SHOP BALANCING OF TURBOMACHINERY ROTORS

by

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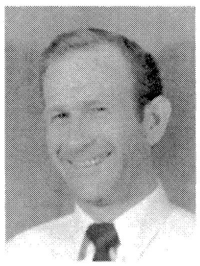

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ABSTRACT

The details of shop balancing flexible shaft rotors are discussed with an explanation of the key parameters involved, such as tolerances and specifications, balance procedures, and documentation. Understanding and controlling these key parameters will result in successful balancing and longer machine service life.

INTRODUCTION

As has been stated many times before, unbalance of rotating machinery parts is the most common cause of vibration. Fortunately, for those in the business of controlling vibration, unbalance is also the simplest form of vibration and, therefore, the easiest to control. "Control" is the key work here, because when a flexible shaft rotor installed in the field vibrates excessively from unbalance, it is often the machine that is in control, rather than the operators. To avoid this dilemma, control must be assumed of unbalance in the repair shops during the rotor assembly stages.

This sounds simple enough, and in fact, is quite simple. However, herein lies the problem. Some people tend to oversimplify or shortcut various parts of the balancing process and end up with a vibrating machine. This happens for a number of reasons, such as tooling error, incorrect methods, making assumptions concerning balance tolerances and specifications, and lack of knowledge concerning the balancing equipment.

The purpose herein is to present the various considerations for achieving high-quality, reliable balance of turbomachinery rotors. Tolerances and specifications, balancing equipment and tooling, balancing procedures, and documentation of results will be discussed in laymen's terms.

TOLERANCES AND SPECIFICATIONS

Tolerances are necessary to provide satisfactory end results. With balancing, the desired end result is a machine that operates at very low levels of vibration upon startup. The lower the level of vibration, the lower the stresses and forces acting upon the machine's rotor, bearings, and support system, which directly relates to machinery reliability and service life. With the balancing technology and methods available today, it is not uncommon for high speed turbomachinery rotors to operate with shaft vibration of 0.5 mils (0.0005 in) or less. To achieve vibration levels this low requires sound balance practices and tight tolerances. When establishing tolerances, there is always a trade-off betweeen practicality and economic feasibility.

Over the years, many balancing tolerances have been established. In the final analysis, all balancing tolerances specify an allowable eccentricity, or offset weight distribution from the rotating centerline, for unbalance divided by journal weight equals eccentricity. With that in mind, some of the most common balance tolerances that are in use today are: VDI Standards (Society of German Engineers) and American Petroleum Institute (API) Standards. For this comparison assume a 500 pound journal static weight for a rigid turbine rotor operating at 7000 cpm (Figure 1).

"VDI" (Society of German Engineers)

VDI balance tolerances assign oz-in/lb of rotor weight values to various rotor classifications. In this example, classification G 2.5 would be applied. Class G 2.5 has an upper limit of approximately 2.14 oz-in/lb and a lower limit of approximately 0.86 oz-in/lb for each 1000 lbs of journal weight. Using the upper limit, this tolerance is:

$$\frac{500 \text{ lb} \times 2.14 \text{ oz-in/lb}}{1{,}000} = 1.07 \text{ oz-in}$$

With this tolerance, the allowable eccentricity is 133.75 μ-in:

$$\frac{1.07 \text{ oz-in}}{(500 \text{ lb} \times 16 \text{ oz})} = 0.00013375 \text{ in}$$

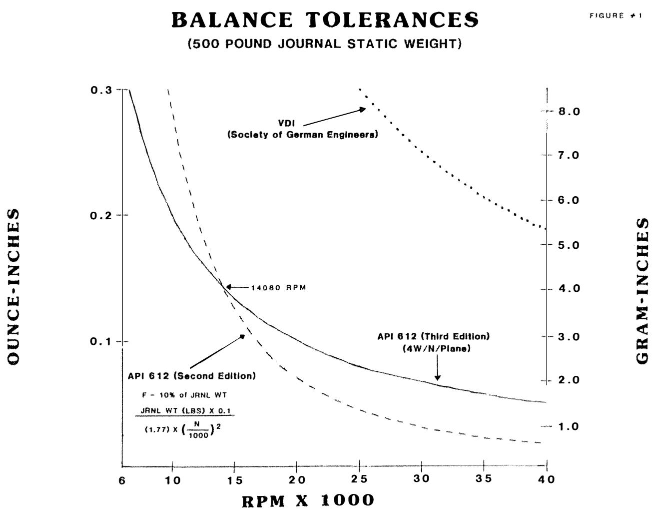

Figure 1. Balance Tolerances (500 lb journal static weight).

API 612 (Second Edition)

API 612 (Second Edition) balance tolerance allows a force equal to ten percent of the journal static weight, or 0.1 G, be applied at the bearing. For this example, this tolerance is:

$$\frac{500 \text{ lbs} \times 0.1}{(1.77)\,(7000/1000)^2} = 0.5765 \text{ oz-inches}$$

With this tolerance, the allowable eccentricity is 72.06 μ-in:

$$\frac{0.5765 \text{ oz-in}}{(500 \text{ lb} \times 16 \text{ oz})} = 0.00007206 \text{ in}$$

API 612 (Third Edition) and API 617 (Fifth Edition)

The most recent balance tolerance adopted by API is 4 W/N, or four times the journal static weight divided by the maximum continuous operating speed (in cpm). For this example, this tolerance is:

$$\frac{4 \times 500 \text{ lb}}{7000 \text{ CPM}} = 0.2857 \text{ oz-in}$$

With this tolerance, the allowable eccentricity is 35.71 μ-in:

$$\frac{0.2857 \text{ oz-in}}{(500 \text{ lb} \times 16 \text{ oz})} = 0.00003571 \text{ in}$$

The minimum shaft vibration values that can be expected using the three different balance tolerances in the preceding examples are summarized in Table 1.

As previously stated, existing API Standards 612 and 617 specify a balance tolerance of ten percent of the journal static weight, or 0.1 G, maximum. The other two tolerances discussed actually use a multiplication factor of the journal weight divided by the rotating speed. For example, the upper limit of class G 2.5 of the VDI Standard is approximately 15 W/N per plane, and the latest API tolerance is 4 W/N per plane. The end result of a tolerance specified in some W/N factor is a specific shaft velocity. Therefore, 15 W/N represents 0.098 in/sec velocity and 4 W/N represents 0.026 in/sec velocity. Because velocity and acceleration are directly proportional with frequency, the upper limit of class G 2.5 of the VDI Standards and the existing API 612/617 tolerance are the same at a speed of 3755 cpm. Further, the existing API tolerance and the newly adopted API tolerance are the same at a speed of 14080 cpm. Below these "crossover" points, the tolerances that specify shaft velocity are "tighter" than the 0.1 G tolerance, and vice versa.

Table 1. Unbalance Results Using VDI and API Specifications for a 500 lb, 7000 cpm Rigid Rotor.

Tolerance (oz-in)	*Displacement (mils)*	*Velocity (in/sec)*	*Acceleration (Gs)*
1.0714 (VDI)	0.27	0.098	0.19
0.5765 (Present API)	0.14	0.053	0.10
0.2857 (New API)	0.07	0.026	0.05

As most major turbomachinery operates below 14000 cpm, it is obvious that the newly adopted API tolerance is a tighter tolerance than either of the others. However, what is not so obvious to the layman is the fact that the tighter tolerance is just as easy for an experienced balancer to achieve as the others. Since this is true, and the tighter tolerance represents a significant reduction in stresses and forces acting on the machine components, why not use it?

BALANCING MACHINES

As classified by the type of support system, there are two basic types of balancing machines, namely "soft-bearing" and "hard-bearing" machines. The "soft-bearing" balancing machine design employs a flexible spring suspension system on which the workpiece is mounted.

The natural frequency of a soft-bearing support system (including the workpiece to be balanced) is very low, so actual balancing is done above this resonance. The unbalance in the rotor results in an unrestrained vibratory motion in the support system, which is normally measured with velocity transducers mechanically connected to the support system.

Hard bearing balancing machines have very rigid support systems in which the natural frequency is normally well above balancing speeds. These machines measure rotor unbalance forces using strain gage-type transducers, and offer several advantages over the soft-bearing design.

- With an equal amount of unbalance, the readings will be fairly linear for various rotors. On a soft-bearing machine this is not so, because the amount of vibration in the spring support system will vary depending on the rotor weight and the configuration.
- Windage from the rotating workpiece and other erratic oscillations are not detected by hard-bearing machines.
- Very large unbalance forces can cause extremely dangerous motion of the spring supported system in soft-bearing machines. Workpieces have been known to hop out of the balancing machine!
- Since the readout of hard-bearing machines is unbalance forces and not vibration of a spring system, the readout will be close to the amount of actual unbalance in a properly calibrated machine.

For the same reasons, the hard-bearing design typically will accept a wider range of rotor weights. The spring support system on soft-bearing machines becomes very limited with lightweight rotors, since the amount of vibration developed is directly affected by mass.

- Finally, some spring support systems have been proven to be non-repetitive in readouts. For example, the readouts vary significantly between runups even though the workpiece is never brought to a complete stop! This type of problem is most serious, for accurate balancing cannot be performed unless readings are repeatable.

Some balancing machines use a driveshaft connected to the workpiece with a universal joint (U-joint). Before this type of drive system can be used to accurately balance a workpiece, the U-joint assembly must be balanced itself. The desired end result is to be able to rotate the U-joint assembly 180 degrees in relation to the workpiece without any variance in the readout, which can become quite troublesome and time-consuming. Belt driven systems are far more accurate and less troublesome.

A few other points concerning balancing machines are worth mentioning. One is to avoid using a machine with antifriction support bearings that have diameters that are equal to the journals of the workpiece. In this instance, any imperfection that results in non-concentricity of the outer race will be interpreted as unbalance by the electronics. Further, since unbalance forces vary as the square of speed, always maintain the same balancing speed from the start to the finish of a job.

In summary, it is not so important that a balance machine be totally accurate, only that it must be repeatable in both amplitude and phase indications. Few, if any, balancing machines remain totally accurate in readout, regardless of the workpiece, so never expect one to do so. Again, it is not important, for the only way to *know* how much unbalance remains is to perform a residual test, which will be discussed later.

TOOLING

The main items to discuss on the subject of tooling are balance mandrels and half keys, since some of the most common problems occur in this area. To begin with, a balance mandrel should be ground between centers to assure concentricity of all diameters throughout its length, as well as a good smooth finish. Afterwards, the mandrel must be precision balanced using a "cheat weight." The desired balance result is that no matter what angular location a trial weight is added, the unbalance readout is always the same. This describes a case where the balance is as close to "zero" residual unbalance as is humanly possible.

Further, the workpiece should be always be mounted with a light shrink fit, never a sliding or loose fit. To emphasize the importance of all of this, consider a blower impeller that operates at 10000 cpm and weighs 75 lb. The tolerance for this is 4 W/N, or 4×75 divided by 10000, which equals 0.03 oz-in. Now, 0.03 oz-in in this case relates to 25 μin, or 0.000025 in of eccentricity, or a total peak-to-peak runout of only 0.00005 in. Therefore, if the workpiece is mounted off center by more than 25 μin, the tolerance is exceeded. With this example, one can imagine how often mandrels that do not meet tolerances are used.

The same "inattention to detail" is common on half keys as well. Some balancing personnel will fill a keyway with a regular, flat-topped half key without ever considering the weight that is missing due to the curvature of the shaft circumference. This "missing" weight can be a considerable amount. For example, consider a 10000 cpm, eight inch diameter by 72 in long compressor shaft, weighing 1000 lb. The shaft has a 1.25 in key on one end, which has a total length of 7.625 in, including the radiused end. The balance tolerance for this shaft would be 0.2 oz-in per plane (4 W/N).

If the balancer completely filed the keyway using a flat topped key, the total weight of the "missing" radiused portion would be approximately 1.4 oz at a four inch radius, or 5.61 oz-in, or 28 times the tolerance, before even getting started! Though this sounds ridiculous, it is commonly done all the time. *All half-keys must completely fill the keyways and be fully radiused to achieve satisfactory results.*

COMPONENT BALANCING

On flexible shaft rotors (those that operate above the first bending critical speed), it is vital to balance all the major components individually before assembly. This is done because, if the rotor is fully assembled, there is no way to know exactly what contribution each component is making to the total measured unbalance vector. Though it is simple to get the entire rotor stack within tolerances on a shop balancer, it must be remembered that only two longitudinal planes on the balancing machine are being measured. Further, during shop balancing, the rotor remains in a rigid mode. If a large unbalance force exists in one of the major components within the rotor, the shaft will flex at this point during high speed operation, and if this happens you're out of business!

Before balancing the bare shaft, be sure that all keyways are filled with half-keys that have the same radius as the shaft itself, as previously discussed. Further, the bearing journals must be checked for eccentricity and all shaft runouts must be phase-related and recorded. If this is all done properly, then the approximate unbalance to expect when the shaft is checked will be known. For example, suppose a compressor shaft has an impeller fit area that is eight inches in diameter and 60 in long. The impeller fit area checks to be non-concentric with the journals by 0.0003 in total indicator readings (TIR). This section of the shaft would weigh approximately 854 lb, and would be offset by 0.00015 in. This would result in an unbalance of 1.0 oz-in/plane, at the indicated high area.

Before mandrel balancing other major components, such as the balancing drum and impellers of a compressor rotor, several things must be considered. First, there must be a precision mandrel as previously discussed, and it should have a keyway to accept each components' key. These job keys are also often overlooked. All keyway clearances should be carefully measured and recorded before balancing is performed. All keys should have a top clearance of approximately 0.004 in to 0.006 in. Excessive key clearances will allow the keys to centrifugate outward during operation, resulting in unbalance of the rotor.

After each component is shrunk on the mandrel, check and record the axial and radial runouts, phase-related to the component keyway. As a general rule, runouts that exceed approximately 0.002 in per foot of diameter are unacceptable.

All components should be balanced to 4 W/N tolerances. As a minimum, a residual unbalance test should be performed on the first of a series of components to check the inaccuracies of the balance machine. An example of this procedure is shown in Figure 2. When balancing on a soft-bearing machine, or if those involved are not fully qualified, experienced balancers, performing a residual unbalance test on each component is recommended.

PROGRESSIVE BALANCING

Progressive or "stack" balancing is necessary, due to the deformation of components during assembly. Components that do not have equal stiffness in all planes, such as those with single keyways, are sure to experience some deformation when shrunk on the shaft. For such components, considerable deformation and resultant unbalance can occur betweeen mandrel balancing using a light shrink fit, and stack balancing on the job shaft with a heavy shrink fit.

Slight deformation of the shaft can also occur due to stresses resulting from the shrunk-on components. These potential problems point out why good documentation is so important prior to balancing. If the amount and location of runouts is known before assembly is done, it is fairly easy to check the components during assembly for straightness and/or excessive stress buildups. As a general rule, "stacked" component runouts should match those taken on the mandrel prior to stacking

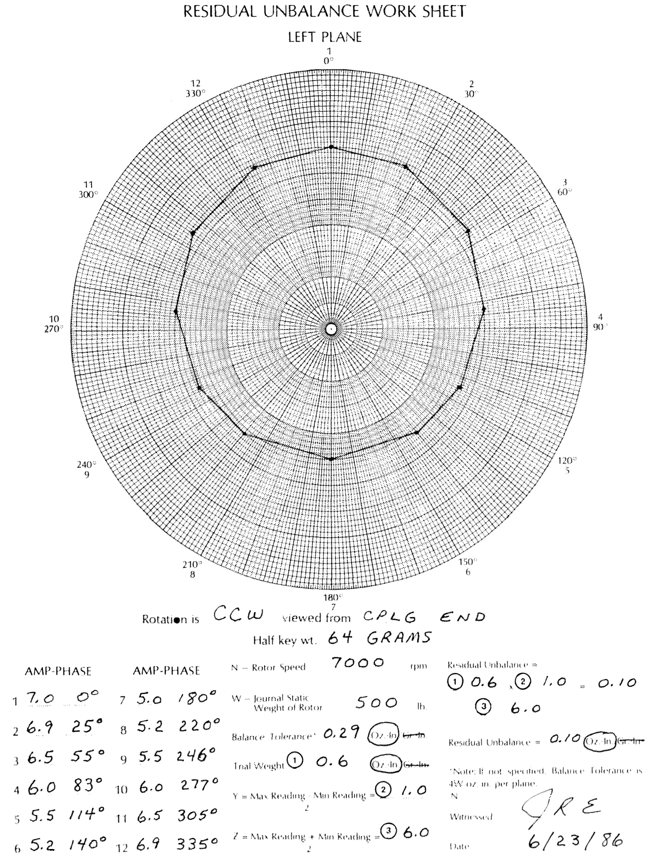

Figure 2. Residual Balance Worksheet.

within 0.001 in TIR, after any existing phase differences are taken into account. Further, any component that results in a change in shaft runout in excess of 0.0003 in is causing unacceptable stresses.

When possible, a rotor should be stacked from the center outward, stacking not more than two components at a time. After assuring that runouts and stresses are within acceptable limits, the balance corrections are made on only the recently stacked components. If a rotor must be stacked from one end, then only one major component at a time is stacked, after which the same procedures are applied. During stack balancing, all shaft half-keys should be left in place until it is necessary to replace them with the job keys, to assure that unbalance due to unfilled keyways is not compensated for in the components.

After the rotor is completely stacked, trim balancing, if required at all, should be very small. In fact, any unbalance that exceeds the 4 W/N tolerance will be from minor deformations and/or stresses, if all of the previously discussed requirements are met. A good rule of thumb here is that the remaining residual unbalance in the rotor should not exceed two times the tolerance prior to trim balancing.

After final trim balancing, a residual unblance test must be performed to verify that the residual unbalance is within specified tolerances (Figure 2). Many shop balancing machines are theoretically calibrated to read out in units of unbalance. However, it is common that the readouts are in error, due to improper calibration, workpiece setup, etc. The twelve-point, residual unbalance test leaves no doubt as to the amount and location of the remaining unbalance. Also, it does not matter that the balance machine readouts may be in error during the test, because the test is performed with known values. The only requirement for the balancing machine is that the readout be consistent for a given amount of unbalance.

During practice runs, the rotor is marked off in twelve equally spaced increments (every 30 degrees) in each correction plane. The trial weight for the test should be one that results in approximately two times the allowable residual unbalance. The trial weight should first be positioned at the heavy spot on the rotor (if known) to assist in selecting the proper readout scale. The rotor is then run up to test speed and the readouts are obtained and recorded. This is repeated for all twelve positions in each plane, allowing polar plotting of the results (Figure 2). The polar plot should approximate a true circle. If not, the most common reason is that the trial weight was not laced at exactly the same radius for each run. Accurate placement of the trial weight is a must, for significant errors may be introduced otherwise. During retesting, if the trial weight is accurately positioned each time and the plot is still not circular, this indicates that the balance machine is producing inconsistant results. A balancing machine that will not readout consistently for two identical unbalance runs cannot be used to determine true residual unbalance.

Once a rotor has been balanced as outlined and well documented by qualified inspectors, further balancing should not be required. Too often, rotors that have been properly balanced to the correct tolerances are removed from storage for "rechecking." When the rotor is rechecked it is likely that the unbalance will exceed tolerances, and therefore be "rebalanced."

Unfortunately, what the balance machine operator does not realize is that he has just balanced out a rotor bow at ambient conditions! When the rotor is installed in the machine and straightens itself during normal operating conditions, it becomes unbalanced again, which results in unnecessary vibration. It is good policy never to balance a rotor that has been properly stack-balanced and documented, unless obvious damage or other sound justification is apparent.

CONCLUSION

The practices outlined are tried and proven to provide smooth running machinery, which in turn leads to longer service life. The tolerances described herein are fairly "tight," but are easily achieved on today's state-of-the-art balancing machines. It is this kind of "attention to detail" that produces additional dividends in equipment reliability.

REFERENCES

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