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BALANCING 3-BEARING ROTORS

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ABSTRACT

Multistage cryogenic pump rotors are vertically oriented and supported radially by three bearings. However, while the rotors are oriented vertically and supported in three planes, they are balanced horizontally in two-plane balancing machines. Since the balancing machines are designed for two planes of support, a third support usually at the far end, is needed to support the rotor.

There are two major issues that may affect the quality of balancing a three-bearing rotor as compared to a two-bearing rotor: The first issue is runout, which is essentially eccentricity that can contribute to unbalance. The shaft runout should be measured with the rotor supported at two locations. For rotors that will have short length ball bearings added after the balancing operation, the two support locations of the balancing machine is typically used, and the far end is left without support. The measurement on the far end could be large, and it is easily mistaken as sag and left without correction.

The second issue is how to adjust the position of the third support when preparing to balance. The support is usually a two-roller device, and the regular adjustment method is to place it under the 3rd bearing location and raise it gradually to touch the rotor such that when the rotor is spun, both rollers are spinning. The result of this adjustment, however, is for the far end to be in a deflected sagged condition, which is different from the operating condition where all three bearings are aligned.

This paper explains the difference between runout and sag, and presents a method to adjust the support such that the rotor can be correctly balanced. The influence of the support adjustment is shown with an example.

INTRODUCTION

Most turbomachinery rotors are supported radially with two bearings consisting of a single span. But in some applications, a rotor will run with three radial bearings consisting of two spans. In some of these three-bearing cases such as turbo-generators, the rotor can be separated into two shafts, which can be built and balanced individually then connected together for the final build and operation. In other cases, such as some vertical pumps [1,2], the rotor consists of a single highly flexible shaft that must be balanced with three supports due to rotor flexibility, which is the focus of this paper. The rotors for cryogenic pumps will be used in this paper, and methods can be generalized to other applications.

A review of Rieger [3] and Darlow [4] indicates that balance calculation methods have been well developed in the literature including multiple planes, measurement points, and multiple spans. However, practical discussion of multi-span balancing has been notably sparse. Thearle [5] and Kroon [6] discuss balancing of turbine-generator sets. Fujisawa and Shiohata [7] discuss balancing of multi-span rotors including an assembled turbine-generator string consisting of five spans. More recently, Kelm, et al. [8] discussed the field balance of a three bearing turbine-generator set. All of these cases mentioned were performed with the machinery fully assembled in the field. Racic and Hidalgo [9] discuss the balancing of a three bearing rotor while supported on two bearings, which is an approach that is not well suited to the problems discussed herein as will be explained.

Figure 1 shows a typical balancing setup of the 3-bearing rotor. The shaft has a motor on the right and an inducer on the left. The two roller bearing supports are part of the low speed balancing machine. The third support (tail support, consisting of two rollers on a rack) is located on the far left and is not part of the balancing machine. It is typically separated from the balancing machine pedestal.

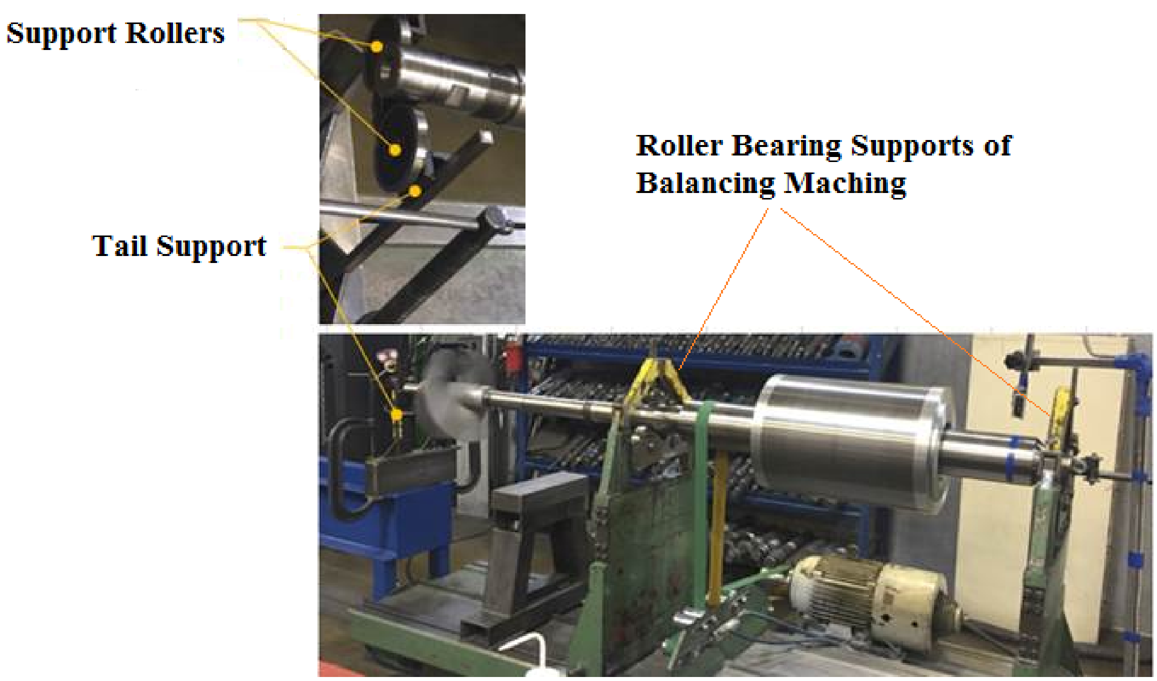

Figure 1: A typical low speed balancing setup of 3-bearing rotors

The rotor shown in *Figure 1* will be used throughout this paper, and some parameters are listed below:

Length: ~110 inches (~2800 mm)
Weight: ~1700 pounds (~800 kg)
Operating Speed: 1800 rpm
Balancing Speed: ~900 rpm

The motor core is shrunk onto the shaft and the motor core and shaft are balanced as an assembly. The balancing planes are usually on the sides of the motor core, and the overhung portion is never used for balancing. All of the installed components are balanced individually before assembly. The inducer has a slight clearance fit at room temperature, which changes to an interference fit under cold operating conditions. The inducer can be added as a progressive balancing step, but this is only performed during trouble shooting. There are two impellers between the inducer and the middle bearing (as shown in *Figure 8*), and they are not installed for the low speed balancing.

One idea discussed during the development of the methods in this paper was to support the rotor with the two end bearings only, like a regular two-bearing rotor. However, even if the balancing machine pedestal is long enough (it is marginal for this rotor as it can be seen in *Figure 1*), it is still not possible due to the long bearing span. For this rotor, the calculated static deflection at the center is close to 100 mils (2.54 mm). It is a concern to statically support the rotor with two end bearings, because it can cause permanent bow. The long bearing span will also drop the first critical speed to about 890 rpm, which is within the balancing speed range. Since the rollers do not provide much damping, the rotor and/or the balancing machine will be damaged if the critical speed is reached.

Balancing the rotor without the tail support, similar to balancing an overhung rotor (see references [10] and [11] for some details), may be viable in rare cases such that the overhung is short enough and light enough without the inducer and impellers. In general, it is not a viable solution for 3-bearing rotors, including the rotor in this paper.

Balancing using two locations other than the bearing locations is not ideal either due to differences in shaft diameter as well as higher surface finish/runout/eccentricity, etc.

Therefore, the third support (tail support) is needed, and this paper discusses how to apply/install it to improve the balancing quality.

SAG AND RUNOUT

Sag:

Sag means static deflection due to gravity in this paper.

Runout:

Depending on the measurement devices, runout can be divided as mechanical runout and electrical runout. Mechanical runout is a measure of the shaft cylindrical surface deviation from a perfectly round surface, concentric with the bearing centers. Shaft bow will show up as runout. Electrical runout is a measure of shaft surface electrical conductivity and magnetic permeability variation.

For the rotors in pumps, since the vibration measurement devices are accelerometers mounted on the outer casing, the electrical runout is not a major concern.

Note that the runout of the rotor in this paper was corrected later.

Runout Check: With and Without Tail Support

The runout of a shaft is usually checked before the balancing at multiple points along the shaft. *Figure 2* shows the cross-sectional drawing with each checkpoint labeled.

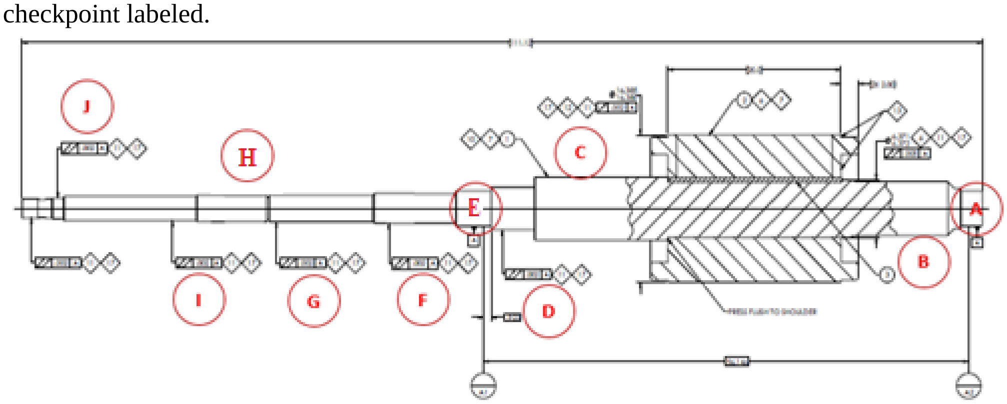

Figure 2: Cross-sectional Drawing with Runout Checking Point Labeled

There are two scenarios when the runout is checked: with and without the tail bearing. For the example rotor in this paper, the addition of the tail bearing results in runout measurements that are quite different.

Scenario one: the rotor is supported on the two main bearings (close to points A and E), and the tail bearing is not used, i.e. from D to J cantilevered. The runout result is shown in *Figure 3*. The runout for points A through F is small, but becoming progressively larger with increased overhung distance. At the tail end, the runout is close to 8 mils peak-to-peak.

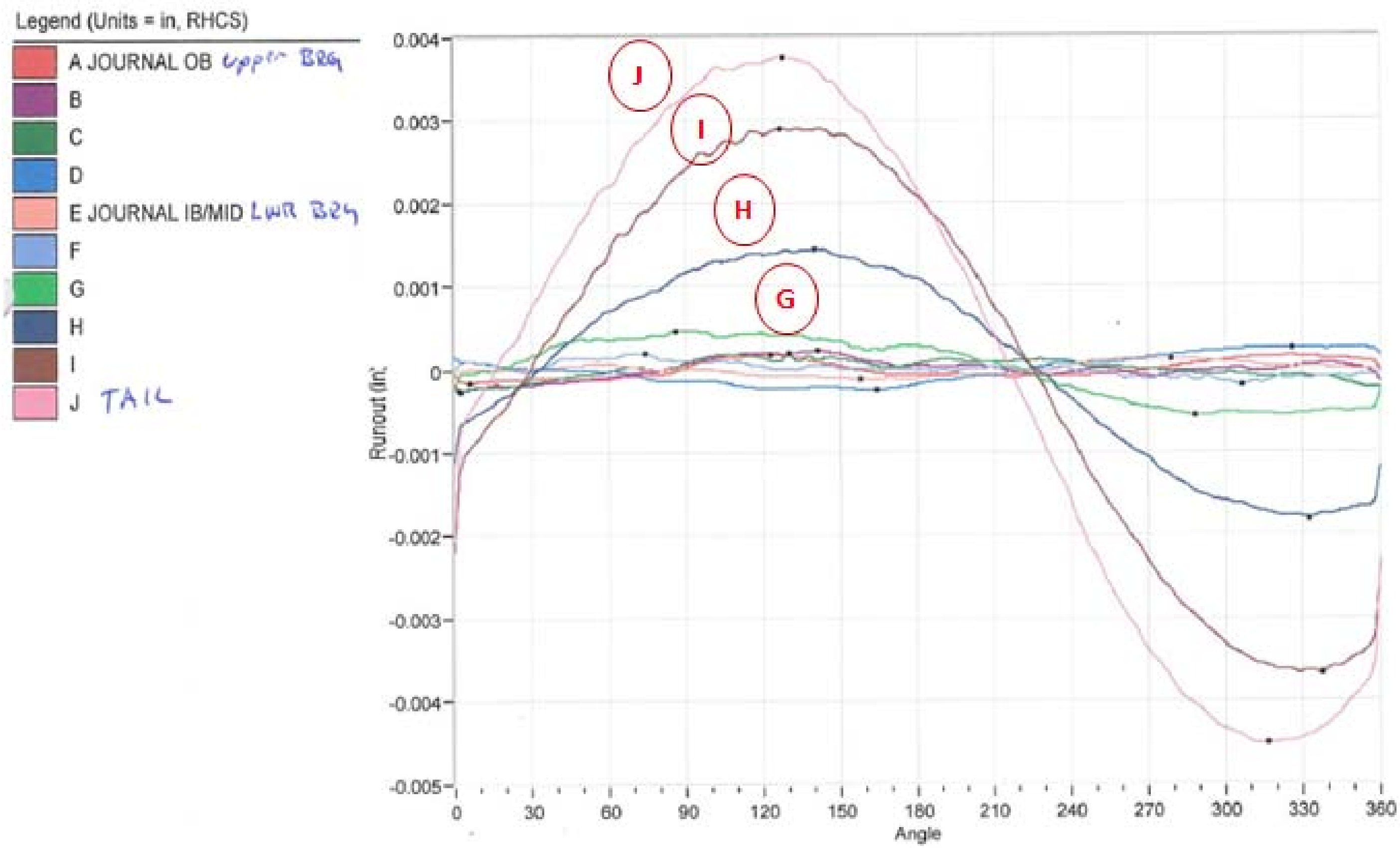

Figure 3: Runout Result at Different Points without Tail Bearing

Scenario two: the rotor is supported on the two main bearings and the tail support is also used (details of how to apply/adjust it will be introduced later). The runout as shown in *Figure 4* becomes much smaller everywhere along the shaft and the maximum is about 1.5 mils peak-to-peak. The sudden reduction of runout due to the tail support can be misinterpreted that the runout (shaft bend) does not really exist, and the rotor is just sagging.

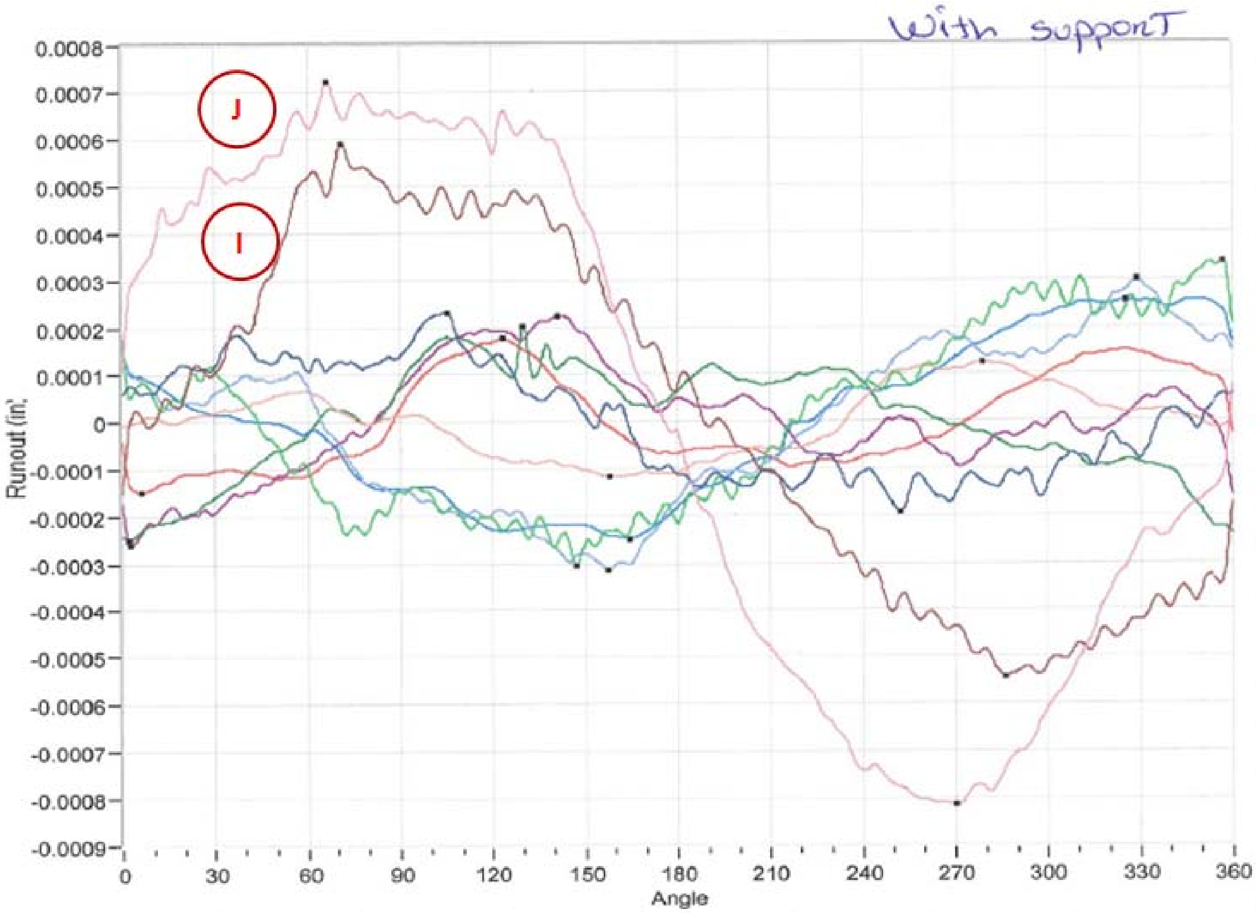

Figure 4: Runout Result at Different Points with Tail Bearing

Explanation: Why It Is Runout, Not Sag

In reality, sag is always present and the runout is superposed on top of the sag deflection. A shaft with an 8 mil peak-to-peak runout will look like the simplified shaft (exaggeratedly) shown in *Figure 5*, where the tail end is on the left. In this situation, the runout measurement would be similar to what is shown in *Figure 3*. Note that the sag is usually not measured, and it is in general difficult to measure.

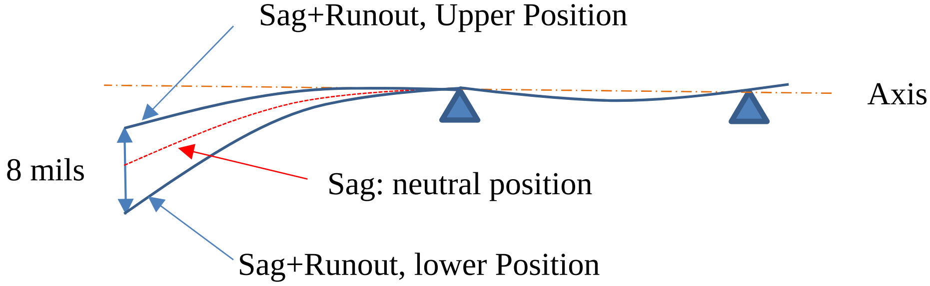

Figure 5: Sag vs. Runnout: No Tail Bearing

When the tail support is added, it is adjusted to ensure good contact between the two rollers and the shaft. This is done by raising the support till both rollers are spinning when the shaft spins, as illustrated in *Figure 6*.

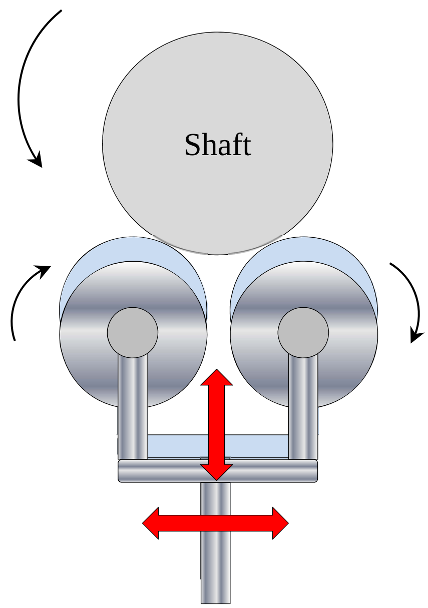

Figure 6: Tail Support Adjustment

In case of runout, the only way to get both rollers spinning continuously is to raise the tail support high enough such that it is above the upper position of the runout, as shown in *Figure 7*. Because of the restriction of the tail support, the runout will be reduced drastically. Thus, the measured runout along the entire shaft will be reduced at all angles, as shown by the measurement in *Figure 4*.

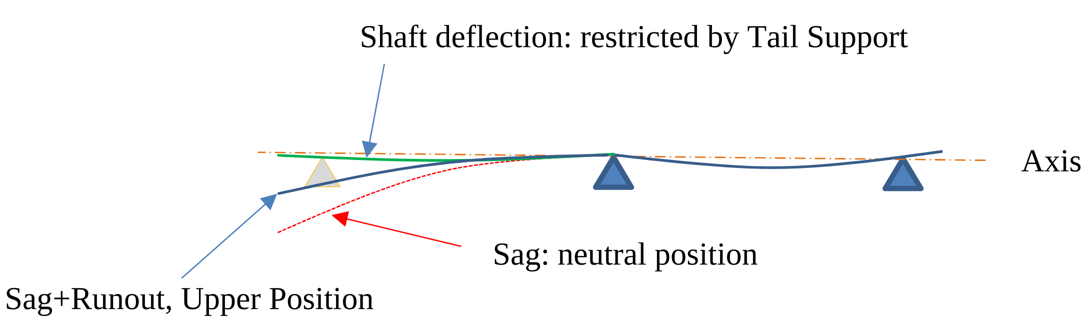

Figure 7: Sag vs. Runout: With Tail Support

Distinguishing Between Runout and Sag Using Tail Support

If the tail support is raised up gradually enough while the shaft spins slowly, then the phenomenon of runout vs. sagging can be observed as follows:

1. If either both rollers spin **continuously** or they do not spin at all at different vertical positions, it is sagging.

2. If the rollers spin and stop (or slow-down) at certain position, it is runout.

3. If only one roller spins all the time and the other does not spin at all, then it is horizontal positioning problem.

Usually it is a combination of sagging, runout and horizontal position to some extent.

ALIGNING THE TAIL SUPPORT

Directly aligning the tail support with the other two might be difficult and unnecessary. Since the shaft can be used as a reference, a few methods are presented in this section using the shaft to adjust the tail support.

Strain gauges

When the rotor is hung upside-down as shown in *Figure 8*, the bending stress at the middle bearing area should be 0. Strain gauges can be applied at this time. When the rotor is placed on the balancing machine, position the tail bearing such that the strain gauges will read 0.

Figure 8: Rotor Hung Up-Down

The advantage of this method is that no additional information or calculation is needed. The disadvantage is that there will be more instrumentation, and the accuracy of the measurement needs to be very high because the strain due to sagging is usually small.

Calculated Static Shaft Deflection with FEA Programs

Most rotordynamics programs provide accurate static deflection, as shown in *Figure 9*, which matches the rotor configuration in *Figure 1* (with inducer). As shown in Figure 10, there is considerable deflection at the far overhung position. Without support, this overhang provides increased load at the center bearing and decreased load at the motor end bearing. There is also a considerable moment on the shaft and any unbalance of the overhung shaft end will indicate a wrong correction in the motor balancing planes.

The static deflection number can be used to adjust the tail support up from the sagging position. Other Finite Element Analysis (FEA) type programs can also be used to generate the static deflection. Note that the balancing configuration may not be the same as the operating configuration, such as when the impellers are not mounted to the shaft. The rotor model needs to match the balancing configuration.

Overall, the advantage of this method is that it is accurate, and no additional instrumentation is needed. The disadvantage is that an analysis program is needed, and the analytical rotor model needs to be properly built / adjusted.

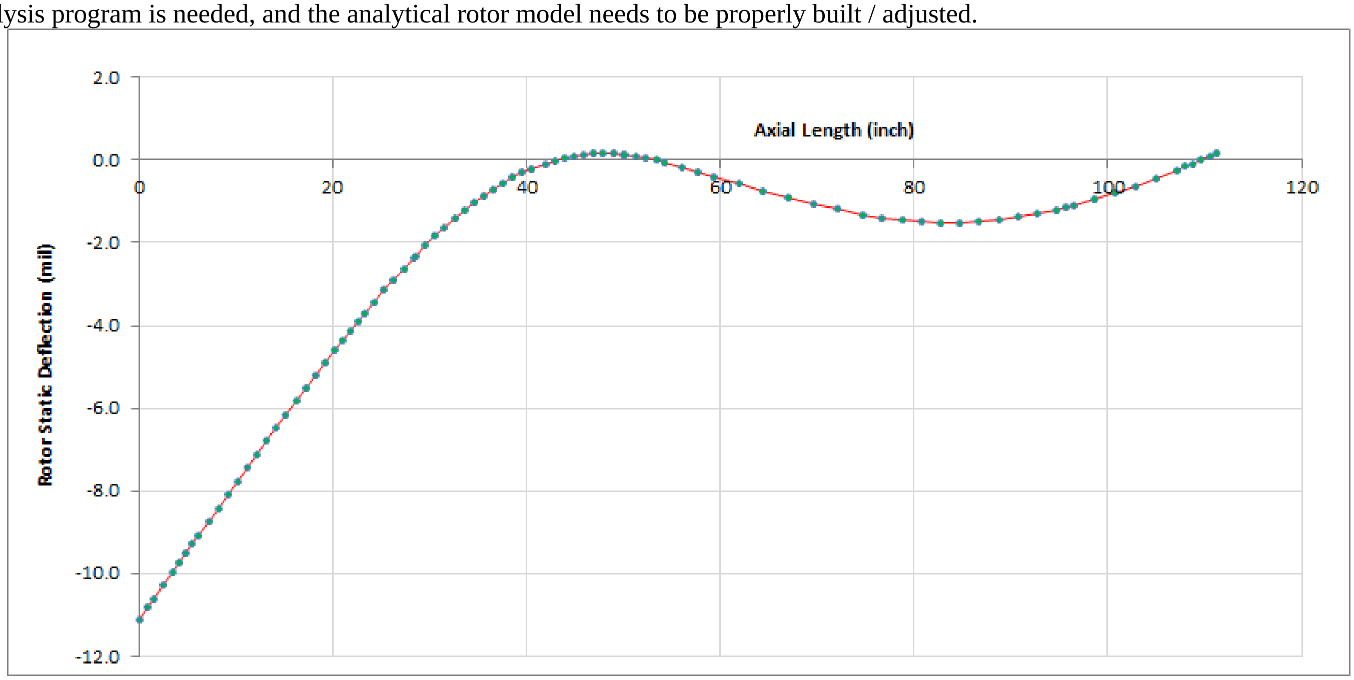

Figure 9: Static Deflection with Inducer (by Rotordynamics Program)

Beam Deflection Equation

If neither of the previous two methods is available, then a rough estimation can be carried out by applying the equation for cantilever beam deflection with uniform distributed load [12]:

$$\delta_x = \frac{w}{2EI}\left(\frac{L^2x^2}{2} - \frac{Lx^3}{3} + \frac{x^4}{12}\right) \qquad (1)$$

where

x is the length from the middle bearing to the tail bearing

δ_x is the deflect/sag at the tail bearing from the centerline

w is the uniform load due to gravity

L is the overhung length (from the middle bearing to the end of the tail-bearing shaft end)

E is Modulus of elasticity
I is Area moment of inertia of cross section

Since this is already an approximation and *x* is close to *L*, the equation can be simplified by substituting *L* for *x* [12] such that

$$\delta_x = \frac{wL^4}{8EI} \tag{2}$$

Since quite often the shaft is stepped down toward the tail, the deflection calculated using the first/largest shaft diameter is closer to the calculated shaft deflection (bare shaft only), but usually larger.

For the example rotor, the static deflection of the bare shaft (i.e. no inducer) calculated by the rotordynamics program is shown in *Figure 10*. The calculated deflection is 9.8 mils using Equation (2) and 9.4 mils using Equation (1), roughly 50% (3 mils) more than the value calculated by the rotordynamics program.

When there is additional mass on the shaft, such as an inducer, the estimated deflection may not be larger, as shown in *Figure 9*.

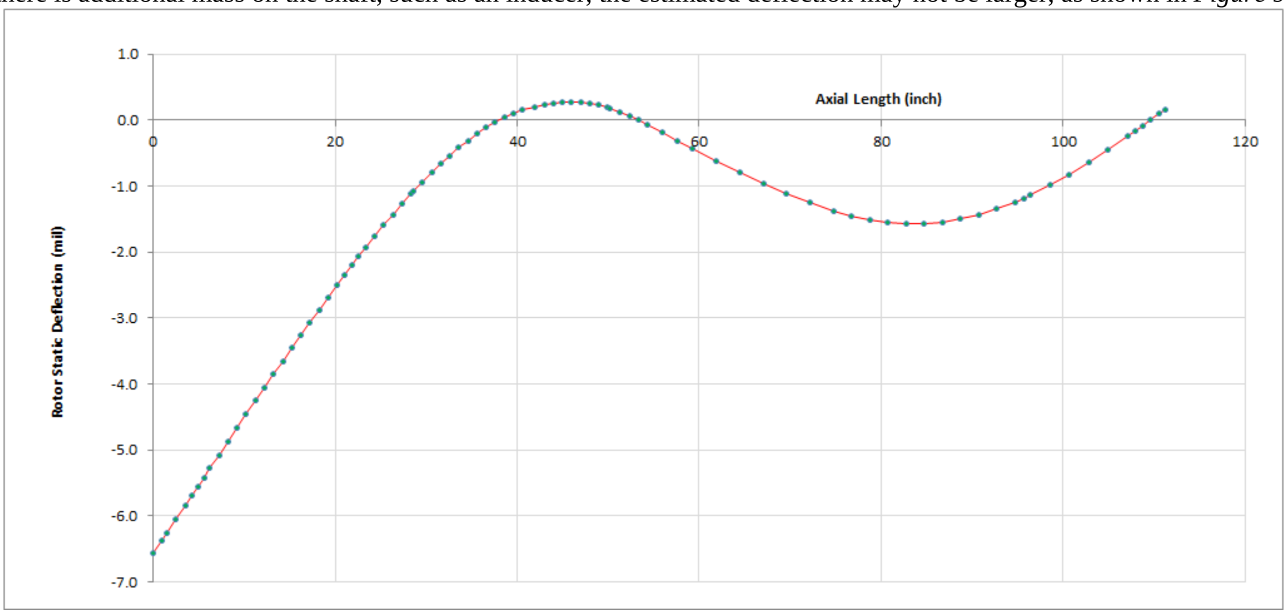

Figure 10: Static Deflection without Inducer (by Rotordynamics Program)

IMPACT ON BALANCING FROM VERTICAL ALIGNMENT OF TAIL SUPPORT

Balancing Acceptance Criteria

Maximum allowable residual unbalance shown in Equations (3) or (4) as specified in many API standards (e.g. API 610 [13]) is often used as the low speed balancing acceptance criteria, and it is equivalent to ISO Grade[1] 0.7 (API 684[14], Paragraph 5.2.7).

In US units:

$$U = 4\frac{W}{N} \tag{3}$$

1: See Reference [15] for definition of ISO Grade

In SI units:

$$U = 6350\frac{W}{N} \qquad (4)$$

where

U is the residual unbalance in oz-in (g-mm)
W is the bearing static load in lbf (kgf)
N is the maximum continuous speed in rpm

For the example rotor, the calculated maximum allowable unbalance is 1.7 oz-in.

Measured Unbalance with Different Vertical Positions

The rotor is setup on the balancing machine and the tail support is first brought to the proper ("Normal") position based on the static deflection. Balancing check is performed and the measured unbalance is recorded. Then tail support is moved downward (Tip down) two times and each repeating the measurements.

Table 1 lists the measured unbalance, angle, and balancing speed. The middle support is called "Tail side", and the support outboard of the motor is called "Motor side". Note that no balancing is done during the unbalance measurements, so the unbalance is larger than the maximum allowable unbalance.

Table 1. Measured Unbalances

Condition	Tail side		Motor side		Speed
	oz-in	angle	oz-in	angle	rpm
Normal support	5.8	137	6.0	291	905
Tip down 0.002"	8.7	147	7.7	306	901
Tip down 0.003"	10.3	143	9.0	305	901

The unbalance angle remains steady but the unbalance changes much for different Tip positions. If plotted, the unbalances are almost linear, as shown in *Figure 11*.

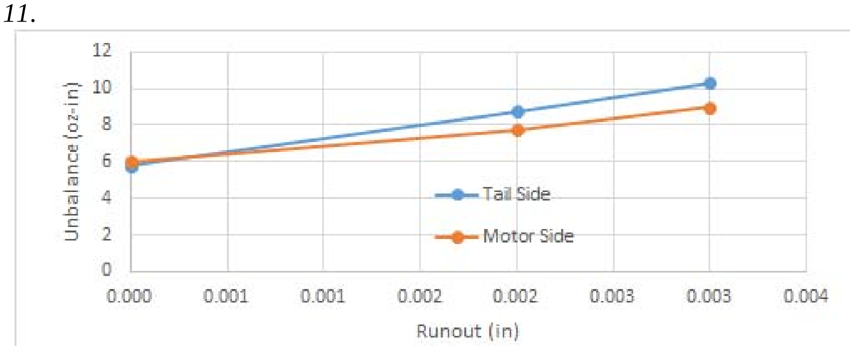

Figure 11: Unbalances with Different Tail Support Positions

From this example, it can be seen that both absolute values (comparing to maximum allowable residual unbalance) and the relative change are large with rather small changes of the tail support position.

RECOMMENDATIONS AND DISCUSSIONS

The purpose of this paper is to improve the low speed balancing quality of 3-bearing rotors. To achieve the best balancing quality, the runout of the shaft needs to be properly measured first, and controlled to a reasonable range. Then the tail support can be installed with the suggested steps below:

1. If the runout exceeds the runout tolerance, then correct the rotor.

2. Measure the runout at the tail support location.

3. Adjust the tail support such that both rollers will spin (or barely spin) at the same time. This will include both horizontal and vertical. This roller position will lift the shaft such that it is half the Total Indicated Runout (TIR) above the neutral sag position.

4. Add one dial indicator on top, and one on the side near the tail support.

5. Raise the tail support such that the top dial indicator changes the additional required amount such that the shaft centerline at the tail bearing is in the same axis as the other two supports and the side indicator remains unchanged. This will require raising the rotor further by the SAG – (TIR/2)

As shown in the previous section, adjustment of the tail support can change the unbalance multiple times of the balancing acceptance criteria. Of course, the impact will be different for different rotors and it might be good practice to get an idea of the impact by varying the tail support position for each type of rotor.

Note that the unbalance in the operating configuration could be quite different from the unbalance in the balancing configuration. The rotor is most often balanced without any component installed between the middle support and tail support. Even though the components (impellers and inducer) are individually balanced, the unbalance for the assembled rotor could change significantly after these components are mounted to the rotor. Balancing with all components installed may not be feasible. A large amount of study work would be required before it becomes a viable solution and a vertical balancing stand could be the only option for performing such work.

Secondly, the balance will always change when operating in service cryogenically. The shaft will always deform some amount due to the cryogenic temperature. This will create a thermal unbalance in operation. In addition, there are numerous fits along the shaft that will change during operation as well including the motor core, inducer, impellers, and rolling element bearings.

Thirdly, the hydraulic force can act as an unbalance force. The hydraulic unbalance for each impeller can be of the same magnitude as the maximum residual unbalance, and together it could be much larger for the cases where there are multiple impellers. The cryogenic rotors discussed herein can be configured with up to 18 impellers.

Fourthly, the magnetic field of the rotor can also act as an unbalance force, and its magnitude can be multiple times of the maximum residue unbalance.

Since the rotor has 3 bearings and it spins in liquid, the vibration induced by the unbalance may not be as large as for 2-bearing rotors, i.e. the system is more tolerant to unbalances. Besides, the vibration measurements are taken from the pump casing using accelerometers, and it is difficult to compare / correlate vibration magnitude with the unbalance, such as using the predicted Bode plots for compressor. So, under the acceptance criteria of the acceleration measurement, the acceptable unbalance is usually much higher than the balancing acceptance criteria, and that could be a reason that this topic has not been a focus of the industry.

CONCLUSION

This paper provides an improved method for balancing vertical 3-bearing rotors in horizontal balancing machines. Rotor strains caused by sag can lead to improper balancing. Rotor strains can be minimized and sag can be properly compensated by positioning a third support bearing using proper procedures and sag prediction.

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