Vibration Generations Mechanisms: Field Balancing of Rigid/Flexible Rotors

Introduction

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. One of the primary causes of vibration in rotating machinery is mass imbalance, this occurs when the principal axis of the moment of inertia is not coincident with the axis of rotation. For a rigid rotor the imbalance is usually eliminated by adding (or subtracting) correction masses in two distinct planes in such a way as to realign and recentre the principal axis. However when using this method, the rotor has to be rebalanced every time its mass distribution changes. This limitation motivates the study of self- compensating balancing devices, in which masses automatically redistribute themselves so as to eliminate any imbalance. Mass unbalance will produce vibration due to the force generated by the eccentric weight. This force will be imposed at the running speed of the shaft, and depends on the amount of eccentric mass m, eccentricity of the weight e_u, and the frequency of rotation ω. In more common terms the unbalance is defined by the eccentric weight, mounting radius, and shaft speed. The observed vibration signature will show elevated amplitudes at 1xRPM and no other significant frequencies when rotor unbalanced is the main fault. 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.

While mechanical imbalance generates a unique vibration profile, it is not the only form of imbalance that affects rotating elements. Mechanical imbalance is the condition where more weight is on one side of a centerline of a rotor than on the other. In many cases, rotor imbalance is the result of an imbalance between centripetal forces generated by the rotation. The source of rotor vibration also can be an imbalance between the lift generated by the rotor and gravity. Machines with rotating elements are designed to generate vertical lift of the rotating element when operating within normal parameters. This vertical lift must overcome gravity to properly center the rotating element in its bearing-support structure. However, because gravity and atmospheric pressure vary with altitude and barometric pressure, actual lift may not compensate for the downward forces of gravity in certain environments. When the deviation of actual lift from designed lift is significant, a rotor might not rotate on its true centerline. This offset rotation creates an imbalance and a measurable level of vibration. Unbalance in a rotor is the result of an uneven distribution of mass, which causes the rotor to vibrate. The vibration is produced by the interaction of an unbalanced mass component with the radial acceleration due to rotation, which together generate a centrifugal force. Since the mass component rotates, the force also rotates and tries to move the rotor along the line of action of the force. The vibration will be transmitted to the rotor's bearings, and any point on the bearing will experience this force once per revolution. Balancing is the process of attempting to improve the mass distribution of a rotor, so that it rotates in its bearings without uncompensated centrifugal forces.

Rotors are classified as being either **rigid** or **flexible.** This Application Note is concerned with **rigid** rotors only. A **rigid** rotor is one whose service speed is less than 50% of its first critical speed. Above this speed, the rotor is said to be **flexible.** A rigid rotor can be balanced by making corrections in any two arbitrarily selected planes. The balancing procedure to flexible rotors is more complicated, because of the elastic deflections of the rotor. A flexible rotor may be nearly perfectly balanced in the shop at low speeds in the balancing machine, but perform poorly when operated in the field environment.

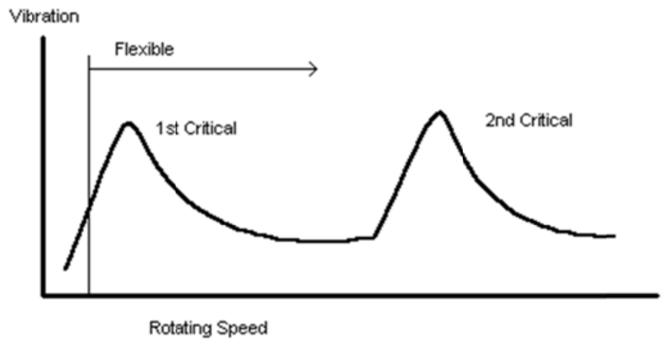

Fig. 4.14 Categorization of a flexible rotor

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. Once a field balance has been completed on a machine (or similar machine) the response data from adding trial weights can be used to calculate future balance corrections using a one shot method (no trial weights required).

Field balancing of rigid/flexible rotor:

Field balancing is the process of balancing a rotor in its own bearings and supporting structure, rather than in balancing machines. Once a balanced rotor has been mounted in its housing and installed in the field, it will not necessarily stay in balance forever. Corrosion, temperature changes, build-up of process material and other factors may cause it to go out of balance again and, thus, start to vibrate. However, unbalance is not the only reason for vibration. Bearing wear, belt problems, misalignment, and a host of other detrimental conditions will also cause it. In fact, experience has shown that vibration is an important indication of a machine's mechanical condition. During normal operation, properly functioning fans, blowers, motors, pumps, compressors, etc., emit a specific vibration signal, or "signature." If the signature changes, something is wrong.

Most rotating components are balanced during the manufacturing process to include balance corrections of individual components (hubs, impellers, etc.) and to make corrections to assemblies of parts (rotors). Corrections made in a manufacturer’s shop will normally be done using a balance machine where the part is either mounted on a shop mandrel or else the entire rotor is balanced. In contrast, field balancing involves using vibration measurements on fully assembled machines that are usually in their final service location, and adding field corrections weights to improve the machine vibration at bearing housings or other locations. 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. Excessive vibration has a destructive effect on piping, tanks, walls, foundations, and other structures near the vibrating equipment. Operating personnel may be influenced too. High noise levels from vibration may exceed legal limitations and cause permanent hearing damage. Workers may also experience loss of balance, blurred vision, fatigue, and other discomfort when exposed to excessive vibration.

Unbalance will generate forces and corresponding vibration response at 1xRPM. For example, if the rotor speed is 3600 RPM, unbalance will produce vibration at 3600 CPM (60 Hz). The FFT plot in figure shows an example of an unbalanced rotor, with vibration shown at the shaft speed and minimal vibration content elsewhere. Since this data was recorded from installed proximity probes and some minor imperfections exist in the shaft surface, there is a small 2xRPM component due to runout.

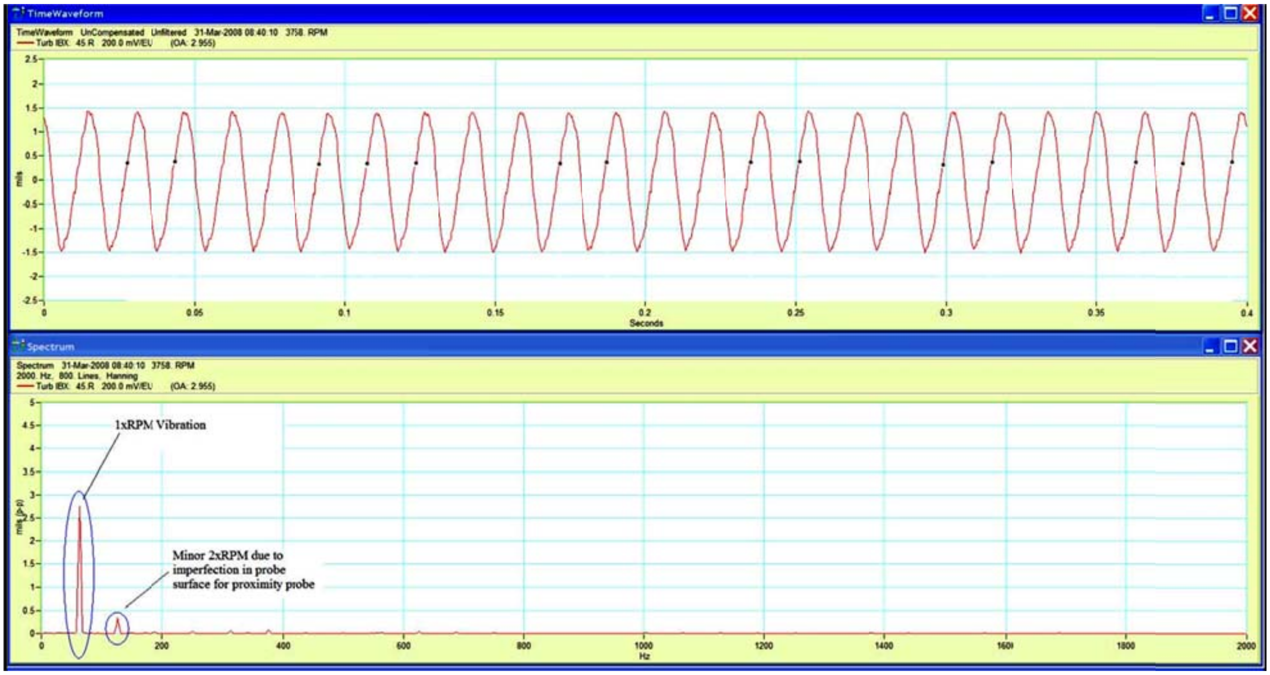

Fig. 4.15 Response of unbalanced rotor (R. D. Kelm, 2008)

Unit of measure: Field balancing can be accomplished using any vibration measurement unit that is proportional to the unbalance force. Common units of measure will include displacement or velocity although there is no technical reason why acceleration could not be used as well. For many users, displacement measurements will result in more logical positioning of trial weights since the displacement phase always identifies the “high spot”.

Balancing Assumptions: There are some basic assumptions that are made when doing field balancing. These include: linear response, accurate/repeatable test measurements, and consistent weight placement. These sound like simple assumptions, but these can produce significant problems during a particular field balance. Variable readings can be even more difficult to identify, and can be caused by thermally induced rotor bows, loading differences, process temperatures, alignment offsets due to variations in casing temperatures, etc. 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. Placement of trial weights and final correction weights can be inaccurate if a care is not used in weight placement. Determining the actual location of the tachometer firing (normally the leading edge of the reflective tape) relative to a position on the rotor where the correction weights are added can produce large phase errors. To eliminate this risk, once a balance process is started, the rotor should be clearly marked with angular positions so that all additional weight additions are properly made relative to the assumed phase angle for the initial trial weights. If the weight additions are done in this fashion (particularly if there is not previous balance data and you aren't trying to reuse the balance response in the future), the possible phase error between the rotor weights and the actual tachometer position is not significant since the balance corrections are all made relative to the trial weights.

In general, field balancing is ideally done until the phase readings become unstable due to the low amplitudes of vibration. However, practical field balancing is frequently finished based on a limited number of balance shots, by time or on the capability and insistence of the balancer. It should be noted that for cases where very high levels of vibration are observed in the initial reference run, the nonlinear response may require starting the balance process over once the vibration is at reasonable levels.

Field Balancing Equipment

Many types of vibration indicators and measuring devices are available for field balancing. Although these devices are sometimes called "portable balancing machines," they never provide direct readout of amount and location of unbalance. Basically, field balancing equipment consists of a combination of a suitable transducer and meter which provides an indication proportional to the vibration magnitude. The vibration magnitude indicated may be displacement, velocity, or acceleration, depending on the type of transducer and readout system used. The transducer can be held by an operator, or attached to the machine housing by a magnet or clamp, or permanently mounted. A probe thus held against the vibrating machine is presumed to cause the transducer output to be proportional to the vibration of the machine. At frequencies below approximately 15 cps, it is almost impossible to hold the transducer sufficiently still by hand to give stable readings. Frequently, the results obtained depend upon the technique of the operator; this can be shown by obtaining measurements of vibration magnitude on a machine with the transducer held with varying degrees of firmness. Transducers of this type have internal seismic mountings and should not be used where the frequency of the vibration being measured is less than three times the natural frequency of the transducer. A transducer responds to all vibration to which it is subjected, within the useful frequency range of the transducer and associated instruments. The vibration detected on a machine may come through the floor from adjacent machines, may be caused by reciprocating forces or torques inherent in normal operation of the machine, or may be due to unbalances in different shafts or rotors in the machine. A simple vibration indicator cannot discriminate between the various vibrations unless the magnitude at one frequency is considerably greater than the magnitude at other frequencies. The approximate location of unbalance may be determined by measuring the phase of the vibration; for instance, with a stroboscopic lamp that flashes each time the output of an electrical transducer changes polarity in a given direction. Phase also may be determined by use of a phase meter or by use of a wattmeter. Vibration measurements in one end of a machine are usually affected by unbalance vibration from the other end. To deter- mine more accurately the size and phase angle of a needed correction mass in a given (accessible) rotor plane, three runs are required. One is the "as is" condition, the second with a test mass in one plane, the third with a test mass in the other correction plane. All data are entered into a hand- held computer and, with a few calculation steps, transformed into amount and phase angle of the necessary correction masses with two selected planes. To simplify the calculation process even further, software has recently become available which greatly facilitates single plane or multi- plane field balancing.

Weight Corrections

Balance weight corrections can be done in a number of ways. When performing a field balance, it is generally desirable to have some method of making trial weight moves that can be easily mounted and removed once final weights are determined. Some trial weights can include:

• Clamp on balance weights (these are commercially available in a wide variety of weights, shapes, installation method and material type)
• Balancing putty (the same stuff used on a shop balance machine)
• Added bolts/washers/nuts
• Engineered weights (dovetail slots, balance plugs, etc.) on machines that have removable balance weights such as power turbines and generators.

Final correction weights are by definition intended to be left permanently on the rotor to correct for unbalance. The final correction weights will include:

- Welded plates
- Engineered weights (same as above)
- Clamp on weights (when welding is not practical)
- Removing weight (drilling holes, grinding, etc.)
- Bolted on weights (added bolts, washers, nuts, etc.)

It would always prefer 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. Particularly on fans, the presence of a lot of fouling may warrant cleaning it off instead of field balancing. In addition, cracks or damage to impellers should normally be repaired prior to making balance corrections. 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. Please review the trial weight selection section for additional guidance in selecting trial weights.

Single Plane Balancing

Single plane balancing is the process of making balance corrections to a rotor in only one axial location. Single plane balancing can be very useful even for rotors that would normally be corrected using a two plane balance method when vibration amplitudes and phase readings allow static or couple corrections alone.

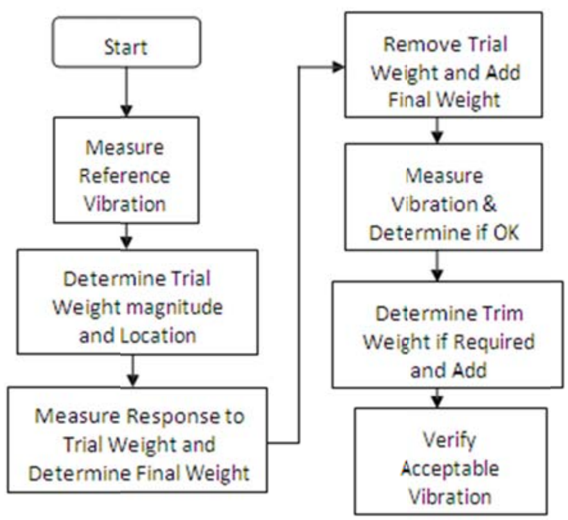

Fig. 4.16 Process of single plane field balancing

Flexible rotor balancing: The physical laws of dynamic balancing dictate that any rigid (stiff) body can be dynamically balanced in any two planes along its axis. Assuming that the rotor in figure below is rigid, and then we would most likely choose to balance in the end planes, as this would allow smaller corrections to be used to achieve a concentric rotating centerline. It should be noted, however, that any (2) of the available (5) planes could be chosen, resulting in equally good balance levels at the journals. The types of rotors which we normally consider to be rigid are electric motor rotors, single stage pumps, fans, coupling spool pieces, etc. Flexible rotors would include multistage pumps, steam turbines, compressors, paper rolls, etc.

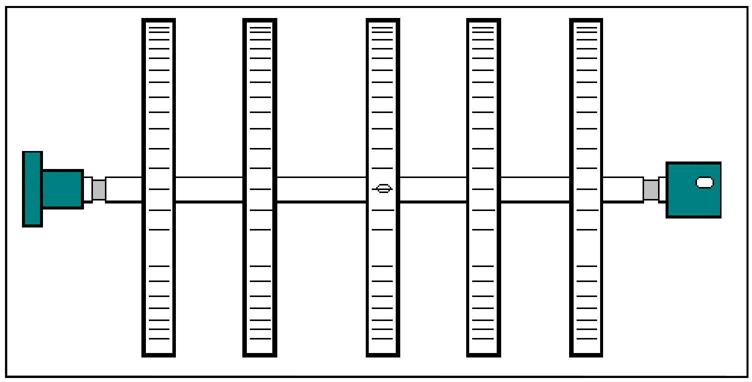

Fig. 4.17 Set of flexible rotors

The reasoning behind multi-plane balancing is grounded in the mode shapes the rotor assumes when approaching critical speeds:

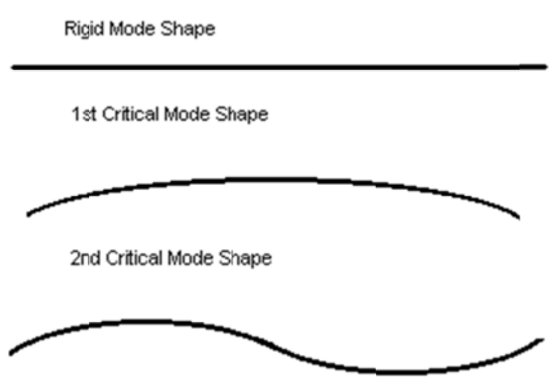

Fig. 4.18 Critical Mode shapes of rotor

Static Balance Tolerances for Flexible Rotors: A normal balance tolerance should be calculated using whatever standard is selected for both left and right correction planes. This tolerance is normally referred to as Uper, which represents the maximum permissible residual unbalance which can be left on the rotor. Since it is not a wise assumption to say that all of the static unbalance exists in the mid planes of a flexible rotor, a mid span static unbalance tolerance (Uper static) should be applied. Several methods for establishing this level of unbalance are discussed below. Most of the current instrumentation on the market today has the capability to read out in either left/right or static/couple modes. If the instrument is limited to only reading in the two plane mode, the static component of the solution may be calculated as shown in the Addendum.

The first method to be discussed utilizes the Total Indicator Runout (TIR) found in the mid span of the rotor. This is sometimes referred to as the GE Method. It can be summarized as:

- If the TIR is 0.0 – 3.0 mils, place 1/3 of the total static balance correction in the midspan balance plane.
- If the TIR is 3.0 – 6.0 mils, place ½ of the total static balance correction in the midspan balance plane.
- If the TIR is >6.0 mils, then 2/3 of the total static balance correction would be placed in the midspan balance plane.

The second method is based on ANSI standard S2.42.1982 “Procedures for Balancing Flexible Rotors”. It is quite complicated, due to the trigonometry and vector math involved, but generally results in a static mid plane correction of 50 – 70%. This method also involves inputting the axial symmetry dimensions of the rotor in the calculation. This paper will not go into details, as the standard may be obtained directly from ANSI (N.I.S.T.) and contains detailed examples.

The third method is simply a field-proven rule of thumb whereby approximately 70% of the static unbalance is removed from the mid span of the rotor on the first balance correction. The rotor is then corrected to uper levels on the end planes. The result is a rotor which is dynamically balanced to calculated uper levels, but the majority of the static was removed in the mid span. This method is by far the simplest, requiring few or no calculations nor rotor runout mapping.

Non-Linearity of Response: Anyone who has done much field balancing will know that balance coefficients will change if the vibration amplitudes start out very high. As a general rule of thumb, any machine with over 0.3 in/sec peak is likely to have non-linear response. Some machines will be non-linear at far lower vibration amplitudes. What this means is that using the standard methods of balancing will result in continual overshoot/undershoot with successive weight calculation as the balance coefficients change unless a method is used to recalculate the balance coefficients as the machine gets smoother. This is the concept of “taking a new O” that has been taught for years in many balance courses.

If the vibration response was linear, the vibration response should increase in a straight line as the unbalance weight is increased. The influence coefficients show that the balance response can vary by a factor of 1.5 to 3.0 as the vibration increases. In addition, the phase lag angles are shown to generally increase with increasing vibration, with an additional phase lag of as much as 45°! With these sorts of variations, it becomes clear why a common rotor kit is not a good demonstrator for field balancing unless it is very well balanced from the start and only small unbalance weights are used for demonstrations. Based on this example, a severely unbalance rotor may result in significant phase lag and large amplitude errors for calculated balance weights until the vibration is reduced to more linear rangesunless it is very well balanced from the start and only small unbalance weights are used for demonstrations. Based on this example, a severely unbalance rotor may result in significant phase lag and large amplitude errors for calculated balance weights until the vibration is reduced to more linear ranges

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