CONSIDERATIONS REGARDING THE DYNAMIC BALANCING OF COMPLEX ROTORS

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Abstract: *Dynamic balancing of the rotor is a fundamental requirement for the smooth operation of any turbomachinery. Unbalance in a machine is recognized as one of the major factors that can lead to machinery malfunction and even catastrophic failure, and may result from its initial manufacturing process or may occur as a result of various operating factors such as machine erosion, thermal effects or unbalance buildup of process material on impellers and surfaces of the rotor. The purpose of this paper is to analyze the theory behind the dynamic balancing of rotors and to describe the process of applying it in the case of a complex rotor. The paper details the types of unbalance in rotors that have been classified, initial unbalance resulting from manufacturing or those that appear while operating, and the process of balancing of rigid rotors (two centrifugal compressor rotors) and a flexible rotor (the main assembly which contains the two centrifugal compressor rotors)..*
Keywords: *dynamic balancing, unbalance, complex rotors.*

1. INTRODUCTION

Rotor balancing is a fundamental requirement for the smooth operation of turbomachinery. Ideally, in the operation of all rotating machinery, the inertia axis of the rotor lies along the rotor spin axis, but in reality, this does not happen and centrifugal moments and forces that are being generated can be transmitted to the bearings or the supporting structure. This unbalance may lead to a motion with large amplitudes that may destroy the shaft, bearings, or the structure. This is why the unbalance of rotors is considered one of the major factors that can lead to machinery malfunction or even failure.

There are many reasons that unbalance may be present in a rotor, the most common being [1]:

a) Blow holes in castings – may be present within the material, undetectable through normal visual inspection and may represent a truly significant unbalance;

b) Eccentricity – exists when the central principal axis of a part does not coincide with its rotating centerline;

c) Addition of keys and keyways – there are few industry-wide standards regarding the addition of keys when balancing components. If two components are balanced without a key, but the two components are then assembled with a key, unbalance will result. Similarly, if both components are balanced with a full key, the assembled units would be unbalanced.

d) Distortion – although a part may be reasonably well balanced following manufacture, there are many influences which may distort or change the shape of a rotor and alter its original balance. Common causes are stress relief and thermal distortion;

e) Clearance tolerances – a common source of unbalance is the stack-up of tolerances possible in the assembly of a machine. Tolerances for the different parts accumulate and produce unbalance;

f) Corrosion and wear – many rotors, particularly fan, blower, compressor and pump rotors, are subject to corrosion, abrasion or wear, that usually do not occur uniformly, resulting in the appearance of unbalance;

g) Deposit build-up – rotors used in material handling may become unbalanced due to the unequal build-up of deposits (dirt, lime, ash, etc.) on the rotor, and the resulting gradual increase in unbalance can quickly become a serious problem;

h) Unsymmetrical configurations – many rotors are manufactured in ways that produce dissymmetry, for example rough surfaces on forgings, core shifts in castings, unsymmetrical parts such as crankshafts, etc;

i) Hydraulic or aerodynamic unbalance – oil trapped in oil galleries, oil trapped in grinding wheels, and cavitation or turbulence can sometimes produce unbalance forces.

All of the above causes of unbalance can exist to some degree in a rotor.

The vector summation of all their effects can be considered as a concentration at a point termed the “heavy spot”. Balancing, then, is the technique for determining the amount of material and location of this heavy spot so that an equal amount of mass can be removed at this location or an equal amount of mass added directly opposite.

Multistage turborotors (compressors, pumps, turbines, etc.) have a residual unbalance due to the assembly of multiple components. Manufacturers usually employ procedures, to insure the initial balancing of their machinery, which generally involve balancing using commercial balancing machines based on either the soft bearing or the hard bearing support methods or using resonant machines. Dynamic balancing usually involves using two planes of correction, and can lead to very high accuracies.

After the rotor has been placed into service, unbalance may appear in the system due to many factors, best described in Table 3.1 by E. J. Gunter & C. Jackson, 1988 [2]:

a. Detectable runout on slow rotation (center of gravity runs to bottom on knife-edges) – Disk or component eccentric on shaft;

b. Measurable lack of symmetry – Dimensional inaccuracies;

c. Detectable runout – Eccentric machining or forming inaccuracies;

d. Detectable angular runout; measured with dial gauge on knife-edges – Oblique-angled component;

e. Detectable runout on slow rotation, often heavy vibration during rotation – Bent shaft; distorted assembly; stress relaxation with time;

f. Visually observable bearing vibration during operation, possible process pulsations – Section of blade or vane broken off;

g. Bearing vibration – Eccentric accumulation of process dirt on surface; Non-uniform process erosion;

h. Shaft bends and throws out center of gravity; heavy vibration – Differential thermal expansion;

i. Rotor machined concentric, bearing vibration during operation; possible process pulsations – Non-homogenous component structure; subsurface voids in casting;

j. Vibration reappears after balancing because of components angular movement; possible vibration magnitude and phase changes – Loose bolt or component slip;

k. Vibration reappears after balancing; apparent angular movement of center of gravity; possible vibration magnitude and phase changes – Trapped fluid inside rotor, possible condensing or vaporizing with process cycle;

l. Bearing vibration; eccentric orbit with possible multi-loops; frequency of vibration is 1, 2 or more per revolution – Ball-bearing wear.

2. THE METHODS

Balancing has been a subject of interest for over one hundred years. The literature in this specific field is extensive, with thousands of references regarding rigid and flexible rotor balancing written, as well as balancing standards developed by various organizations.

The methods used for this study are those set by the Romanian Research & Development Institute For Gas Turbines – COMOTI and IRD Mechanalysis, Inc., in accordance with the international standards ISO 1925 – 2001 Mechanical vibration – Balancing – Vocabulary, ISO 19499 – 2007 – Mechanical vibration – Balancing – Guidance on the use and application of balancing standards, ISO 1940-1:2003 – Mechanical vibration – balance quality requirements for rotors in a constant (rigid) state – Part 1 – Specification and verification of balance tolerances, and ISO 1940-2:1997 – Mechanical vibration – balance quality requirements for rotors in a constant (rigid) state – Part 2 – Balance errors.

2.1. Basic principles of balancing

Balancing is the process by which we determine the amount and angular location of the heavy spot so we can either add an equal amount of mass to the opposite side of the rotor or remove mass at the heavy spot. We know that the more unbalance we have, the greater the force and, thus, the greater the amplitude of vibration. For this reason when balancing in place, we use the amplitude of vibration to help us determine how much unbalance we have. In addition, we use the position of a reference mark on the part as seen by an analyzer strobe light to help us find the location of the unbalance.

2.2. Dynamic unbalance

It is perhaps the most common type of unbalance and is defined simply as unbalance where *the central principal axis and the rotating centerline do not coincide or touch*. This type of unbalance exist whenever static and couple unbalance are present, but where the static unbalance is not in direct line with either couple components. As a result, the central principal axis is both tilted and displaced from the rotating centerline. Generally, a condition of dynamic unbalance will reveal comparative phase readings which are neither the same nor directly opposite one another. This type of unbalance can only be solved by making corrections in a minimum of two planes.

2.3. Two-plane balancing techniques

The choice of balancing technique will depend on several factors such as unbalance configuration, length-to-diameter ratio, balance speed compared to operating speed, rotor flexibility and amount of cross-effect.

Two-plane balancing techniques are:

a. Separate single plane approach – used when the rotor length to diameter ratio is large;

b. Simultaneous single-plane approach – used when the rotor length to diameter ratio is large and the original unbalance vector indicates a predominantly static or dynamic unbalance configuration;

c. Force/Couple Derivation – used in overhung rotor configurations and some standard rotors;

d. Two-plane vector calculations: either a graphical method or by using an automatic balancing instrument or programmable hand calculator.

2.4. Cross-effect

Also called "correction plane interference", can be defined as the effect on the unbalance indication at one end of a rotor caused by unbalance at the opposite end. Because of cross-effect, the unbalance indications observed at each end of a rotor do not truly represent the unbalance in their respective correction planes. Instead, each indication will be the resultant of unbalance in the associated correction plane plus the cross-effect from the opposite end. At the start of a balancing problem, there is no way of knowing the amount and phase of cross-effect. In addition, the amount and phase of cross-effect will be different for different machines. For this specific study, a two-plane balancing method has been used, together with an IRD Mechanalysis Inc. balancing instrument.

Each component of the assembly has been balanced individually to reduce the final assembly residual unbalance as much as possible. The limits for the residual unbalance of each component were calculated using formulas specific to the type of rotor placement.

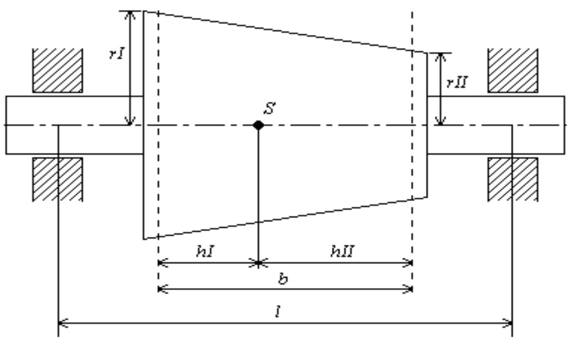

Fig. 2.4. "Long-asymmetrical" rotor placement

$$U_{lim} = \frac{9549 \cdot G \cdot m}{n} \text{ gmm} \quad (1)$$

Where: U_{lim} is the total unbalance limit G is the balance quality depending on rotor type; m is the rotor mass; n is the working speed of the rotor.

This total unbalance limit is then split in two limits, one for each bearing. This split depends on knowing the rotor center of mass (marked with "S" in Fig.2.4.), but if it is not known, an educated approximation will suffice.

$$U_{lim1} = U_{lim} \cdot \frac{h_I}{b} \text{ gmm} \quad (2)$$

$$U_{lim2} = U_{lim} \cdot \frac{h_{II}}{b} \text{ gmm} \quad (3)$$

Where: U_{lim1} / U_{lim2} are the unbalance limits for each bearing; U_{lim} is the total unbalance limit previously calculated; h_I / h_{II} are the distances between each correction plane and the center of mass; b is the distance between the correction planes (marked by the dashed lines in Fig.2.4.).

To find out the limit of the unbalance mass, we only need to divide the unbalance limit of each bearing by the radius at which the corrections are made (r_I and r_{II} in Fig.2.4.).

The dimension "l" in Fig.2.4. is not used in any calculation for this rotor placement. Its only purpose here is to accurately position the balancing machine bearings according to the rotor bearings, because it is recommended, if possible, to place the rotor exactly how it would be placed in the machinery that uses it.

It is important that the rotor is leveled, so it could spin parallel to the ground, to insure a normal distribution of unbalance to each bearing.

Then, a piece of reflective tape is placed on the rotor, thus determining the phase angle 0 position.

Next, three calibration runs are needed to calibrate the balancing instrument with the rotor being balanced.

The first run measures the amplitude of the unbalance (in µm) using two sensors (one on each of the two soft bearings on which the rotor rests).

The next two runs are needed to measure the change in amplitude when a calibration weight is added, first only to the left plane, second only to the right plane.

This is done to accurately determine the weight needed to balance the rotor by calculating how many grams are needed for each µm of amplitude. Since the reflective tape marks the 0 angle position, the angular position of the peak amplitude can easily be determined for each correction plane.

Corrections were made to two of the five components (specifically the two centrifugal compressor rotors) by placing weights opposite of the determined angular position for each heavy point of the two correction planes until the unbalance was reduced to a value below the calculated residual unbalance limits, thus confirming the position where mass must be removed to balance the rotor.

To prevent removal of too much material, the right plane of each rotor was left untouched to be able to use them as balancing planes when the final assembly would be balanced

Using the previously mentioned formulas, (1),(2) and (3), the residual unbalance limits have been calculated for the two centrifugal compressor rotors, working at 22000 RPM, with the balancing grade G2.5:

-1st stage (mass 45 kg): $U_{lim} = 48.8\ gmm$, $U_{lim1} = 18.3\ gmm$ and $U_{lim2} = 30.5\ gmm$;

-2nd stage (mass 35 kg): $U_{lim} = 38\ gmm$, $U_{lim1} = 13.3\ gmm$ and $U_{lim2} = 24.7\ gmm$.

3. RESULTS

After balancing, the residual unbalance for the first stage (balanced at 528 RPM) is 14 gmm at 161° in the left balancing plane, with 1.11 µm amplitude of vibration in the left bearing (front of rotor), meaning 23% lower than the limit, and 24 gmm at 326° in the right balancing plane, with 1.86 µm amplitude of vibration in the right bearing (back of rotor), meaning 21% lower than the limit.

The residual unbalance for the second stage (balanced at 573 RPM) is 9.1 gmm at 0° in the left balancing plane, with 0.824 µm amplitude of vibration in the left bearing (front of rotor), meaning 31% lower than the limit, and 13.2 gmm at 45° in the right balancing plane, with 0.906 µm amplitude of vibration in the right bearing (back of rotor), meaning 46% lower than the limit. As previously stated, material was removed only in the left planes.

For the main assembly the residual unbalance limits have been calculated as $U_{lim} = 166.7\ gmm$, $U_{lim1} = 96.5\ gmm$ and $U_{lim2} = 70.2\ gmm$, considering the assembly mass 105 kg, balancing grade G2.5 and working speed of 22000 RPM.

Unlike the centrifugal compressor rotors, the main assembly is flexible and has critical speeds that need to be taken into consideration. In this case, we have found the first critical speed to be between 500 RPM and 600 RPM, where the amplitude of vibration jumps to very high values (65-70 µm) and returns to more reasonable values (10-15 µm) at 700 RPM.

After balancing (at 734 RPM), the residual unbalance of the main assembly is 42 gmm at 97° in the left plane, with 0.58 µm amplitude of vibration in the left bearing (front of assembly), meaning 46% lower than the limit, and 47.2 gmm at 213° in the right plane, with 0.97 µm amplitude of vibration in the right bearing (back of assembly), meaning 32% lower than the limit.

CONCLUSIONS

In this study we have emphasized the importance of combining balancing techniques, having patience and not rushing to remove material as soon as the heavy point is found.

Individual parts of complex assemblies must be balanced but not necessarily have material removed from them.

One must always have a picture of the final assembly in mind to have a better understanding of how the parts will all be put together.

Some residual unbalances might cancel each other out, or at least decrease in magnitude, should the part placement in the final assembly be made, if possible, considering the position of the heavy point for each part.

In some cases the heavy point shifts to another angular position in the final assembly planes of correction.

This mix of procedures and techniques could be applied to many other complex assemblies like multistage axial compressor disks or multistage axial turbine disks, as well as other assemblies that require both individual part balancing and final assembly balancing.

BIBLIOGRAPHY

1. “Dynamic Balancing Handbook Form #2049”, IRD Mechanalysis, Inc., 6150 Huntley Road, Columbus, Ohio 43229 USA, October 1990.

2. Edgar J. Gunter, Charles Jackson, “Rotordynamics”, Chapter 3 “Balancing of Rigid and Flexible Rotors”, 1988.

Neville F. Rieger, “Balancing of Rigid and Flexible Rotors”, The Shock and Vibration Information Center, United States Department of Defense, 1986.