Balancing of aeroderivative turbine

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Abstract

GE's Machinery Diagnostic Services team was invited to investigate vibration issue on an aero derivative gas turbine installed on a power plant. Machine train consist of a generator and aero derivative turbine. The unit was under commissioning and it wasn't possible to have it at nominal speed for a long time because of high vibration on Power Turbine bearings. Vibration testing was carried out using ADRE 408 DSPi (Dynamic Signal Processing Instrument) to hook up vibration data from the existing Bently Nevada's 3500 Series vibration monitor system. Relative vibration data were collected from the Bently Nevada's Rack 3500 monitor buffered outputs. Each of the Machine Train's bearings is monitored by Bently Nevada XY proximity probe pair. Absolute velocity transducer were also installed on generator bearings and bearing #4.The shaft rotates counter clockwise, when viewed from the driver (Gas Turbine) to the driven. The first fire of this Unit was carried out the 25 May 2010. The unit had many probes in alarm. The generator DE was running in alarm and after a period of running at idle began to creep from 86um pp to 96um pp, the unit was shut down with normal shutdown sequence. During the start-up that followed, the unit tripped on high level vibration at Power Turbine area during the run up. The unit was showing the symptoms of high unbalance so the attempt to balance it on coupling was done. However the attempts were not successful and response of the system was found not repeatable and nonlinear so the inspection of the coupling was performed. It was found that transportation wax was not removed from coupling prior assembly. After removing the wax from the coupling (1-2kg) the unit was started successfully

1 Machine Train layout

Machine train as shown in Fig. 1 consists of a gas turbine driven a Generator. Each of the machine train's bearing is monitored by displacement transducers mounted on the machine casing in a plane (XY) perpendicular to the rotor axis of the machine to observe radial motion of the shaft. The XY pairs of non-contacting proximity probes are mounted at 45-degrees left (Y-probe) and 45-degree right (X-probe). A pair of vertical and horizontal seismic probes was temporary mounted on bearing #4. The machine train diagram is shown in Fig 1. The shaft rotates counter clockwise, when viewed from the driver to the driven

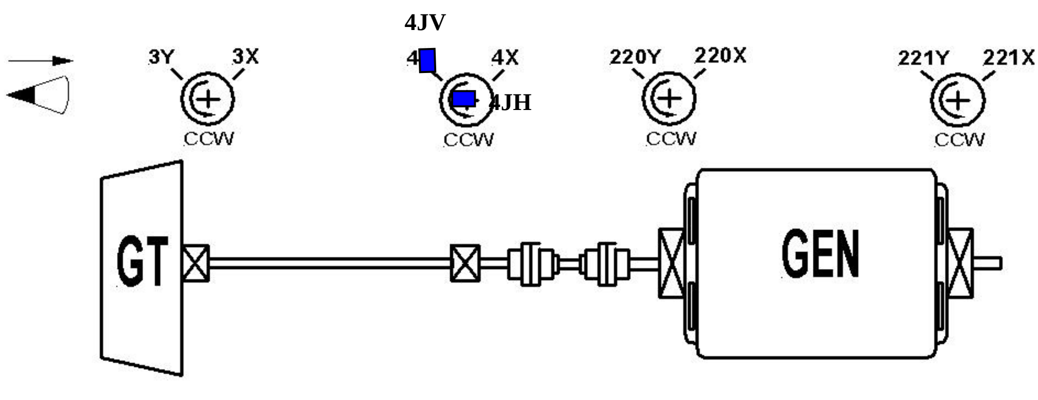

Figure 1: Machine train layout

2 Discussion

2.1 What is the issue ?

During first start-up of the unit the machine reached full speed and then the level of vibration started to increase slowly around bearing #4 and # 220. It was then decided to shutdown the unit. After the second start up, very high vibration levels on bearings #4 and #220 were noticed and synchronization speed couldn’t be reached because of high level of vibration at Power Turbine area during run up. This behaviour is not common and it indicates that something happened on the machine between the both runs. It is an indication of non-linearity of the phenomenon but it was not possible to conclude on the modification

The commissioning team checked twice the coupling mounting and the alignment but proven ineffective. The DCS trends showed only overall vibration trends so diagnostic system was needed to obtain more informative characteristics. A first set of vibration data was recorded using ADRE 408 DSPi (Dynamic Signal Processing Instrument) connected to vibration signals from the existing Bently Nevada’s 3500 Series vibration monitor system. During the start up the vibration level exceeded the vibration limits. For bearings #3 and #4 the alarm level was 160 um pp and the danger level was 230um pp. The shaft vibration for bearing #4 reached **385um pp** and the casing vibration was **70mm/s pk** (Fig.2 & Fig.3) at 2900rpm. The vibration was well above the limits

2.2 Data analysis

The high level of seismic probes indicated that an important amount of the energy of the rotor displacement was transferred to the casing. From Bode plot (Fig.2), it could be noticed that 1X was the main component of the vibration signal and 1X was increasing with the speed. The behaviour of the machine looked to be the same during start up and shutdown which means that the behaviour of this phenomenon was repeatable

At this point of the investigation, there was a doubt about the cause of the high vibration level since it seemed that from one run to another one the behaviour was not linear. However, the comportment of the machine seemed to be repeatable between a startup and the shutdown associated.

Moreover, the orbit shapes (Fig.4) was showing several flat portions from 1200rpm to 3000rpm which could be associated to rub condition. It is always difficult to know if the rub is the cause of the high vibration or if the rub is the consequence

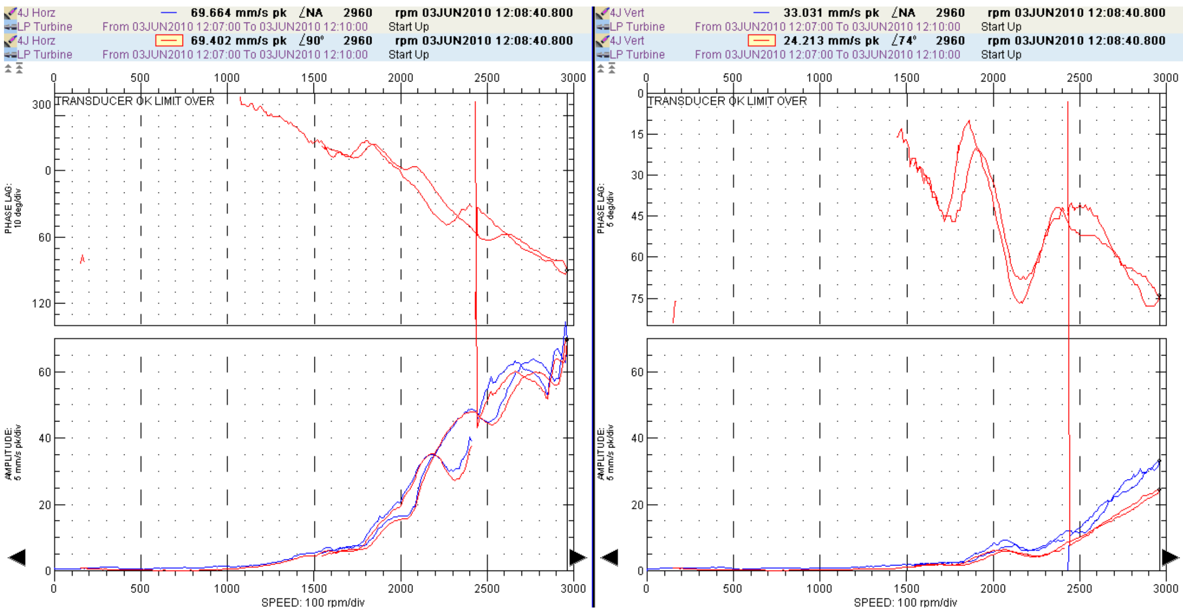

Figure 2: Bode plot, direct and 1X for velocity probes 4JH and 4JV (bearing 4)

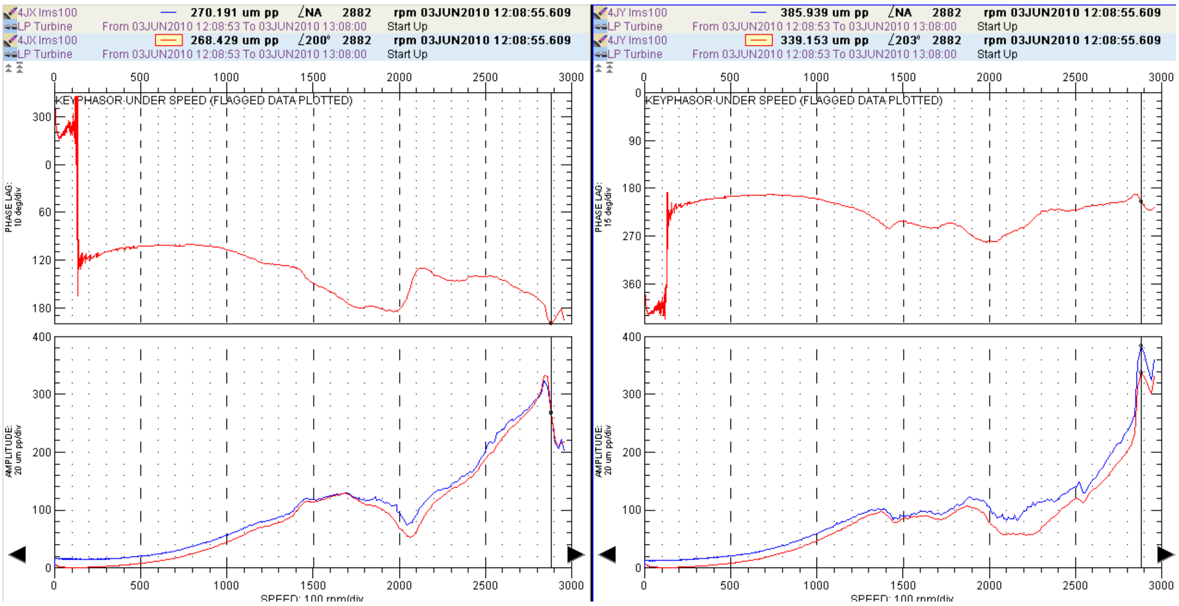

Figure 3: Bode plot, direct and 1X compensatedfor probes 4JH and 4JV (bearing 4)

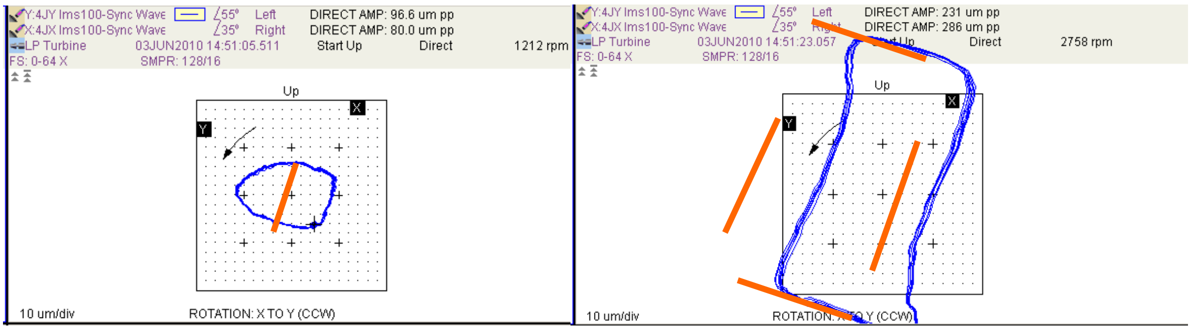

Figure 4: Direct orbit for bearing 4 showing rub condition

From all the previous observations, it was decided to try to balance the unit.

Selecting a trial weight can be a dangerous proposition. In many cases, the vibration on the machine is already high, and adding an excessive trial weight could increase the vibration amplitudes. If the vibration is already high, it is good not to make a lot of trial runs without reducing the vibration. So a good practice is to at least get in the correct quadrant for the trial weight location and add enough weight to get some response. In all cases, the ideal trial weight would be one that provides enough effect to calculate a final weight from, yet small enough that, if the phase lag is much different than expected, the vibration won't significantly increase. This selection is not easy unless a reasonable amount of information is known about the machine. Adequate trial weight would be considered to be enough weight to produce a 10% change in vibration vector based on the original vibration.

Not enough information was available to estimate correctly a trial mass and the typical trial mass was not known for the first trial run. It was decided to add a trial weight of 100g on the coupling in order to decrease the unbalance. A small effect on the angle was observed but the amplitude of the vibration didn't change. Another run with a trial weight of 300g was done to try to have an influence on the amplitude. The vibration was not affected. Then a calculation was carried out to try a last time to balance this unit. The last run had no impact on the vibration

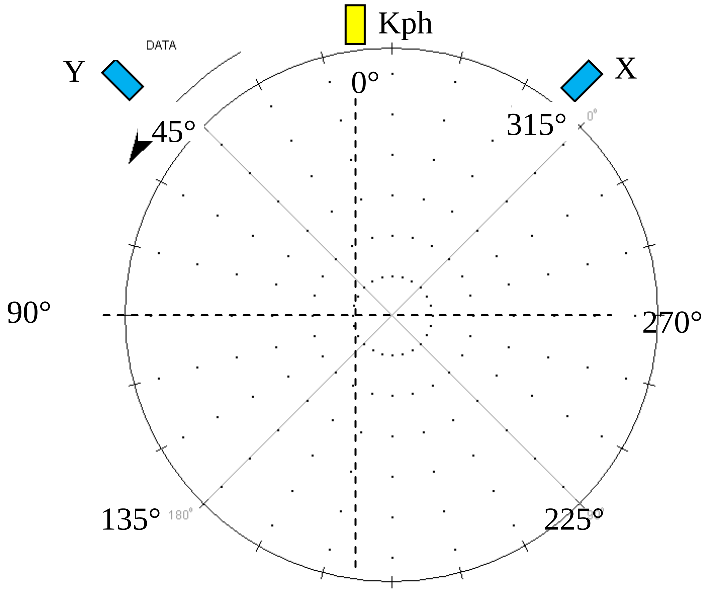

Figure 5: Balancing plane

- Initial run (Blue) was recorded by generator OEM team.
- 1st Run (Red): A trial weight of 100g was added at 0° from the Generator Kph. This weight had an insignificant effect on the vibration amplitude but it changed the angle by 90°
- 2nd Run (Green) a trial weight of 200g was added at 20°CCW from the Generator Kph. The 100g at 0° was not removed. The weight had an insignificant effect on the vibration amplitude and on the vibration angle.
- 3rd Run (Blue) 200g and 100g weights were removed. 80g at 50° CCW was added. This modification had no effect on the unbalance.

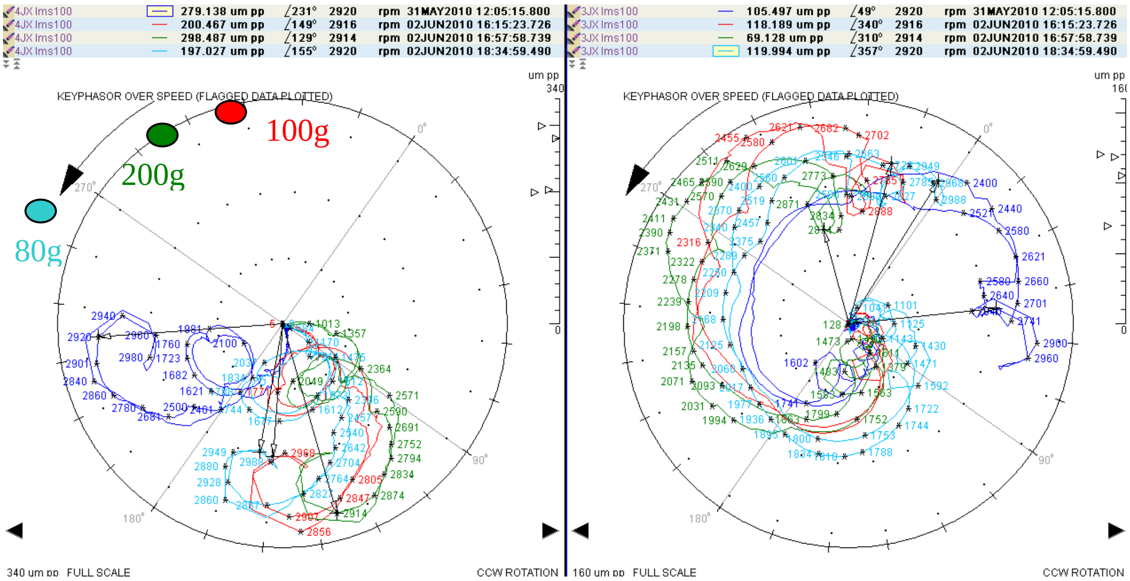

Figure 6: Polar plot for probes 4X and 3X. Several runs

After these three runs, the balancing was stopped since the weights had no influence on vibration level. The next day, it was decided to remove all the weights and to do two other runs. The goal was to check if the behaviour of the machine was repeatable or not

On the following polar plot (Fig.7), three runs are plotted

- Initial run (Blue) was recorded by generator OEM team.

- 4th Run (Red): all the weights were removed, so the machine was in the same condition than the 31st of May run. The angle was shifted by 90°. The machine had not the same behaviour in the same condition. It was concluded that behaviour was nonlinear and something moved or changed between the 1st run and the 4th run.

- 5th Run (Green) A trial weight of 100g was added at 220°CCW from the Generator Kph. The weight had again NO effect on the vibration amplitude and on the vibration angle.

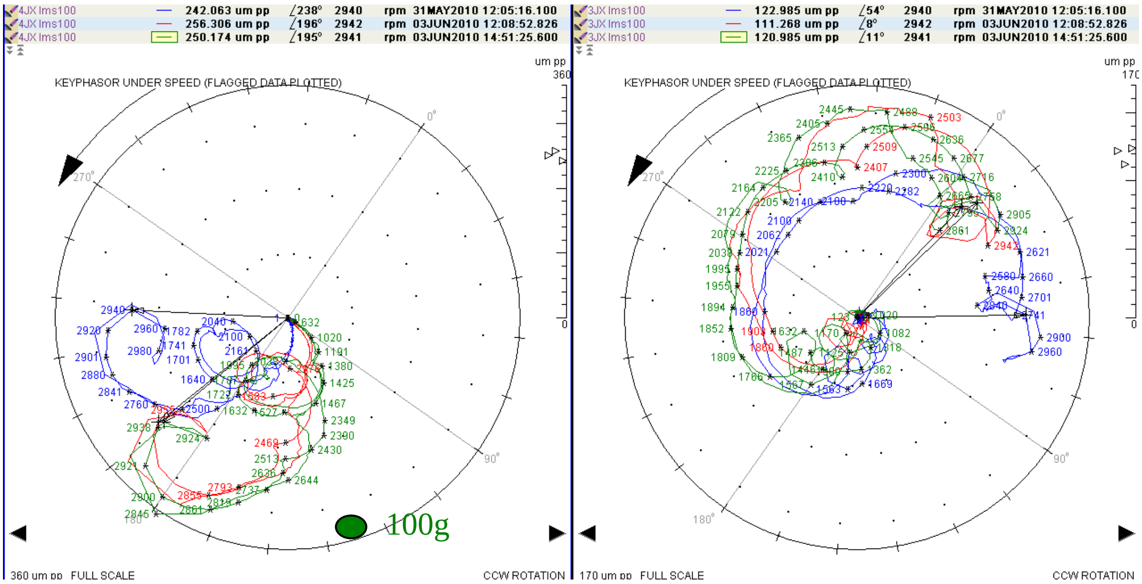

Figure 7: Polar plot for probes 4X and 3X. Three runs

The last run had no impact on the vibration. Therefore, it was decided to stop balancing activity. At this point of the diagnostic, the malfunction was not known and it was difficult to give any conclusion. Since the coupling had already been checked several times, the next logical step would have been to open the bearing #4. This type of intervention would have delayed the commissioning of the unit and the cause of that high vibration would not have been found.

2.1 Results from Data analysis

Before to potentially open the bearing #4 and since this machine behavior was very strange, a meeting was organized between engineering, factory and field representative. This discussion was very helpful since this case was known

There are some basic assumptions that are made when doing field balancing. These include: linear response, accurate/repeatable test measurements, and consistent weight placement. Linear response of a system simply states that if we get a response of 10 µm pp for a certain weight size that we should get a 20 µm pp response for twice that amount of added weight This will normally be mostly true unless the rotor is severely out of balance Accurate and repeatable vibration amplitudes is another assumption. It is absolutely mandatory to assure that the same machine conditions are used during vibration measurement for each balance run including rotor speed, machine load, heat levels, etc

Prior to attempting any balance corrections, a proper vibration analysis should be done to determine the likelihood that the machine is in fact out of balance. Making balance corrections to a machine with some other fault can in many cases reduce the vibration amplitudes. However, if balance corrections are made to a machine that is not out of balance to start with, the forces generated by the fault will still exist even though balance corrections may reduce the amplitude at some measurement points

From field and factory experience it was decided to disassembly the coupling between the generator and the turbine to check if the shipping protection wax was removed before the start of the unit. At each end of the coupling, one kilogram of wax was found. For this unit, the typical mass, which is used to balance it, is 20g. This wax (Fig.8) was the cause of the high vibration level and it was a very important unbalance (around 1kg / 2kg)

After removing the transportation wax the last run was done to verify the level of vibration. The machine was able to reach full speed with a low level of vibration (Fig.9)

Figure 8: Picture of the was which was found in the coupling

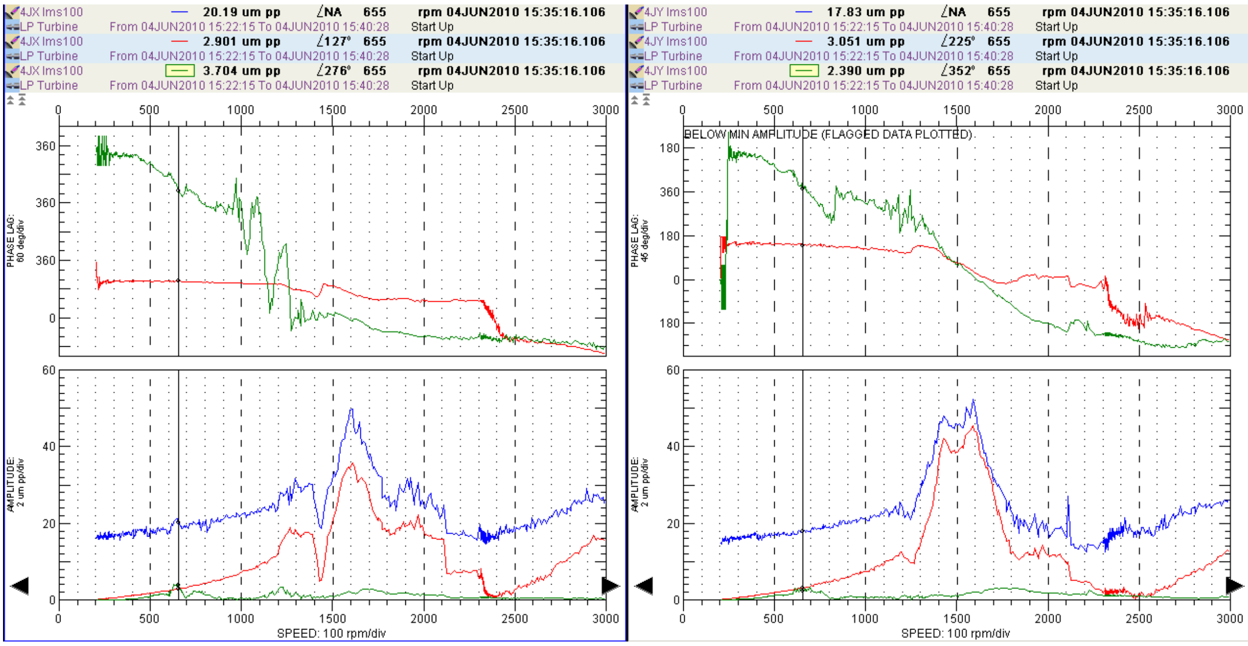

Figure 9: Bode plot, Direct, 1X compensated and 2X compensated for probes 4JX and 4JY after the was was removed

3 Conclusion

A successful diagnostic is also depending on the cooperation between different teams involved in the root cause analysis. It is always important to share knowledge in order to save precious time on site and determine quickly the correct cause of any malfunction.

Vibration in rotating machinery is commonly the result of mechanical faults including mass unbalance, coupling misalignment, loose components, and many other causes. Improving the levels of vibration should always include elimination of the source of vibration and not addressing the symptom by making balance corrections. When trying to balance a machine someone should always have in mind if balancing is the correct action or if the results of the balancing are making sense

Unfortunately, other common faults can also generate high levels of vibration at 1xRPM including coupling misalignment, looseness, rotor bows, and a variety of other sources. In some cases, these faults will produce other symptoms that can suggest corrections other than balancing should be done. Yet in many cases, balancing may be the chosen course of action for lowering vibration amplitudes even though it is not the source of vibration

Once it is determined that balance corrections should be made, the balancing process includes measuring reference vibration, adding trial weights, observing the response due to trial weights, and using the response characteristics to determine the location of balance correction weights to reduce vibration to an acceptable level

This case shows something important when dealing with balancing. It is always good practice to inspect the rotor for causes of unbalance to determine if there is a good reason for the rotor being out of balance and to help identify other problems that could exist. In some cases, previous balance weights may have come off due to being poorly installed or improper materials (corrosion). The other benefit of this type of visual inspection is the benefit of looking at previous balance weights that have been used as a mental reference point for the selection of trial weight magnitudes