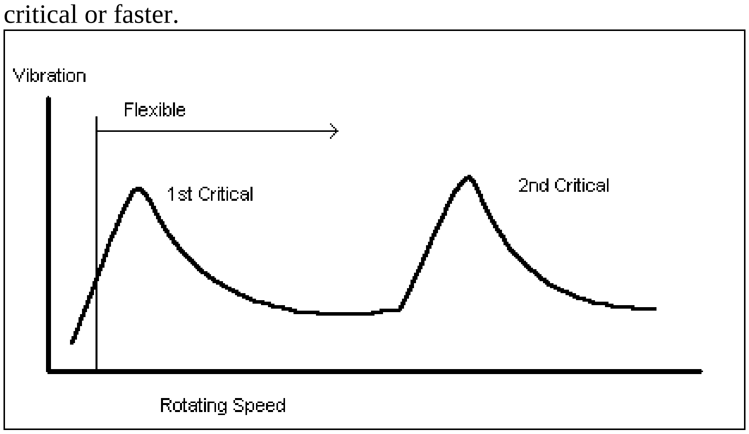
Flexible Rotors – Techniques and Tolerances for Balancing

Introduction

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. This paper will define flexible rotors and provide proven methods for dynamically balancing them in the shop environment. The balance tolerance criteria for flexible rotors and its application at low speeds will also be discussed.

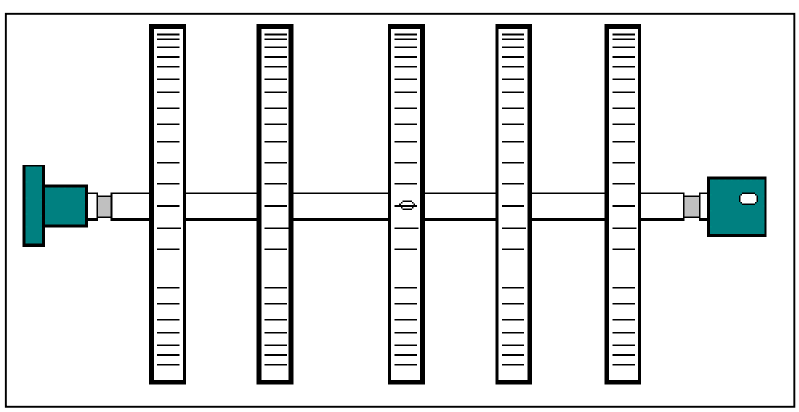
What is a flexible rotor?

A rotor is generally considered to be flexible in nature when it operates close to or above its natural frequency (critical speed). The rule of thumb is to consider a rotor flexible if it operates at 70% of 1st critical or faster.



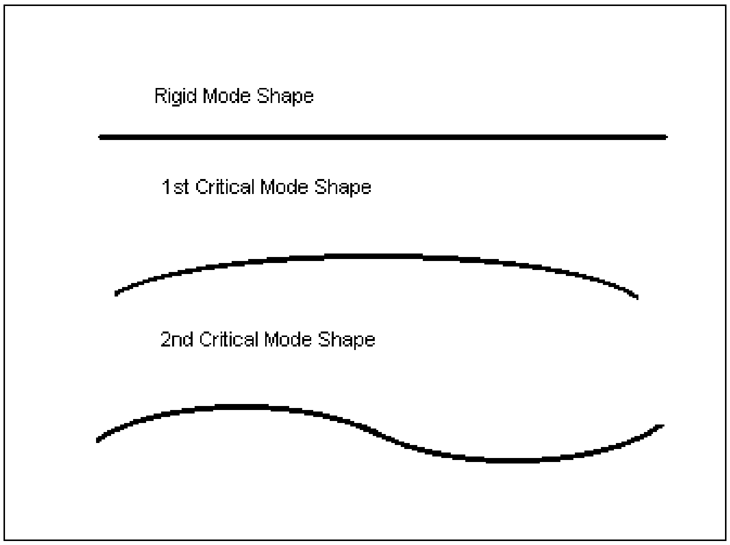
Flexible vs. Rigid – Why worry?

The physical laws of dynamic balancing dictate that any rigid (stiff) body can be dynamically balanced in any two planes along its axis. <u>Assuming that the rotor in Diagram 1 below is rigid</u>, 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.



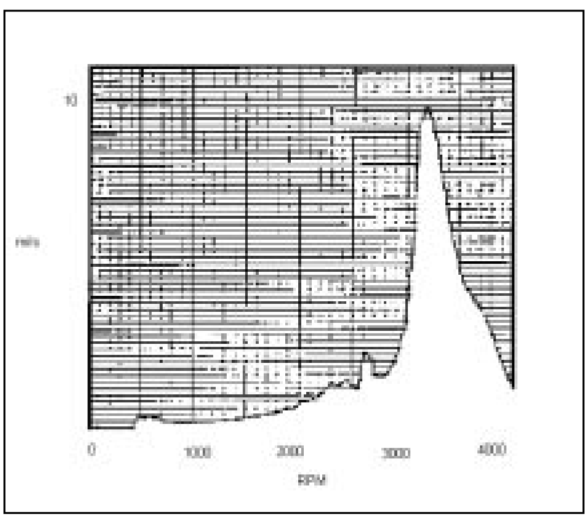
The types of rotors which we normally consider to be rigid are electric motor rotors, single stage pumps, fans, coupling spool pieces, etc. Flexible rotors would include multistage pumps, steam turbines, compressors, paper rolls, etc.

If we assume, however, that the rotor in the illustration is flexible, then we present a much different problem in the low speed shop balancing environment – one requiring multi-plane balance techniques. The reasoning behind multi-plane balancing is grounded in the mode shapes the rotor assumes when approaching critical speeds:

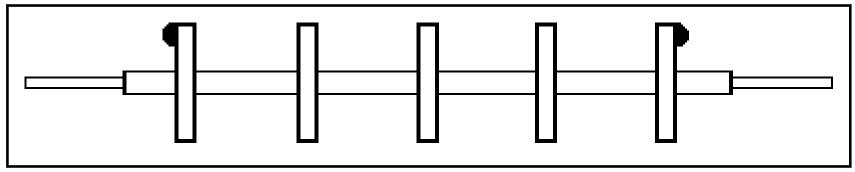


A case history to consider:

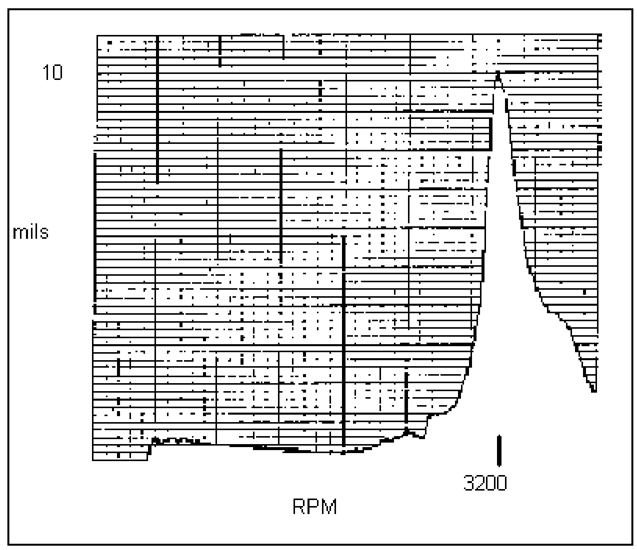
A multistage rotor like the one pictured above was operated in unrestrained bearings from rest to 4,000 RPM. The resultant Bode' Plot shown below indicated a 1st critical at about 3,200 RPM with a vibration amplitude at the bearings of 10 mils.



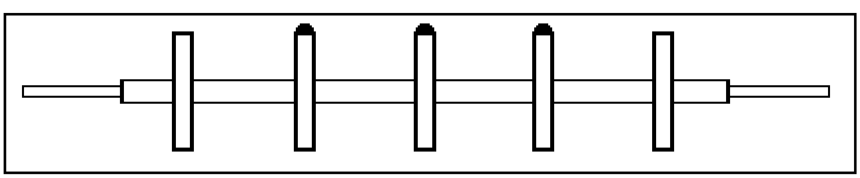
Low speed dynamic balancing was performed on the rotor using the two end planes as correction locations (shown below). The rotor was balanced to a level of 0.014 gram-inches and 0.016 gram-inches left & right respectively.

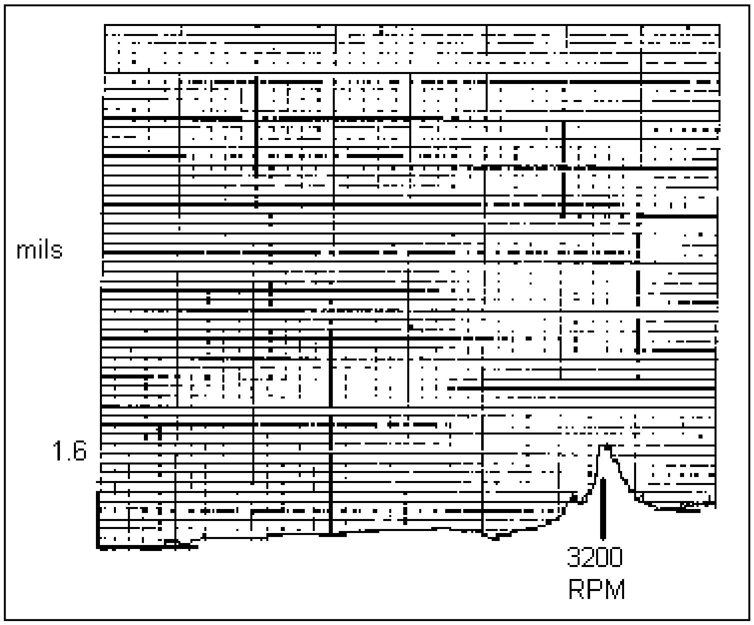


The rotor was operated through the critical speed range once more and the resulting Bode' Plot below revealed virtually no change in the vibration at the bearings. This is the inherent danger in low speed balancing of flexible rotors – the rotor residual unbalance was well within even the tightest of tolerances, yet probably flexed in excess of 20 mils along the mid span when operated through the critical. This rotor would, most likely, have rubbed and catastrophically failed.



The mode shape of the rotor when passing through the 1st critical indicates that any static unbalance left in the rotor in the mid span would tend to further excite the resonance. With this in mind, the temporary weights used to balance the part on the end planes as shown above were removed and a method known as static/couple derivation was employed in order to remove the static unbalance in the mid span. The static/couple method is simply a means of deriving the static component and the couple component of dynamic unbalance and is shown in more detail in the addendum. With the rotor now balanced in the planes shown below to 0.018 gram-inches of static, the rotor was again rotated through the critical speed range, resulting in the Bode' Plot shown on page 4.





It should also be noted that the rotor does not have the couple correction made to it at this point, and is, therefore, not balanced to the previous levels read at the journals in the balancing machine. It is quite obvious that the static unbalance had a major impact on the vibration level at the first critical. This rotor will now be finished by balancing the remaining dynamic (mostly couple) unbalance at the end planes. In summary, the rotor is now dynamically balanced, but the static was removed in the mid span (which contains the center of gravity of the part) and the couple removed on the end planes.

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 U_{per}, 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 ($U_{per\ static}$) 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.

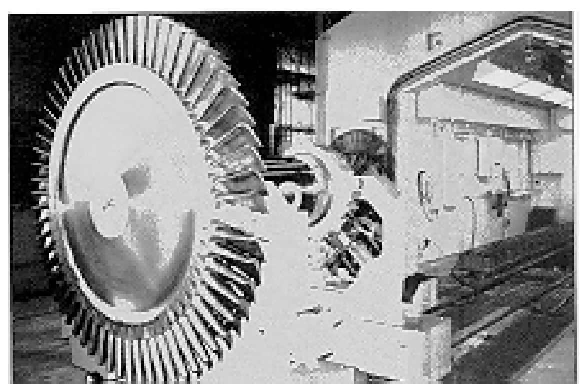
If the TIR is >6.0 mils, then 2/3 of the total static balance correction would be placed in the midspan balance plane.

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 U_{per} levels on the end planes. The result is a rotor which is dynamically balanced to calculated U_{per} 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.

Low Speed Shop Methods vs. Operating Speed Balancing

The very best place to balance any rotating element is in its own environment. It should also be apparent that low speed shop balancing does not in any way approximate the speed, load, torque, bearing conditions, etc. that the rotor will experience in the field. There are several fine facilities in the U.S. which offer "operating speed balancing", which means primarily that they can run the rotor in a vacuum chamber up through the critical speeds and observe the magnitude of shaft flexure. A typical "at speed" facility is pictured below.While this is most certainly better than trying to guess what the rotor will do in the field, it is also time consuming and quite expensive. And, for the common compressor, boiler feed pump, multistage turbine or paper roll, it has been proven that the static/couple technique applied at low speeds in the shop is more than sufficient to minimize rotor deflection through critical speeds.



Picture courtesy of TurboCare Houston Facility

Summary

It is quite possible to balance a flexible rotor at low speeds (rigid mode) in the shop utilizing the simple technique of applying static component balancing to the mid span of the part.

References:

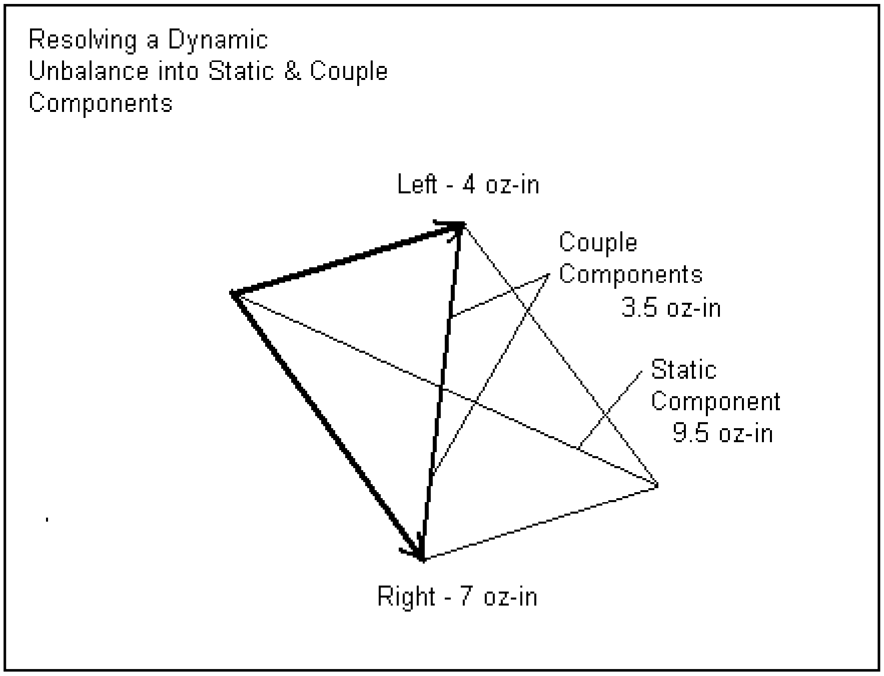
"Balancing Rotors Using the Static/Couple Method", Mr. Dick Baxter, Rich II Resources, Ltd, Centerburg, Ohio

"Procedures for Balancing Flexible Rotors", ANSI Standard S2.42.1982

"Why Operating Speed Balance", TurboCare Technical Information Memorandum #112, 8/94

"Slow Speed Balancing of Assembled Rotors to Minimize Rotor Critical Response", Mr. Randy Fox, IRD Mechanalysis, 1989

Addendum



If the balancing instrument is not equipped to resolve static and couple from the dynamic answer, then the vector method above must be employed. Simply plot the left and right amount vectors pointed toward their respective angular locations, connect the ends with a line and find the mid point of that line. Construct parallel lines to the left/right vectors and where they meet on the midpoint line is the termination of the static vector.