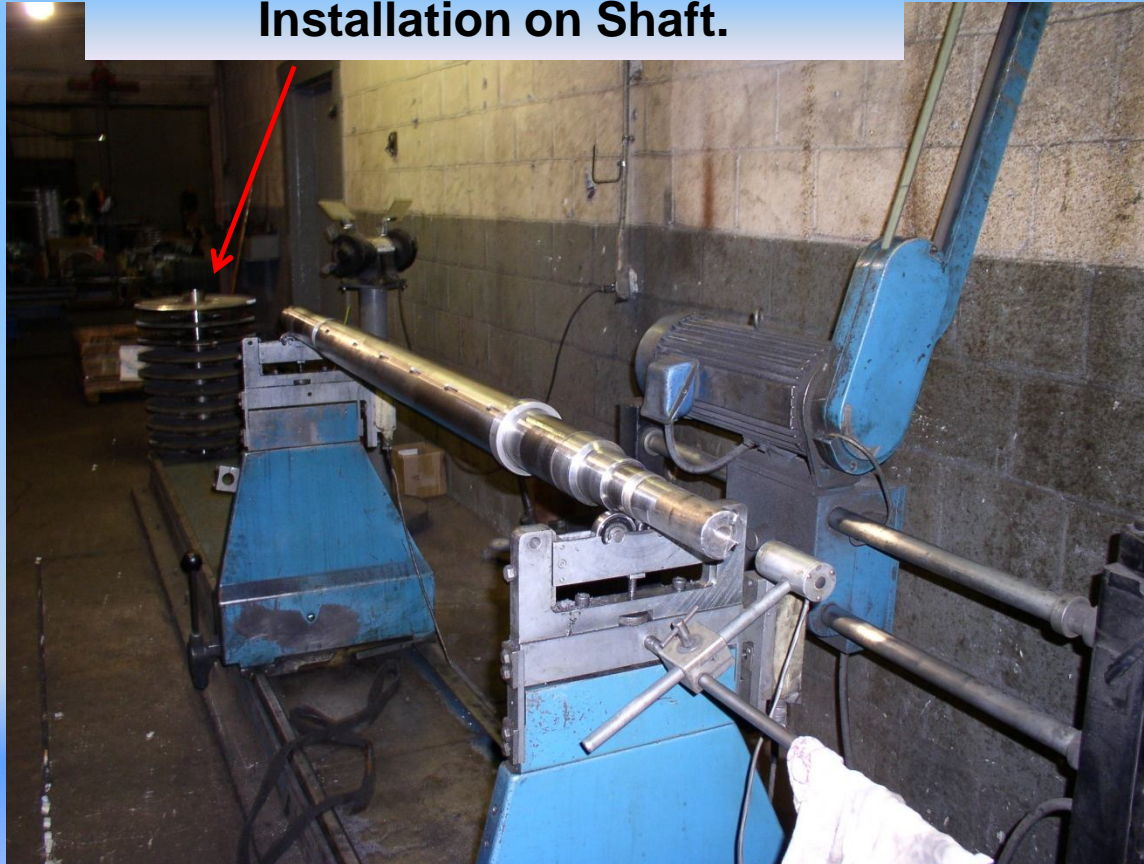


Figure 41. Keys Transfer Torque to the Impeller Hubs.

## Blower Reassembly:

**Bare Shaft in Balancing Machine**

**Impellers Stacked, Ready for  
Installation on Shaft.**

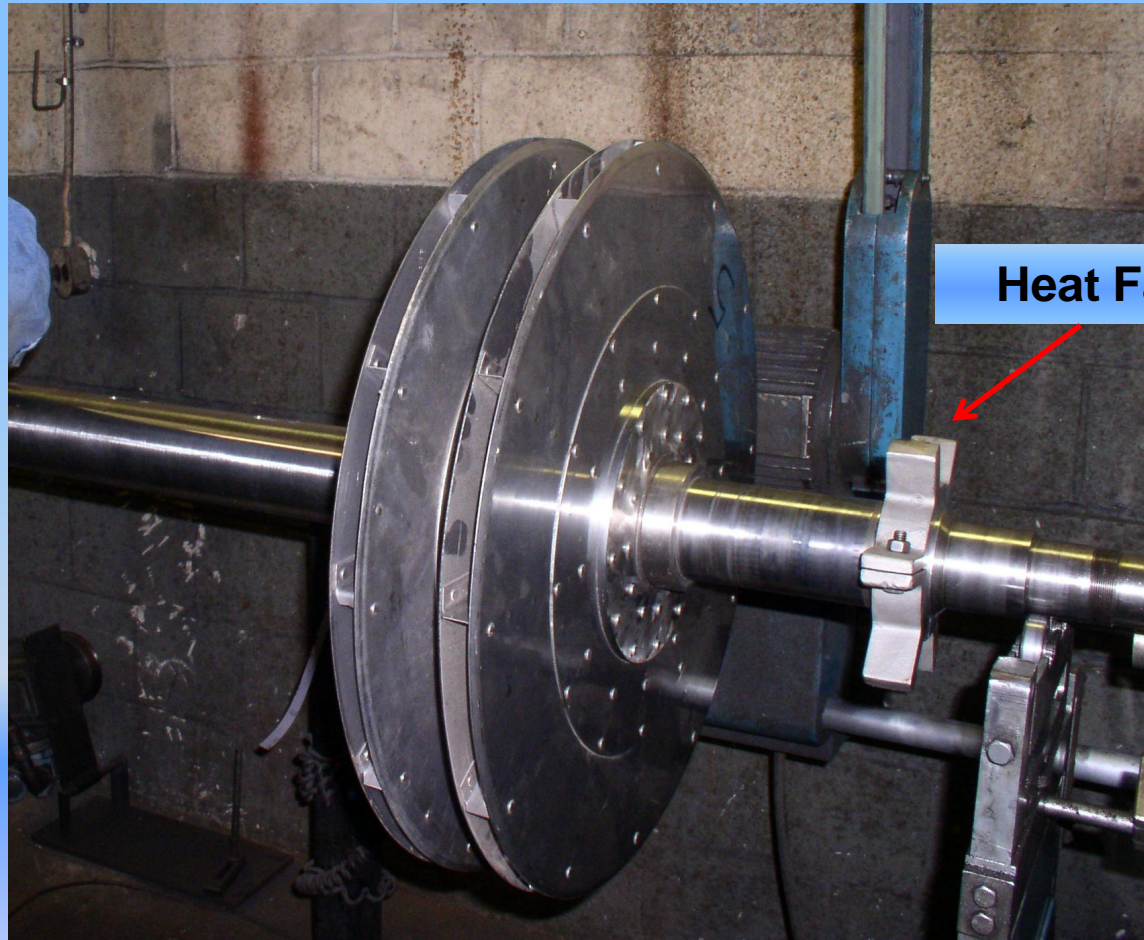


**Figure 42. Bare Shaft in Balancing Machine. Impellers Are Stacked at the End of the Balance Machine.**



## Blower Reassembly:

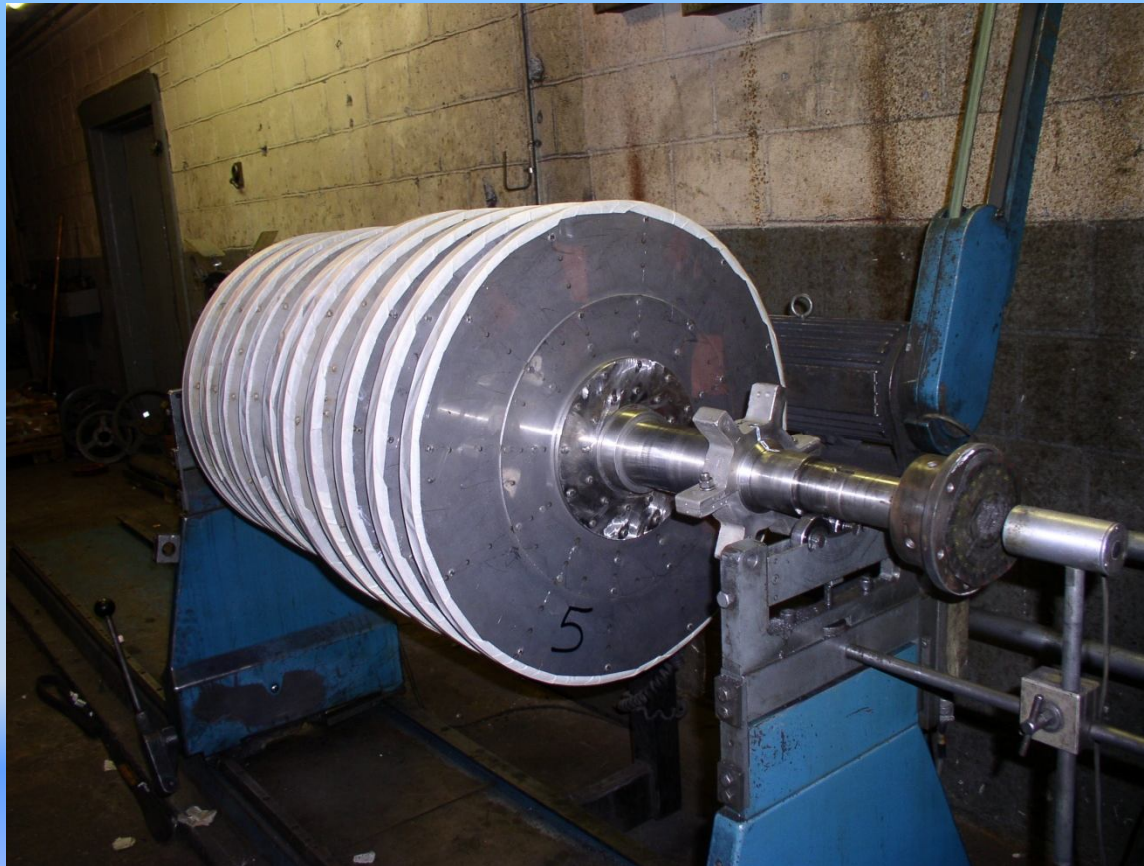
**First Two Impellers and Heat Fan  
Installed on Shaft.**



**Figure 43. Impellers Being Installed on Shaft.**

## Blower Reassembly:

**Rotor Stacked – Impellers Taped to Prevent Pumping Air (Reduce Wind Resistance).**



**Figure 44. Rotor Stacked, Impellers Taped For Wind Resistance For Balancing.**

# Blower Balance Report

## Balance Report

Date: 08-02-07

Balance speed: 425 RPM

Balanced for Operating speed 3600 RPM

Half Key Weight:

Near side = 0 gm

Far side = 0 gm

Initial Unbalance

Near Vibration:

4.74 mils at 276 degrees

Near Velocity:

0.105 in/sec

Near Unbalance:

10.1294 oz-in or 287.1685 gm-in

Far Vibration:

4.19 mils at 289 degrees

Far Velocity:

0.093 in/sec

Far Unbalance:

12.3922 oz-in or 351.3191 gm-in

Finish Unbalance

Near Vibration:

0.48 mils at 8 degrees

Near Velocity:

0.011 in/sec

Near Unbalance:

0.8950 oz-in or 25.3737 gm-in

Far Vibration:

0.38 mils at 188 degrees

Far Velocity:

0.009 in/sec

Far Unbalance:

0.7970 oz-in or 22.5955 gm-in

Weight

Near: 26.11 grams or 0.92 ounces

Far: 35.13 grams or 1.24 ounces

ISO-1940 G 2.5 recommended maximum unbalance: 54.4733 gm-in

Near side: 27.2367 gm-in Far side: 27.2367 gm-in

weight 460 lbs shaft/10 impellers/spacers

]

(c) 2007 Dynamics Research Corp.

This unit has been precision balanced using our State-of-the-Art Dynamics Research Corp. COMPUTER Balancer, United STATES PATENT NO. 5,627,762 + 5,412,583, AND OTHER PATENTS PENDING.

Balance Report of Incoming and Final Balance.

Balance correction Near Plane.

Balance correction Far Plane.

# Blower Balance Report

Data from Balance Report used to calculate the unbalance forces at each bearing.

Serial No. 712311455

	Near Plane		Ubal Force lbf	Far Plane		Ubal Force lbf
Initial Unbalance	10.129	oz-in	233.017	12.392	oz-in	285.070
Finish Balance	0.895	oz-in	20.589	0.797	oz-in	18.334
Force Reduction			212.428			266.736
Percent Reduction			91.16%			93.57%
	grams	oz		grams	oz	
Correction Weight	0.89	0.03		2.26	0.08	
ISO G 1.0	19.184	gr-in	Near Side	9.592	Far Side	gr-in
	0.675	oz-in		0.338	oz-in	
	15.539	lbf		7.770	lbf	
Rotor Weight	450	lbf				



# Blower Balance Report

Data from Balance Report used to calculate the unbalance forces at each bearing.

Serial No. 712311455

	Near Plane		Ubal Force lbf	Far Plane		Ubal Force lbf
Initial Unbalance	10.129	oz-in	233.017	12.392	oz-in	285.070
Finish Balance	0.895	oz-in	20.589	0.797	oz-in	18.334
Force Reduction			212.428			266.736
Percent Reduction			91.16%			93.57%
	grams	oz		grams	oz	
Correction Weight	0.89	0.03		2.26	0.08	
ISO G 1.0	19.184	gr-in	Near Side	9.592	Far Side	gr-in
	0.675	oz-in		0.338	oz-in	
	15.539	lbf		7.770	lbf	
Rotor Weight	450	lbf				

## Rotor Inspection

### Rotor and Bearing Housing Inspection:

**Shaft was replaced due to undersized fits.**

Drive End Shaft Bearing Fit	<b>Undersized 0.0062 inch</b>
Drive End Shaft Housing Bore	<b>Oversized</b> (no measurement on report)
Non Drive End Shaft Bearing Fit	<b>Undersized 0.0064 inch</b>
Non Drive End Housing Bore	<b>Over sized</b> (no measurement on report)
Max Shaft Wear @ Packing 0.030 in	<b>Undersized</b> (no measurement on report)
Impeller shaft fits	<b>0.010 inch variation in fit diameters.</b>

## Conclusions:

- ❑ **The design of the blower rotor was very flexible:**
  - **Shaft Diameter to bearing span ratio of >20:1.**
  - **The rotor response is primarily controlled by the stiffness of the shaft.**
  - **The shaft stiffness is relatively low due to the long bearing span.**
  - **The bearings are located close to the 1<sup>st</sup> mode nodal points and provide little control of the 1<sup>st</sup> critical.**
- ❑ **Vibration amplitudes of the five blowers tested were well above industrial standards and guidelines published by the Blower OEM.**
- ❑ **Vibration data on DB19 indicated unbalance of the blower rotating assembly and misalignment as the primary forcing functions. Worn bearing fits and oversized housing bores would have increased unbalance response.**
- ❑ **Flexing of the skids (frames) supporting the motor and blower was clearly evident in the ODS models. The skid was redesigned by equipment owner using thicker structural elements, additional stiffening and machining of mounting pads coplanar.**



## **Conclusions:**

- ❑ The modal test of the blower and motor frame assembly showed a very responsive bending mode of the frame at the motor end within the operating speed range. The natural frequency mode shape was rocking of the motor in the vertical and axial directions. All five blowers tested had high amplitude motor vibration in the axial and vertical directions. It was discovered that stiffener plates/gussets originally provided by the OEM had been removed by maintenance personnel to gain access to hold down bolts. These gussets had not been replaced.**
- ❑ Bearing defects were identified by vibration data on several of the motors and blower bearing housings.**
- ❑ Inspection of the piping and bellows flexible connectors showed evidence of excessive pipe strain at the blower's inlet and discharge nozzles due to un-restrained thermal growth of the piping as the blowers cycled on and off. The piping was not adequately supported to prevent excessive pipe strain of the blower nozzles.**
- ❑ Permalign data measured on DB18 and DB19 showed over 20 mils relative horizontal movement of the motor and blower during shutdown and cooling to ambient temperature. There were also excessive alignment changes when adjacent blowers cycled on and off caused by thermal growth of the piping pushing/pulling the blowers.**

## Conclusions:

- ❑ **DB 19 vibration test data showed extremely high amplitude vibration during startup and shutdown. A rotor critical speed was indicated at 2400 RPM. Mid-span rub was suspected to act as a third bearing raising the rotor critical from about 1450 RPM to about 2400 RPM. Confirmed by inspection and rotor-bearing model that packing was acting as a third bearing.**
- ❑ **Other options for sealing are available other than packing which include carbon ring and mechanical seal.**

## **Recommendations:**

**The following recommendations were provided to plant management to improve reliability of the Centrifugal Blowers.**

- Blower overhaul, rotating assembly balancing and mechanical run test.**
  - **Develop a rotor inspection process and inspection measurements.**
  - **Rotor multi-plane dynamic balance of rotating assembly Per ISO 1940-1 Balance Quality G1.0.**
  - **Mechanical run test, 1 hour minimum after temperatures stable.**
  
- Repair Facility Audit.**
  - **Incoming disassembly and inspection process.**
  - **Tolerance for replacement components**
  - **Access to OEM drawings and specifications.**
  - **Assembly process – part inspection, shaft and impeller dimensions, allowable runout, balancing process, training of personnel, condition of measuring tools and machine tools.**
  - **Post assembly run test capability.**
  
- Modal Test of First Fabricated Skid (New Design).**
  - **Insure that no structural natural frequencies are with +/- 10% of 1X, 2X run speed.**
  - **Modify skid design if test results indicate need.**

## **Recommendations:**

**The following recommendations were provided to plant management to improve reliability of the Centrifugal Blowers.**

- ❑ Install permanent Accelerometers on the Motor & Blower Bearing Housings.**
  - ❑ Cable signals to NEMA enclosures located outside the blower area.**
  - ❑ Consider connecting Accelerometers to IMI Model 682A05 Bearing Fault Detectors which can be monitored by PLC.**
  
- ❑ Blower Nozzle Loading (Pipe Strain)**
  - Conduct Piping Study**
  - Study to provide recommendations for piping supports and flexible piping connections to the blower nozzles.**
  - Obtain allowable blower nozzle loading from OEM.**

## **Action Taken by Management:**

- 1. Witnessing of Blower Disassembly, Repair, Balancing, Reassembly and Run Test was Performed at Authorized Repair Shop.**
- 2. Plant management decision to begin overhauling Centrifugal Blowers in-house. This would provide more control over dimensional accuracy of fits, balancing and assembly procedures.**
- 3. Inspection form developed to document the blower shaft fit dimensions and fit runouts, impeller radial/axial runout.**
- 4. The Skid Frame was Redesigned using Thicker Elements, Bracing and Blower Mounting Points were Machined Co-Planer.**
- 5. The Cork isolation material was changed to meet the blower OEM's specification.**
- 6. A piping study was conducted. Supports were added to fix the piping at the blowers and reduce pipe strain on the blower nozzles.**
- 7. Ambient alignment Targets were Determined based on Permalign Measurements.**

## References

1. Atra-Flex Flexible Couplings; <http://www.atra-flex.com/>
2. ME'scopeVES; <http://www.vibetech.com/>
3. [http://rodyn-inc.com/html\\_aboutRodyn.html](http://rodyn-inc.com/html_aboutRodyn.html)
4. DyRoBeS Software; <http://www.dyrobes.com/>
5. Bennett, Edward, D'Orsi, Nicholas, Japikse, David, Karon, David, Osborne, Colin; **A Lexicon Of Turbmachinery Performance**, Concepts ETI, Inc., Wilder Vermont, Oct 28-Nov 1, 1991, page 1-15, 1-16.
6. Flexicraft Industries; Metal Expansion Joint Instructions, [http://www.flexicraft.com/Metal\\_Expansion\\_Joints/](http://www.flexicraft.com/Metal_Expansion_Joints/)
7. **Ludeca permalign:** [http://www.ludeca.com/prod\\_permalign.php](http://www.ludeca.com/prod_permalign.php)

End