

STIFFNESS & MACHINE VIBRATION

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OUTLINE

- I. How Stiffness Effects Machine Vibration – Equation Of Motion & Dynamic Resistance.
- II. Stiffness Defined – Hooke's Law.
- III. Stiffness – Constant Or Variable? Factors That Change Stiffness.
- IV. Material Stiffness – Elasticity & Stress-Strain
- V. Geometric Stiffness – Examples Using Simple Structures
- VI. Dynamic Stiffness – How Stiffness Responds To A Dynamic vs A Static Force.
- VII. When Stiffness Goes Bad – Looseness & Resonance.
- VIII. Case Histories Involving Changes In Stiffness.

EQUATION OF MOTION – VIBRATING SYSTEM

- The equation of motion for a vibrating system is shown below. This equation is for a single degree of freedom system undergoing forced vibration.

The diagram shows the equation of motion for a single degree of freedom system undergoing forced vibration. The equation is $\ddot{x} = \frac{F_0 \cos \omega_f t - kx - C\dot{x}}{m}$. Callouts in yellow boxes identify the terms: 'Measured Vibration (acceleration, \ddot{x})' points to the left side; 'System mass (m)' points to the denominator; 'Dynamic Force ($F_0 \cos \omega_f t$)' points to the first term in the numerator; 'Spring (Stiffness) Force (kx)' points to the second term; and 'Damping Force ($C\dot{x}$)' points to the third term.

$$\ddot{x} = \frac{F_0 \cos \omega_f t - kx - C\dot{x}}{m}$$

Measured Vibration (acceleration, \ddot{x})

System mass (m)

Dynamic Force ($F_0 \cos \omega_f t$)

Spring (Stiffness) Force (kx)

Damping Force ($C\dot{x}$)

FACTORS THAT EFFECT MACHINE VIBRATION

- From the equation of motion, we see that the mass, stiffness & damping present in a vibration system all act to **reduce or retard** its motion.
- As all these three factors act to reduce the system vibration, we can think of their summation as the “**dynamic resistance**” of our vibrating system.
- Thus, in general, when either the mass, stiffness, or damping are increased, our machine vibration (motion) tends to be reduced.
- From the equation of motion, we also see that the **dynamic force** present in a vibrating system acts to **increase** its motion. Thus, in general, when the dynamic force(s) are increased, our machine vibration (motion) tends to increase.
- A simple way to sum up our conclusions from the equation of motion are as follows:

$$\mathbf{Vibration\ (motion)} \sim \frac{\mathbf{Dynamic\ Force}}{\mathbf{Dynamic\ Resistance}}$$

VIBRATION LEVELS .VS. DYNAMIC RESISTANCE

- Many vibration analyst's assume that when the vibration levels on a machine rise it is due to an increase in dynamic force(s) at or on the machine.
- While an increase in dynamic force may indeed be the culprit, often times the force levels are essentially unchanged from the prior survey, but instead a change in **dynamic resistance** has occurred.
- The most common change in dynamic resistance experienced in the field from survey to survey is a change in stiffness.
- Common stiffness changes observed in the field include a loose bolt, a cracked weld, a loose or missing pipe hanger or pipe stand, cracked or defective grout within the base, etc.
- ***While these changes in stiffness are very difficult (if not impossible) to precisely identify if one is analyzing vibration data from afar, they are often self evident and easily found for the observant and curious analyst who is actually at the machine in question. Being there matters.***
- Many times identifying **exactly** what machine component or part is loose (stiffness reduction) and exactly where it is loose is accomplished by using a combination of your eyes, your fingers, your knowledge of the machine and the forces it produces, and your vibration analyzer.
- Using your vibration analyzer, identify the areas of the machine where higher levels are identified and in what directions, then use your sense of sight and touch to help nail down more precisely the exact locations where your machine stiffness might have been compromised (reduced).

STIFFNESS DEFINED – HOOKE'S LAW

- The stiffness or rigidity of a structure can be defined by Hooke's Law expressed as follows:

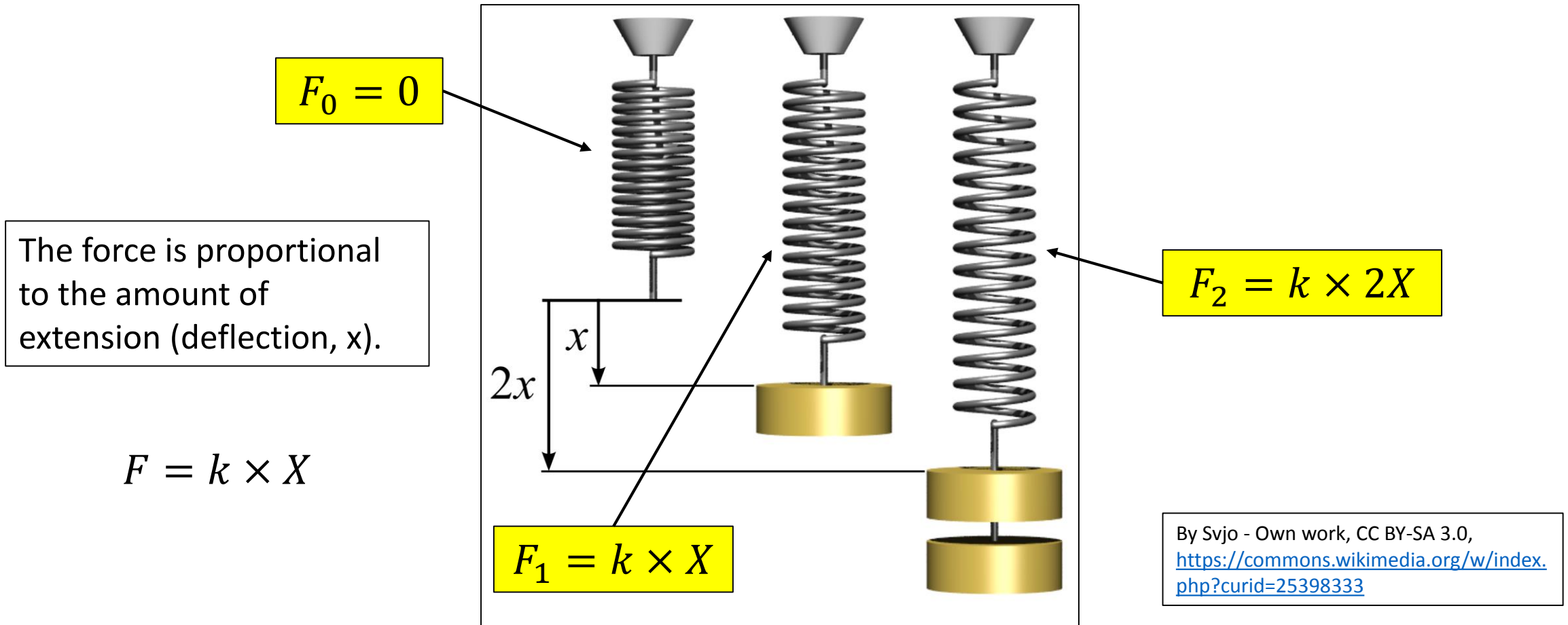
$$\text{Stiffness} = k = \frac{\text{Force } (F)}{\text{Displacement or Deflection } (X)}$$

this formula is most often seen as $F = k * x$

where F = spring force & x = the displacement or deflection of the spring.

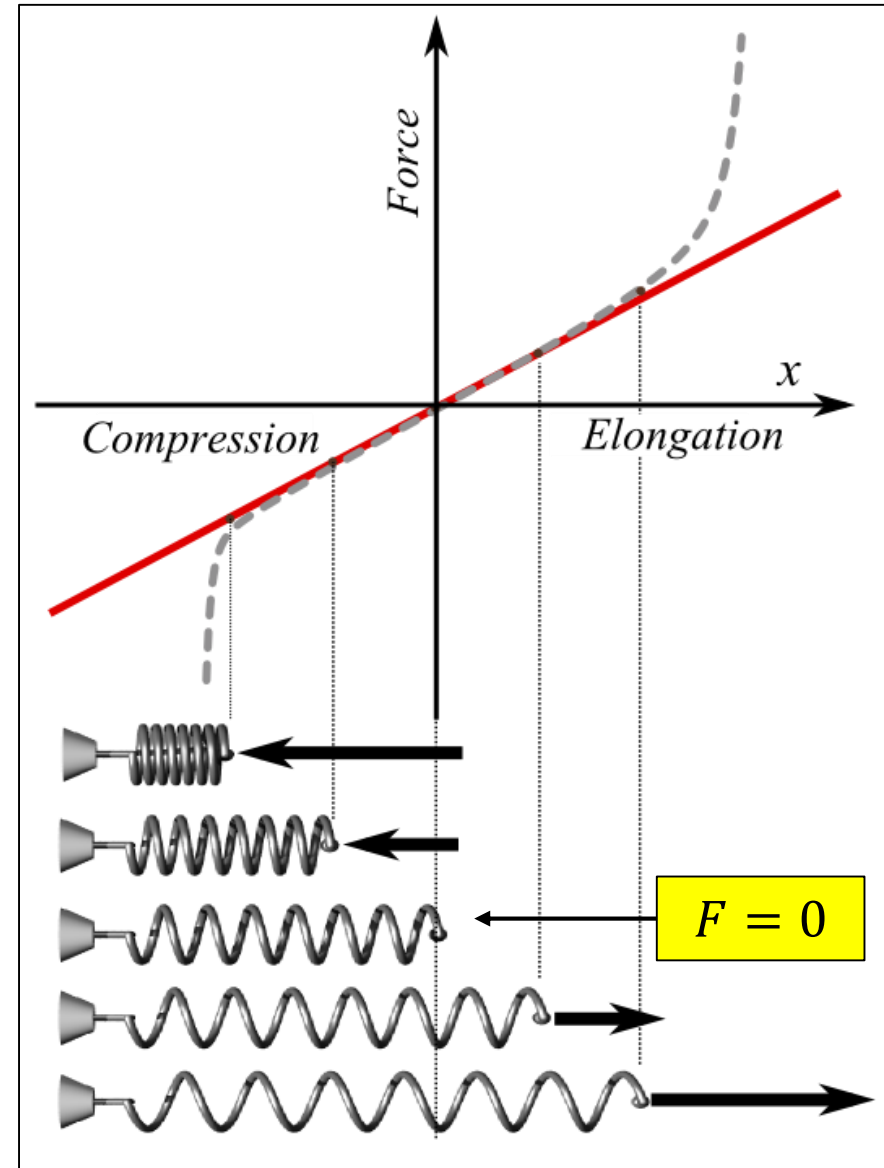
- The mathematical expression for stiffness is credited to the 17th century British physicist, Mr. Robert Hooke who first published it in 1676.

GRAPHIC OF HOOKE'S LAW – EXTENSION SPRINGS



EXPERIMENTAL DATA - HOOKE'S LAW

- Plot of applied force F vs. elongation X for a helical spring according to Hooke's law (**red line**) and what the actual experimental data might look like (**dashed grey line**).
- At bottom, pictures of spring states corresponding to points on the plot are shown; the middle one is in the relaxed state (no force applied).
- Note how beyond a certain lower and upper force limit, Hooke's law doesn't hold (plot is non-linear).
- ***Thus, the idea that stiffness is a constant is only true within a certain force range for a given spring.***



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WHAT FACTORS CHANGE MACHINE STIFFNESS?

- 1) **Material** type used in the structure (ie: steel, aluminum, concrete, wood, fiberglass, etc).
- 2) **Geometry** of the structure. What is the base, width, height, etc of the structure.
- 3) **Dynamic force level.** A machine or structure's stiffness is only constant within a given force range (see above slide). This is true for nearly if not all materials.
- 4) **Frequency Of Vibration.** As we will see later, the dynamic stiffness of any structure is dependent on the frequency of the dynamic force applied.
- 5) **Where The Force Is Applied.** Depending on where and in what direction the dynamic force is applied to a given structure or machine, it will be met by a different amount of stiffness or resistance to motion.
- 6) **Temperature.** At higher temperatures, most materials exhibit lower levels of stiffness. At lower temperatures, most materials exhibit higher levels of stiffness.

MATERIAL STIFFNESS - ELASTICITY

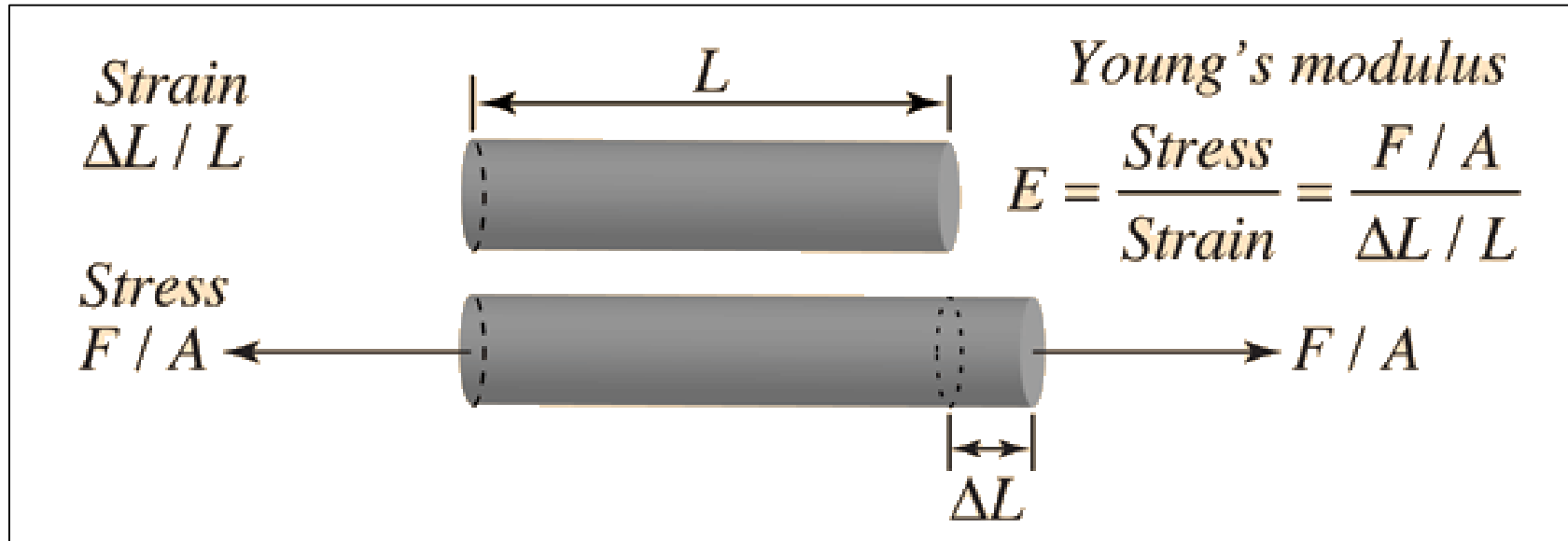
- The concept of a structure's stiffness is related to the idea of a material's elasticity.
- ***Generally when we speak of a structure's stiffness we are referring to the combination of its material & geometric stiffness.***
- The elasticity (material stiffness) of a given material (steel, aluminum, etc) is defined by its stress-strain curve and by the following formula:

Stress = Modulus of elasticity (E) * Strain (e)

or Stress = S = E * e = Force/Area

- Another name for modulus of elasticity is **Young's modulus** named after the 19th century British scientist Mr. Thomas Young who contributed greatly to our understanding of the stress-strain relationship within materials.

ILLUSTRATION OF YOUNG'S MODULUS (ELASTICITY)



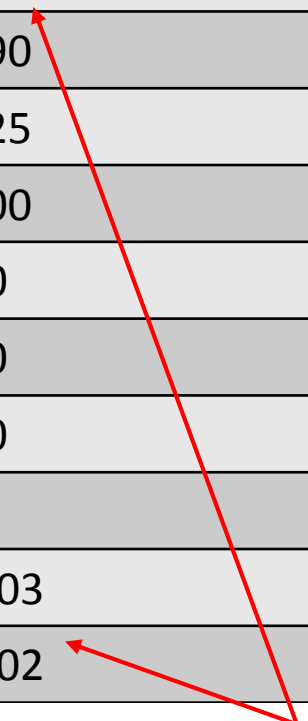
- Young's modulus (material stiffness) changes with temperature.
- At lower temperatures, materials tend to become more stiff, and at higher temperatures, materials tend to become less stiff (more flexible).

- **Stress = Force / Area** and **Strain = $\Delta L / L$ = Change in length / Original length**
- At a strain value of 1 or 100%, the part or material has doubled its original length due to the applied stress.
- The higher a material's Young's modulus value (E), the more stress has to be applied to achieve identical values of strain.
- High Young's modulus = stiff material. Low Young's modulus = flexible material.

Reference: [hyperphysics – Elasticity, Elastic Properties](#)

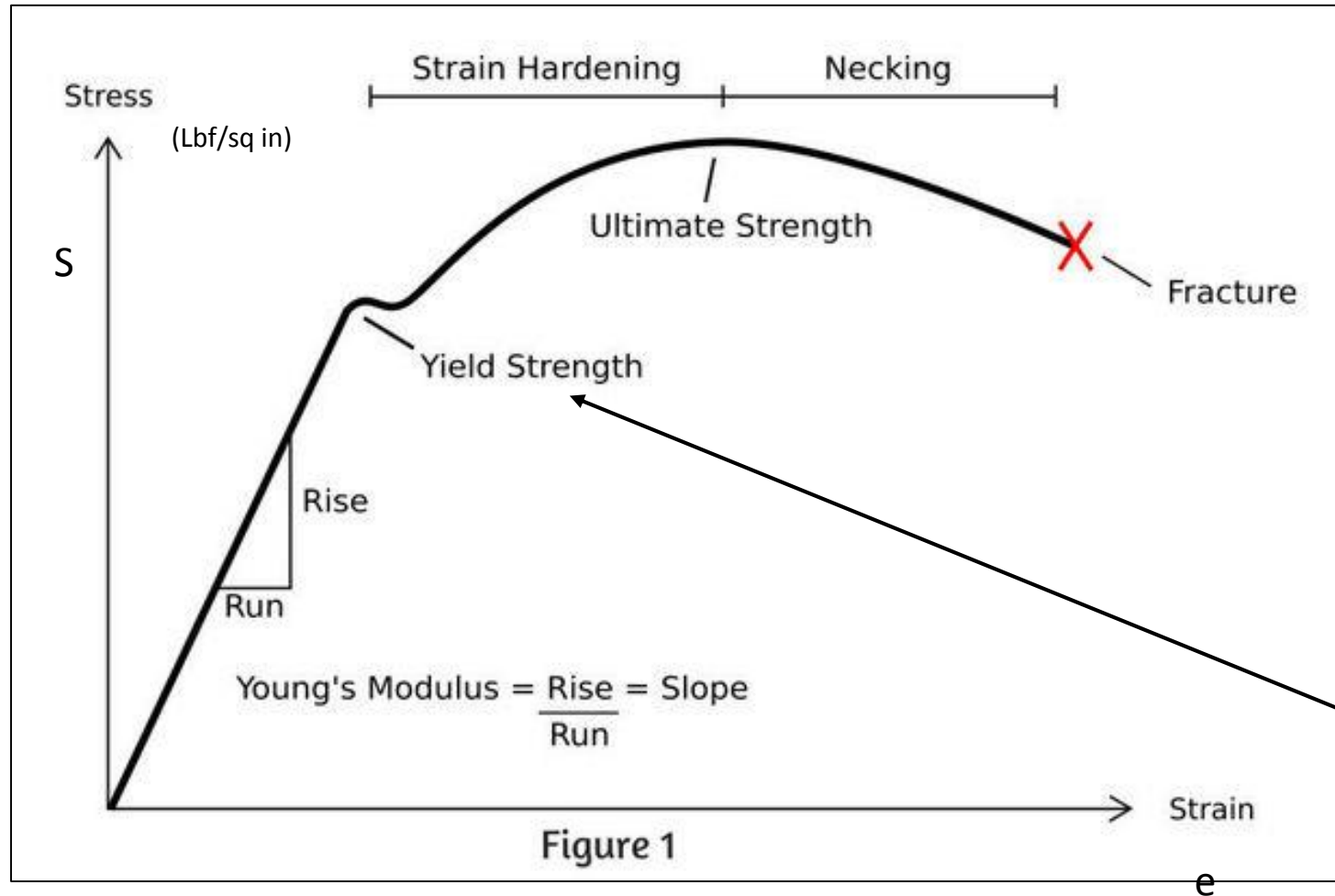
YOUNG'S MODULUS FOR COMMON MATERIALS

<u>MATERIAL</u>	<u>YOUNG'S MODULUS (Gpa)</u>	<u>YOUNG'S MODULUS (KPSI)</u>
DIAMOND	1,220	176,950
CARBON STEEL (MILD)	210	30,460
STAINLESS STEEL	190	27,560
CAST IRON	125	18,130
TITANIUM	100	14,500
ALUMINUM	70	10,150
GLASS	70	10,150
WOOD (ALONG GRAIN)	10	1,450
PLASTIC	2	290
CORK	0.03	4
RUBBER	0.02	3



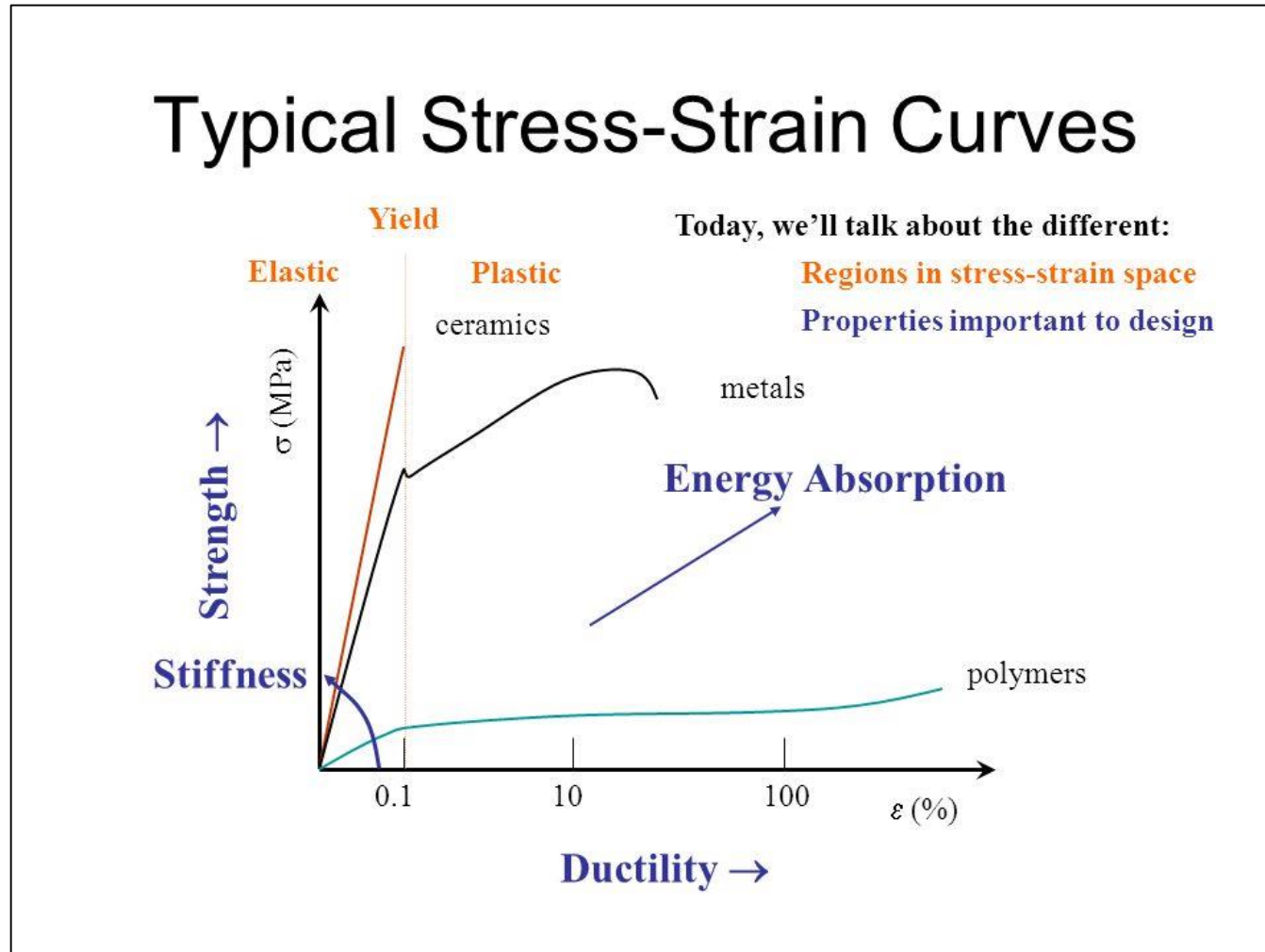
Note the vast difference in material stiffness between steel and either rubber or cork. Now you can see why rubber or cork are commonly used in machine isolators (reduces stiffness greatly ~ 10,000 times less).

STRESS-STRAIN CURVE



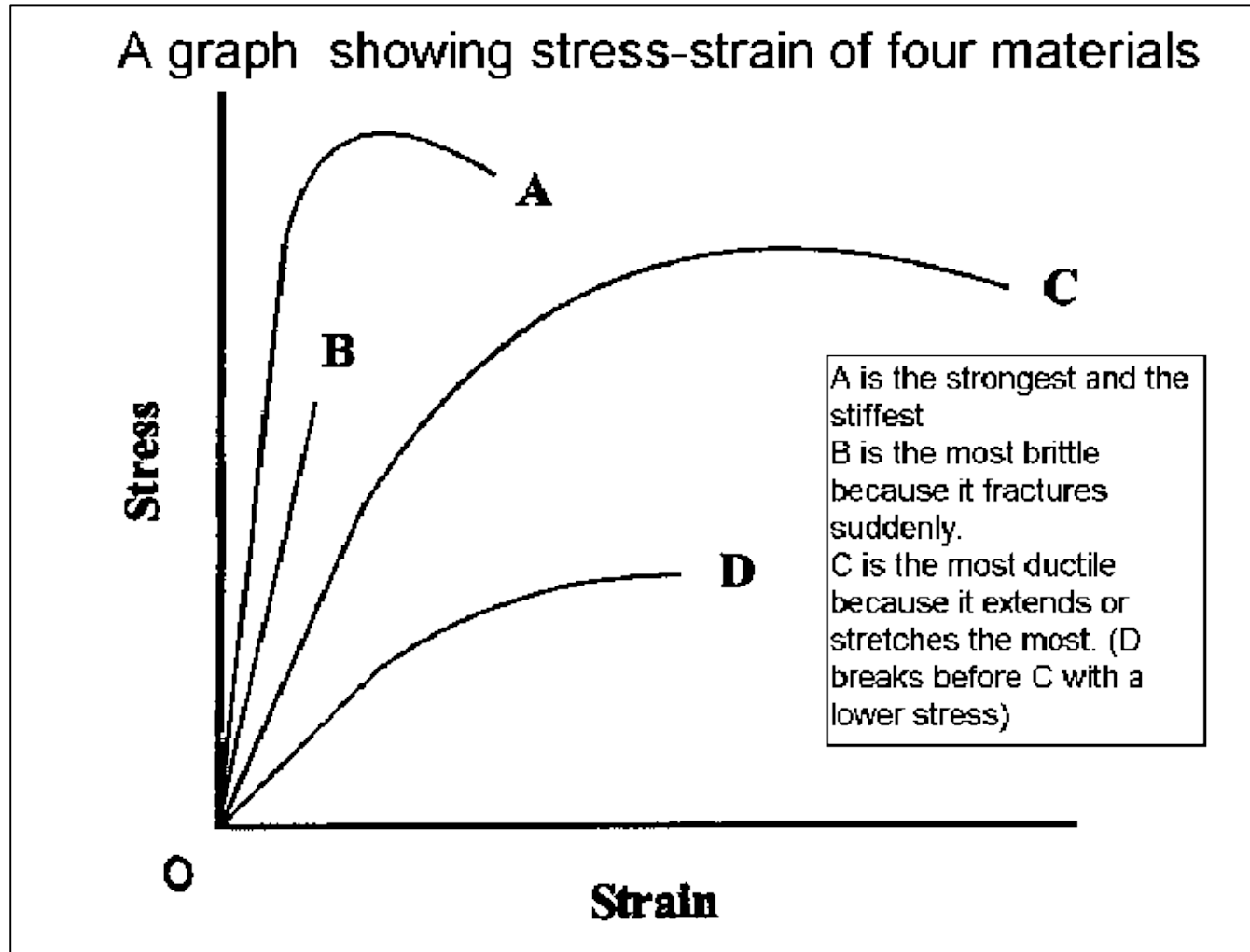
- The plot at left is a typical stress-strain curve for a ductile material like some steels and aluminum.
- The Y or vertical axis is the stress applied to the material while the X or horizontal axis is the amount of strain or the % change in its length that occurs as a result of the applied stress.
- Strain = change in length / original length or $\Delta L / L$.
- The formula we showed earlier for Young's modulus $S = E \cdot e$ is only applicable for applied stresses up to the material's yield strength.

STRESS-STRAIN CURVES FOR DIFFERENT MATERIALS 1



- The plot at left shows stress-strain curves for three different general materials (from left to right): ceramics, metals & polymers.
- Note how the slope of the curve (material stiffness) for ceramics is steep in comparison to either that for metals or polymers.
- Note how the curve for metals and polymers becomes non-linear (plastic) once the yield strength is reached.
- Note how little if any plastic region exists for ceramics. This is another way of saying that ceramics are brittle (they don't yield, they break when you exceed their yield stress).

STRESS-STRAIN CURVES FOR DIFFERENT MATERIALS 2

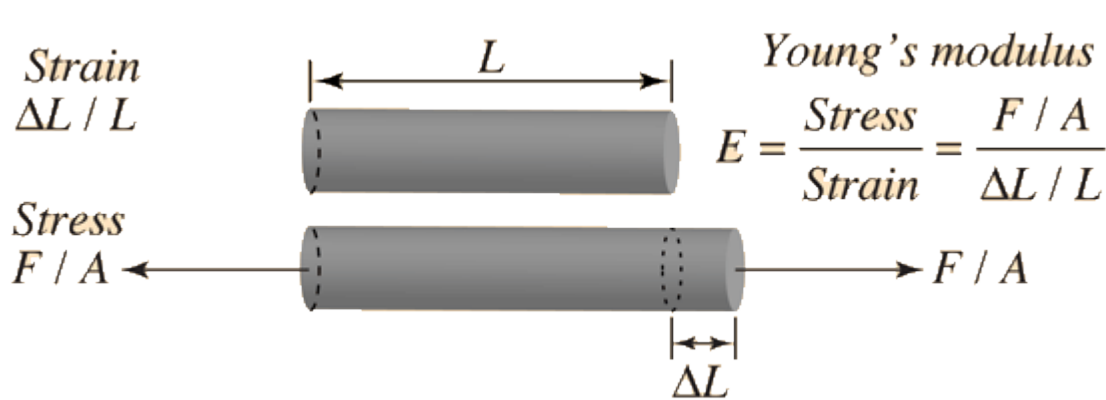


- The plot at left shows four hypothetical stress-strain curves for different general materials (A, B, C & D).
- Material **A** would be considered the strongest & stiffest of the 4-ea materials shown as its stress-strain curve was the steepest and it endured the highest stress before yielding.
- Material **B** would be considered a stiff & brittle material with a steep stress-strain curve, but it fractured suddenly with little if any yielding.
- Material **C** would be considered the most ductile due to its long yielding prior to fracture.
- Material **D** would be similar to C but less strong as it yields and fractures at lower stress levels.

GEOMETRIC STIFFNESS

- The **total stiffness** of a structure or machine is the combination of its **material stiffness** and its geometry or **geometric stiffness**.
- The **geometric stiffness** of a structure is dependent on the layout, design, and dimensions of the structure.
- Changes to the structure's length, width, height, etc directly affect the geometric stiffness of a structure.
- The geometric stiffness of a structure is reflected in its “moment of inertia” (I) and other factors such as its length, height, width, etc.
- The moment of inertia is often thought of as a measure of a beam or structure's ability to resist bending; this resistance is very dependent on the direction of bending.
- A structure having a high moment of inertia in a certain direction will have high stiffness in that direction.
- $I = \frac{bh^3}{12}$ (rectangular beam of base (b) and height (h)) and $I = \frac{\pi r^4}{4}$ (circular beam of radius (r))
- Examples of simple structures with formulas for their stiffness are shown in the slides that follow.

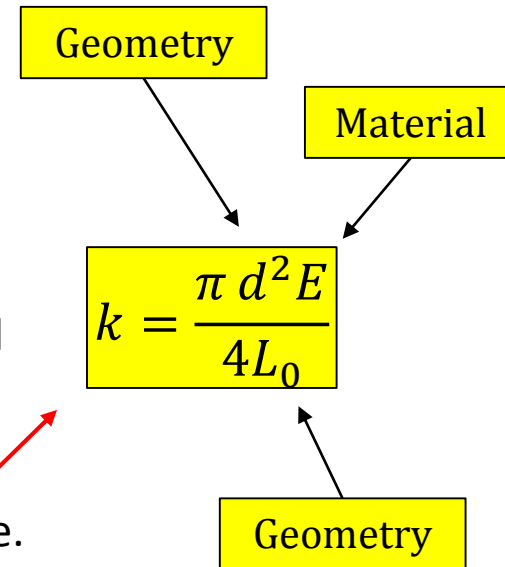
GEOMETRIC STIFFNESS – AXIAL (NORMAL) STRESS



$$k = \frac{F}{\delta} \quad [\text{general form}]$$

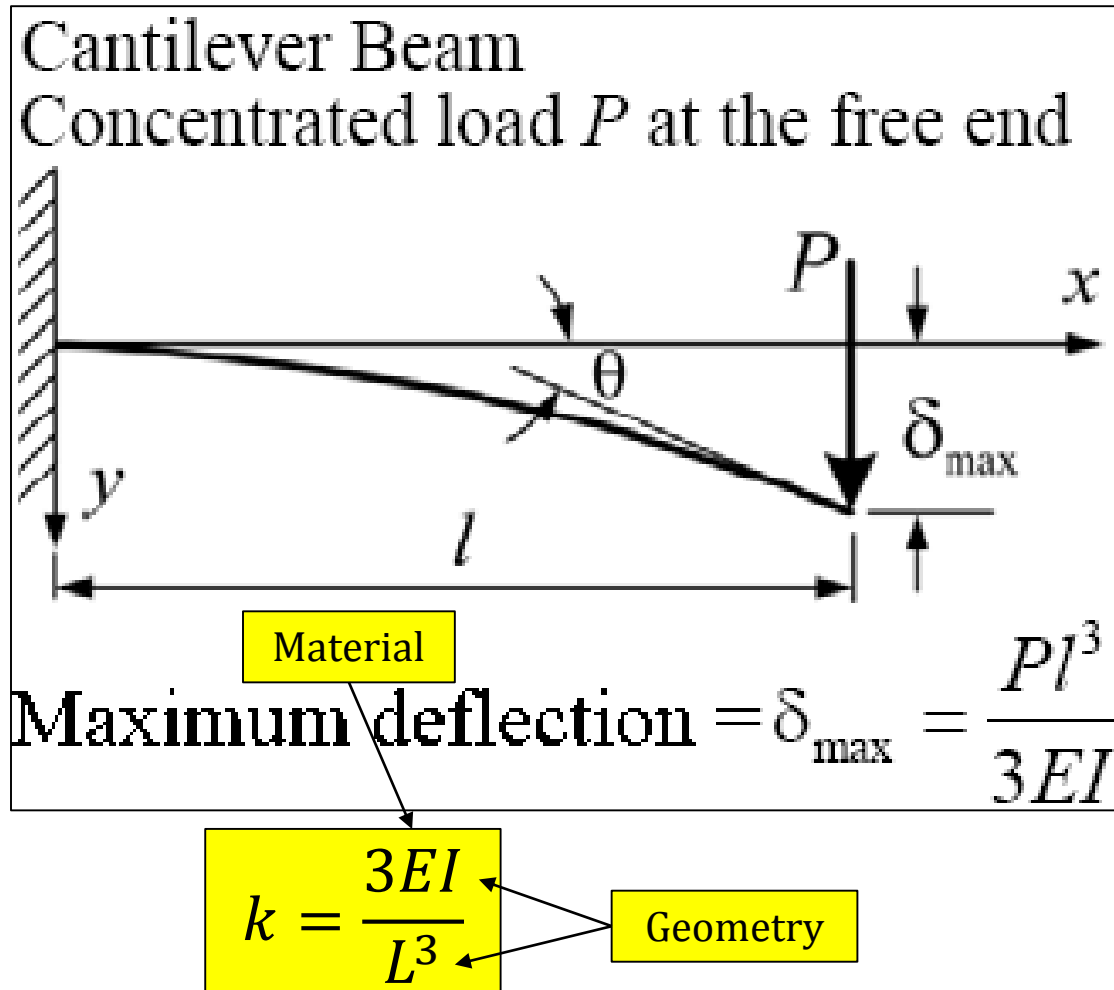
$$= \frac{AE}{L_0} \quad [\text{normal stress form}]$$

Where $A = \frac{\pi d^2}{4}$ (circular rod) and



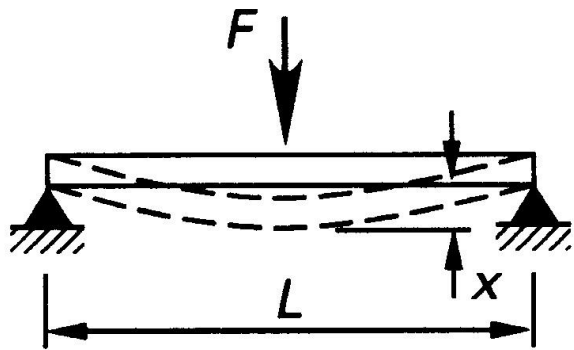
- The deflection is ΔL (tension) = $\delta = x$ and the strain $e = \Delta L / L$ or rearranging $\Delta L = e * L$.
- From Hooke's Law we know that $F = kx$ where $x = \delta = \Delta L =$ deflection, and $F = P =$ Applied force.
- Rearranging Hooke's Law, $k = F / x = F / \Delta L$
- Hooke's Law for normal stress is $S = F / A = E * e = E * \Delta L / L$ or rewritten this becomes $F = A * E * e = A * E * \Delta L / L$
- Thus, the formula for stiffness becomes: $k = F / x = F / \Delta L = A * E * \Delta L / L / \Delta L = \mathbf{k = A * E / L}$
- Note that the **material** plays a role here in the term "E" which is Young's modulus for the beam material in question (steel, aluminum, etc).
- The majority of the terms in this equation though relate to the **geometry or geometric stiffness** of the beam (length L, cross sectional area A, etc).

GEOMETRIC STIFFNESS – CANTILEVER BEAM



- The deflection is $\delta = \frac{Pl^3}{3EI}$
- From Hooke's Law we know that $F = kx$ where $x = \delta$ = deflection, and $F = P$ = Applied force.
- Rearranging this equation and inserting our formula for deflection above, we find the following for the stiffness $k = F / x = 3EI/l^3$
- Note that the **material** plays a role here in the term "E" which is Young's modulus for the beam material in question (steel, aluminum, etc).
- The majority of the terms in this equation relate to the **geometry** or geometric stiffness of the beam (length, cross sectional area, etc).
- The moment of inertia (I) would vary according to the cross sectional geometry of the beam (square, rectangular, circular, tube, etc).

GEOMETRIC STIFFNESS – SIMPLY SUPPORTED BEAM



Maximum
Deflection (x)

$$\frac{FL^3}{48EI}$$

Stiffness (k)

$$\frac{48EI}{L^3}$$

Material

$$k = \frac{48EI}{L^3}$$

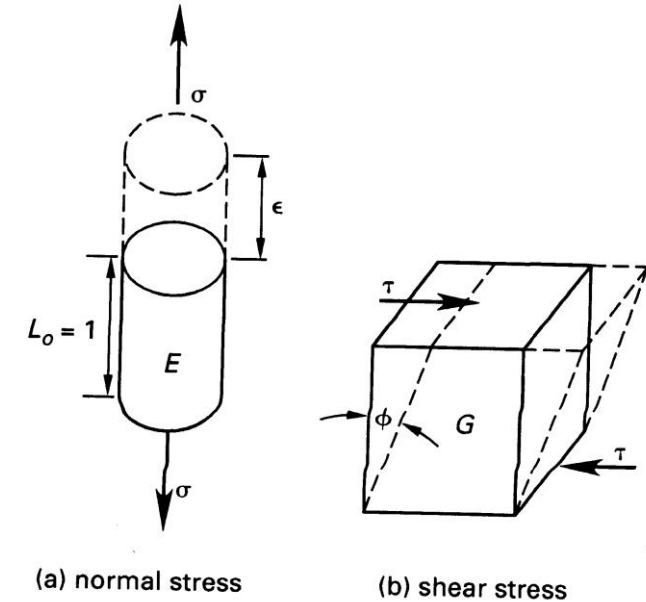
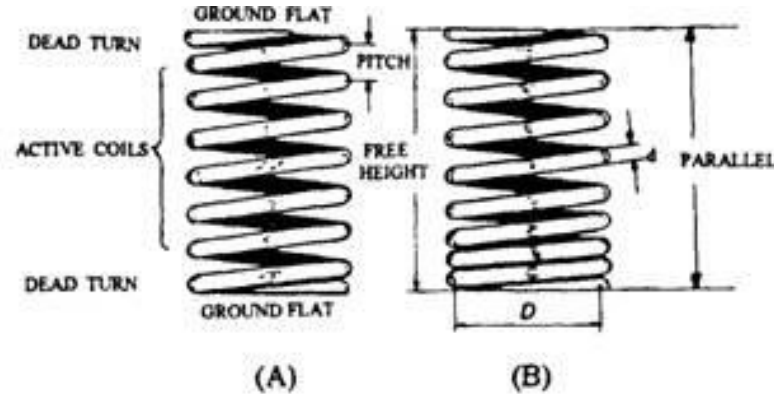
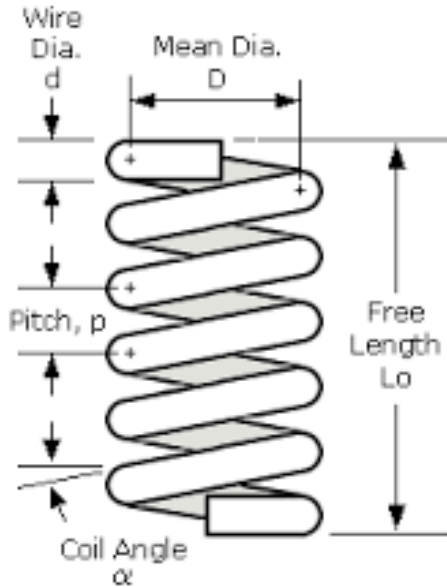
Geometry

- As in the prior example for a cantilever beam, note in this example of a simply supported beam that the material plays a role here in the term “E” which is Young’s modulus for the beam material in question (steel, aluminum, etc).
- The majority of the terms in this equation relate to the geometry or geometric stiffness of the beam (length, cross sectional area, etc).
- The moment of inertia (I) would vary according to the cross sectional geometry of the beam (square, rectangular, circular, tube, etc). It is solely dependent on the beam’s geometry.

A GREAT EXAMPLE OF GEOMETRIC .VS. MATERIAL STIFFNESS: A COMMON SPRING .VS. LENGTH OF METAL WIRE IN TENSION

- Compare a common steel spring versus a length of steel wire of the same thickness for a great example of how geometry plays a major role in determining the stiffness of any given mechanical system.
- The formula for both a spring and for a length of metal wire in tension are shown in the next slide.
- Both the spring and wire are in tension, share a common material (steel), a common wire diameter (d), and a common length (L_0). **Only their geometry is different.**

COMPARE THE STIFFNESS OF A WIRE VERSUS A SPRING



SPRING STIFFNESS

$$k_s = \frac{G d^4}{8D^3 n_a}$$

- k = spring stiffness
- G = Shear modulus of spring material
- d = spring wire diameter
- D = spring coiled diameter
- n_a = number of active spring coils

The shear modulus “ G ” is similar to the modulus of elasticity “ E ” only the deflections are occurring in the shear direction on the material (see above).

WIRE STIFFNESS

$$k = \frac{F}{\delta} \quad [\text{general form}] \quad k_w = \frac{\pi d^2 E}{4L_0}$$

$$= \frac{AE}{L_0} \quad [\text{normal stress form}]$$

Where $A = \frac{\pi d^2}{4}$

CALCULATE & COMPARE STIFFNESSES – WIRE .VS. SPRING

- Suppose we have a **steel wire** in tension of diameter $d = 1/8''$ and total length $L_0 = 18-7/8''$.
- The stiffness of this steel wire will be calculated using the formula:
- Where $E =$ Modulus of elasticity (steel) = 30,000 kpsi
- Thus, $k_w = \frac{\pi d^2 E}{4L_0} = \frac{\pi(1/8'')^2(30,000,000)}{4(18.875)} = \underline{19,505 \text{ lb/in}}$
- Suppose we have a **steel spring** made of the same wire of diameter d and total length L_0 with a coil diameter of $D = 8d = 8 \times 1/8'' = 1''$ and with 6 active coils. The total spring length is identical to that of the wire above and is verified by the formula for active spring length as follows:
- $L_a = L_0 = \pi \times D \times n_a = \pi \times 1'' \times 6 = 18.84'' \sim 18.875''$
- The stiffness of this spring will be calculated using the formula:
- Where $G =$ Shear Modulus (steel) = 11,500 kpsi
- Thus, $k_s = \frac{G d^4}{8D^3 n_a} = \frac{11,500,000(1/8'')^4}{8(1'')^3(6)} = \underline{58.5 \text{ lb/in}}$
- Comparing the wire versus the spring stiffness we see that $k_w / k_s = 19,505/59 = 331$.
- Thus, by simply changing the geometry of steel wire from a straight wire in tension to a coiled spring we reduced its stiffness by a factor of over 300 times!
- **Where stiffness is concerned, remember that geometry matters!**

$$k_w = \frac{\pi d^2 E}{4L_0}$$

$$k_s = \frac{G d^4}{8D^3 n_a}$$

STIFFNESS .VS. FREQUENCY OF DYNAMIC FORCE: “DYNAMIC STIFFNESS”

- The stiffness of any structure to dynamic forces is highly dependent on the frequency of the dynamic force applied. There is no better example of this than the phenomena called resonance.
- Resonance of a machine or its structure occurs when the frequency of any dynamic force applied to a machine or structure is coincident with a natural frequency of the machine or structure.
- I like to think of a natural frequency as a local minimum of a machine or structure’s dynamic stiffness.
- Accurate measurements of a machine or structure’s dynamic stiffness are made when we collect modal data on them.
- Collection of modal data from a machine or structure involves the application of a known dynamic force at an exact location and measuring the response or motion of the structure or machine to that applied force. **Cause (dynamic force) & Effect (vibration/motion).**
- **When we make modal measurements on a machine or structure, we measure its dynamic stiffness.** This stiffness can and will vary according to the frequency of vibration, location along the structure, and direction of force application.
- The formal terms for the transfer function and its inverse “dynamic stiffness” are shown in the next slide. Examples of dynamic stiffness measurements (modal measurements) are shown in following slides for a vertical pump.

FORMAL TERMS FOR THE TRANSFER FUNCTION & ITS INVERSE “DYNAMIC STIFFNESS” BY UNIT OF MEASURE

The following are formal terms for the transfer function & its inverse “dynamic stiffness” by unit of measure chosen by the analyst:

TRANSFER FUNCTION	
Compliance	Displacement/Force
Mobility	Velocity/Force
Inertance	Acceleration/Force

“DYNAMIC STIFFNESS”	
Apparent Stiffness	Force/Displacement
Impedance	Force/Velocity
Apparent Mass	Force/Acceleration

The appropriate units should be chosen depending on the frequencies of vibration being measured.

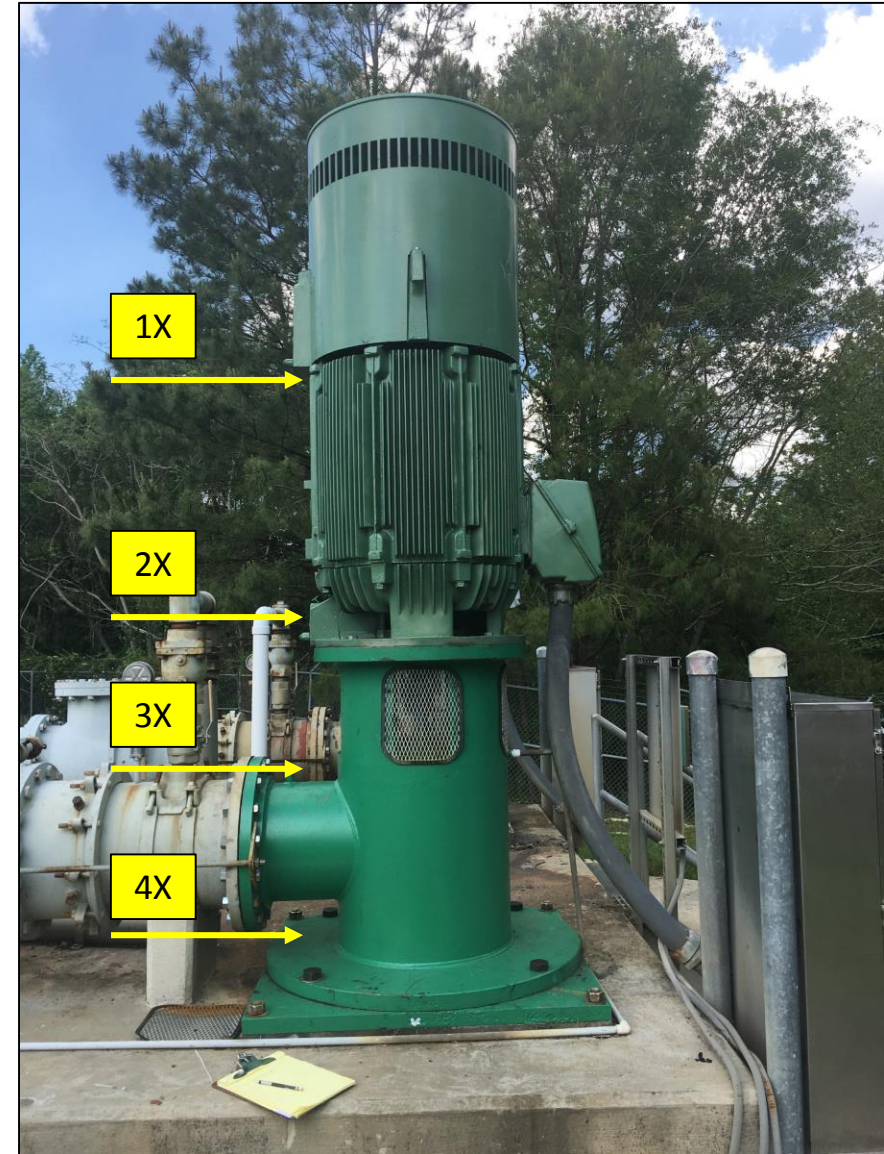
STIFFNESS

HOW IT IS A VARIABLE VERSUS:

- FREQUENCY
- LOCATION
- DIRECTION

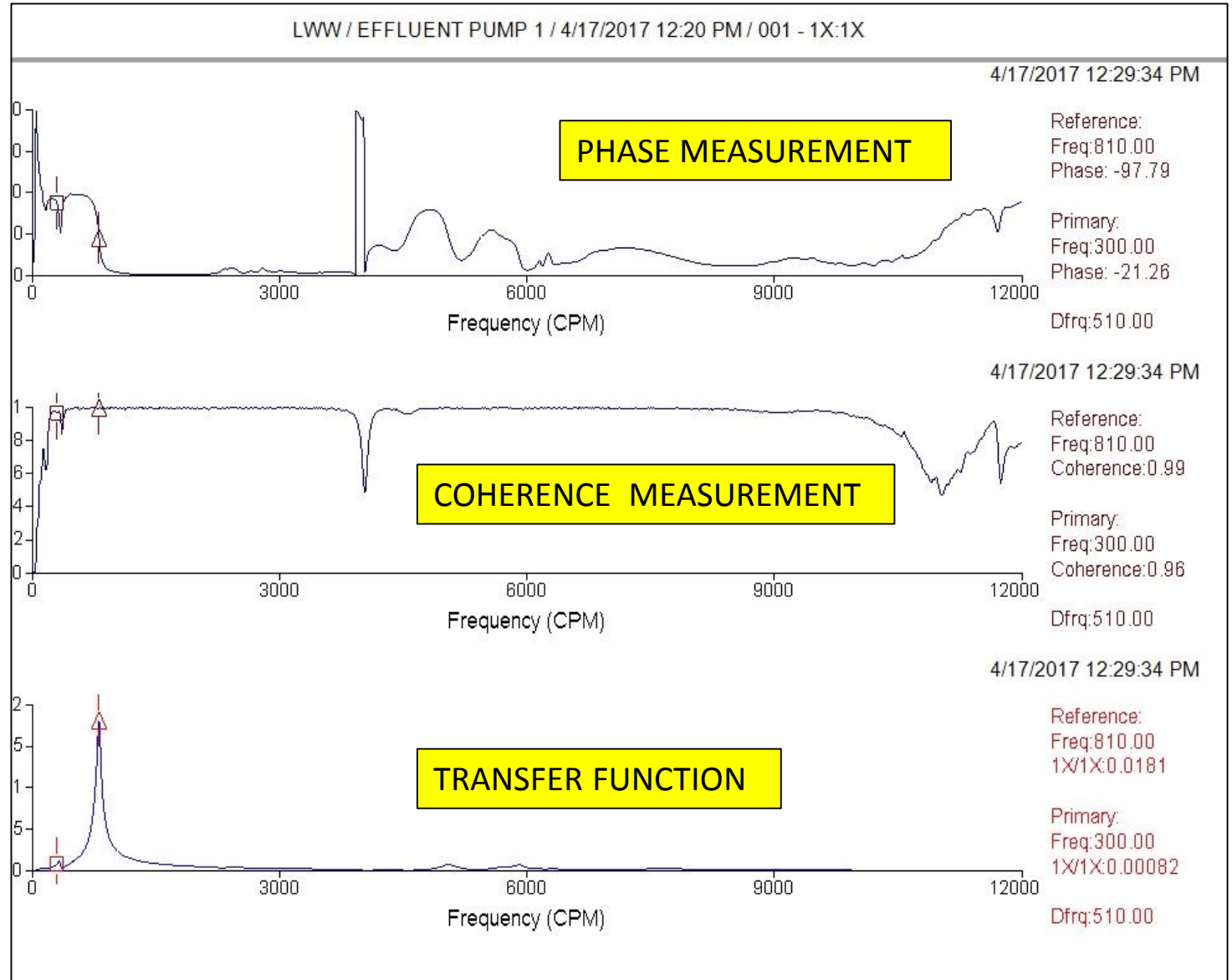
VERTICAL PUMP

- Modal vibration data was collected on the vertical pump at right.
- Data was collected in both the discharge direction (X) and the transverse discharge direction (Y).
- Data was also collected at various points and elevations along the machine as follows:
 - 1) 1X = Top Motor Bearing, Discharge Direction
 - 2) 2X = Bottom Motor Bearing, Discharge Direct.
 - 3) 3X = Top Pump Bearing, Discharge Direct.
 - 4) 4X = Pump Base, Discharge Direct.
- What follows is a presentation of some of the modal data collected from this pump with the goal of illustrating that stiffness is dependent on location, direction, and the frequency of the dynamic force.



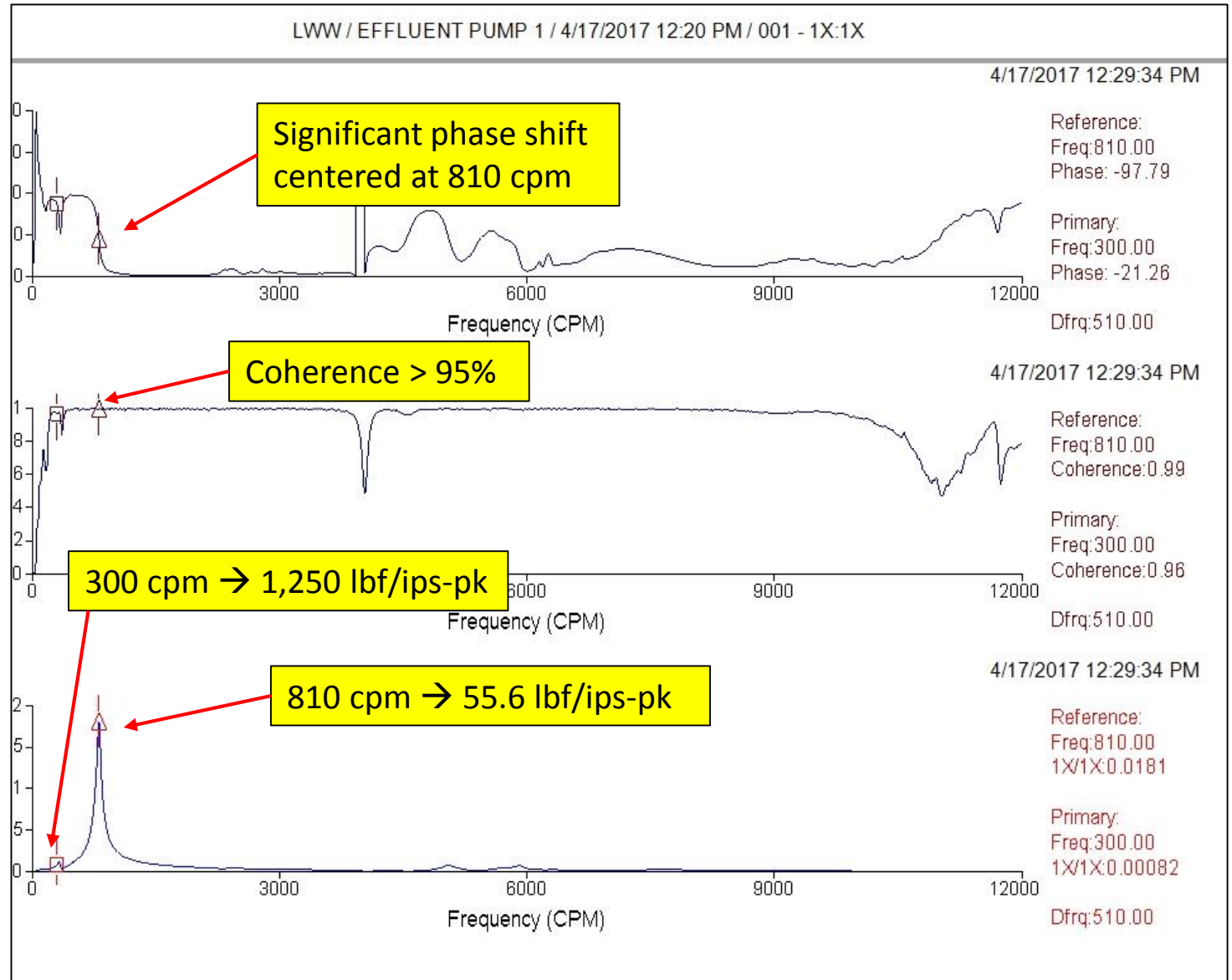
DYNAMIC STIFFNESS – VERTICAL PUMP

- The plots at right are modal vibration measurements from the top bearing of a vertical pump in the discharge direction.
- The bottom plot is known as the **transfer function** for the measurement which is a ratio of how much response (vibration) occurs as a result of the force applied.
- The middle plot is known as the **coherence function** which can be thought of as a measure of the quality of the modal measurement. Values of coherence > 90% are required while values > 95% or so are desired. When coherence values are lower than this, suspect problems such as looseness, bad wiring connections, transmitted vibration, etc.
- The top plot is the **phase function** which is a timing measurement between the force input and vibration response.



DYNAMIC STIFFNESS – FREQUENCY

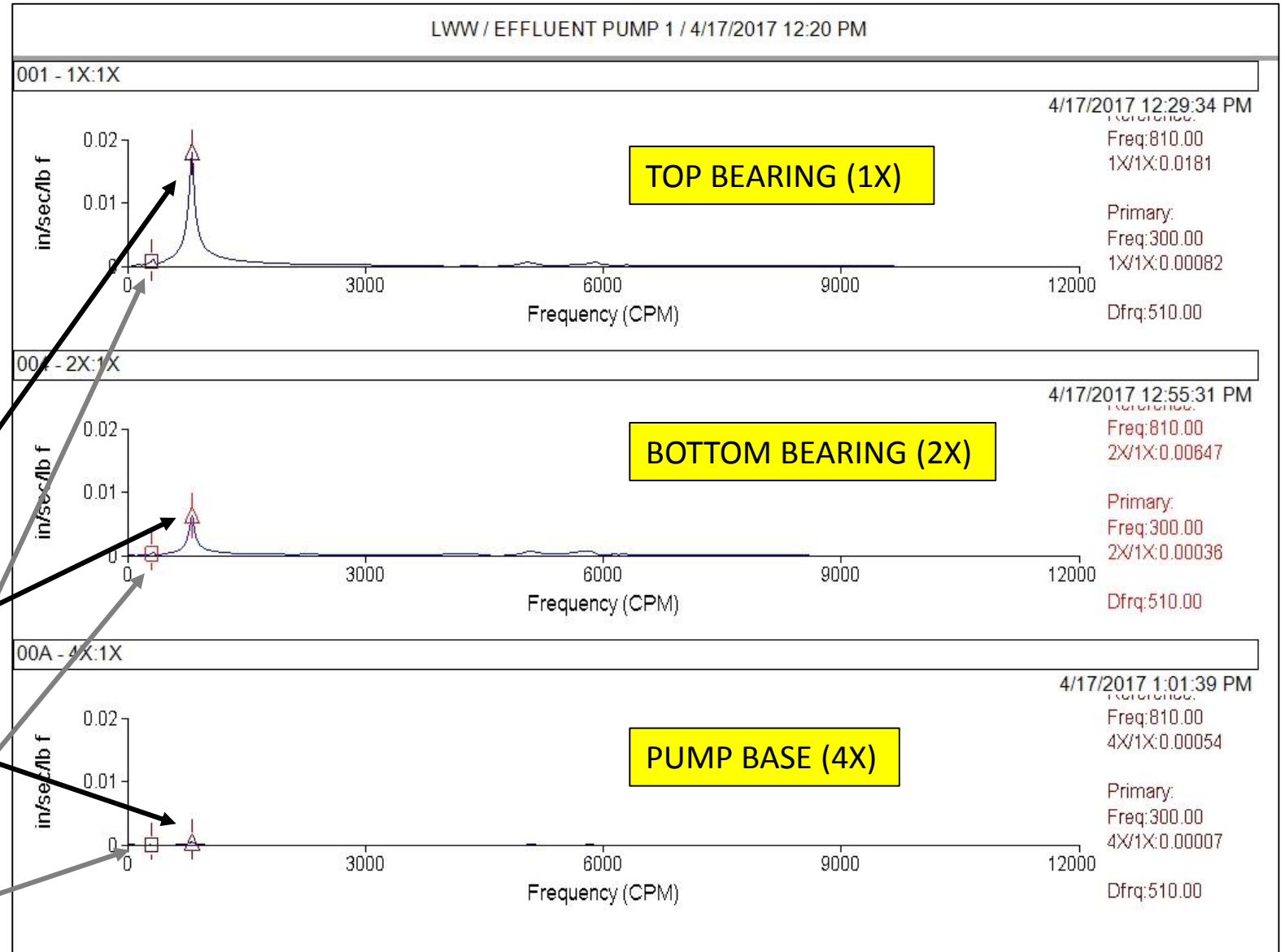
- The transfer function measurement shown is in units of mobility, ips-pk/lbf which is the inverse of the “dynamic stiffness”, lbf/ips-pk.
- High mobility measurements equate to high flexibility and low stiffness.
- Note how the mobility of the top bearing at or near ~ 810 cpm is very high (0.018 ips/lbf) in relation to all other nearby frequencies (0.0008 ips/lbf) – **810 cpm is a natural frequency of the vertical pump.**
- Inverting these mobility measurements, we get a dynamic stiffness of 55.6 lbf/ips-pk at 810 cpm versus 1,250 lbf/ips-pk at 300 cpm – **this great difference in stiffness is the primary reason why we get so much more vibration (motion) when operating at or near a natural frequency.**
- ***Do you see now how stiffness is not a constant but highly dependent on the frequency of the dynamic force?***



DYNAMIC STIFFNESS – LOCATION

- The plot at right is from the same vertical pump in the same direction (discharge) but at different locations (1X = top bearing, 2X = bottom bearing, and 4X = pump base).
- Note how stiffness at the same frequency varies greatly with location (lbf/ips-pk).**

- Stiffness 1X (810 cpm) = 55.6
- Stiffness 2X (810 cpm) = 154
- Stiffness 4X (810 cpm) = 1,852
- Stiffness 1X (300 cpm) = 1,250
- Stiffness 2X (300 cpm) = 2,778
- Stiffness 4X (300 cpm) = 14,286



DYNAMIC STIFFNESS - DIRECTION

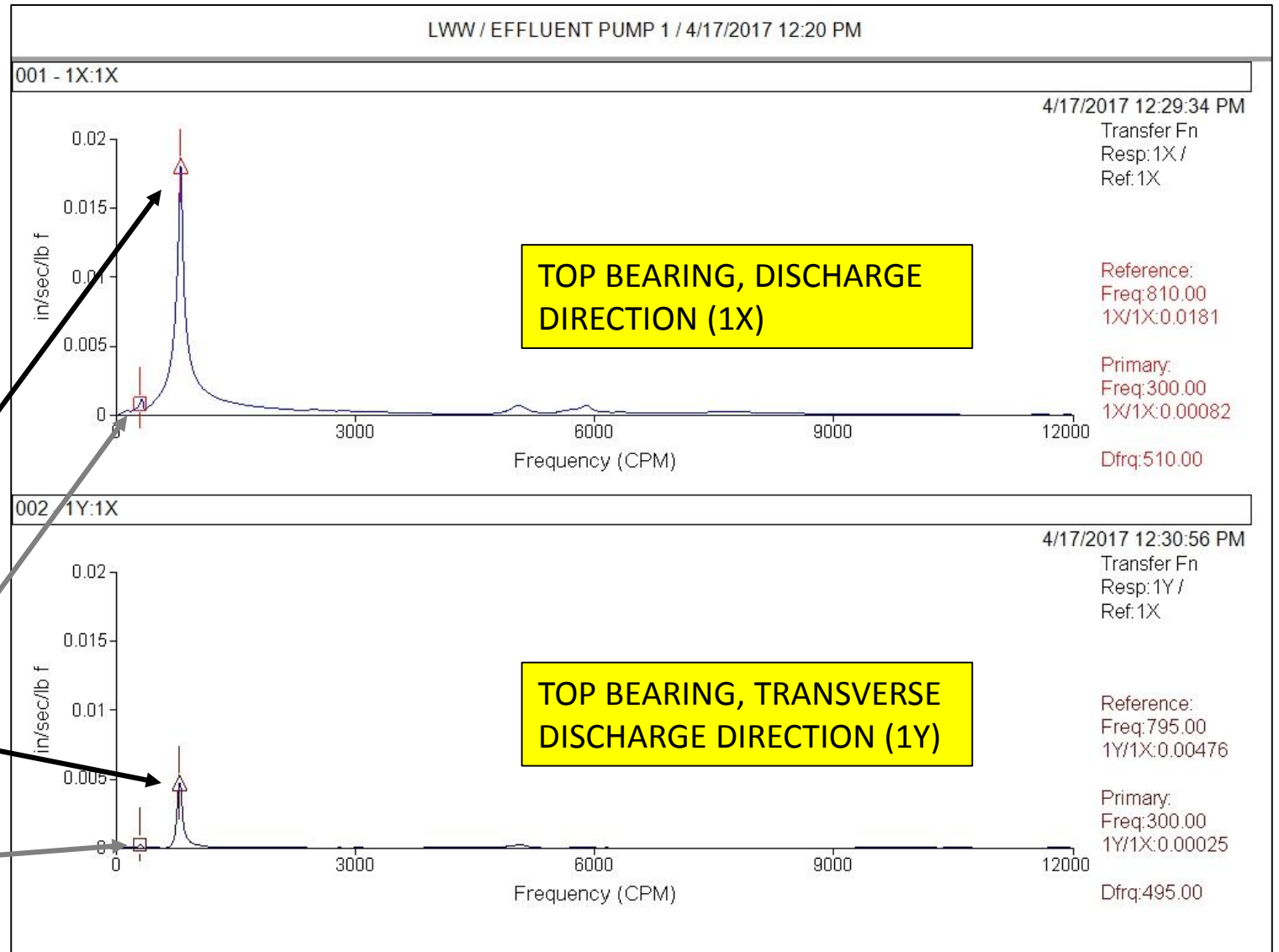
- The plot at right is from the same vertical pump at the same point (top bearing), but in different directions (discharge direction, 1X & transverse discharge direction, 1Y).
- **Note how stiffness at the same location or point varies with direction (lbf/ips-pk).**

• Stiffness 1X (810 cpm) = 55.6

• Stiffness 1Y (810 cpm) = 208

• Stiffness 1X (300 cpm) = 1,250

• Stiffness 1Y (300 cpm) = 4,000



WHEN STIFFNESS GOES BAD

- LOOSENESS
- RESONANCE

WHEN STIFFNESS GOES BAD - LOOSENESS

- Looseness between machine components can be thought of as a loss or deterioration of stiffness.
- This lack or loss of stiffness is commonly at a point along a machine especially affecting a certain direction (horizontal, vertical, or axial) or at a certain component.
- It often occurs when the fasteners or welds, etc that hold different machine components together fail or are undermined in some way.
- Common examples of looseness on machinery would include:
 - 1) A loose anchor bolt or base bolt at a machine base.
 - 2) A cracked weld within a machine frame.
 - 3) Compromised fits (bearing to housing, bearing to shaft, excessive bearing clearance, housing to machine frame, etc).
 - 4) Loose machine supports/structure includes structure supporting a machine as well as supporting connected piping, ducting, etc that the machine is attached to.
 - 5) Cracked shafts, frames, housings or other machine components.

IDENTIFYING LOOSENESS (LOSS OF STIFFNESS)

- When looseness is present, the stiffness of the machine is reduced, and the vibration (motion) at the machine will almost always increase.
- **Increases in vibration at the machine speed (1x rpm) are the most common, but increased vibration at any dynamic force present is possible when looseness is present.**
- Unlike resonance which is usually specific in the vibration frequency or region it amplifies, looseness can and often does cause a **general increase** in vibration at many frequencies.
- When looseness is present, our spectral vibration data will often show the appearance of multiples or harmonics of the running speed and of other sources of dynamic force.
- In addition to the vibration data collected, your presence on-site with your eyes, ears, and fingers is often key to nailing down **exactly** where and between what components the looseness exists. **Being there really matters if accuracy is desired.**
- Once looseness is suspected from your vibration data, use your analyzer as the “Geiger counter” in combination with your eyes, ears and fingers to better locate the exact source or location of the looseness.

IDENTIFYING LOOSENESS (LOSS OF STIFFNESS)

- Common vibration characteristics or symptoms when looseness (loss of stiffness) is present:
 - 1) Higher than normal 1x rpm vibration than in prior surveys.
 - 2) Higher than normal vibration at the vane-pass, lobe-pass, or gear-mesh frequency especially if such an increase is either **directional** or **localized** in nature.
 - 3) Higher than normal vibration at just about any other frequency of vibration especially when the vibration is directional in nature.
 - 4) The presence of vibration at multiples of the above frequencies or at 1/2x rpm or 1/3x rpm and multiples.
 - 5) Asymmetric motion in the waveform data.
 - 6) Clipping or impacting motion in the waveform data.
 - 7) A significant change in amplitude or phase at any vibration frequency with a small movement of the sensor (often between machine components).
 - 8) ODS analysis and Motion Amplification techniques are particularly adept at identifying **exactly** where looseness is located and between which two components, etc.

WHEN STIFFNESS GOES BAD - RESONANCE

- Beware that many of these symptoms of looseness can also be observed when a resonance is present.
- The most common difference between the effects of resonance versus looseness:
 - 1) Resonance **usually** causes significant amplification of vibration at one frequency whereas looseness can cause amplification at one or many frequencies.
 - 2) The increased amplitude of vibration due to resonance is usually more significant than that due to looseness.
 - 3) A significant phase shift approaching 180 deg is associated with resonance (below versus above natural frequency).
- Be aware also that a condition of looseness (loss of stiffness) can lead to resonance as the system natural frequency(s) that are excited in a state of resonance are highly dependent on the stiffness:

$$f_n = \sqrt{k/m}$$

CASE HISTORIES INVOLVING CHANGES IN STIFFNESS

CASE HISTORY 1

POSITIVE DISPLACEMENT PUMP

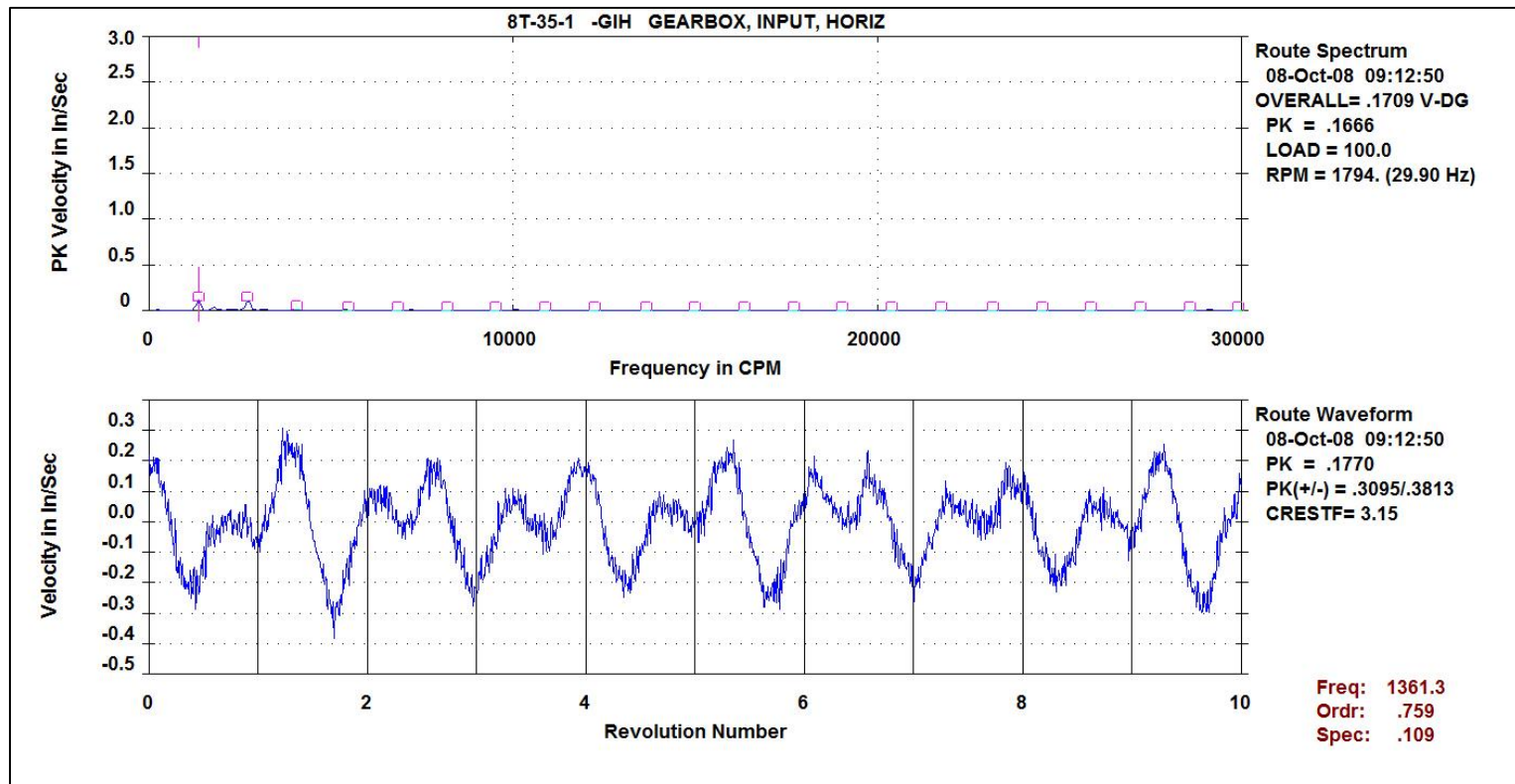
CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

- The first case history involves a horizontal positive displacement pump driven by a 4-pole induction motor thru a small gearbox.
- The motor operates at $\sim 1,790$ rpm while thru gearbox reduction the pump operates at ~ 120 rpm and generates vane-pass vibration at $\sim 1,350$ cpm (11x rpm).
- The entire machine is supported by a fabricated steel base bolted to the concrete floor.
- A history of looseness problems were experienced by this machine that will be described in the following slides.



CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

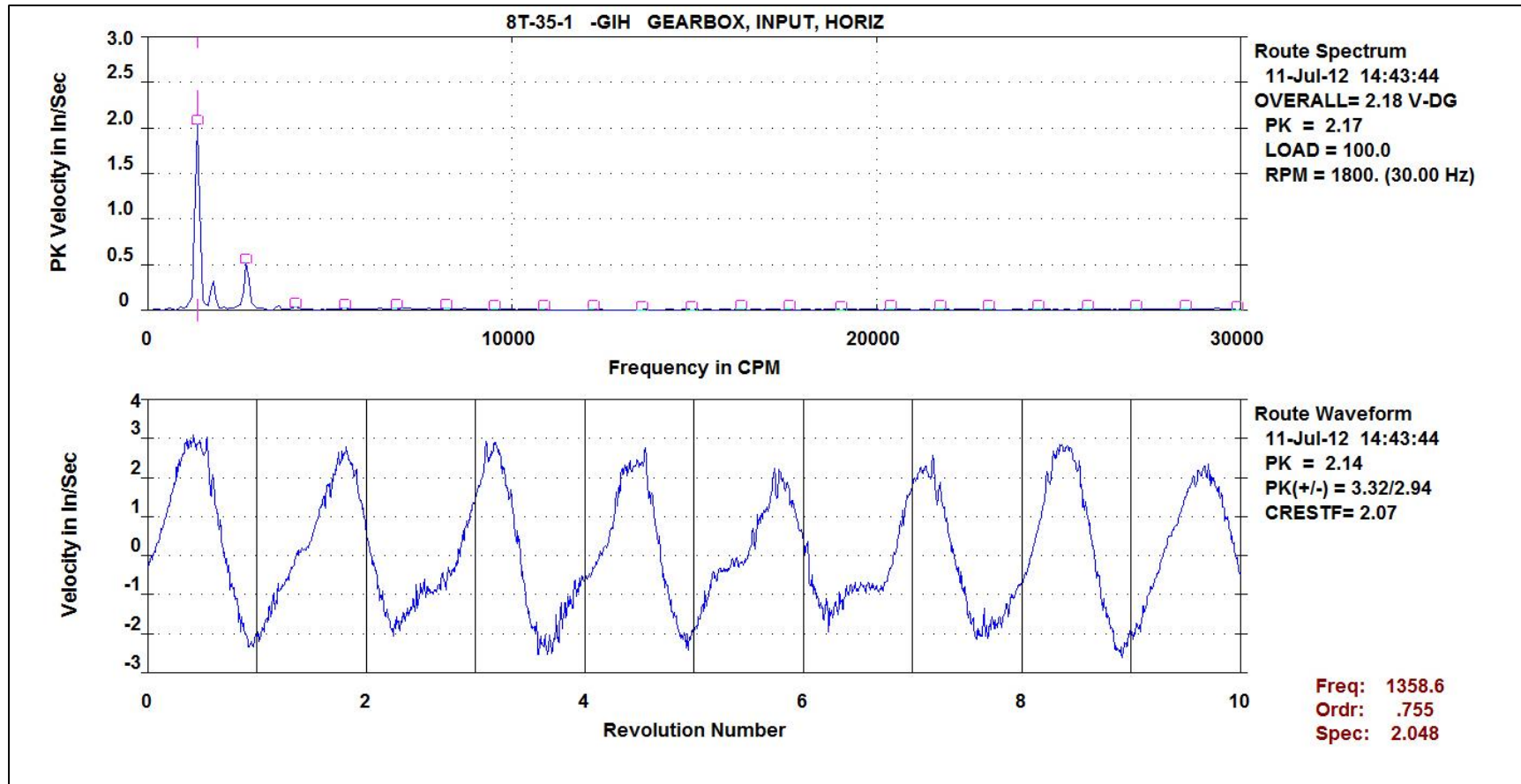
- 10/8/08 – Reference vibration data on pump.
- Spectral & waveform data shown at gearbox, horizontal point (representative point throughout machine history).



- Overall levels are low (~ 0.17 ips-pk) with no significant impacting seen in waveform data.
- Dominant frequency of vibration is at pump vane-pass frequency & 2x vane-pass (1,350 cpm & 2x).

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

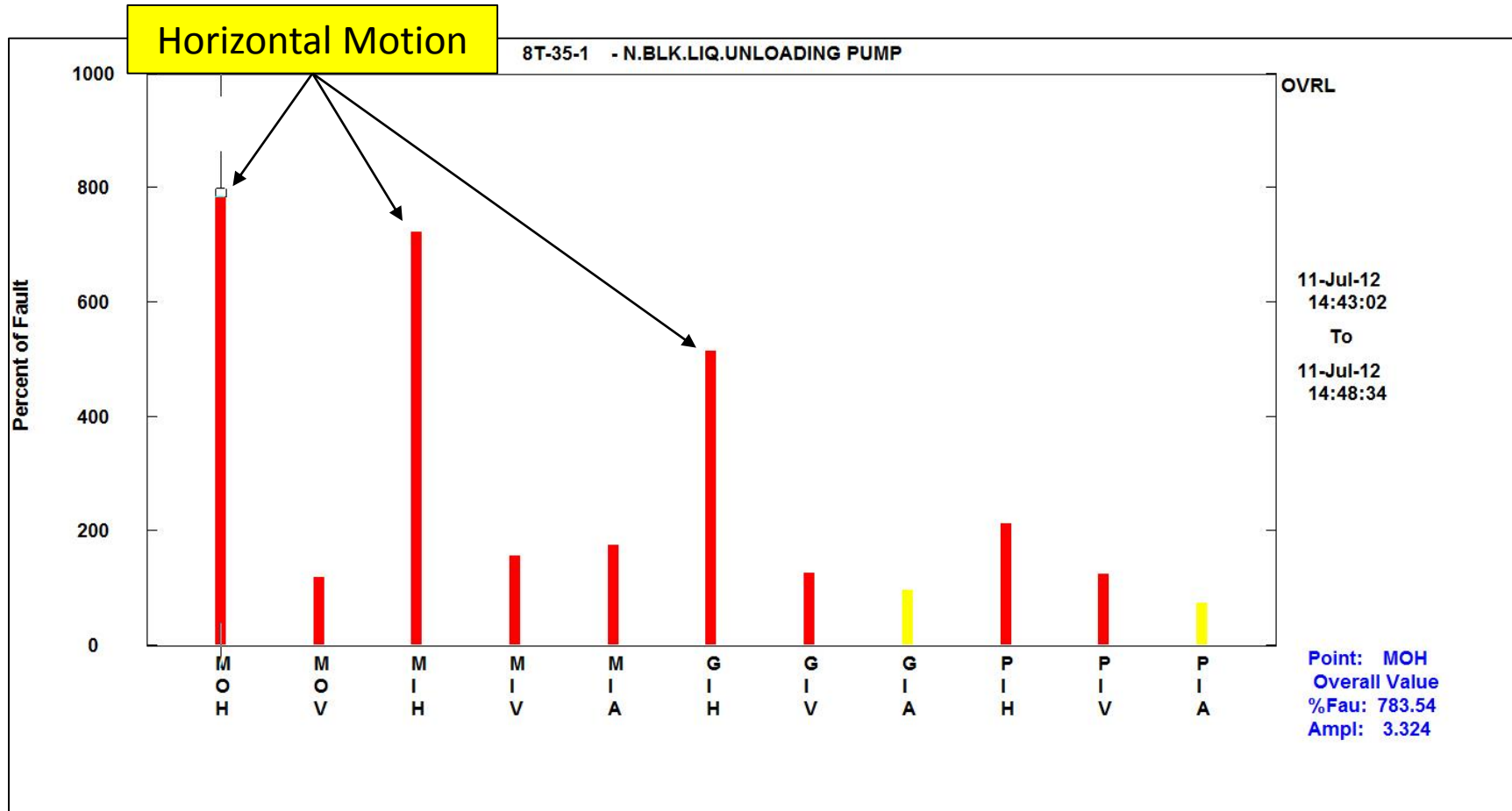
- 7/11/12 – Loose machine base & potential pump problems showing up at vane-pass.



- Overall levels are severe (~ 2.2 ips-pk).
- Dominant frequency of vibration is at pump vane-pass frequency & 2x vane-pass (1,350 cpm & 2x).
- Note how the effects of our loose base don't show up as high 1x rpm or 2x rpm, but instead at elevated levels of vane-pass (1,350 cpm).
- Vane-pass was the dominant dynamic force.

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

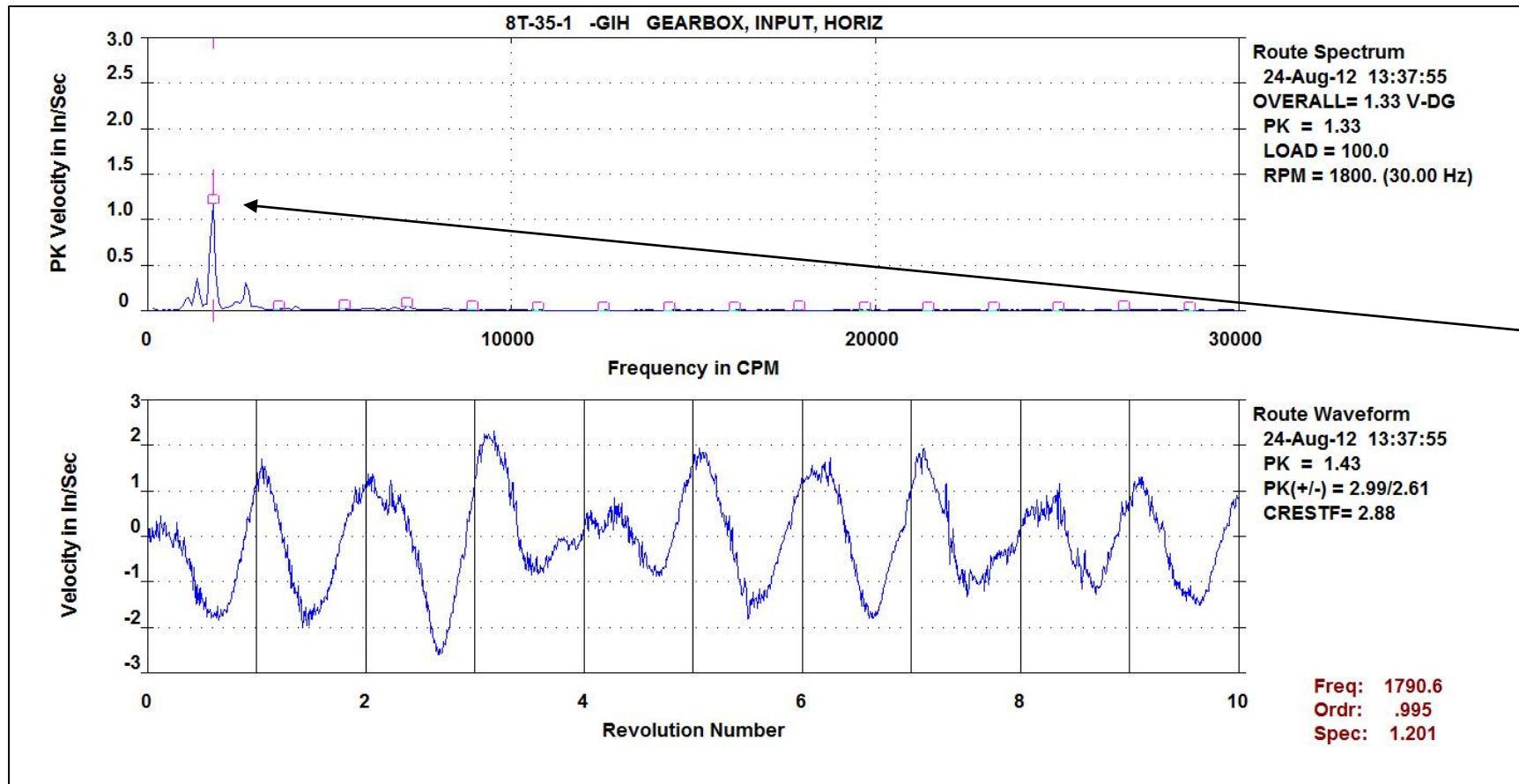
- 7/11/12 – Loose machine base & potential pump problems showing up at vane-pass.



- Overall levels are severe (~ 3.3 ips-pk).
- From the profile plot of overall vibration levels at left note how **directional** our vibration levels are.
- Dominant vibration occurring in the **horizontal direction** with highest levels at the motor.
- The source of the vane-pass frequency is at the pump where the vibration levels are lowest (PIH, PIV, etc).

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

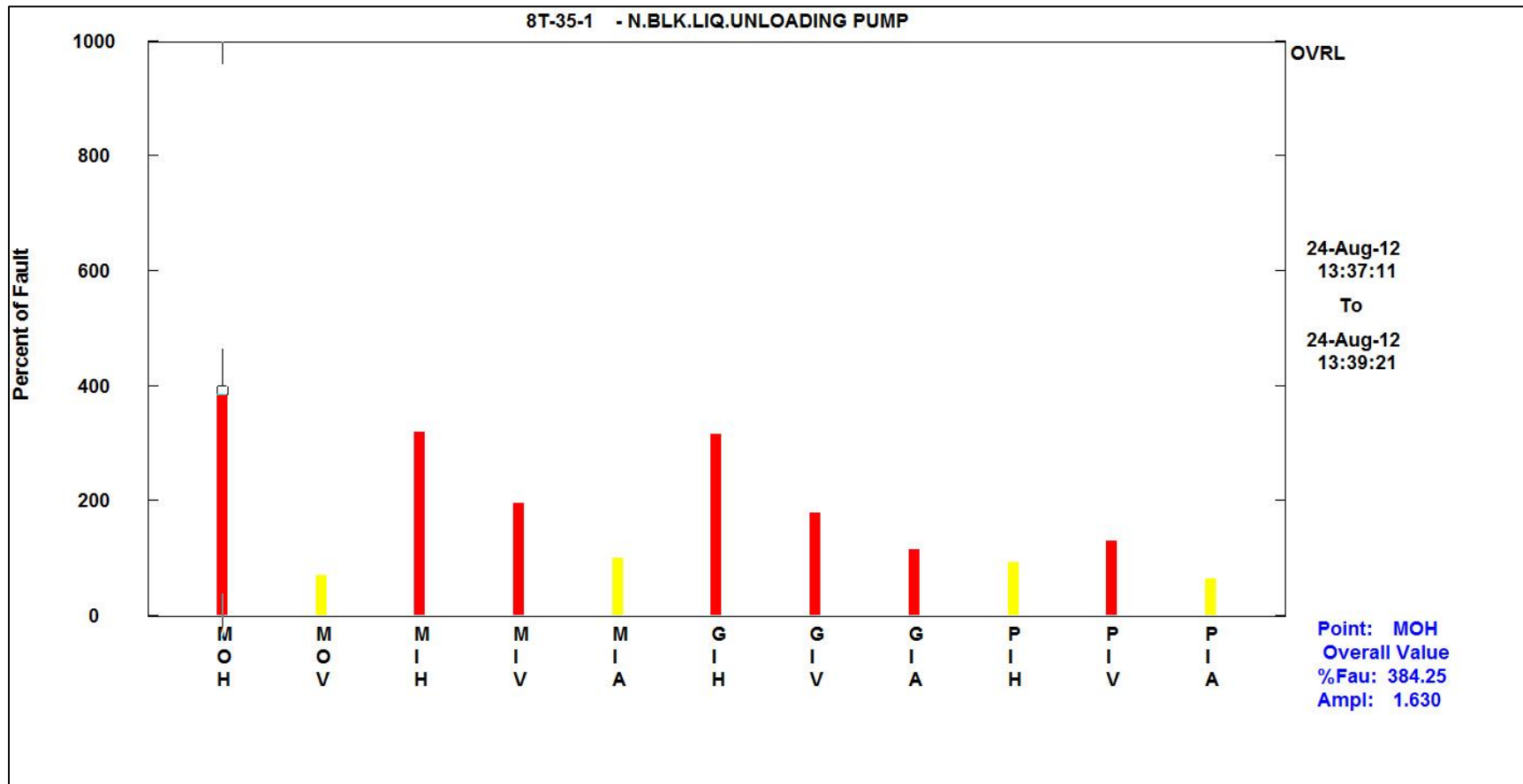
- 8/24/12 – Pump changed, but machine base problem remains.



- Overall levels are **lower** following the pump change (dynamic force reduced, but remain severe (~ 1.3 ips-pk)).
- Dominant frequency of vibration is now at **motor speed (1,790 rpm)**.
- Some level of alignment or coupling problems were also suspected.

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

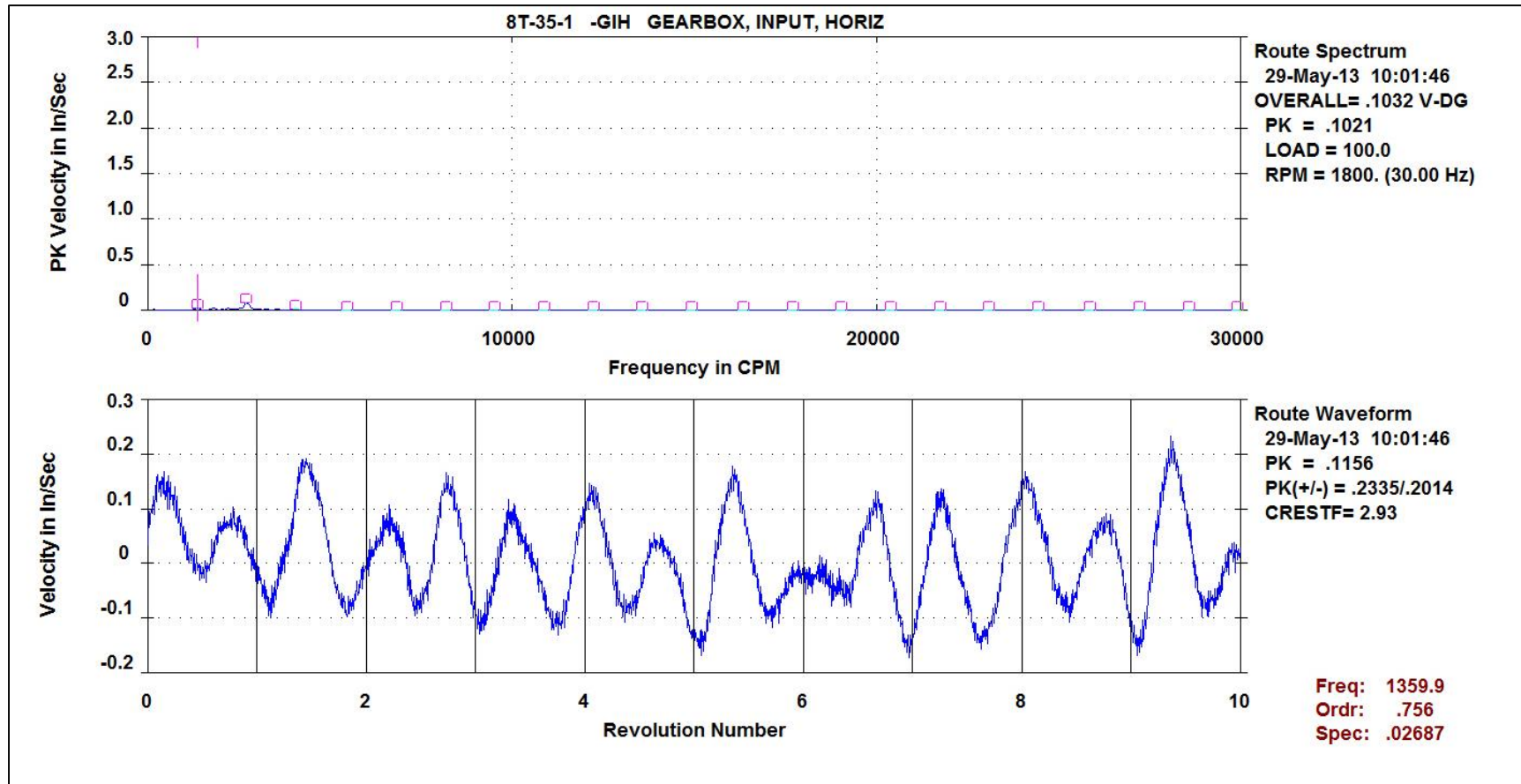
- 8/24/12 – Pump changed, but machine base problem remains.



- Overall levels are lower following the pump change (dynamic force reduced, but remain severe (~ 1.3 ips-pk)).
- Dominant frequency of vibration is now at motor speed (1,790 rpm).
- Horizontal levels remain highest, but not as dominant as before pump change.

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

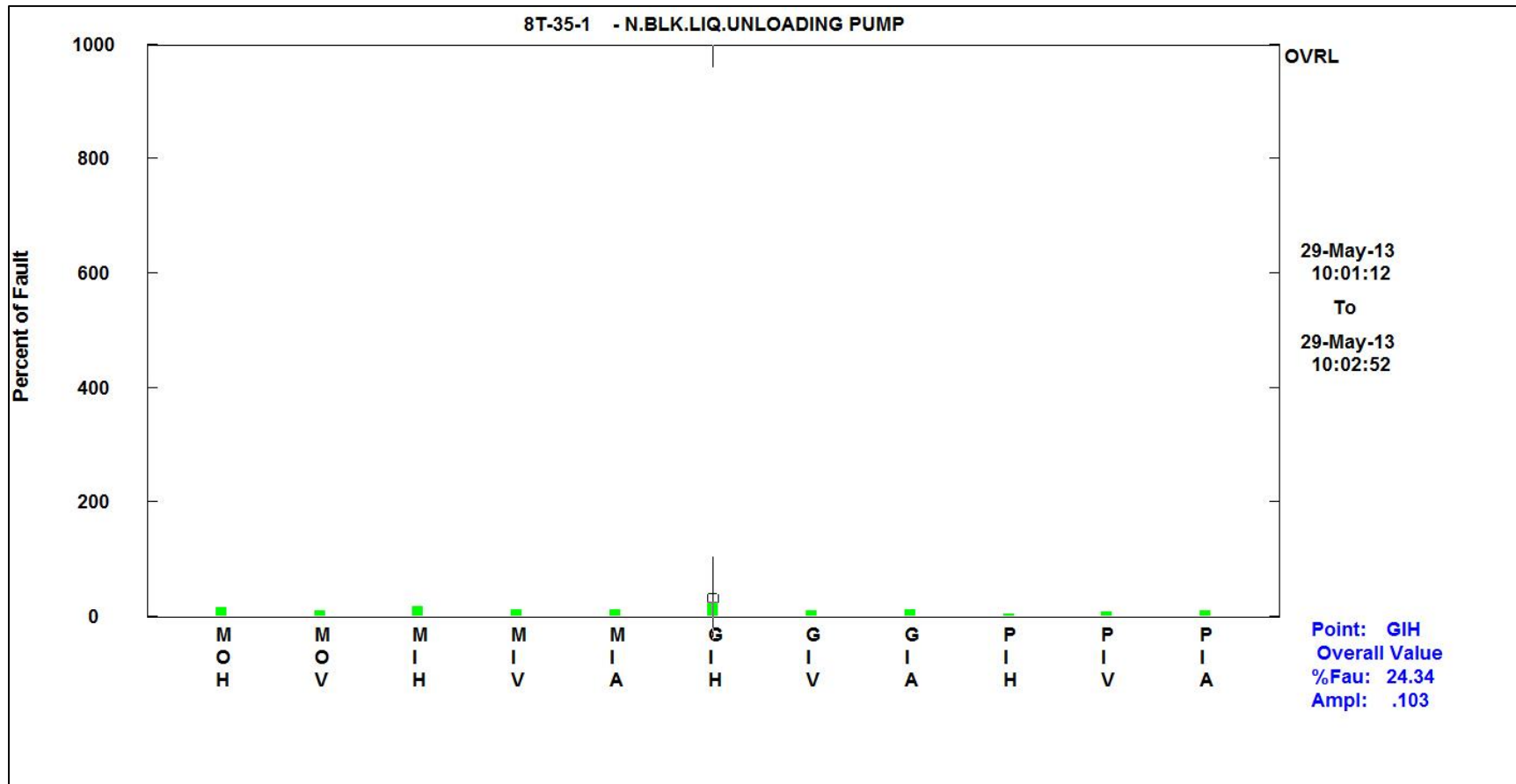
- 5/29/13 – Machine base repaired. All anchor bolts tightened up & base grouted into concrete floor.



- Overall levels are significantly lower following the machine base repairs (~ 0.10 ips).

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

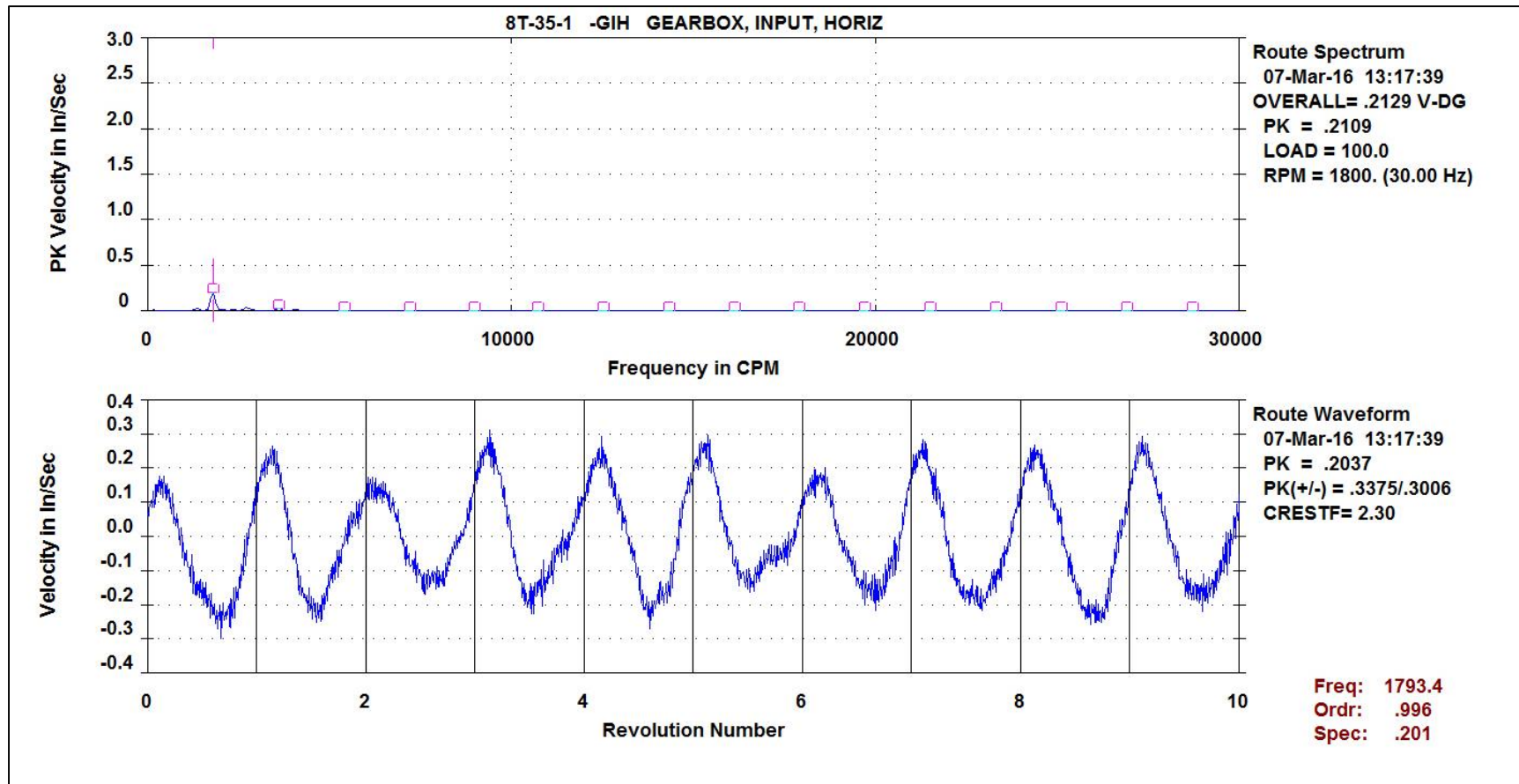
- 5/29/13 – Machine base repaired. All anchor bolts tightened up & base grouted into concrete floor.



- Overall levels are significantly lower following the machine base repairs (~ 0.10 ips).
- Note how following repairs to the machine base, vibration levels across the entire machine are significantly lower.
- Note also how now our horizontal levels don't stand out at all versus the others.

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

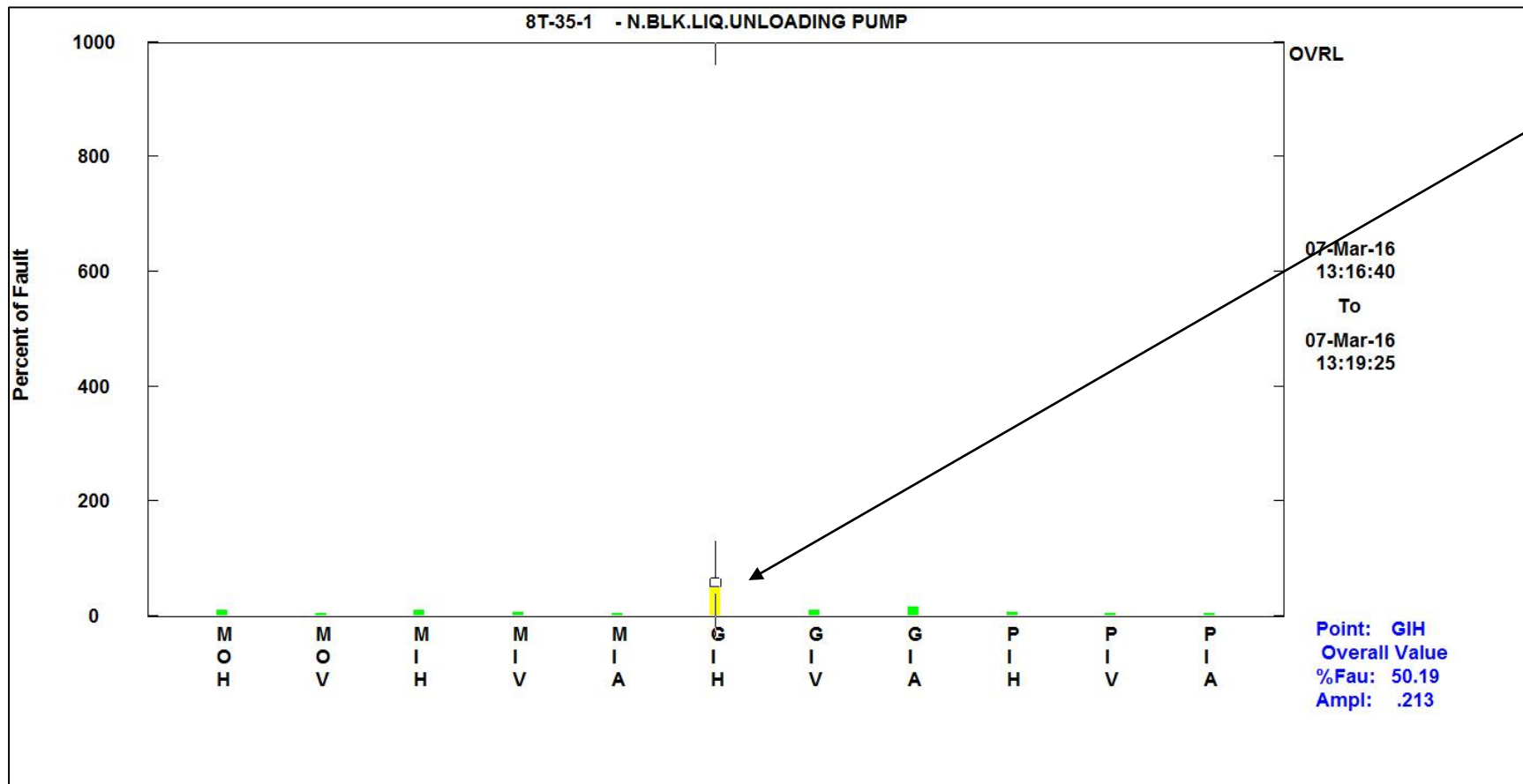
- 3/7/16 – Three years later, a new base problem begins (grout cracking and deteriorating at spots under machine base).



- Overall levels have doubled since machine base repairs (~ 0.21 ips).

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

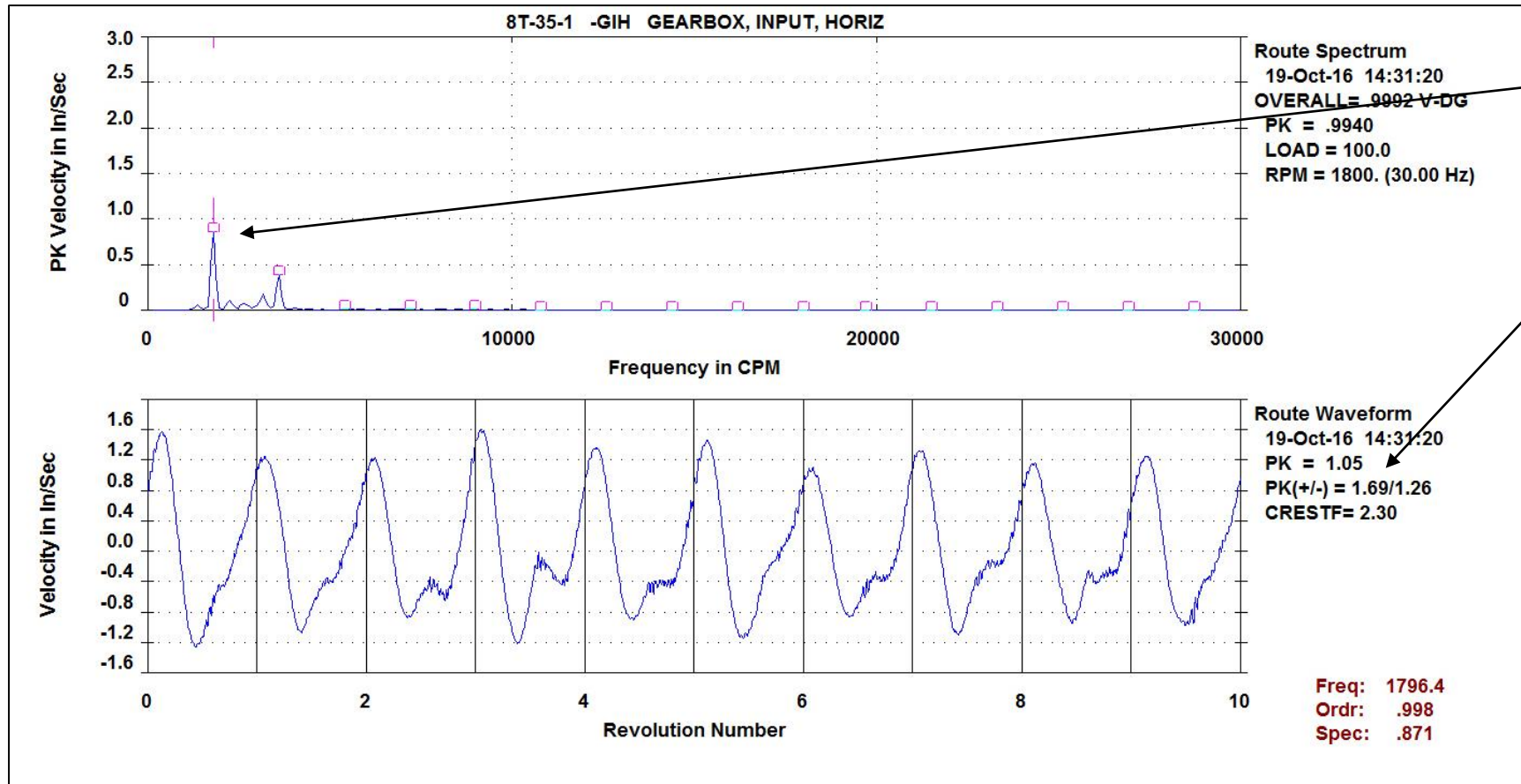
- 3/7/16 – Three years later, a new base problem begins (grout cracking and deteriorating at spots under machine base).



- Note how increase in vibration levels is occurring primarily at one measurement (**localized**).
- Levels at the gearbox, input, horizontal measurement (GIH) are elevated while all others remain ok.
- Remember that a characteristic of both looseness and resonance is **directional** vibration.

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

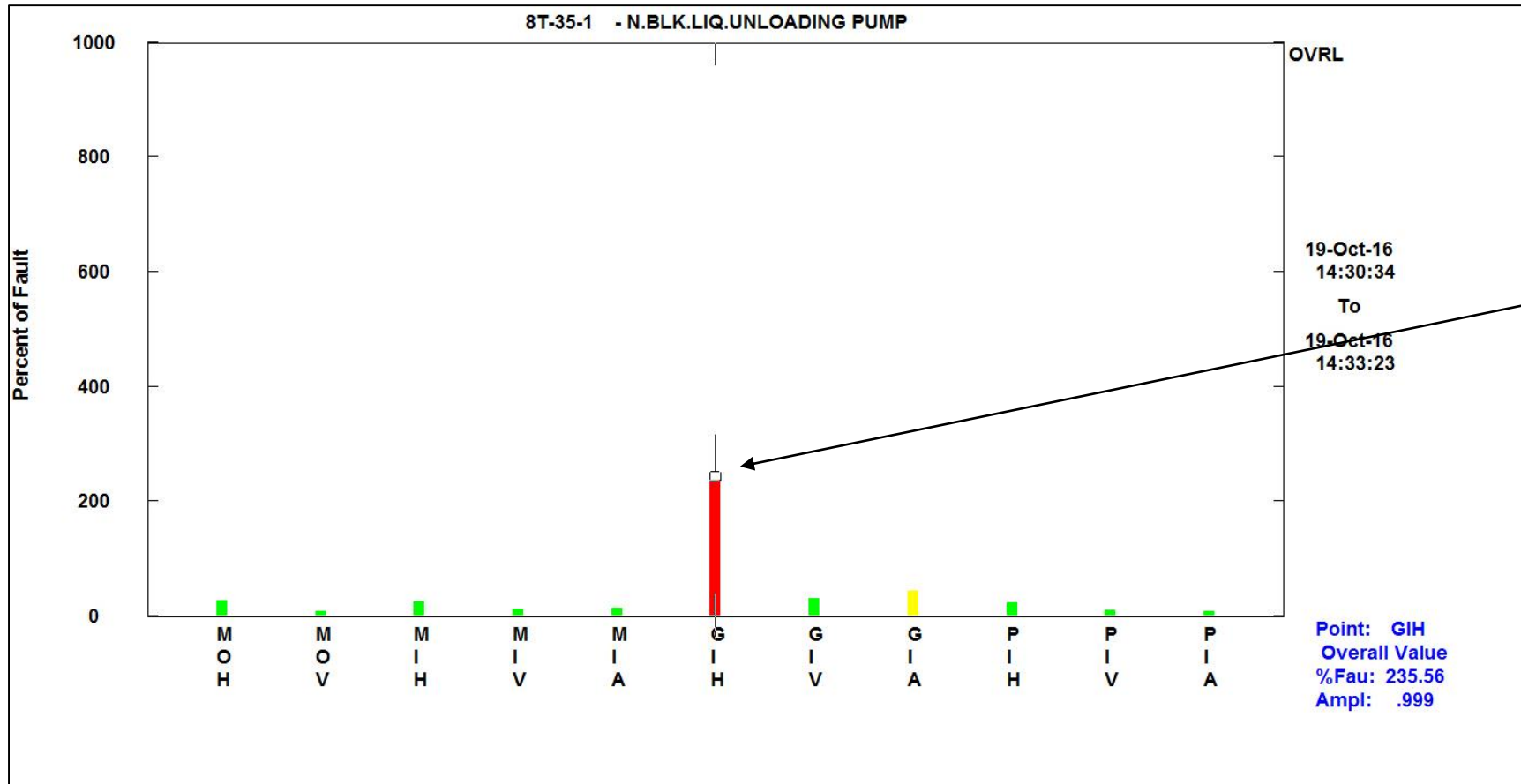
- 10/19/16 – New base problem continues to deteriorate.



- Overall levels continuing to climb (~ 1 ips).
- Dominant vibration occurring at motor 1x & 2x rpm.
- Asymmetric waveform (+1.69 versus -1.26).
- **On-site inspections noted significant movement and flexure of the machine base at or near the gearbox with much less movement at the machine base near either the motor or pump – being there matters.**

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

- 10/19/16 – New base problem continues to deteriorate.



- From this profile plot of the overall levels versus machine measurement point, note how the vibration is very **localized**.
- Majority of vibration is isolated at the gearbox in the horizontal direction (GIH).
- Further inspection found the grout had cracked and deteriorated under the machine base at the gearbox location reducing the stiffness at this spot (not generally) along the machine.

CASE HISTORY 1 – POSITIVE DISPLACEMENT PUMP

- This case history does a good job at illustrating two characteristics of looseness:
 - 1) Looseness is often **directional** in nature
 - 2) Looseness is often **localized** in nature.
- It also underscores the importance a solid foundation and machine base has on your machine's vibration levels.

CASE HISTORY 2

BELT-DRIVEN SCREEN

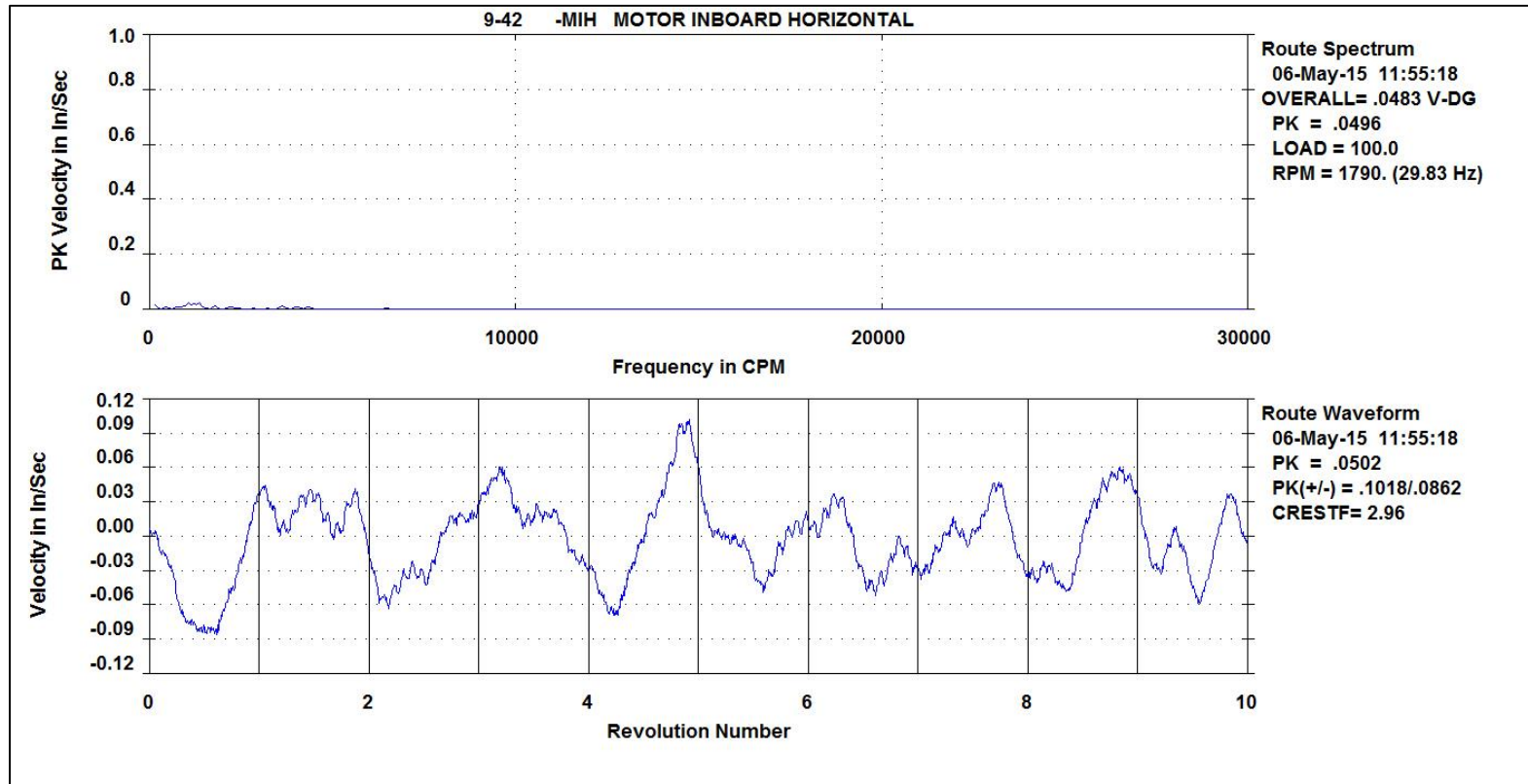
CASE HISTORY 2 – BELT-DRIVEN SCREEN



- The second case history involves a horizontal screen driven by a 4-pole induction motor thru a belt-drive.
- The motor operates at $\sim 1,780$ rpm while the screen operates at $\sim 2,000$ rpm.
- The entire machine is supported by a fabricated steel base bolted to the concrete floor via 4-ea anchor bolts (no grout).
- Just looking at the side and end-on views of this machine ask yourself in which direction is the stiffness high and low.
- Taking into account that fact that this machine is essentially a centrifuge generating high radial forces, where and in what direction might the vibration levels be high?

CASE HISTORY 2 – BELT-DRIVEN SCREEN

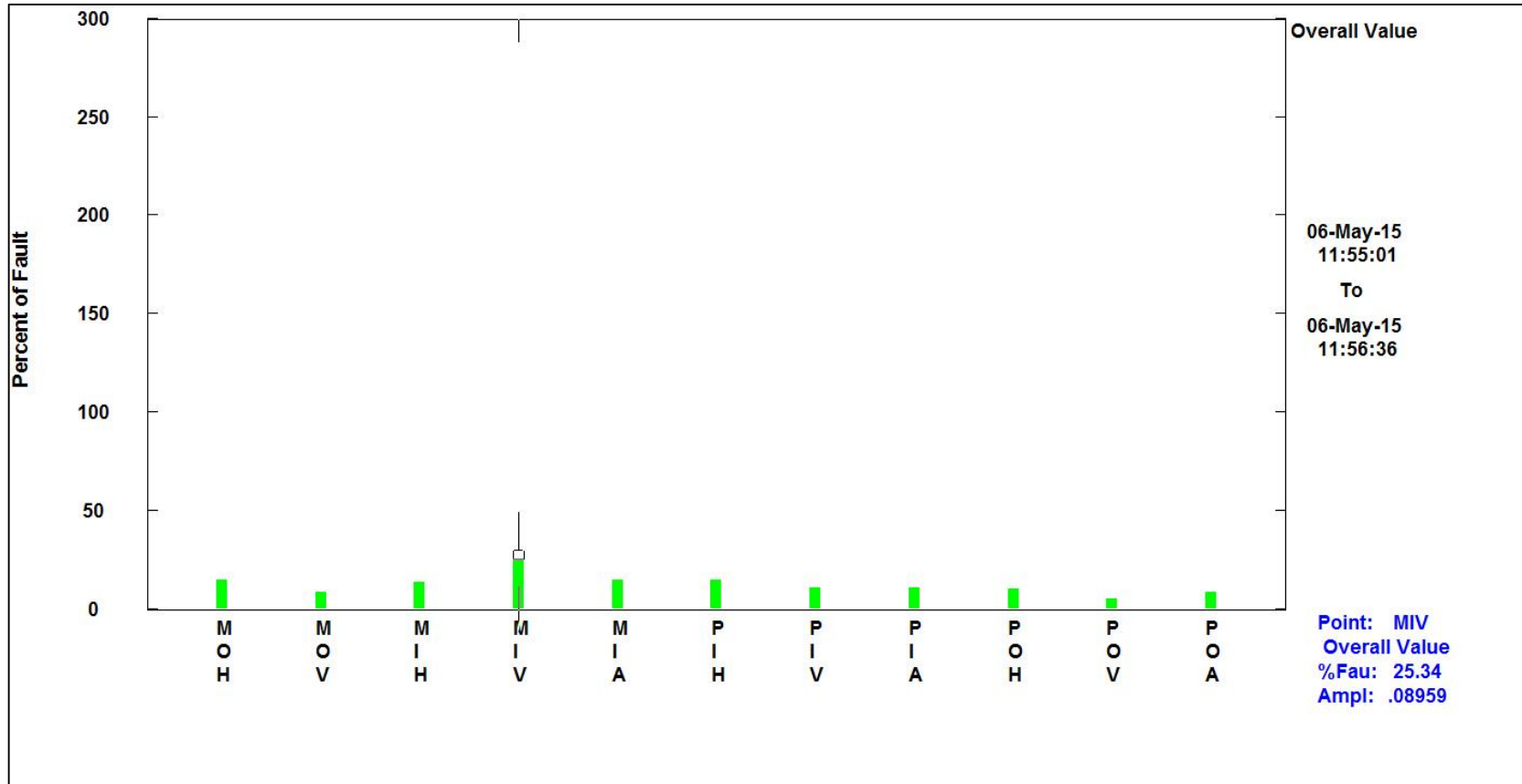
- 5/6/015 – Reference vibration data on screen.
- Spectral & waveform data shown at motor, inboard, horizontal point (representative point thru machine history).



- Overall levels are very low (~ 0.05 ips-pk).

CASE HISTORY 2 – BELT-DRIVEN SCREEN

- 5/6/015 – Reference vibration data on screen.

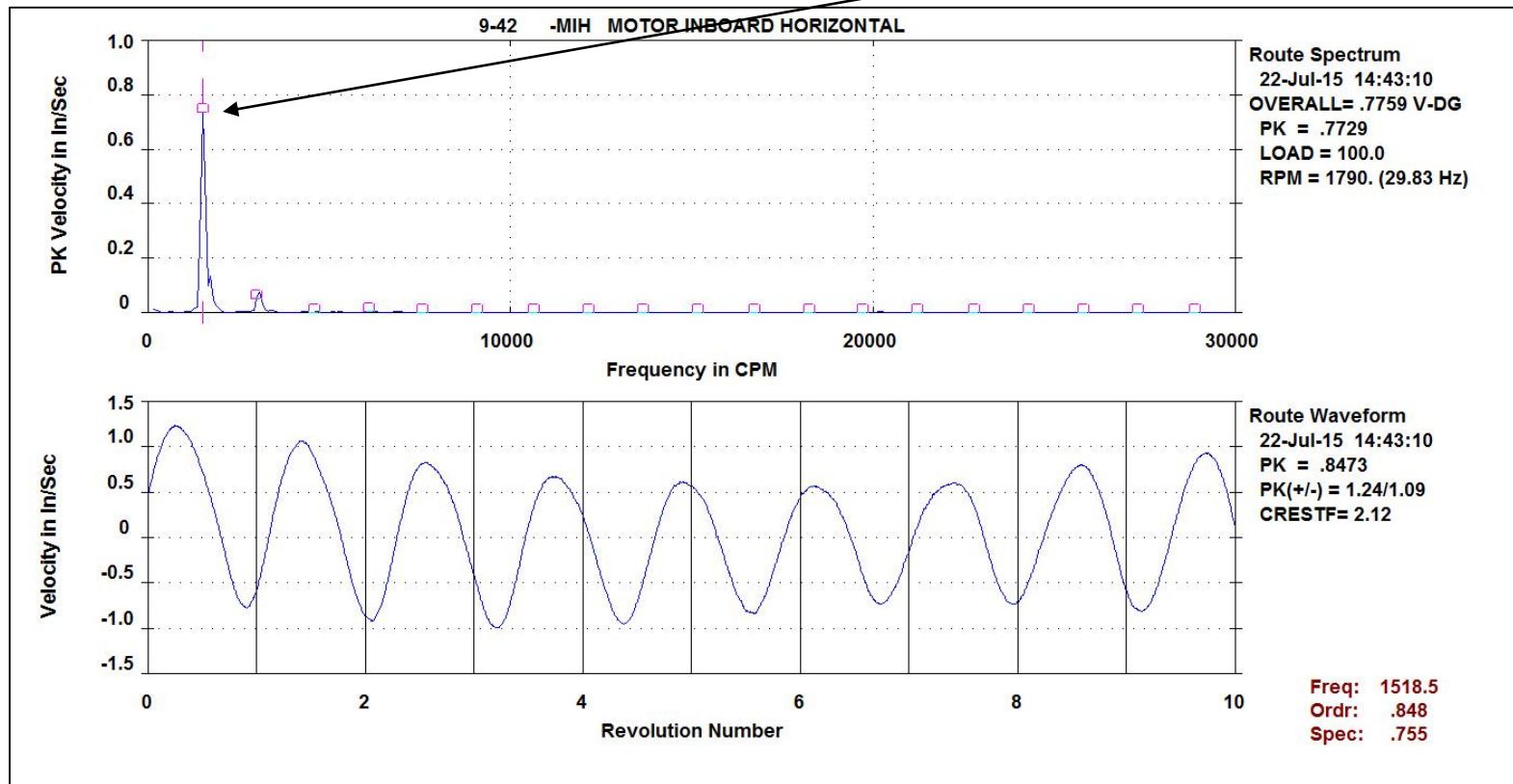


- Overall levels are very low with highest level at motor, inboard, vertical (MIV) @ 0.09 ips-pk.

CASE HISTORY 2 – BELT-DRIVEN SCREEN

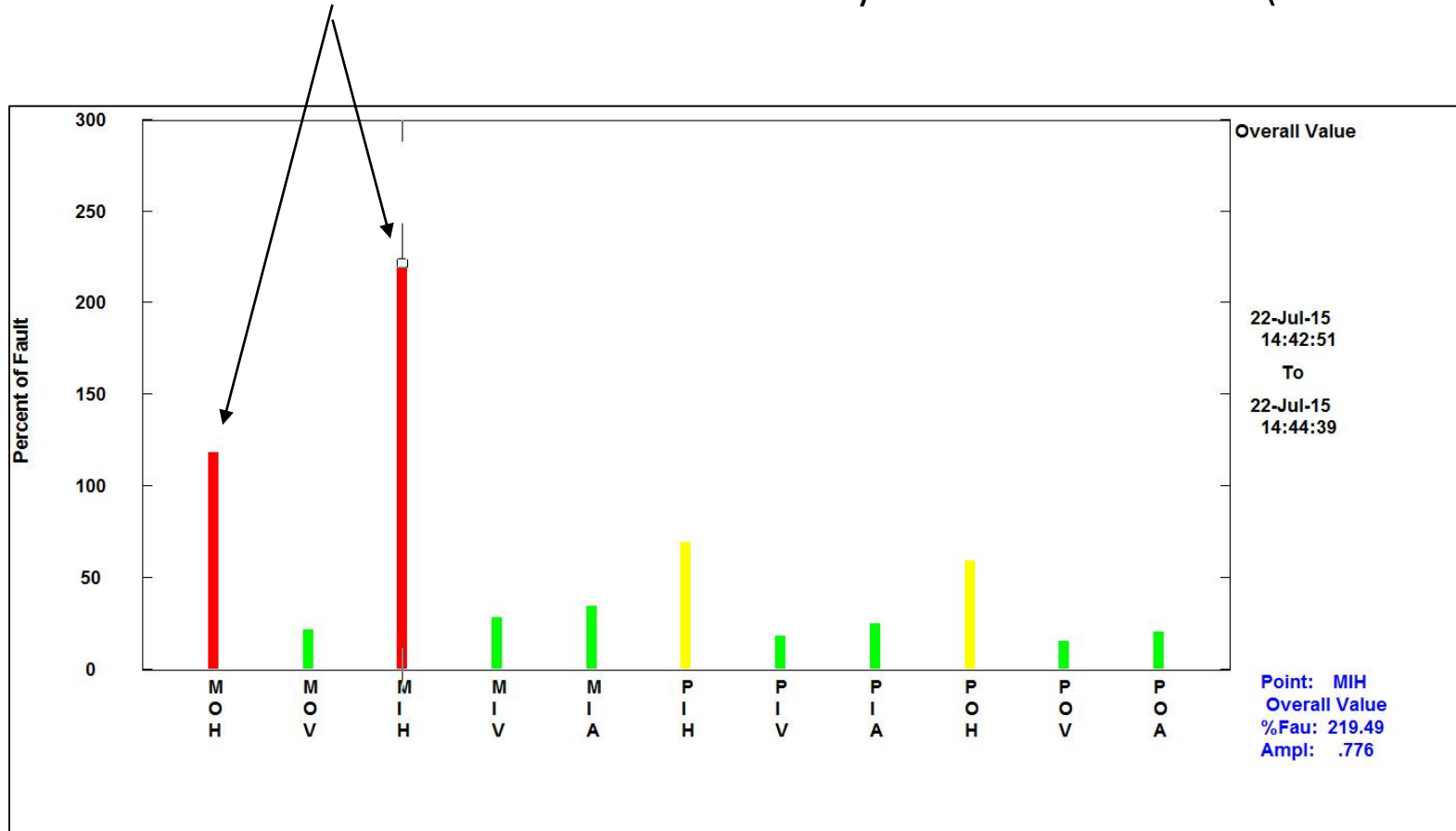
- 7/22/15 – Loose anchor bolt(s) at machine base.
- Significantly higher vibration levels than baseline (0.77 ips-pk).

- Dominant vibration is at $\sim 1,520$ cpm.
- The motor speed is $\sim 1,790$ rpm while the screen speed is $\sim 2,000$ rpm.
- The source of vibration at $\sim 1,520$ cpm was unknown – resonance was suspected with excitation from the motor speed and/or a multiple of belt frequency.
- On-site inspection following collection of the vibration data identified the loose anchor bolts – **being there matters.**
- At least one of the anchor bolts was loose in the concrete.



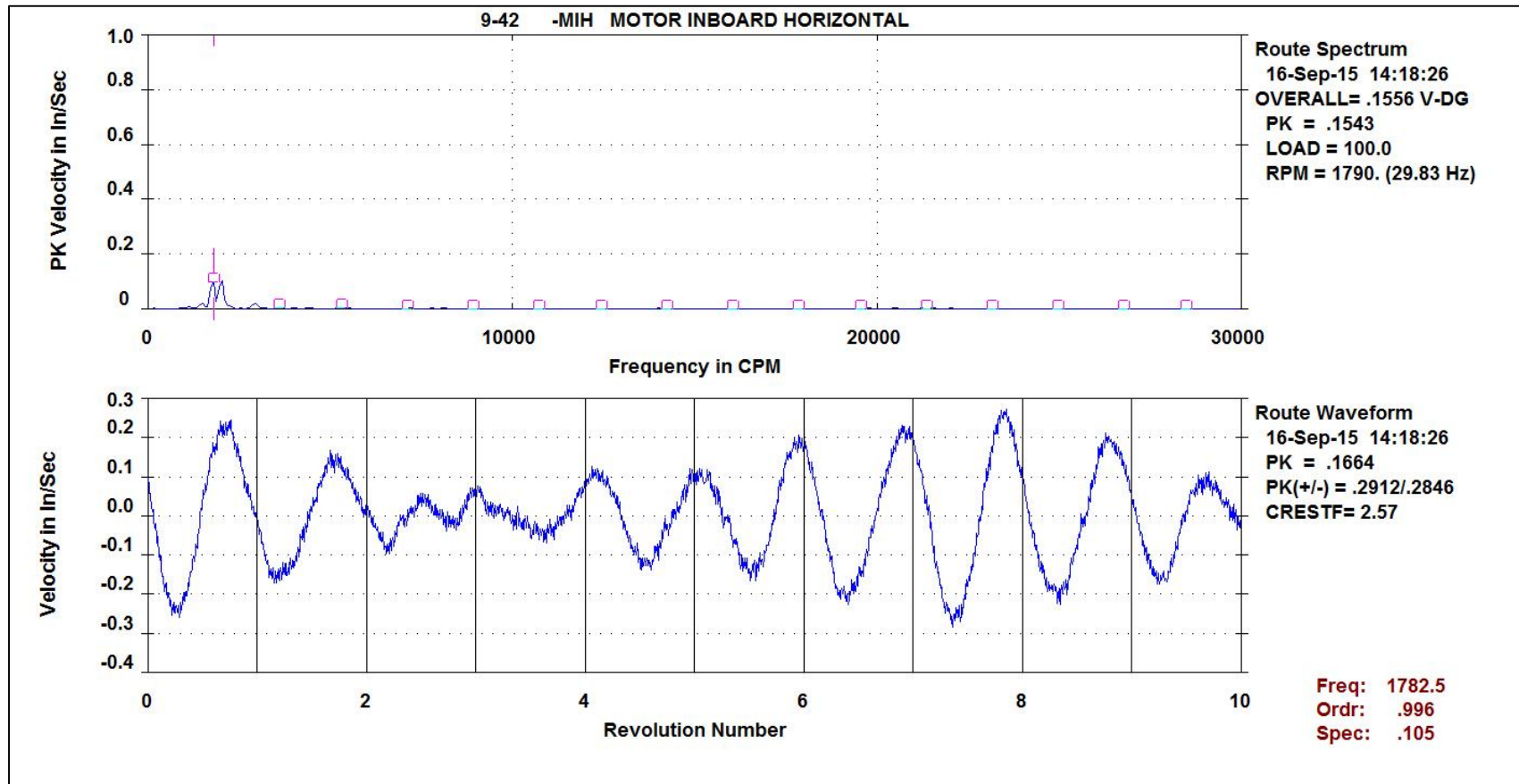
CASE HISTORY 2 – BELT-DRIVEN SCREEN

- 7/22/015 – Loose anchor bolt(s) at machine base.
- Profile of overall vibration levels shows very **directional** vibration (horizontal motion).



- Highest levels at the motor, inboard, horizontal (MIH).
- Much lower levels in both the vertical & axial directions.
- On-site inspection following collection of the vibration data identified the loose anchor bolts.
- At least one of the anchor bolts was loose in the concrete.

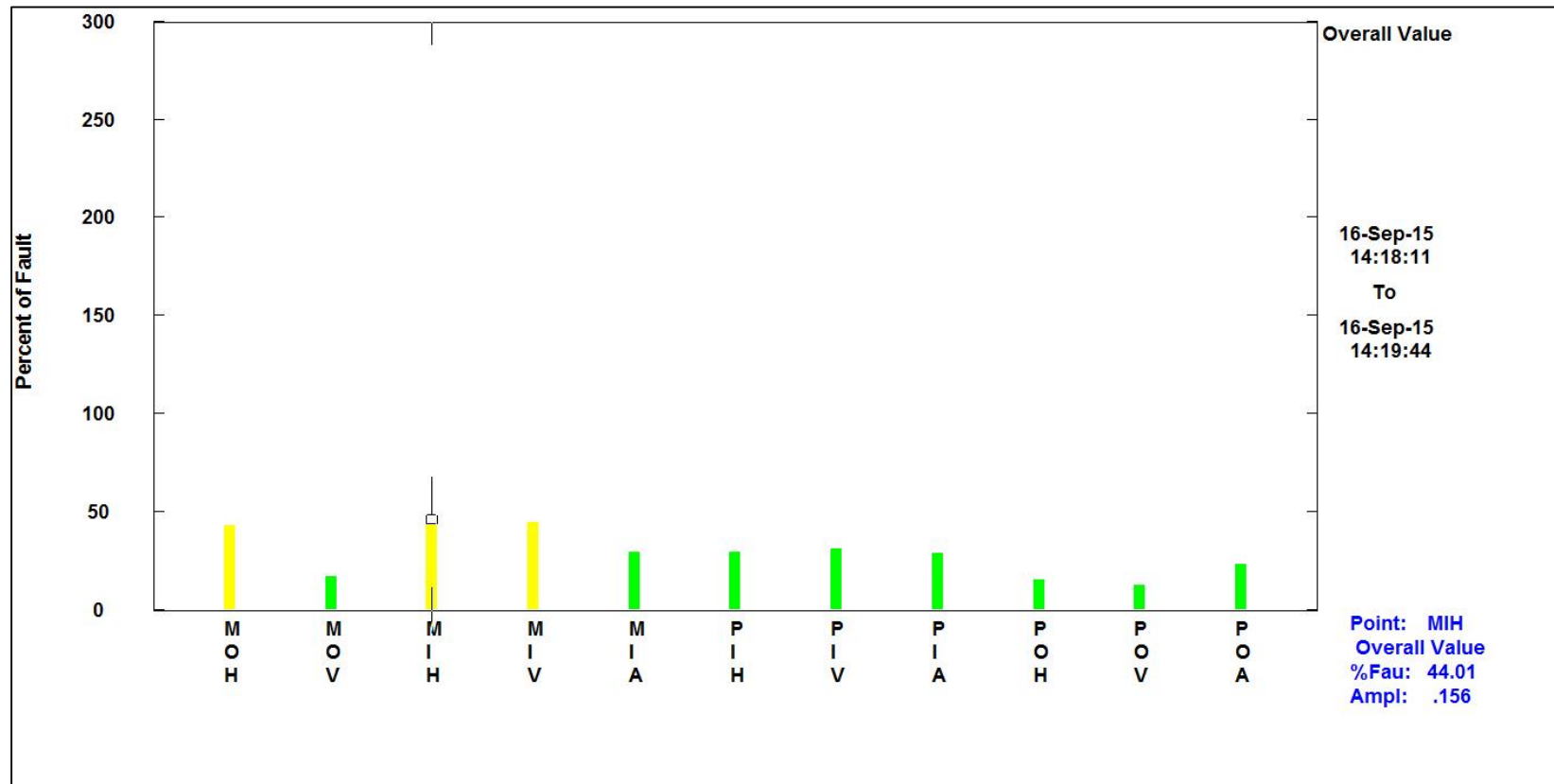
CASE HISTORY 2 – BELT-DRIVEN SCREEN



- 9/16/15 – All anchor bolts tightened up.
- Significant drop in overall vibration levels (0.77 ips → 0.16 ips) and elimination of prior vibration at ~ 1,520 cpm..

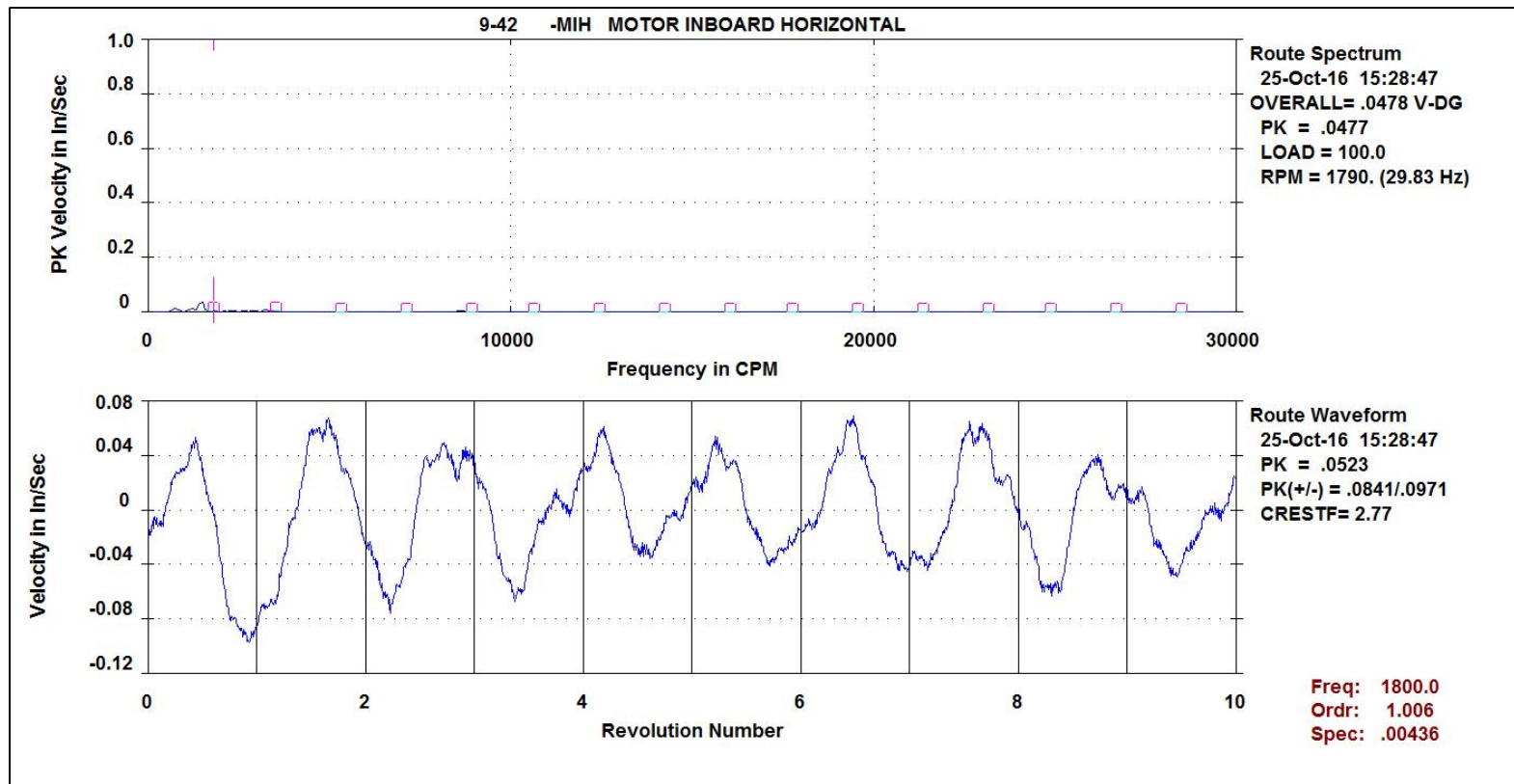
CASE HISTORY 2 – BELT-DRIVEN SCREEN

- 9/16/15 – All anchor bolts tightened up.
- Profile plot of overall vibration levels shows highest level at 0.16 ips-pk and directional vibration eliminated.



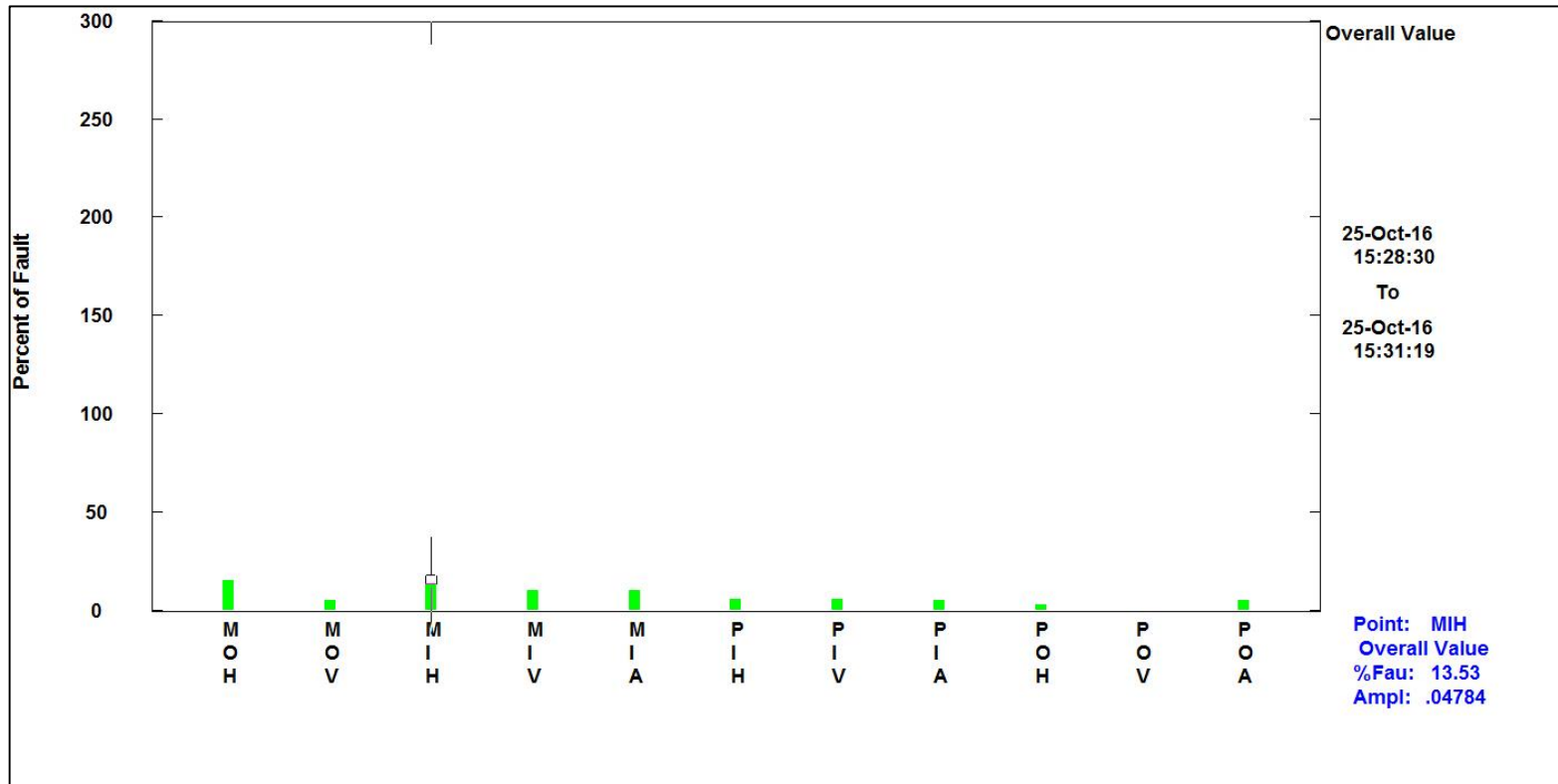
CASE HISTORY 2 – BELT-DRIVEN SCREEN

- 10/25/16 – All anchor bolts tightened up again and double-nutted to resist loosening.
- Drop in OA levels again: 0.16 ips \rightarrow 0.05 ips.



CASE HISTORY 2 – BELT-DRIVEN SCREEN

- 10/25/16 – All anchor bolts tightened up again and double-nutted to resist loosening.
- Highest level now at 0.05 ips-pk.



CASE HISTORY 2 – BELT-DRIVEN SCREEN

- As the only repairs known to be performed were cleaning of the screen and tightening of the anchor bolts, the question remained – what was the source of dominant vibration at $\sim 1,520$ cpm when the anchor bolts were loose.
- My suspicion was the elevated vibration at 1,520 cpm was due to excitation of a natural frequency in the horizontal direction set up by the reduction in stiffness from the loose anchor bolts as simply tightening the anchor bolts eliminated the problem.
- The suspected natural frequency (1,520 cpm) was close enough to either the motor speed or a multiple of belt frequency for excitation by either of these forces to be possible.
- **The moral of this story is that one loose anchor bolt on your machinery can result in major increases in its vibration levels due to the significant changes in stiffness that occur. The worst of both worlds can set up – looseness causing resonance.**

CASE HISTORY 3

BELT-DRIVEN FAN 1

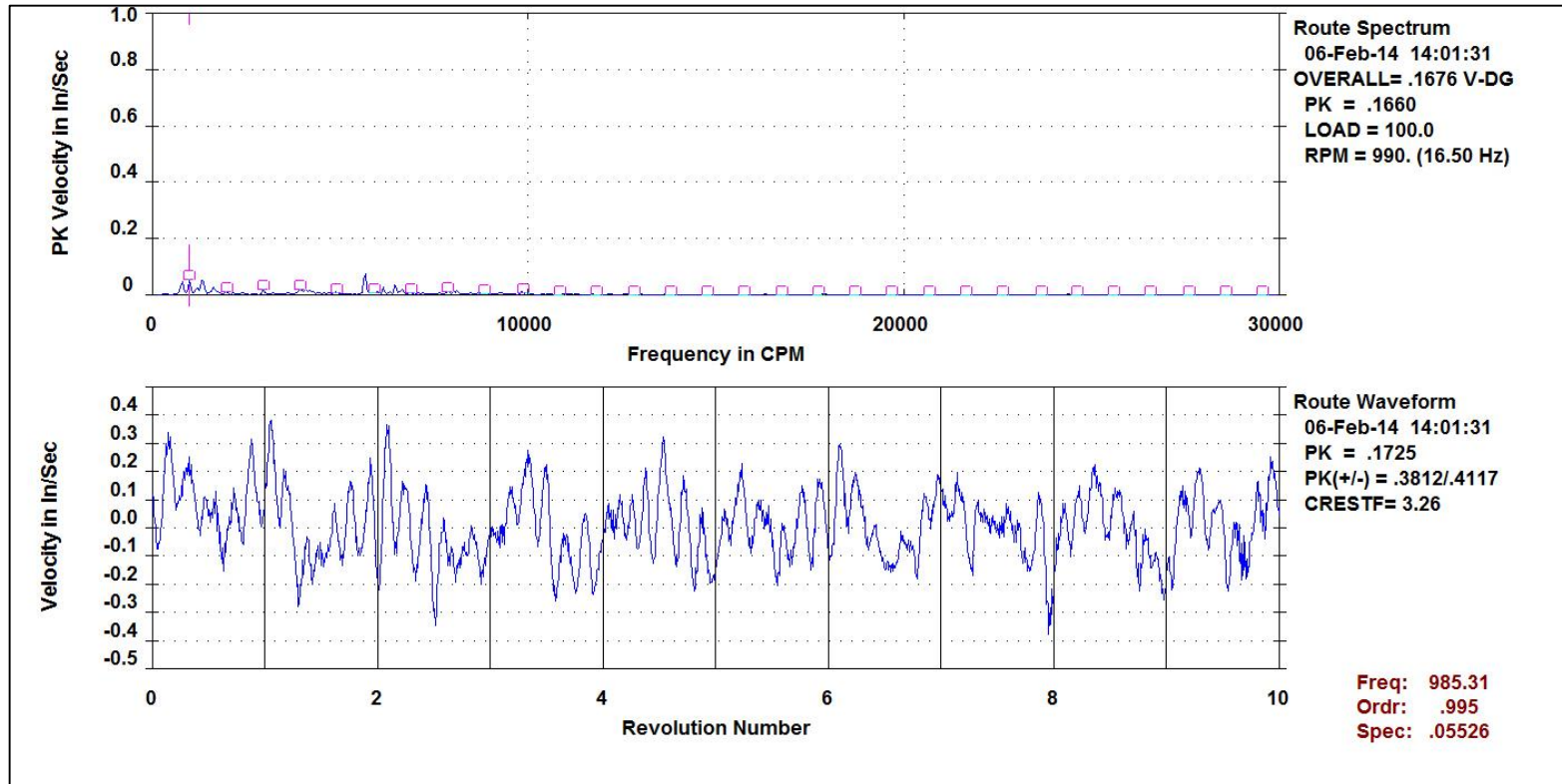
CASE HISTORY 3 – BELT-DRIVEN FAN

- The third case history involves a centrifugal fan driven by a 4-pole induction motor thru a belt-drive.
- The motor operates at $\sim 1,780$ rpm while the fan operates at $\sim 1,000$ rpm.
- The fan is supported by a fabricated steel base bolted to the concrete floor via 6-ea anchor bolts.
- High vibration at the fan speed was occasionally encountered at this machine, but a good cleaning of the fan usually reduced levels to a satisfactory level.
- In the following case history, more than just normal buildup on the fan wheel was at work.



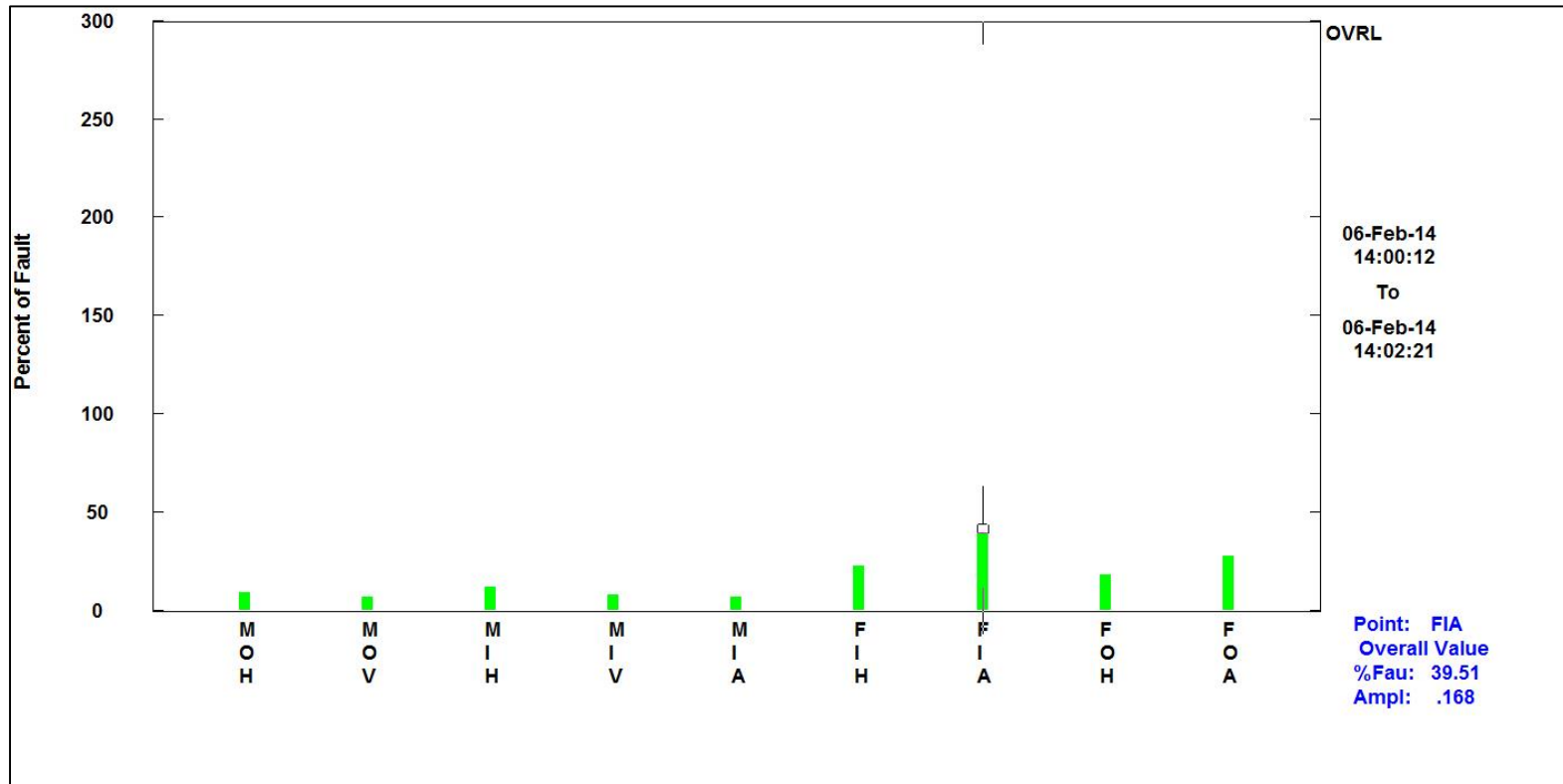
CASE HISTORY 3 – BELT-DRIVEN FAN

- 2/6/14 – Reference vibration data on fan. Low and acceptable vibration levels (0.17 ips-pk).
- Spectral & waveform data shown at fan, inboard, axial point (representative point thru machine history).



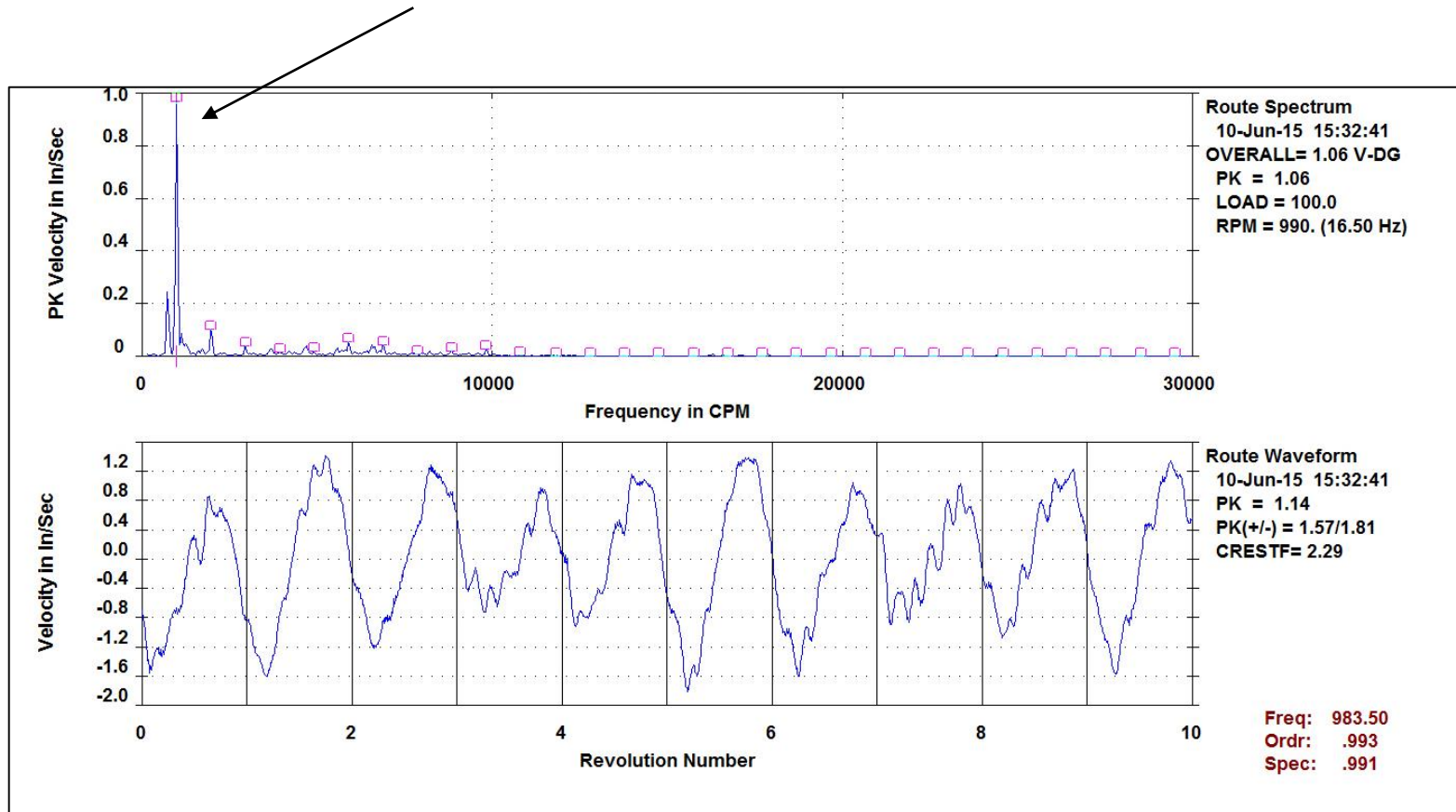
CASE HISTORY 3 – BELT-DRIVEN FAN

- 2/6/14 – Reference vibration data on fan. Low and acceptable vibration levels (0.17 ips-pk).
- Parameter profile of overall vibration levels shows fan, inboard, axial point highest at 017 ips-pk.



CASE HISTORY 3 – BELT-DRIVEN FAN

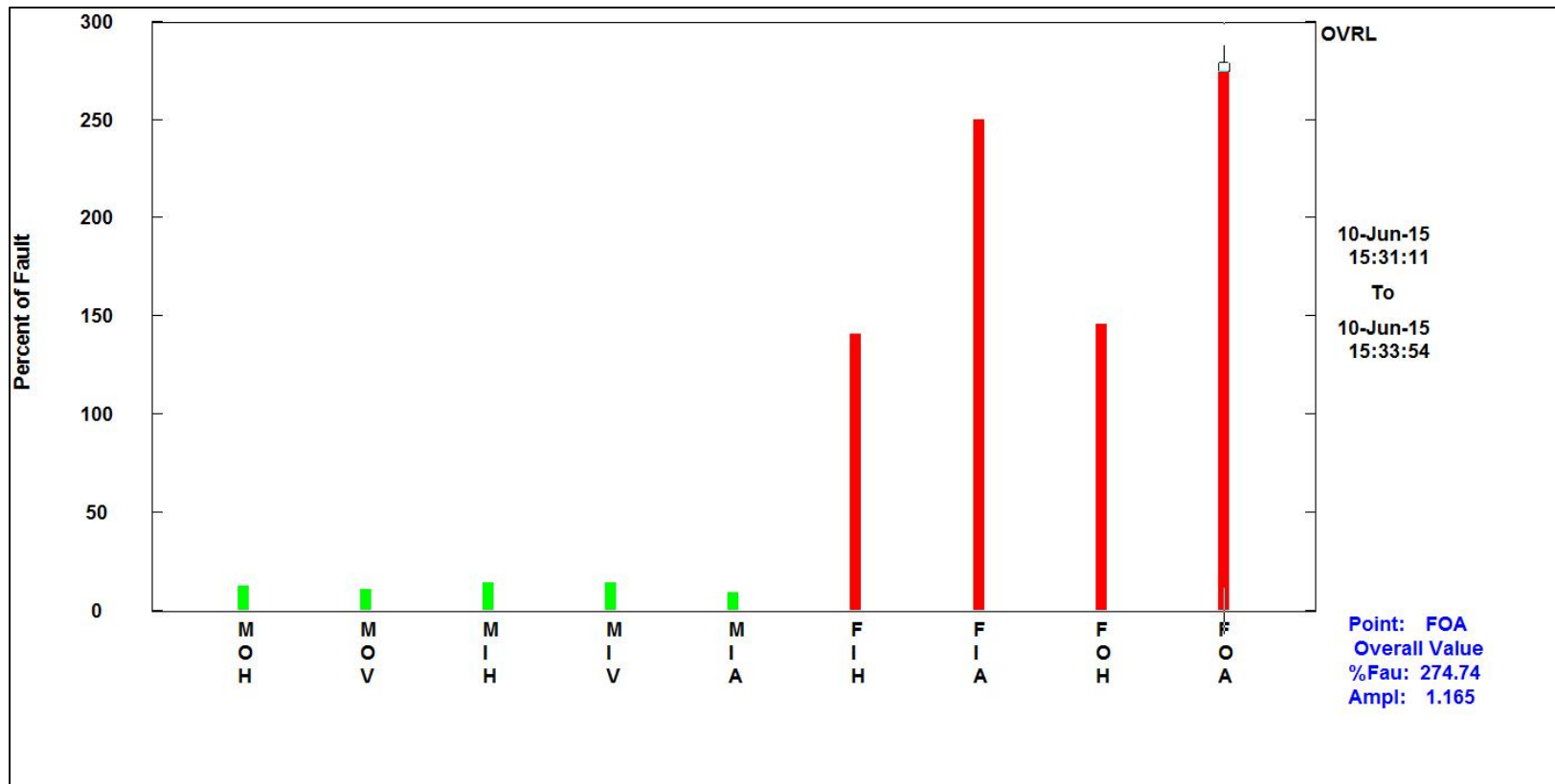
- 6/10/15 – High vibration noted at fan.
- Dominant vibration occurring at 1x fan rpm (984 cpm).



- As this was an over-hung fan, elevated axial vibration at 1x fan speed would be expected if unbalance in the fan wheel were present.
- To the **off-site analyst**, this vibration data would have been saying – unbalance.
- To the **on-site analyst**, the vibration levels in excess of normal prompted an on-site inspection that quickly revealed loose anchor bolts – **being there matters.**

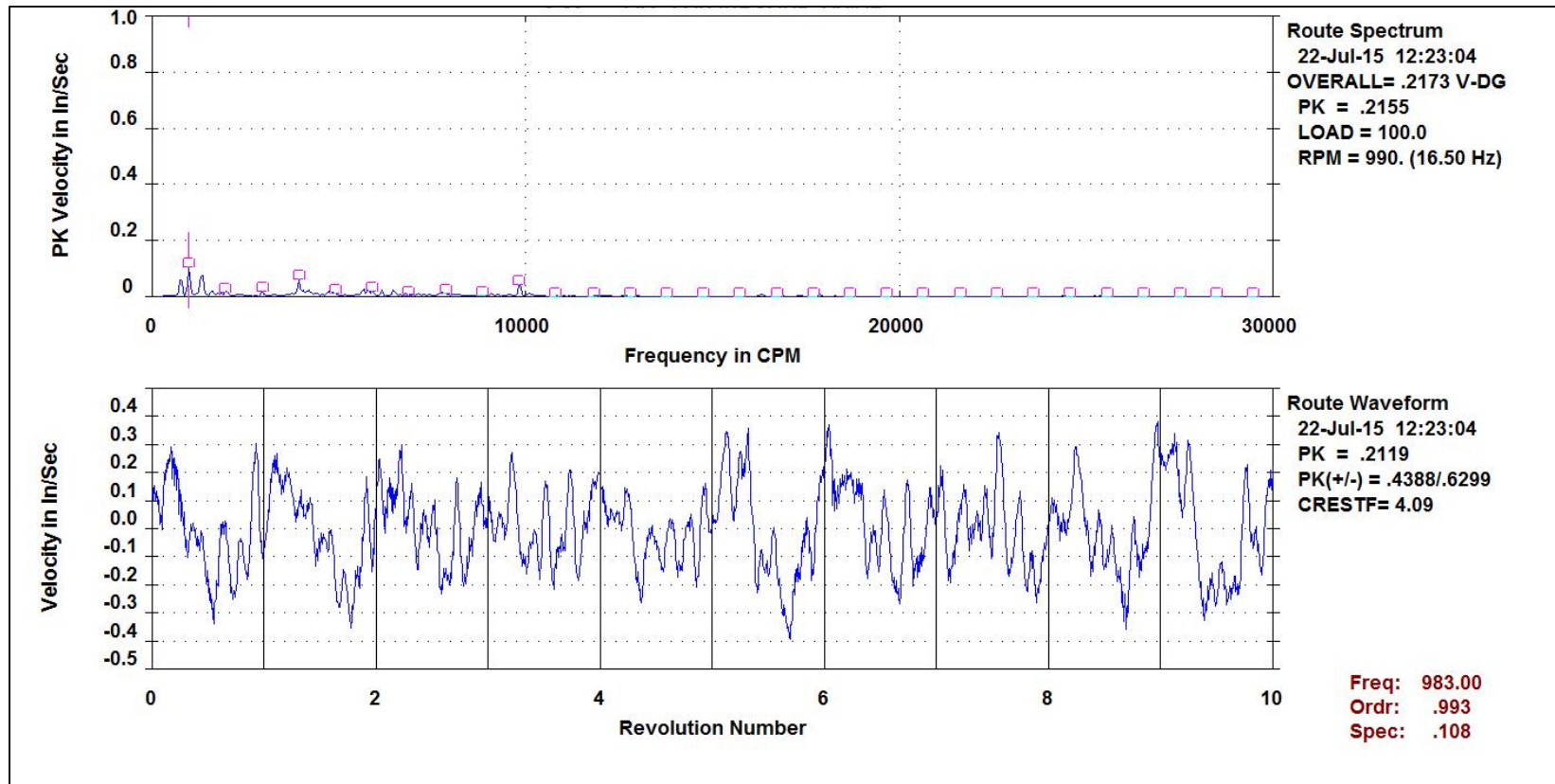
CASE HISTORY 3 – BELT-DRIVEN FAN

- 6/10/15 – Loose anchor bolts identified.
- Parameter profile plot of overall vibration levels shows dominant vibration at fan with highest levels in axial direction (FOA @ 1.2 ips-pk). This is a common pattern seen when unbalance is present on an over-hung fan wheel.



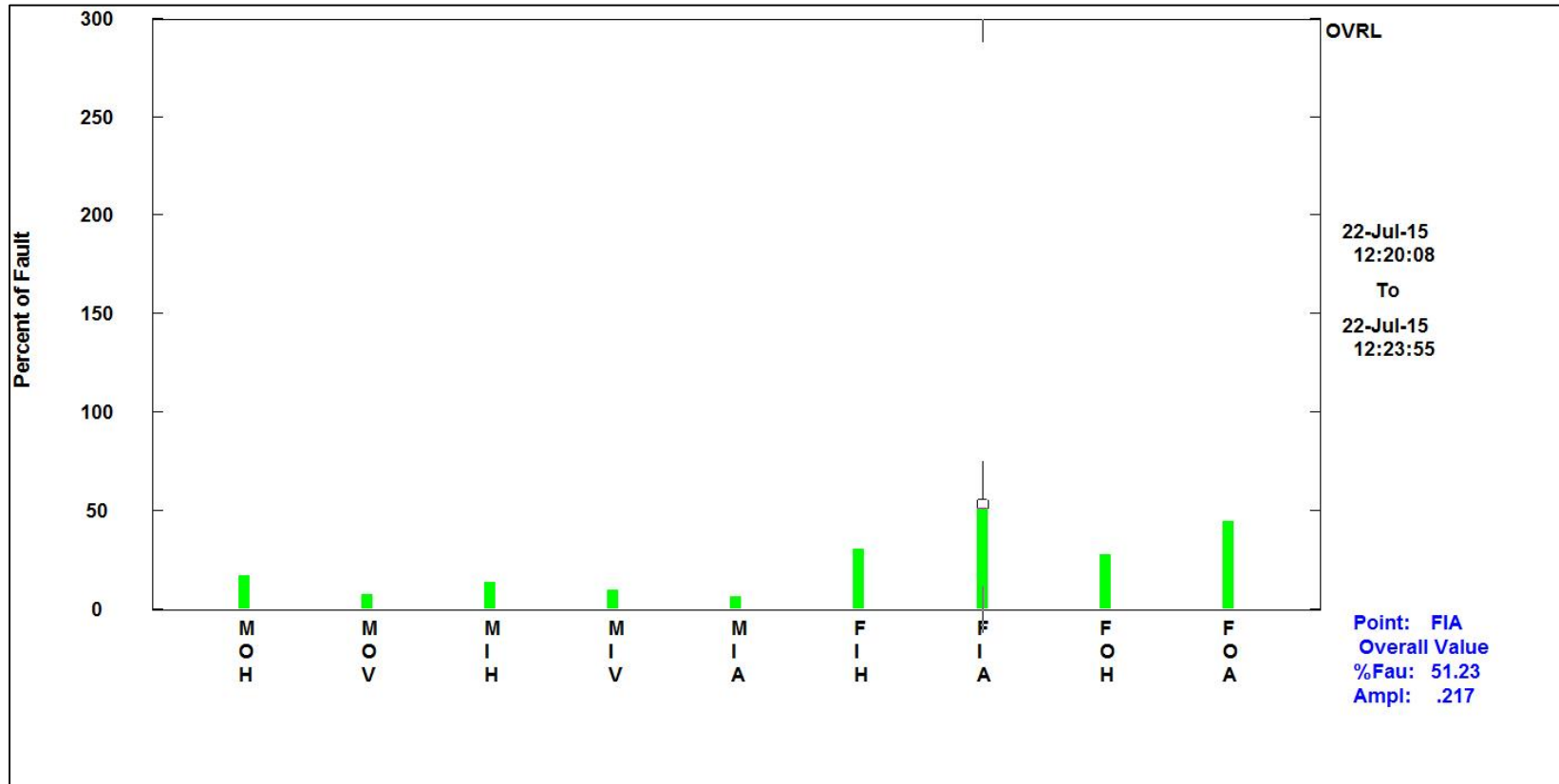
CASE HISTORY 3 – BELT-DRIVEN FAN

- 7/22/15 – All anchor bolts tightened.
- Vibration at fan speed (983 rpm) now low and acceptable at 0.11 ips-pk.



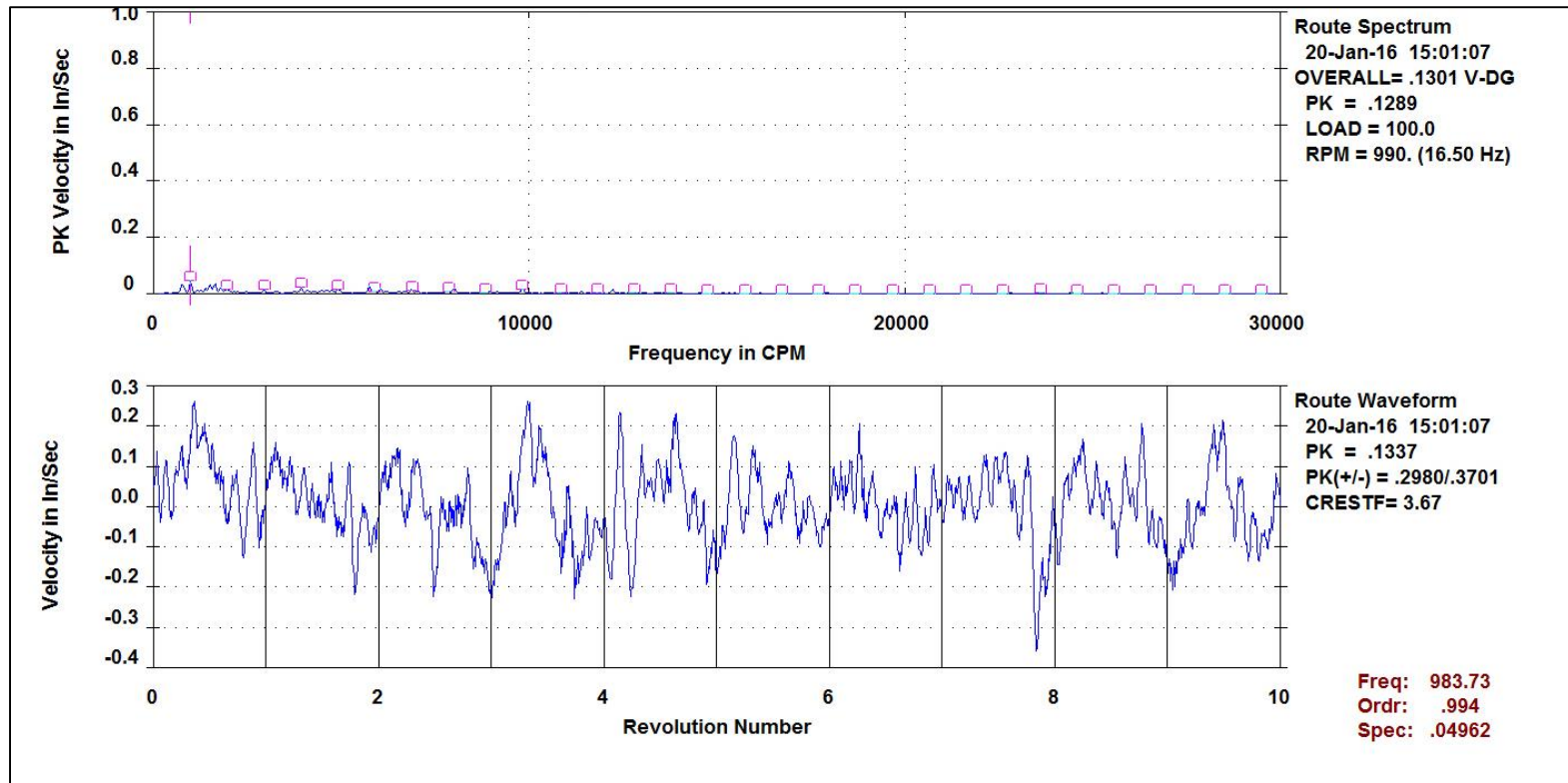
CASE HISTORY 3 – BELT-DRIVEN FAN

- 7/22/15 – All anchor bolts tightened.
- Parameter profile plot of overall vibration levels shows highest level at 0.22 ips-pk (fan, inboard, axial – FIA).



CASE HISTORY 3 – BELT-DRIVEN FAN

- 1/20/16 – Additional anchor bolts added at fan, sheave-end.
- Vibration levels now low & acceptable at 0.13 ips-pk with dominant vibration at fan 1x rpm (984 cpm).
- Significant improvement in vibration levels (from 1.2 ips → 0.13 ips).



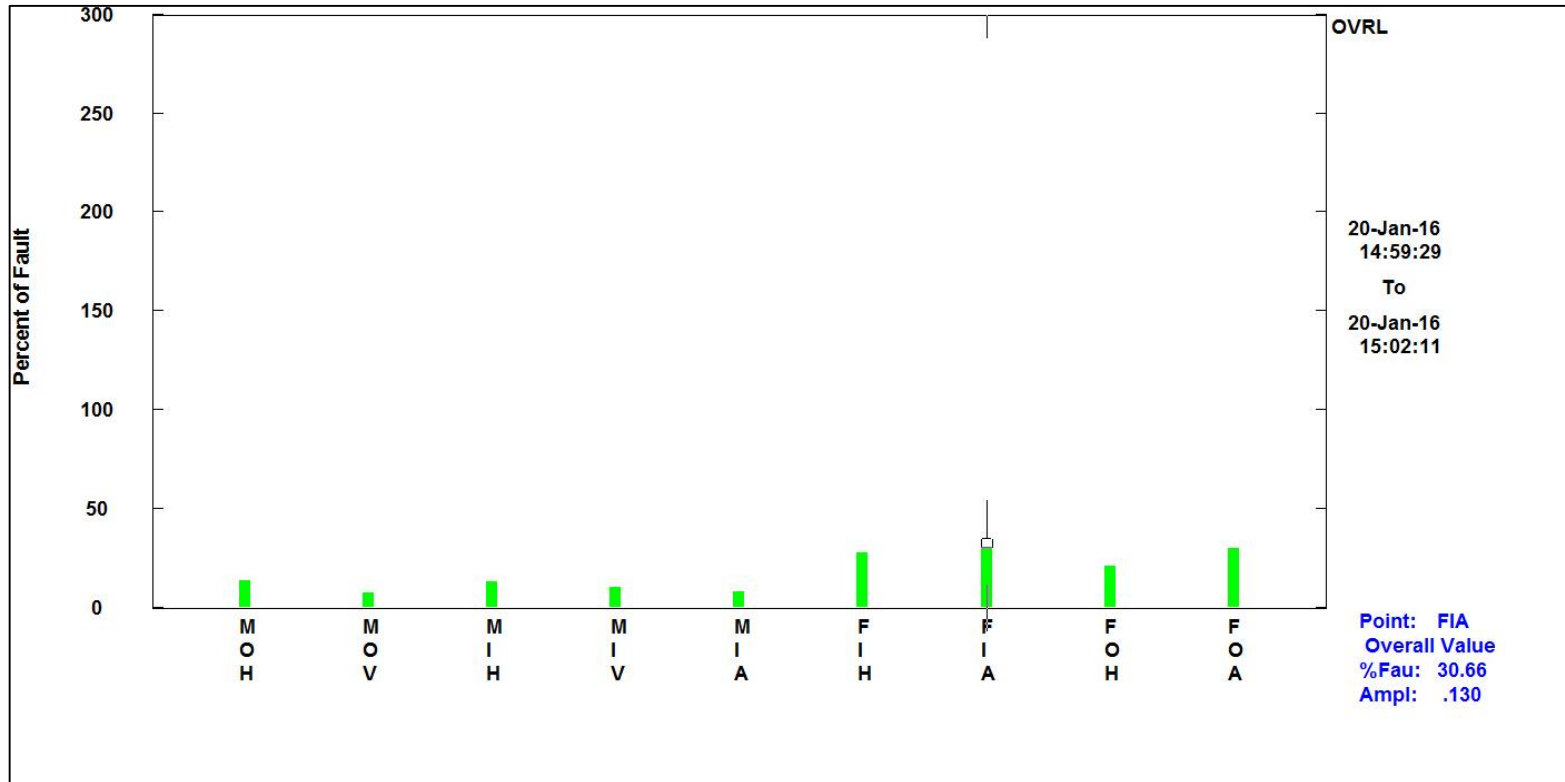
CASE HISTORY 3 – BELT-DRIVEN FAN

- The two red arrows point to the exact locations where additional anchor bolts were added to the fan base.
- Anchor bolts at this location were already present at a nearby sister fan that had experienced no trouble.
- Why these anchor bolts hadn't been installed at this fan in the past was unknown.



CASE HISTORY 3 – BELT-DRIVEN FAN

- 1/20/16 – Additional anchor bolts added at fan, sheave-end.
- Vibration levels now low & acceptable at 0.13 ips-pk maximum with dominant vibration at fan 1x rpm (984 cpm).
- Significant improvement in vibration levels (from 1.2 ips → 0.13 ips).



CASE HISTORY 3 – BELT-DRIVEN FAN

- The moral of this story again is that being there matters.
- In this case history what looked just like overhung fan unbalance in the vibration data in fact turned out to be a case of looseness.
- The loose anchor bolts were at just the **wrong** place for the fan where a small amount of looseness resulted in a significant reduction in stiffness in the axial direction at the fan.
- The reduction in stiffness caused by the loose anchor bolts allowed a lot more vibration to occur from the existing residual fan unbalance.
- **Later impact testing at this fan while it was down during an outage found a natural frequency in the axial direction just above the running speed of 990 rpm, so looseness had caused resonance of the fan assembly in the axial direction.**

CASE HISTORY 4

BELT-DRIVEN FAN 2

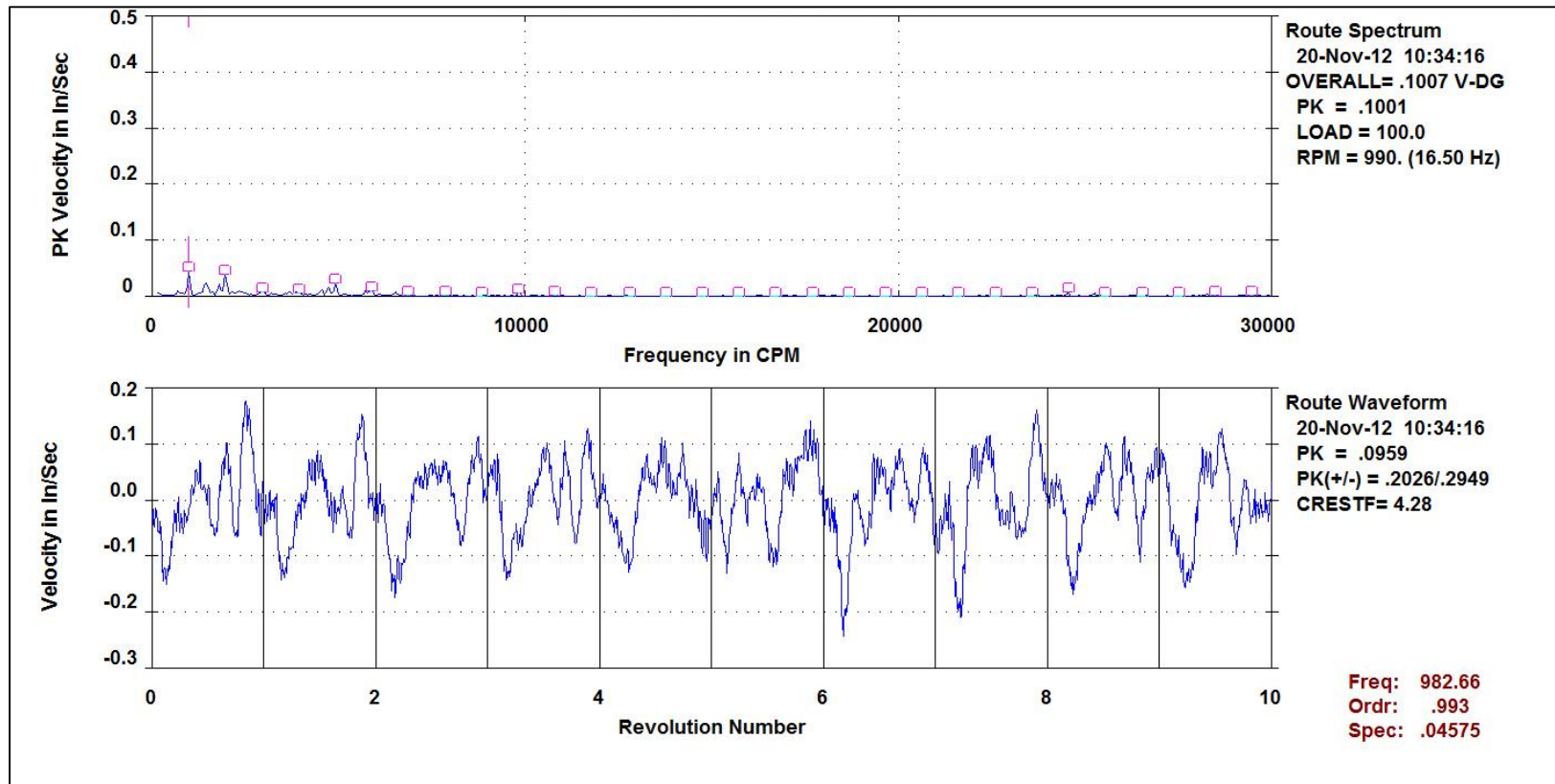
CASE HISTORY 4 – BELT-DRIVEN FAN

- The fourth case history also involves a centrifugal fan driven by a 4-pole induction motor thru a belt-drive.
- The motor operates at $\sim 1,790$ rpm while the fan operates at ~ 990 rpm.
- The fan is supported by a fabricated steel base bolted to the concrete floor via 6-ea anchor bolts.
- High vibration at the fan speed was occasionally encountered at this machine, but a good cleaning of the fan usually reduced levels to a satisfactory level.
- In the following case history, more than just normal buildup on the fan wheel was at work.



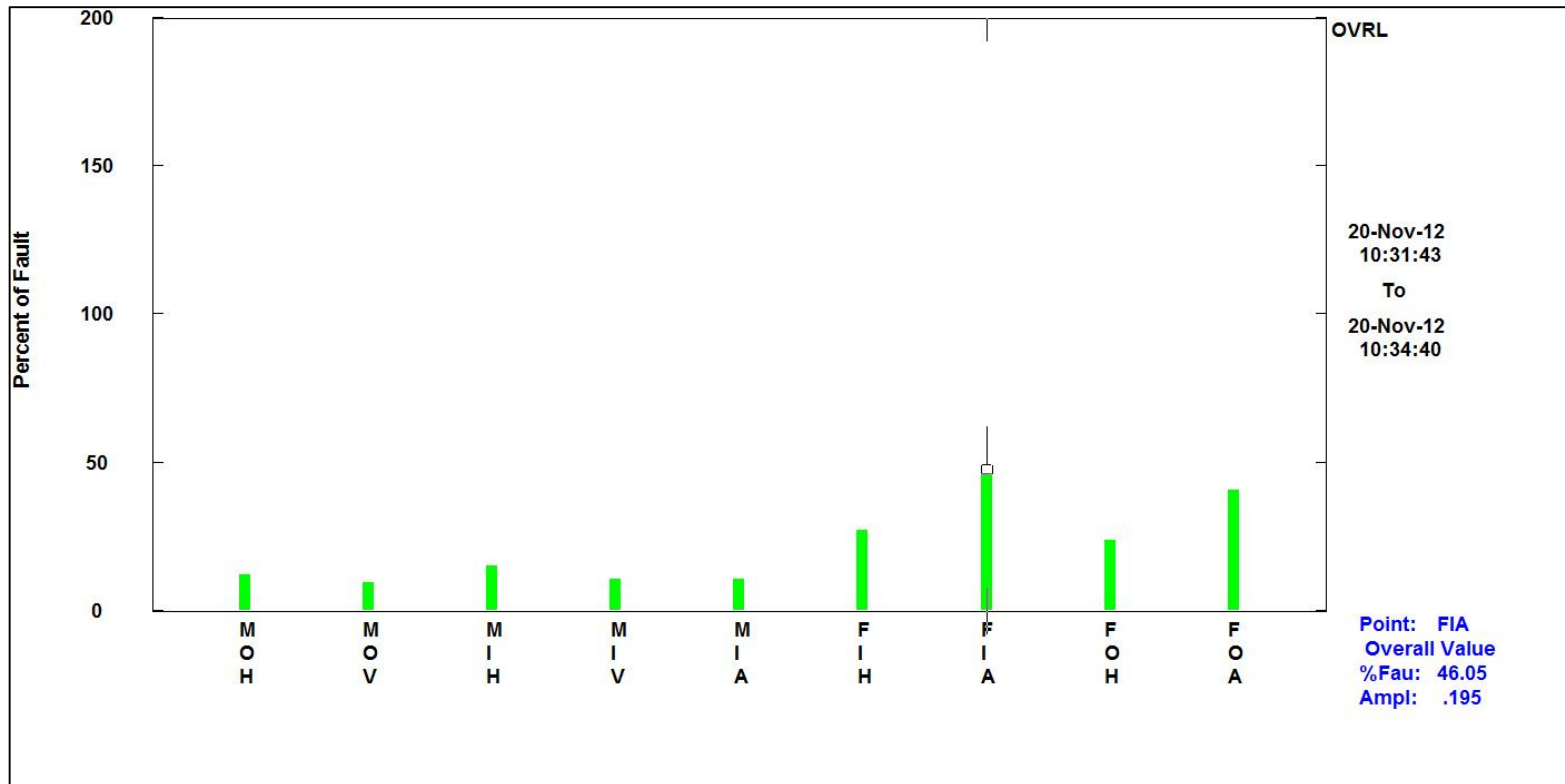
CASE HISTORY 4 – BELT-DRIVEN FAN

- 11/20/12 – Reference vibration data on fan. Low and acceptable vibration levels (0.10 ips-pk).
- Spectral & waveform data shown at fan, outboard, horizontal point (representative point thru machine history).



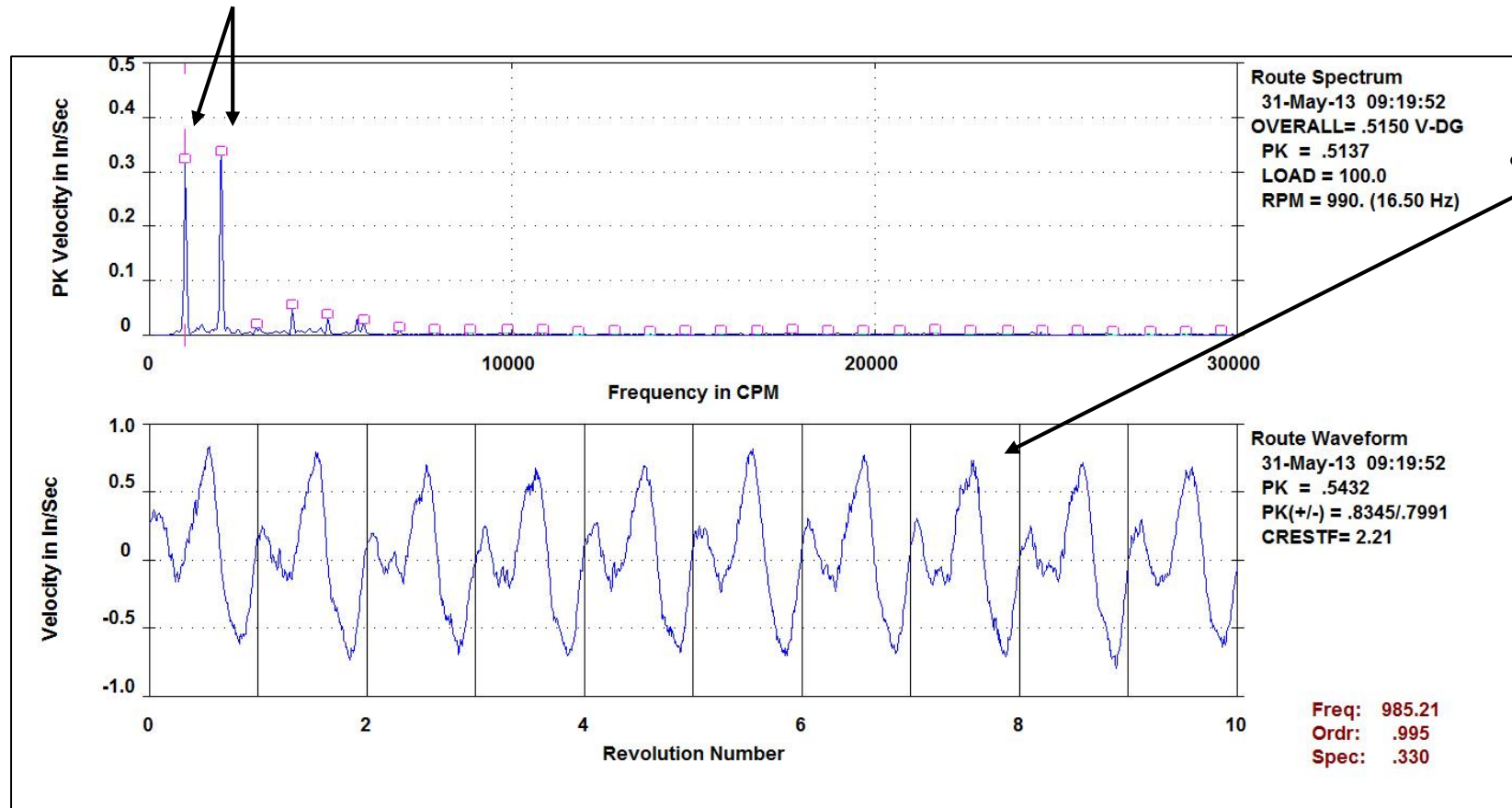
CASE HISTORY 4 – BELT-DRIVEN FAN

- 11/20/12 – Reference vibration data on fan.
- Profile plot of overall vibration levels shows 0.20 ips-pk highest level across machine.



CASE HISTORY 4 – BELT-DRIVEN FAN

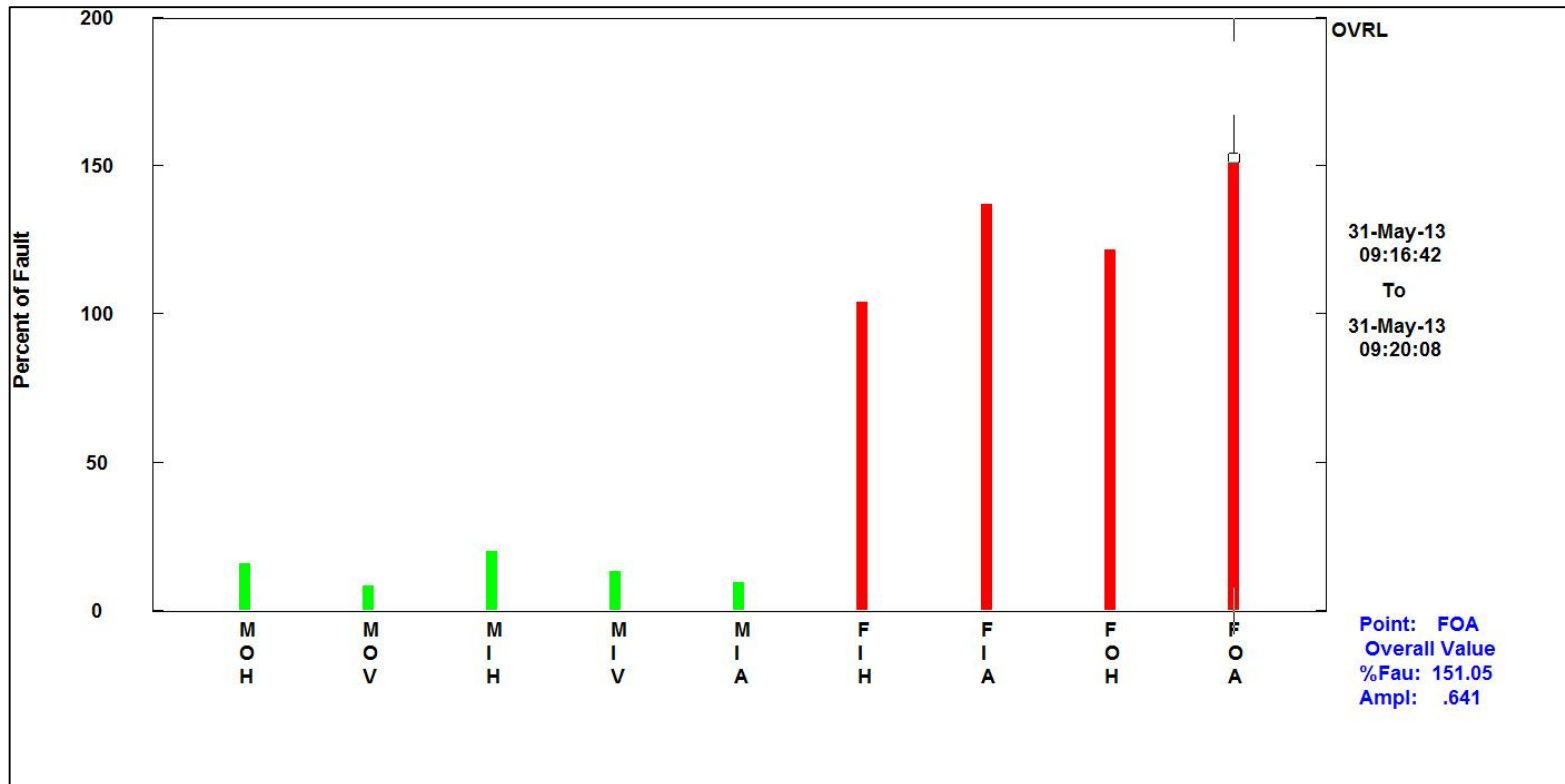
- 5/31/13 – Cracked welds identified at left side of fan frame near outboard bearing.
- Much higher vibration levels (0.51 ips) with dominant vibration at 1x rpm & 2x rpm of fan (type “B” looseness).



• Note the distortion of the waveform.

CASE HISTORY 4 – BELT-DRIVEN FAN

- 5/31/13 – Cracked welds identified at fan frame near outboard bearing; unbalance at fan wheel also possible.
- Much higher vibration levels with dominant vibration at 1x rpm & 2x rpm of fan (type “B” looseness).
- Note how problem is only at fan measurements, no issues seen at motor (low levels there).



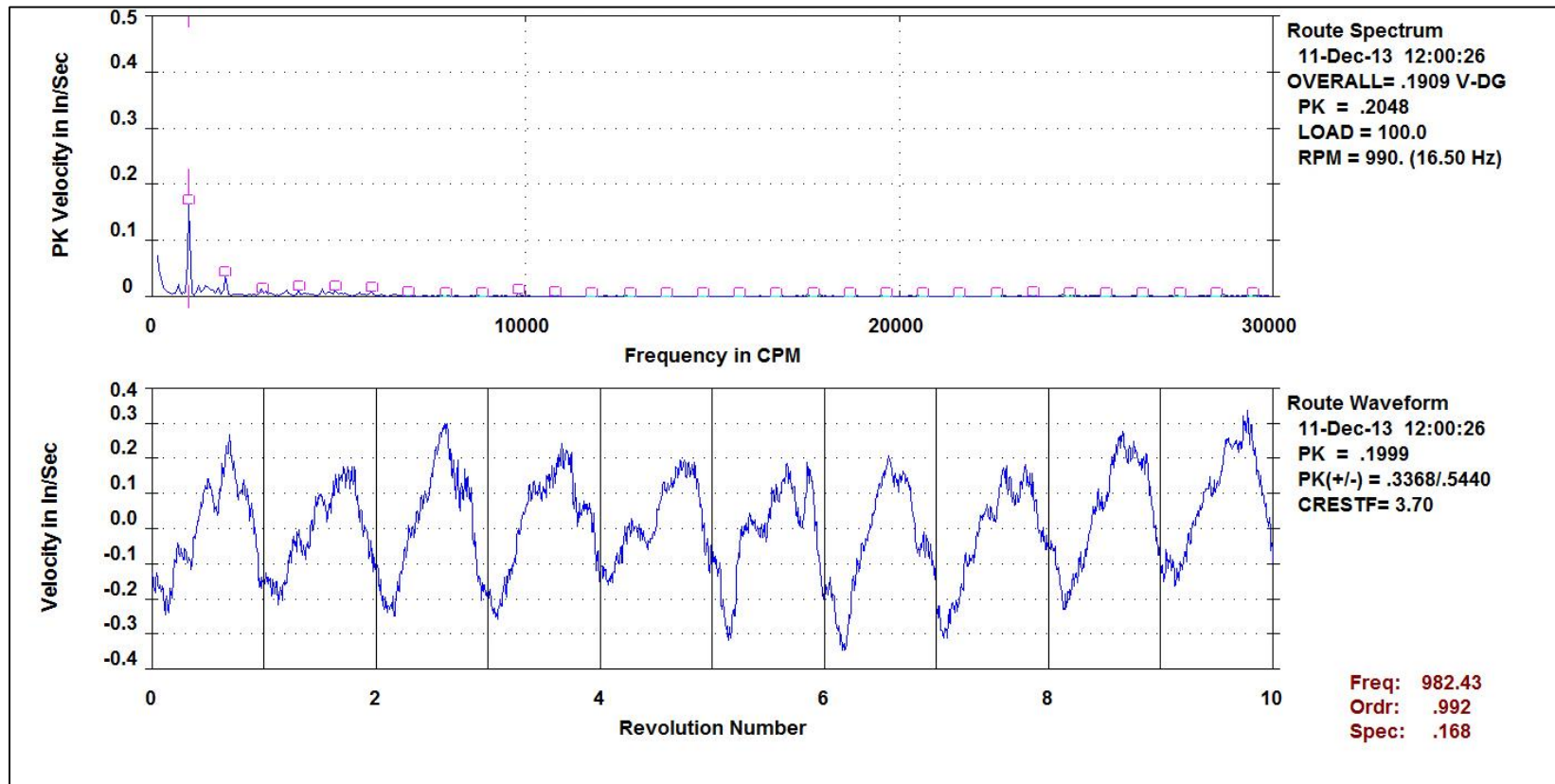
CASE HISTORY 4 – BELT-DRIVEN FAN

- 5/31/13 – Cracked welds identified at fan frame near outboard bearing; unbalance at fan wheel also possible.
- Much higher vibration levels with dominant vibration at 1x rpm & 2x rpm of fan (type “B” looseness).
- From photos at right you can see where these cracks were welded up. The fan shroud attaches to its supporting structure (angle and channel) via welds. A few of these welds were completely cracked and broken on the left side of the shroud only.
- **The cracked welds reduced the stiffness of the fan ONLY on the left side creating ASYMMETRIC stiffness from left to right (potential source for elevated 2x rpm).**



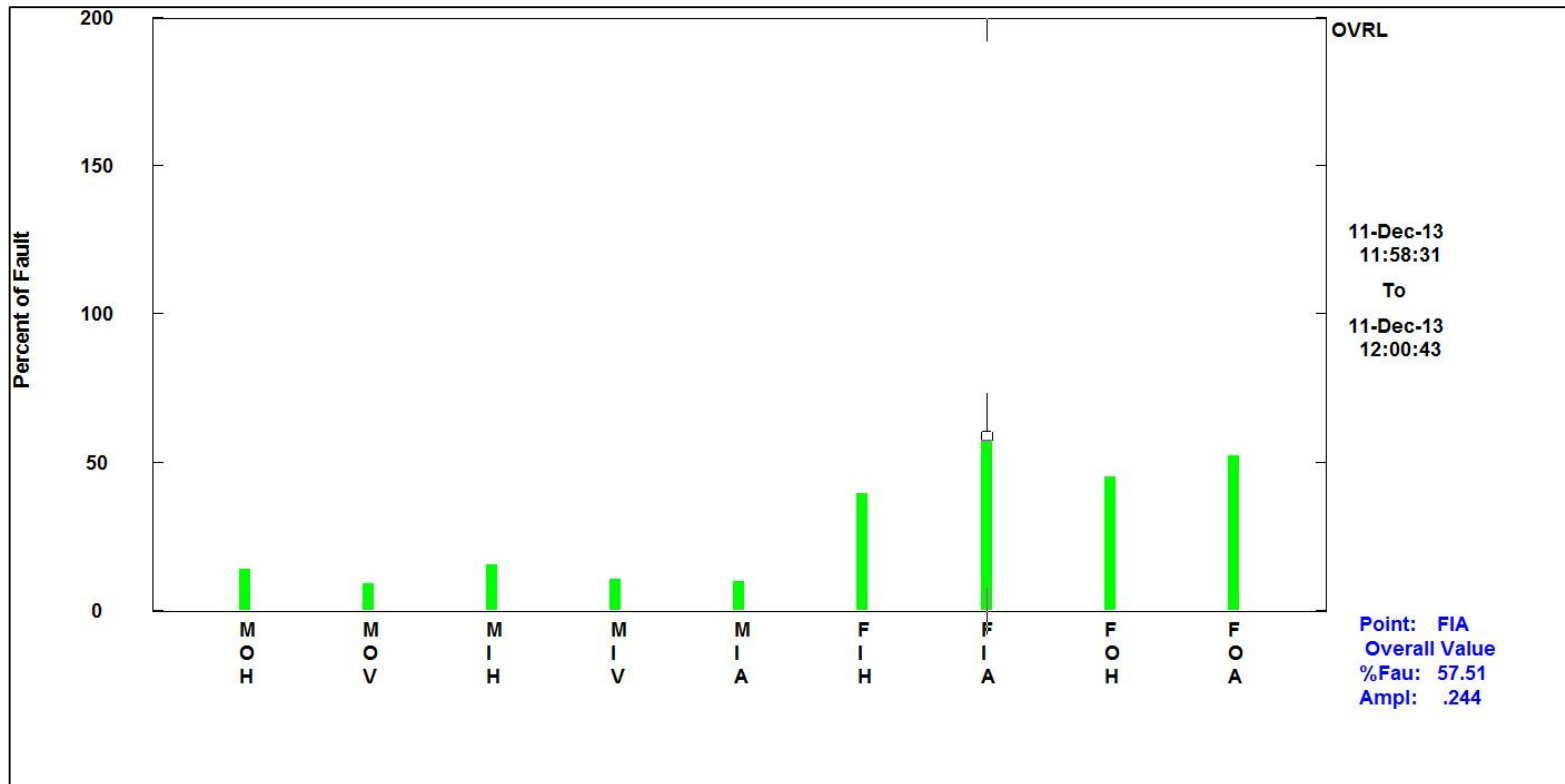
CASE HISTORY 4 – BELT-DRIVEN FAN

- 12/11/13 – Cracked welds repaired and fan wheel cleaned.
- Much lower vibration levels and significant reduction at 2x rpm.



CASE HISTORY 4 – BELT-DRIVEN FAN

- 12/11/13 – Cracked welds repaired and fan wheel cleaned.
- Profile plot of overall vibration levels shows highest level at 0.24 ips-pk at fan, inboard, axial.



CASE HISTORY 4 – BELT-DRIVEN FAN

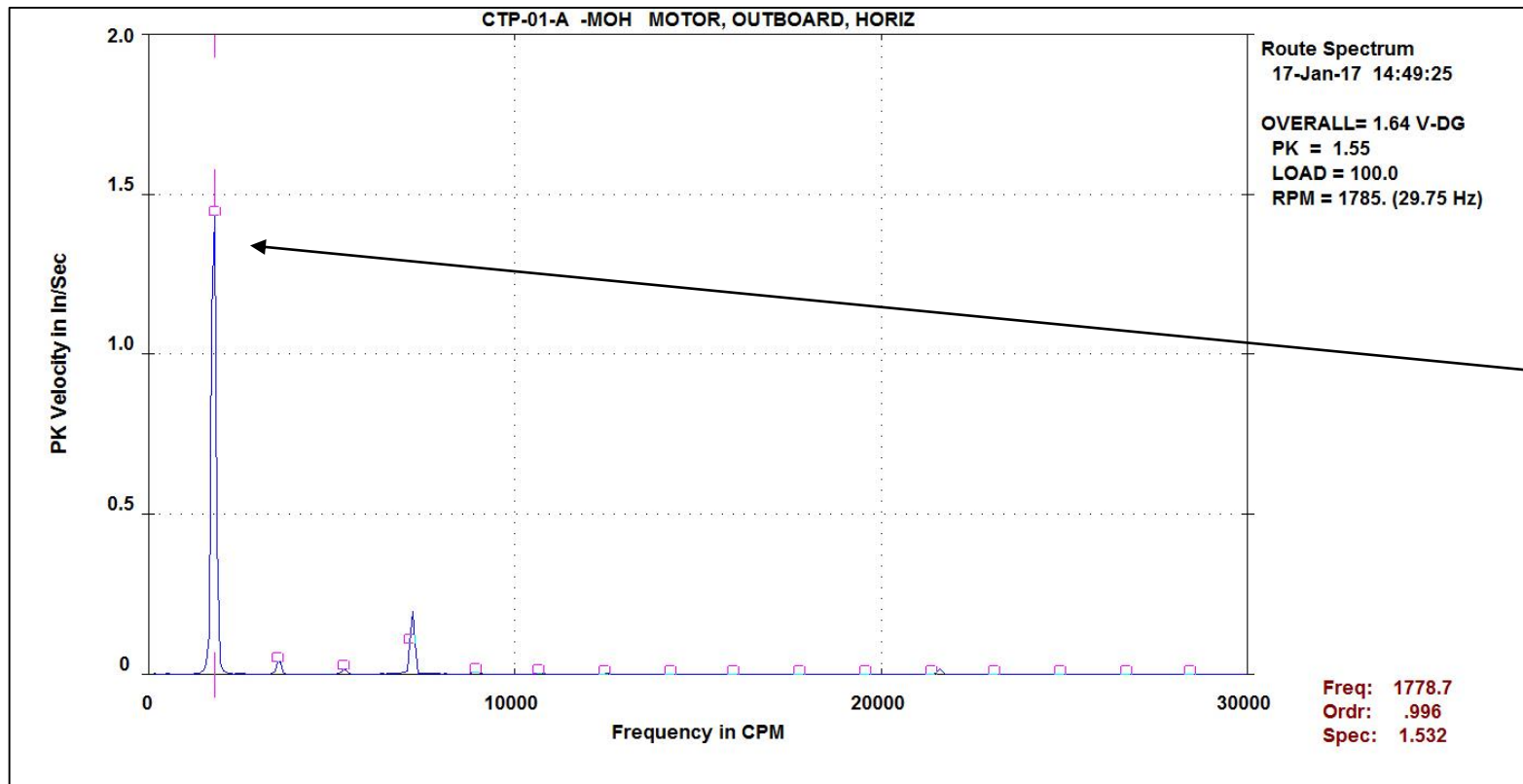
- Moral of this story:
 - 1) Reductions in stiffness are not always due to loose bolts. Inspect for cracked welds or broken components, etc.
 - 2) Asymmetric stiffness can be a source of vibration at 2x rpm.

CASE HISTORY 5

SMALL CONDENSER PUMP

CASE HISTORY 5 – SMALL CONDENSER PUMP

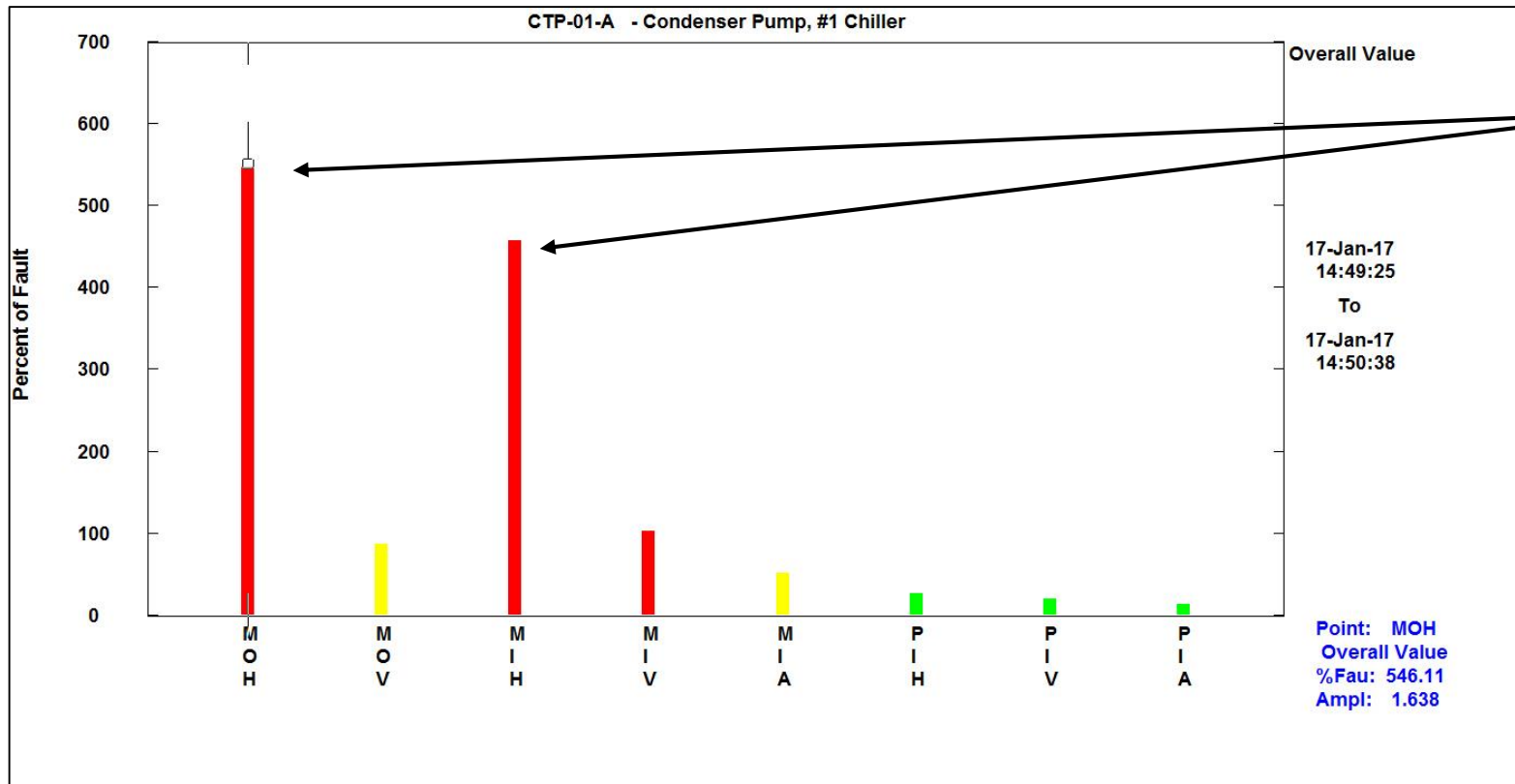
- 1/17/17 – Routine vibration survey noted severe horizontal vibration levels at motor. Levels much higher than previous levels.
- Loose anchor bolts at the motor-end of the machine base had been identified from the prior survey and repaired by the plant.



- Spectral data showed dominant vibration at the machine speed of ~ 1,780 rpm.

CASE HISTORY 5 – SMALL CONDENSER PUMP

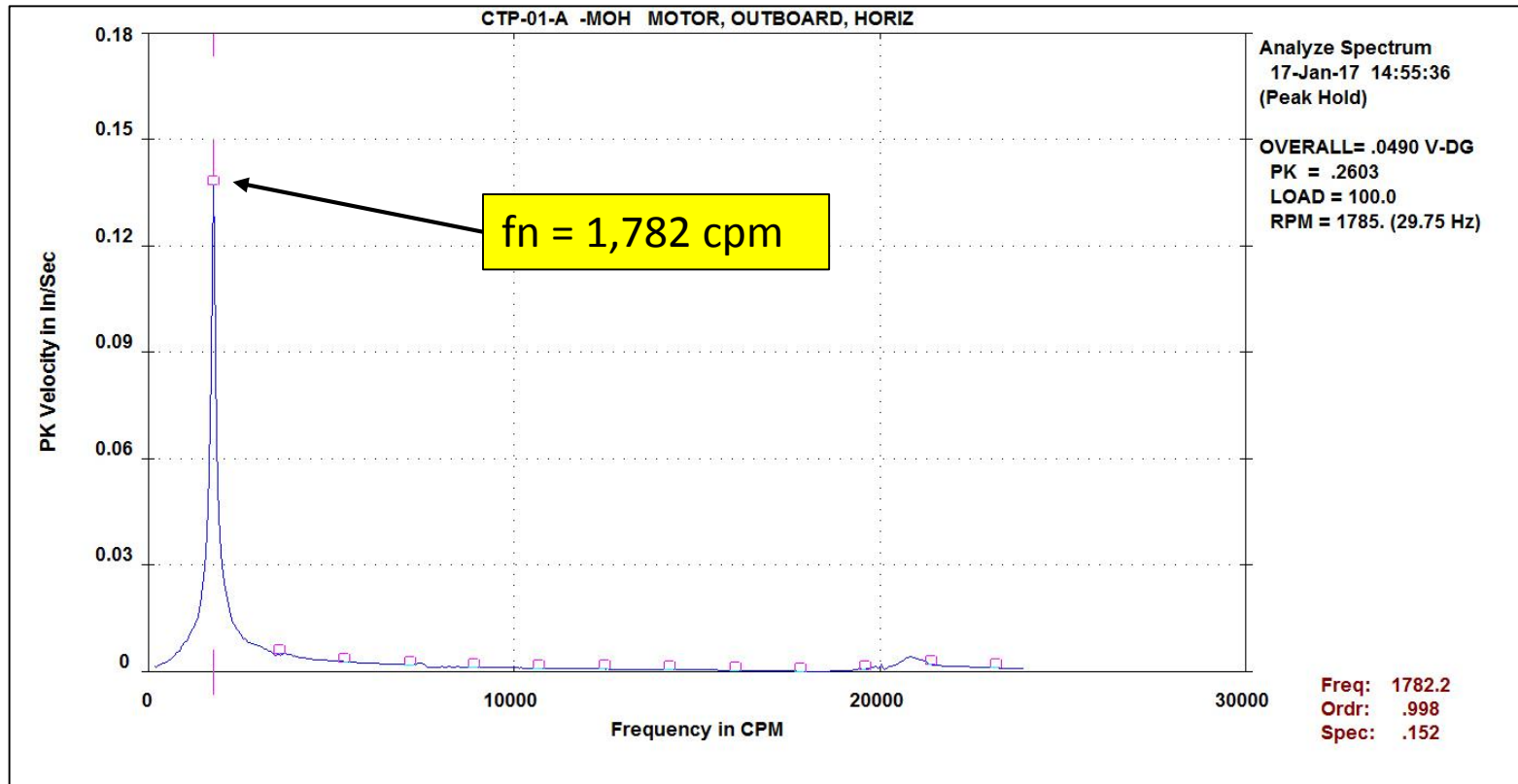
- 1/17/17 – Routine vibration survey noted severe horizontal vibration levels at motor. Levels much higher than previous levels. Loose anchor bolts at the motor-end of the machine base had been identified from the prior survey and repaired by the plant.



- Profile plot of overall vibration levels showed **very directional vibration (horizontal) and very localized vibration (motor only)**.
- No problems seen at any pump measurement.

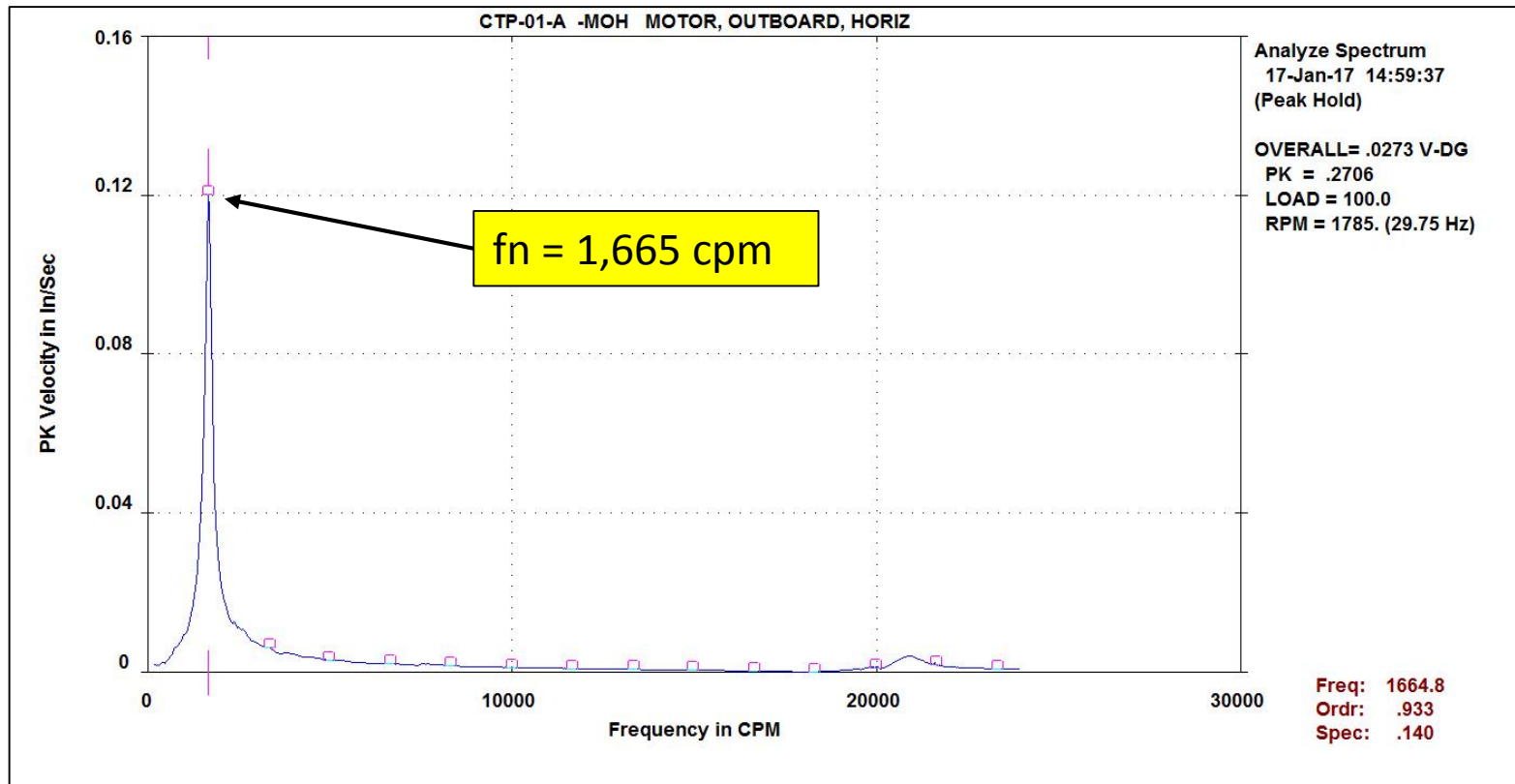
CASE HISTORY 5 – SMALL CONDENSER PUMP

- 1/17/17 – A quick bump test of the motor outboard in the horizontal direction using the Peak-Hold function on the analyzer identified the reason for the increased vibration levels – a natural frequency at 1,782 cpm was identified in the horizontal direction.



CASE HISTORY 5 – SMALL CONDENSER PUMP

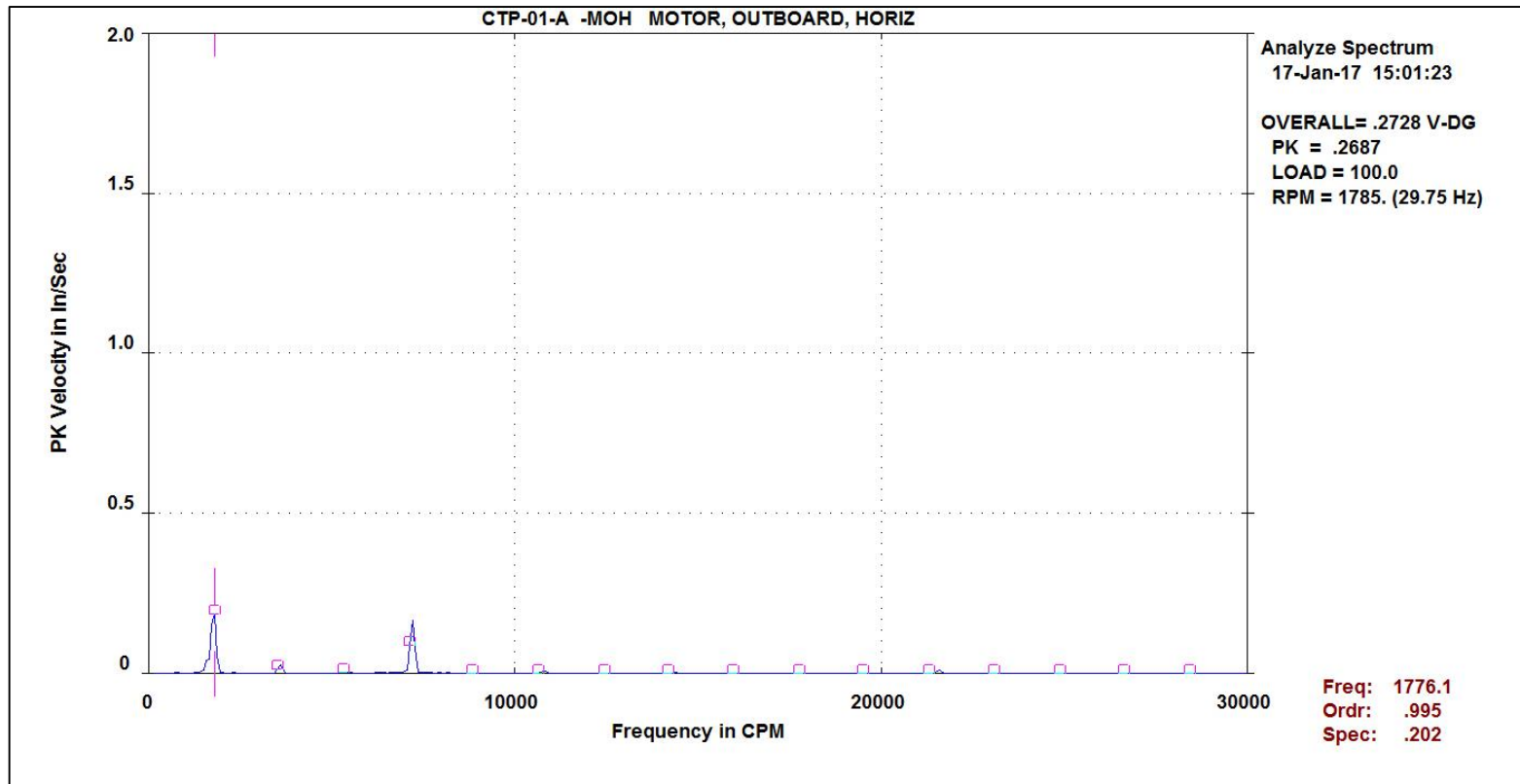
- 1/17/17 – As the prior levels were not nearly this high **and** it was known that the only repairs performed was to tighten all anchor bolts, the decision was made to slightly loosen one of the two anchor bolts at the motor outboard end (where the vibration was).



- A follow-up bump test confirmed a reduction in the natural frequency from 1,782 cpm (tight anchor bolt) to 1,665 cpm (loose anchor bolt).

CASE HISTORY 5 – SMALL CONDENSER PUMP

- 1/17/17 – Follow-up vibration data collected after the one anchor bolt was loosened confirmed the unconventional “repair” worked. **Our motor vibration levels dropped from 1.63 to 0.27 ips-pk by loosening one of the anchor bolts at the motor end.**



- The tightening of the anchor bolts raised our horizontal natural frequency right into resonance with the machine speed of ~ 1,780 rpm.
- By loosening one anchor bolt at the motor end, we reduced the stiffness in the horizontal direction at the right location and moved our natural frequency sufficiently away from the machine speed to avoid resonance.

CASE HISTORY 5 – SMALL CONDENSER PUMP

- Morals of this story:
 - 1) Every now and then lowering the stiffness is better than increasing it.
 - 2) Be aware of the location of any natural frequencies near your running speed, vane-pass frequency or any other dominant dynamic forces and do what you can to avoid them.

CASE HISTORY 6

CENTRIFUGAL PUMP

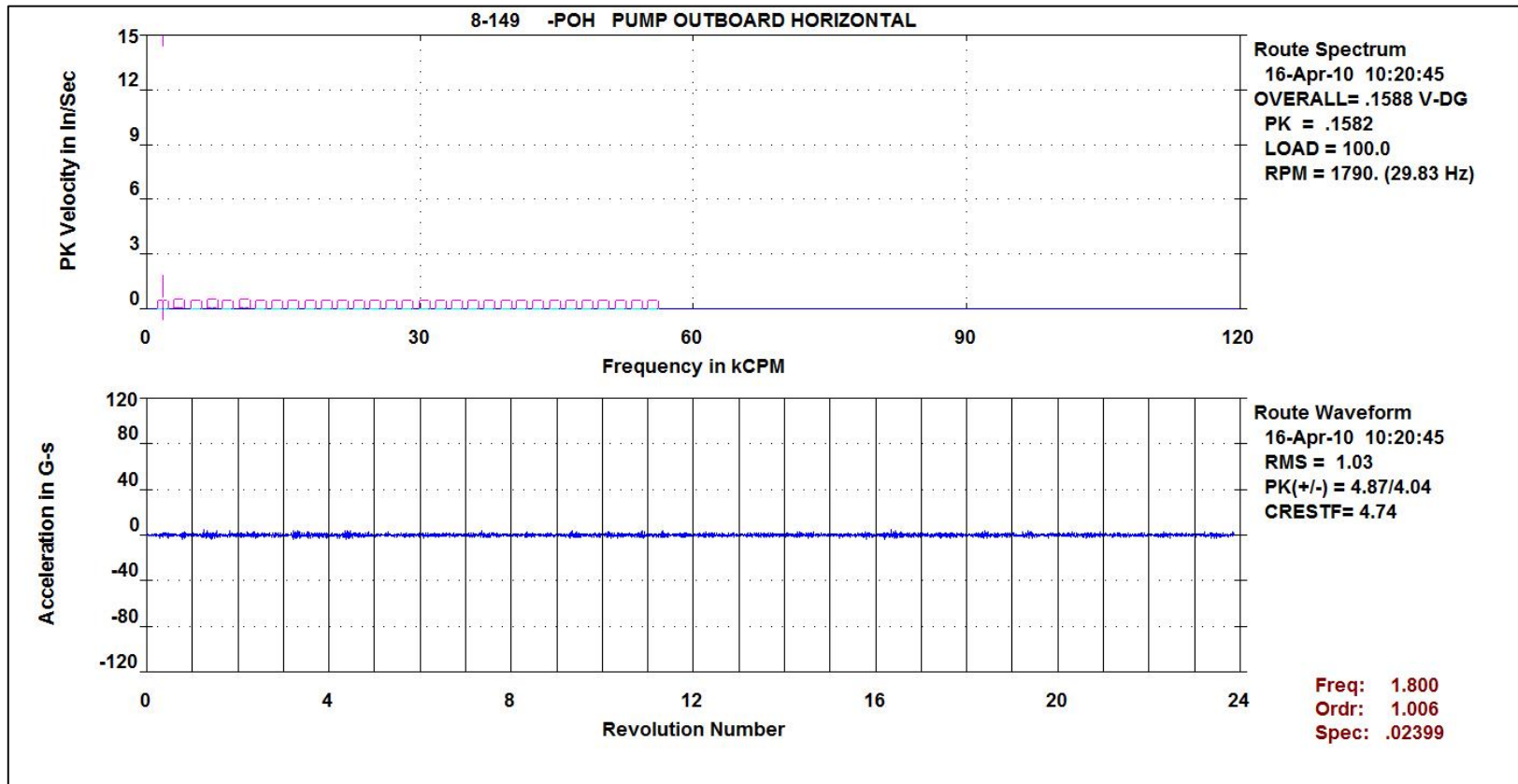
CASE HISTORY 6 – CENTRIFUGAL PUMP

- The sixth case history involves a horizontal, centrifugal pump driven by a 4-pole motor operating at $\sim 1,790$ rpm.
- Photo at right shows the view from the pump outboard end of this machine.



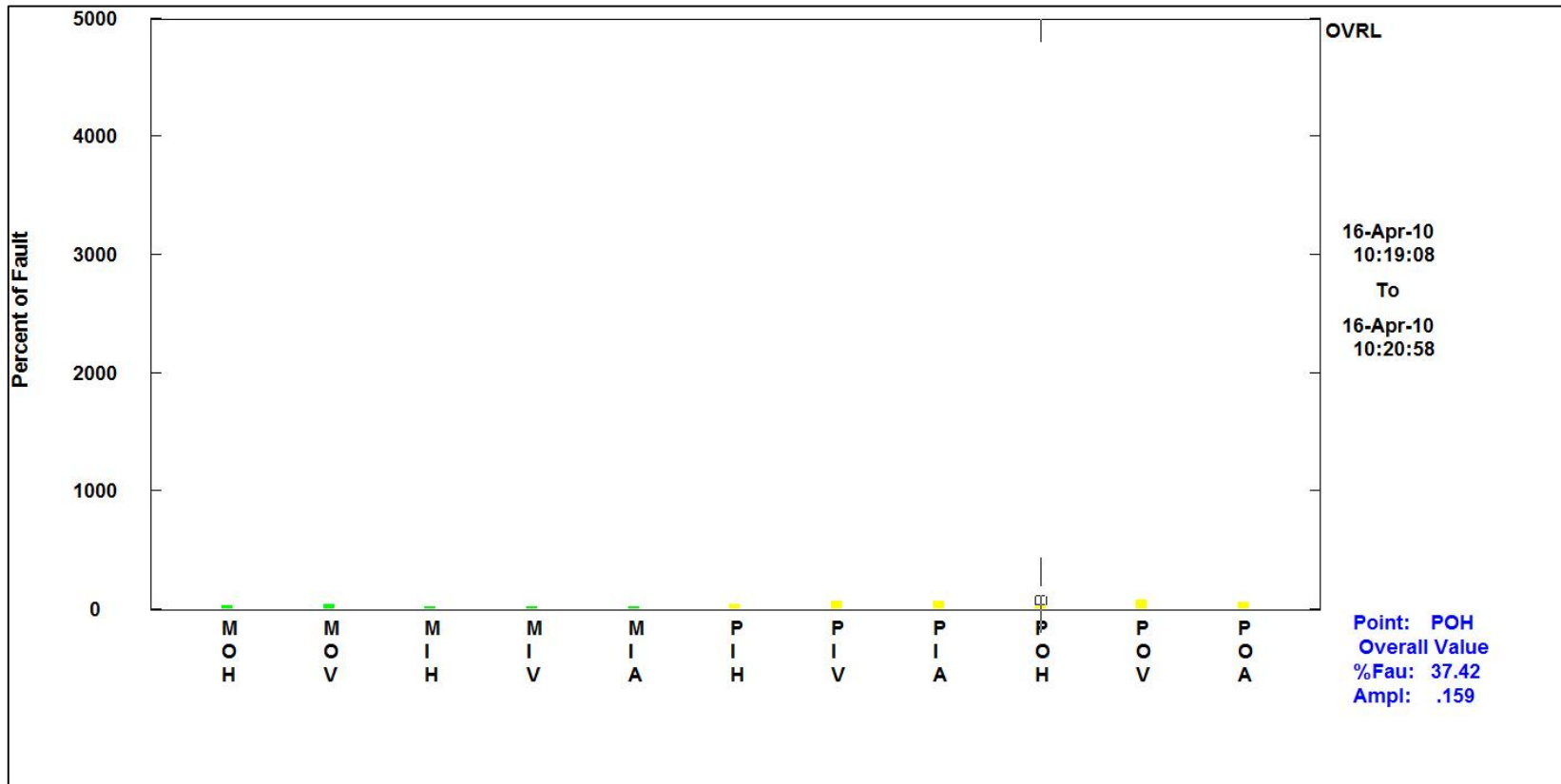
CASE HISTORY 6 – CENTRIFUGAL PUMP

- Route data collected a few months prior to trouble.
- The scale at left of the spectral data is extreme I know, but for a reason.



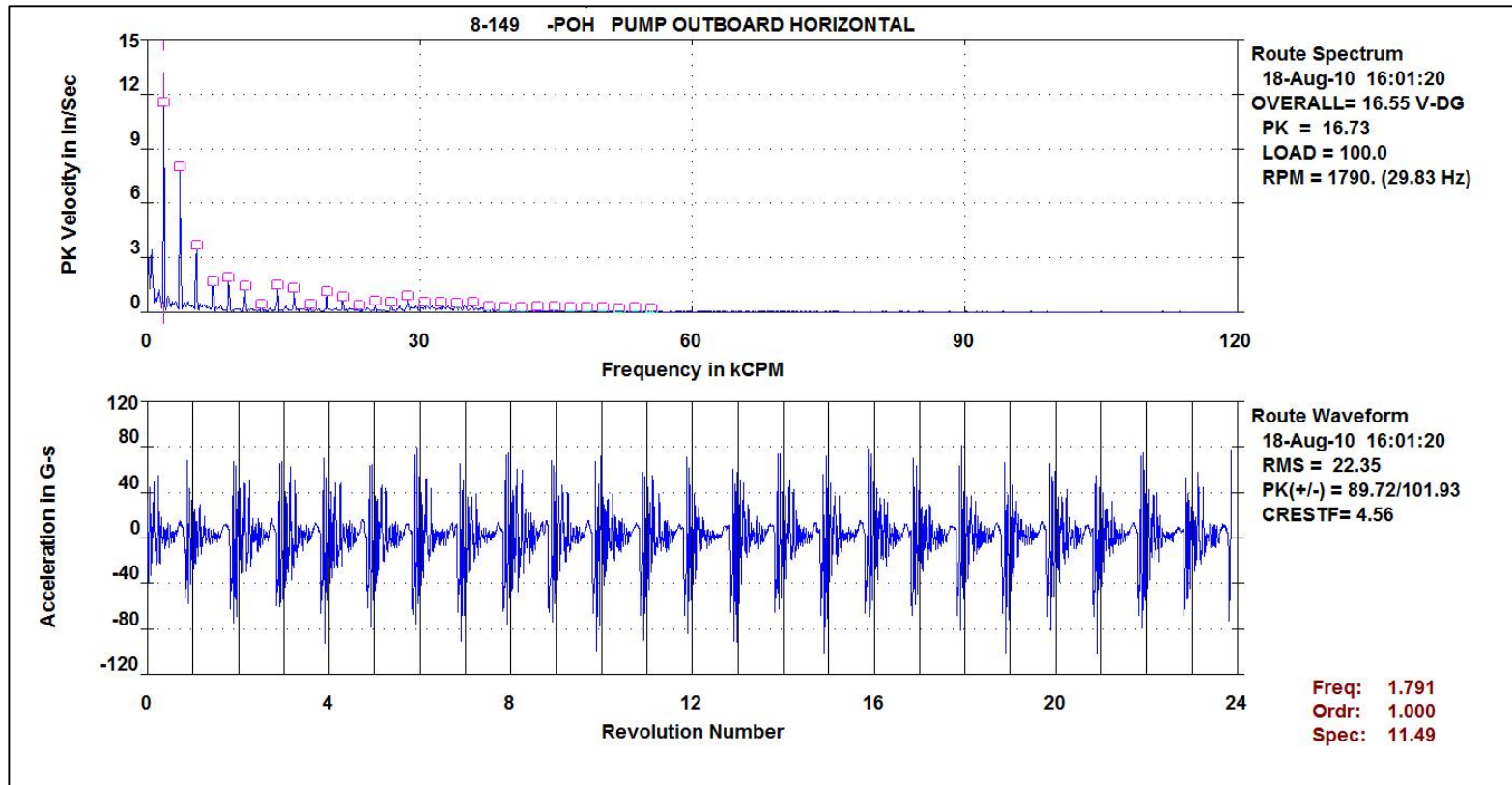
CASE HISTORY 6 – CENTRIFUGAL PUMP

- Route data collected a few months prior to trouble at this pump.
- Profile plot of overall vibration levels. Scale at left is extreme for a reason.



CASE HISTORY 6 – CENTRIFUGAL PUMP

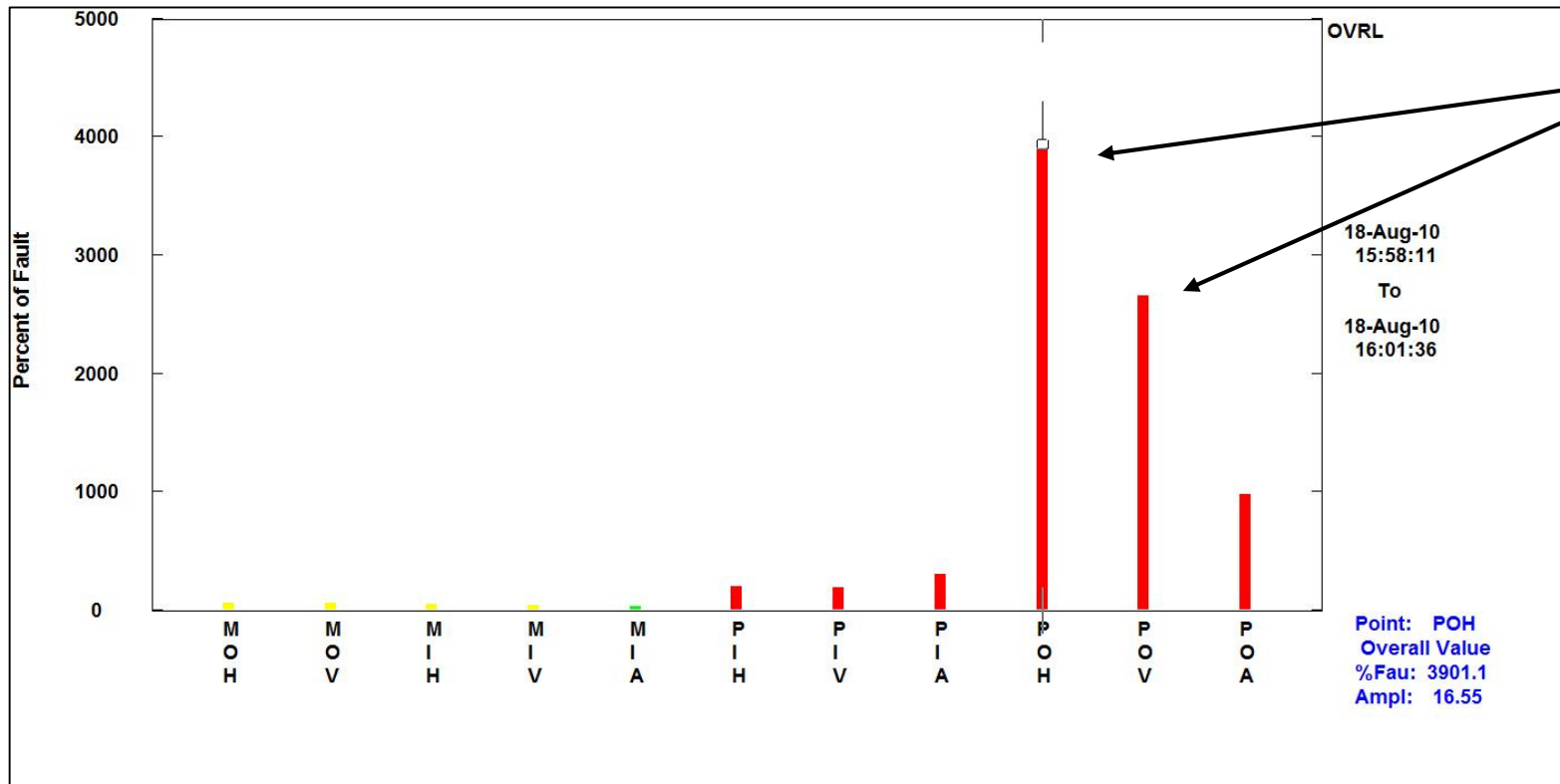
- 8/18/10 – Extreme looseness at pump outboard bearing and bearing fault stage 4 (excessive clearance at outboard bearing). Little if any warning of this problem due to failed shaft seal and resulting contamination of outboard bearing.



- Severe vibration levels at 16.6 ips and spectral data shows multiples of machine speed of ~ 1,790 rpm with raised noise floor.
- The vibration was so bad I had to hold my sensor in place even though it had a flat magnet on a flat ferrous surface.
- Note in the waveform data the strong impacting at the machine speed (102 g's-pk!!).

CASE HISTORY 6 – CENTRIFUGAL PUMP

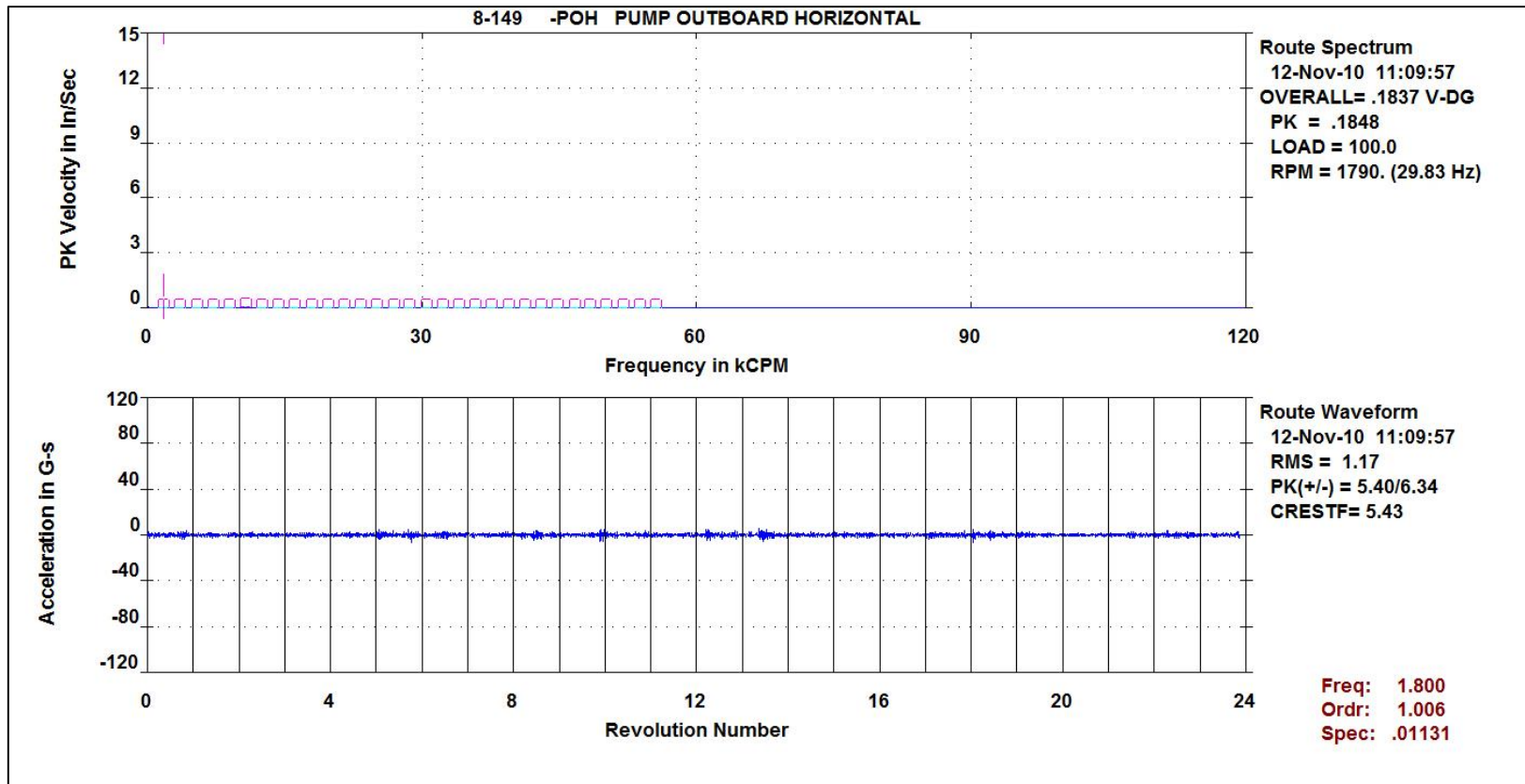
- 8/18/10 – Extreme looseness at pump outboard bearing and bearing fault stage 4 (excessive clearance at outboard bearing). Little if any warning of this problem due to failed shaft seal and resulting contamination of outboard bearing.



- Profile plot of overall vibration levels showed **localized vibration concentrated at pump outboard bearing.**

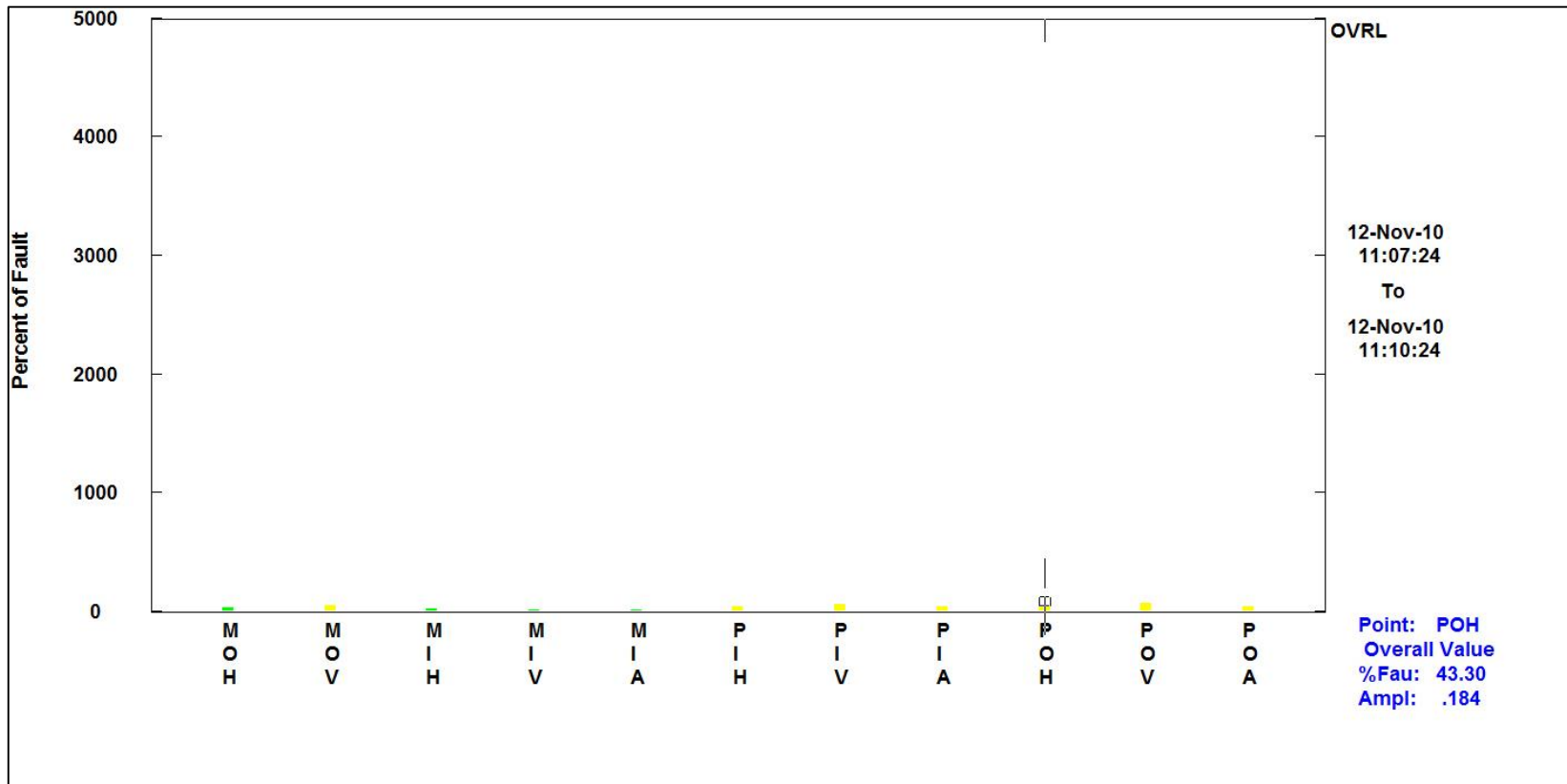
CASE HISTORY 6 – CENTRIFUGAL PUMP

- 11/12/10 – Follow-up vibration data after pump bearing changed and seal repaired.
- Note the significantly lower levels. Spectra OA levels 16.6 ips → 0.18 ips. Waveform levels 102 g's-pk → 6 g's-pk.



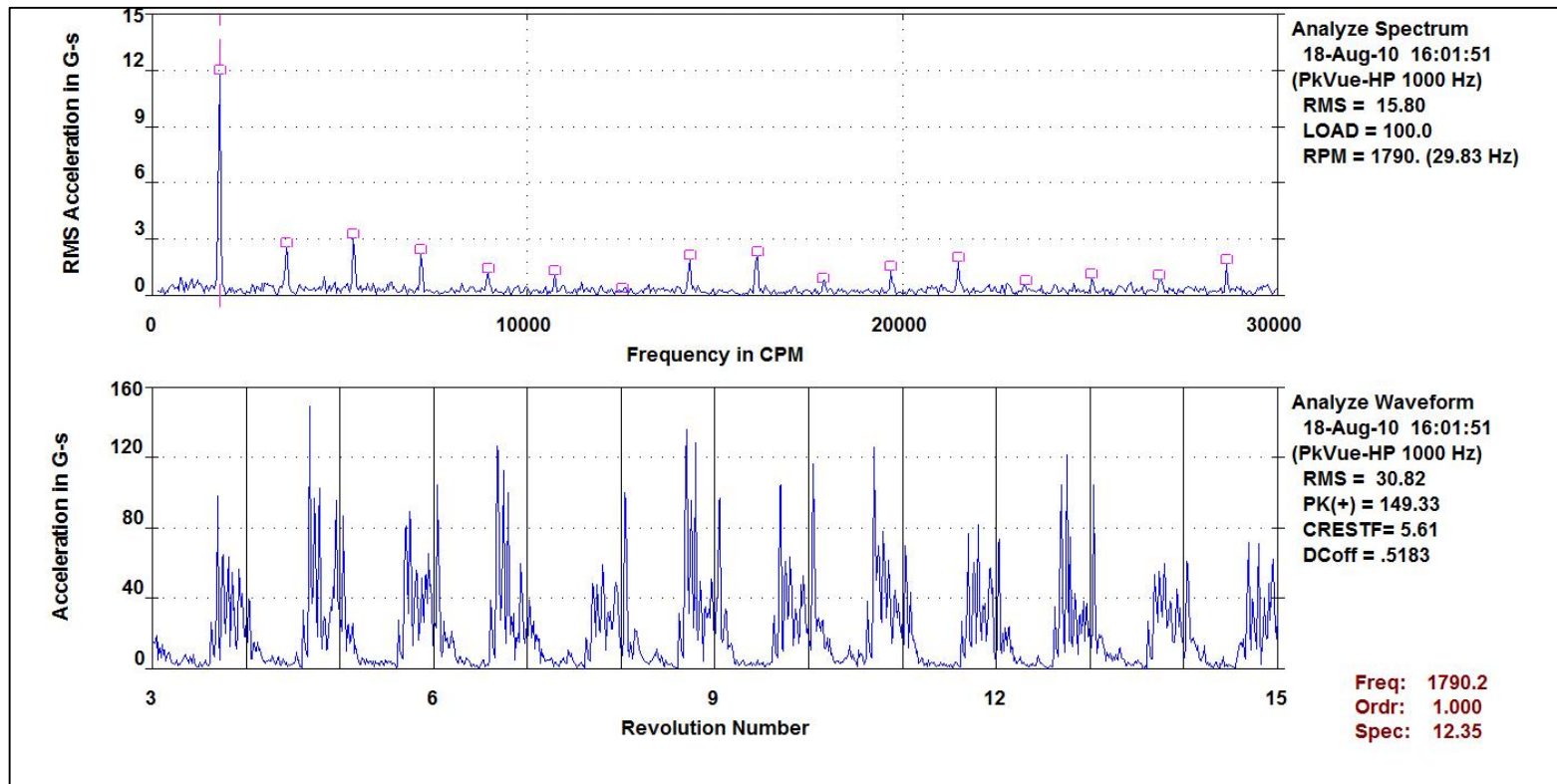
CASE HISTORY 6 – CENTRIFUGAL PUMP

- 11/12/10 – Follow-up vibration data after pump bearing changed and seal repaired.
- Profile plot of overall vibration levels shows a significant drop in levels. All vibration levels low and now not concentrated at pump outboard as before.



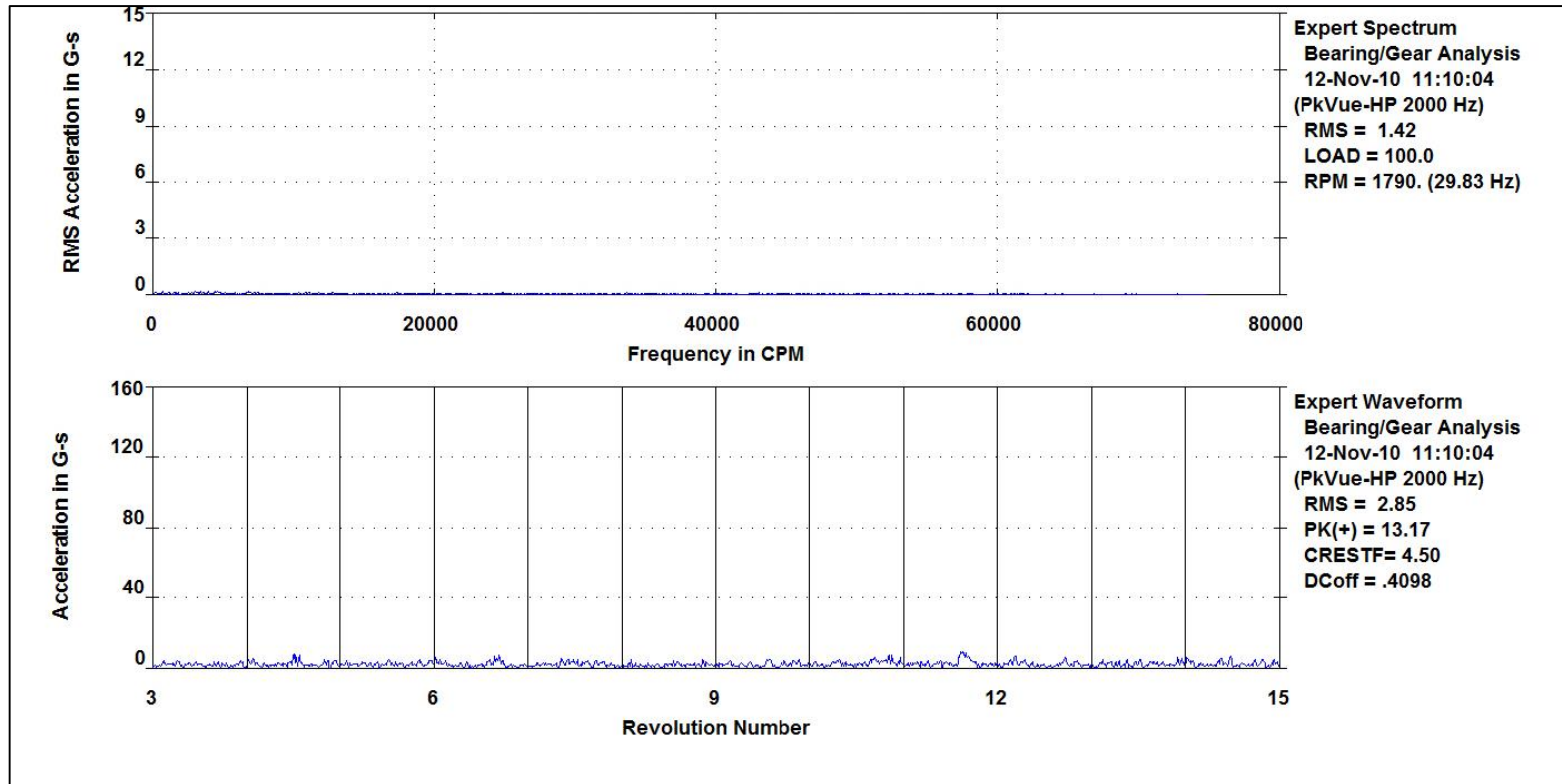
CASE HISTORY 6 – CENTRIFUGAL PUMP

- 8/18/10 – Peakvue spectra and waveform data of loose pump outboard bearing (stage 4 bearing failure).
- Dominant vibration at machine speed and multiples.
- Severe Peakvue waveform levels at 149 g's-pk!!!



CASE HISTORY 6 – CENTRIFUGAL PUMP

- 11/12/10 – Follow-up vibration Peakvue data after pump bearing changed.
- Note the significantly lower levels with identical scaling at spectra.
- Peakvue waveform levels go from 149 g's-pk to 13 g's-pk.

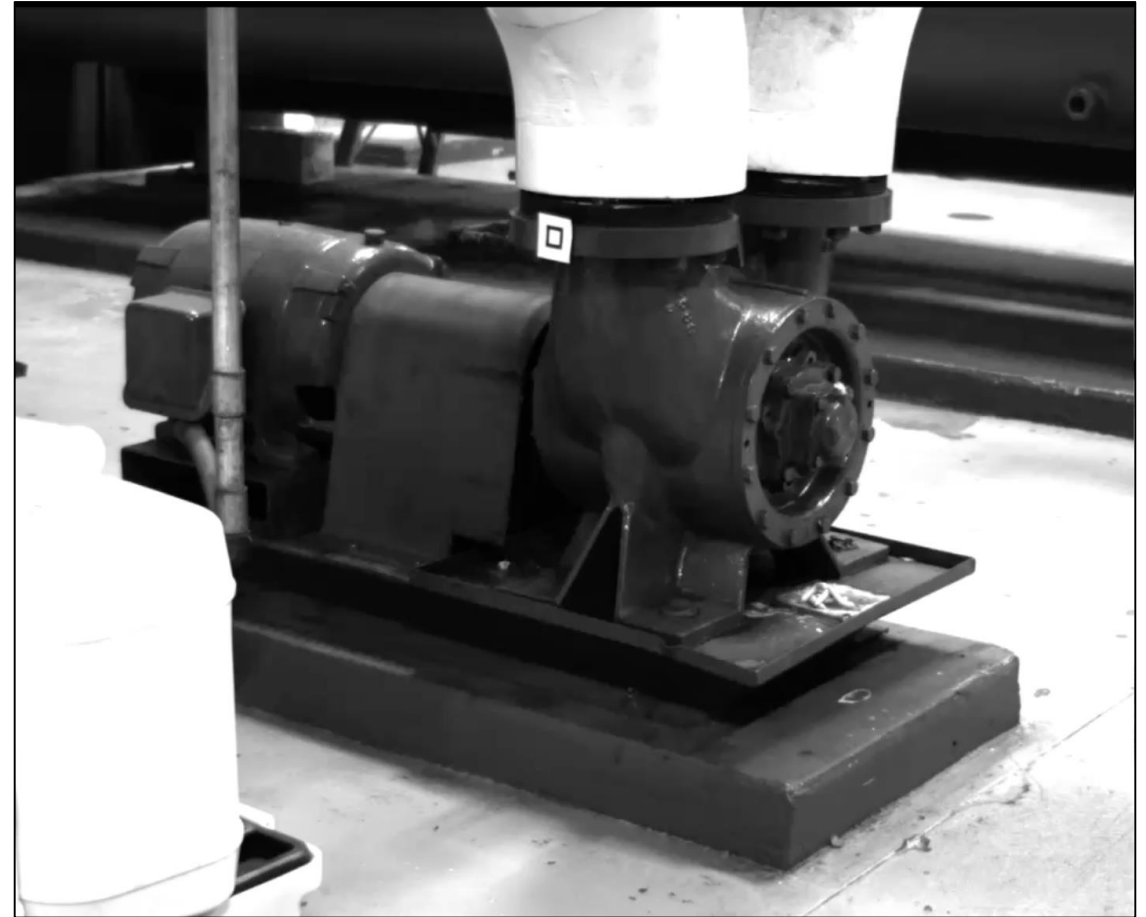


CASE HISTORY 7

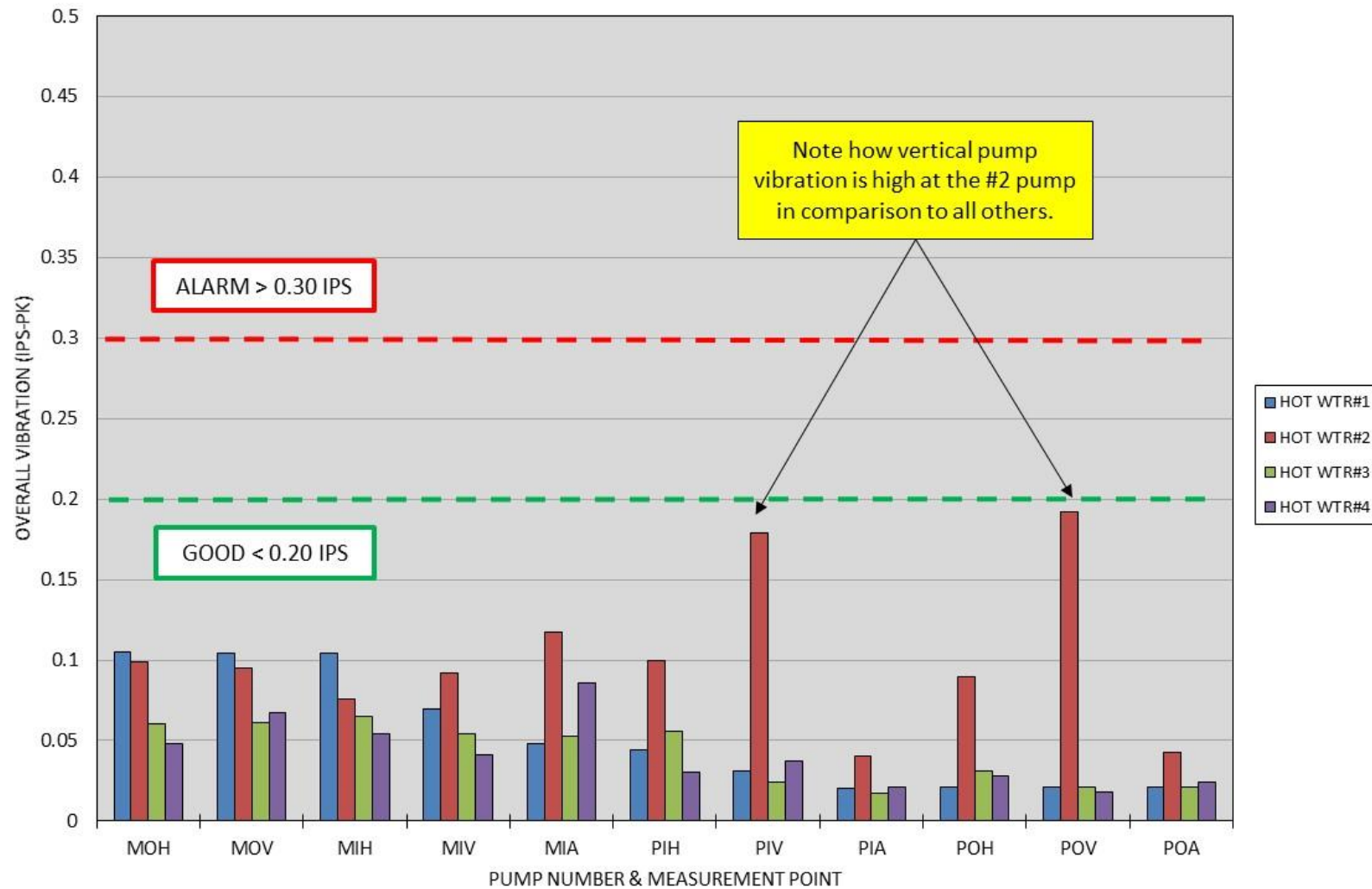
CENTRIFUGAL PUMP WITH VERTICAL VIBRATION PROBLEMS

CASE HISTORY 7 – PUMP WITH VERTICAL VIBRATION

- The seventh case history involves a horizontal, centrifugal pump driven by a 4-pole motor operating at $\sim 1,790$ rpm.
- Photo at right shows the view from the pump outboard end of this machine.
- Routine vibration surveys repeatedly noted unusually high vertical vibration levels at a particular hot water pump at a plant.
- This small pump was one of four Hot Water Pumps in the boiler room of a plant.
- Only this #2 Hot Water Pump continued to show elevated vertical vibration levels.



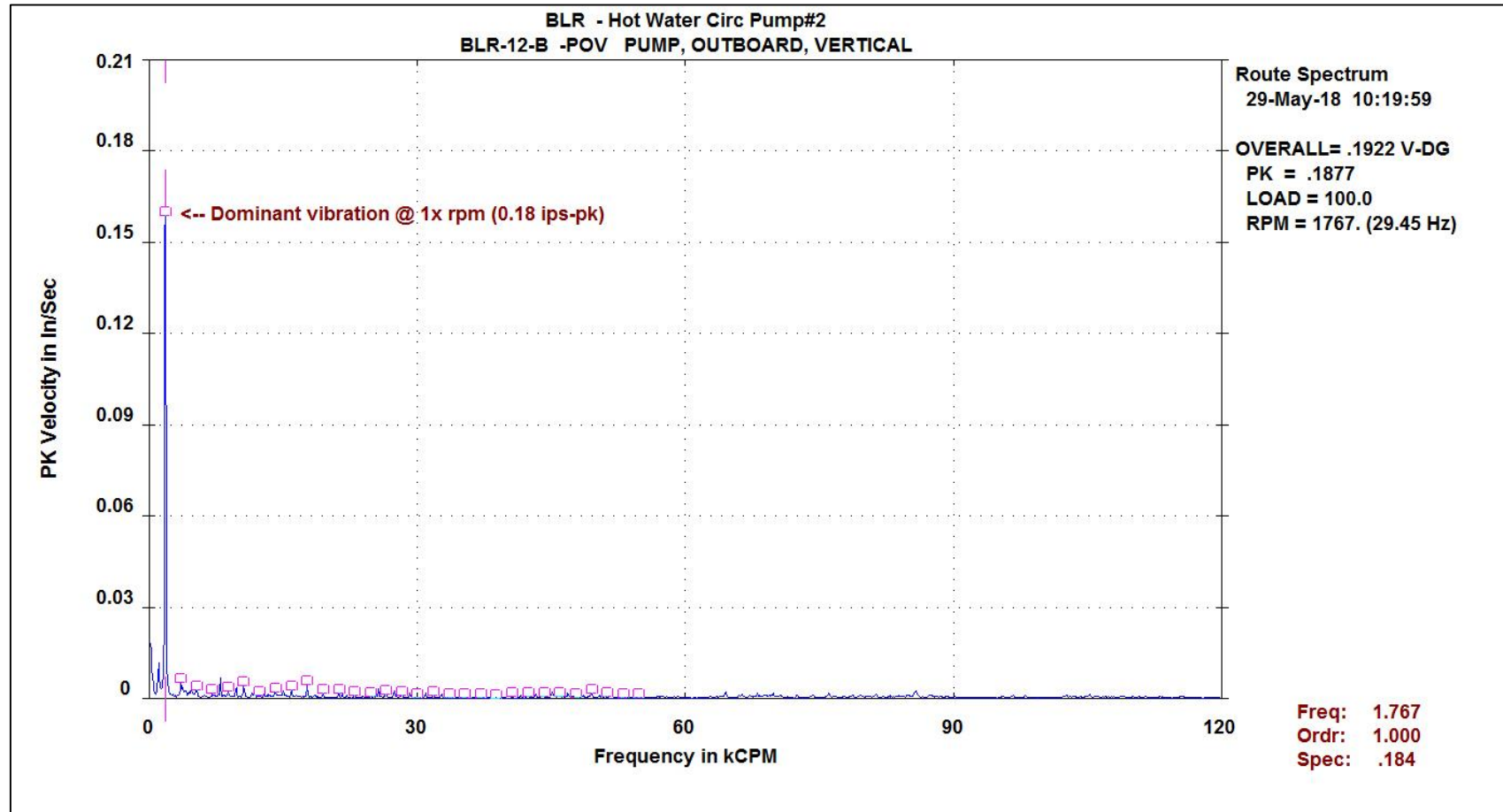
BOILER ROOM, HOT WATER PUMPS,
OVERALL VIBRATION LEVELS, MAY 29TH, 2018



CASE HISTORY 7 – PUMP WITH VERTICAL VIBRATION

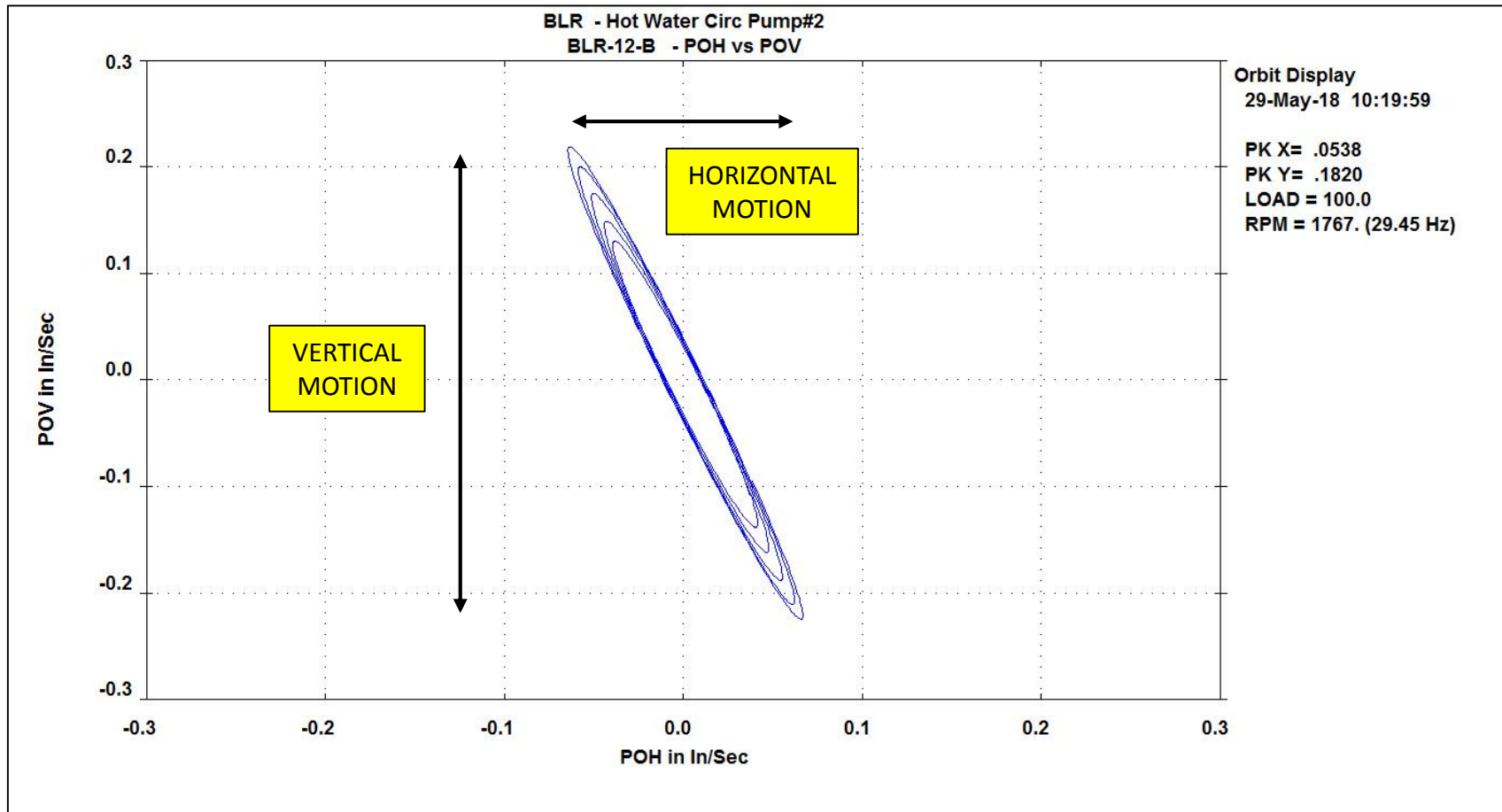
- Profile plot of overall vibration levels shows high vertical vibration levels only at #2 hot water pump – why?
- No problems seen at the #2 **motor** measurements.
- Elevated pump vertical vibration not seen at any other hot water pump even though on paper they are all the same pumps operating at the same speeds and on the same type of bases, etc.

CASE HISTORY 7 – PUMP WITH VERTICAL VIBRATION



- Dominant frequency of vibration at the points of highest vibration was at 1x rpm (1,767 rpm).

CASE HISTORY 7 – PUMP WITH VERTICAL VIBRATION



- Orbit plot of the pump outboard bearing filtered at the running speed 1x rpm (~ 1,767 rpm).
- Note how the overwhelming amount of motion is in the vertical direction.
- Compare ~ 0.4 ips-pk-pk vertical (POV) .vs. 0.12 ips-pk-pk horizontal (POH).
- Note how the orbit plot provides a good visual sense for what is going on at this pump.
- Remember how a symptom or fingerprint of looseness is directional vibration.

CASE HISTORY 7 – PUMP WITH VERTICAL VIBRATION

- So from the comparative profile plots we knew the #2 hot water pump stood out from all others as having unusually high vertical pump vibration levels.
- From the pump spectral data in the vertical direction we knew that the dominant frequency of vibration was at the pump speed of $\sim 1,767$ rpm.
- Orbit data from the pump bearings confirmed the directional nature of the pump motion (vertical).
- But these hot water pumps were mounted on rigid not isolated bases, so we would expect our vertical stiffness to be higher than our horizontal stiffness. Something unusual was going on.
- From the vibration data observed thus far we had good reason to suspect either resonance or looseness was present particularly at the pump, but where exactly was the looseness located?

CASE HISTORY 7 – PUMP WITH VERTICAL VIBRATION

- Additional inspections were performed on the pump including a motion amplification (MA) study.
- The MA study found only the sides (right & left) of the pump base were supported and held with the center portion left completely unsupported thereby lowering the vertical stiffness significantly at the exact location thru which the line of action of the forces at the pump were acting (shaft centerline).
- The anchor bolt locations were relatively tight, but the area between the anchor bolts were unsupported and loose.
- The permanent solution is to grout in the base properly. The temporary solution recommended to the plant was to add SS shims as necessary between the pump base & concrete pedestal to eliminate the looseness.
- I look forward to collecting follow-up data on this pump.

CASE HISTORY 7 – PUMP WITH VERTICAL VIBRATION

Morals of this story:

- Looseness is often directional in nature (remember the profile plots and vertical orbits).
- Looseness is often localized (remember the profile plots showing elevated vertical vibration at the pump but not at the motor).
- Motion Amplification was effectively used not to identify that a problem existed (we already had good cause to suspect either looseness or resonance from our route data) but to help pinpoint the exact location of the looseness.
- Being there matters again. Effectively analyzing and pinpointing the exact location of this problem would have been much more difficult if not impossible if I hadn't been there on-site.

THE END