The Trouble with Balancing, Part I, June 2005

Mass Balancing is a well-known and almost mature technology. If it worked consistently every time with excellent results, then it could be called mature. The reality is that mass balancing does not always produce the desired results when balancing a machine in-situ for physical reasons that are beyond the control, or even knowledge, of the person performing the procedure. Mass balancing works better on a balancing machine where conditions are better controlled. Unfortunately, we can't leave the rotor there. To do any useful work, the balanced rotor must be installed in a housing, moved to its permanent location, and connected to something else. Therein lies some of the trouble. Other troubles are in the methods of balancing, of which there are several.

The influence-coefficient method, in particular, is most often used, but it requires both linearity and reciprocity of the physical system to have any hope of converging to a solution. We usually enter a balance task assuming that the system is linear and that reciprocity applies. This article is intended to point out how to recognize a balance job going bad and how to recover. For the non-balancer, this article is intended to develop an appreciation for the dynamic world of rotating machines where Newton's Laws do not fully apply (Figure 1).

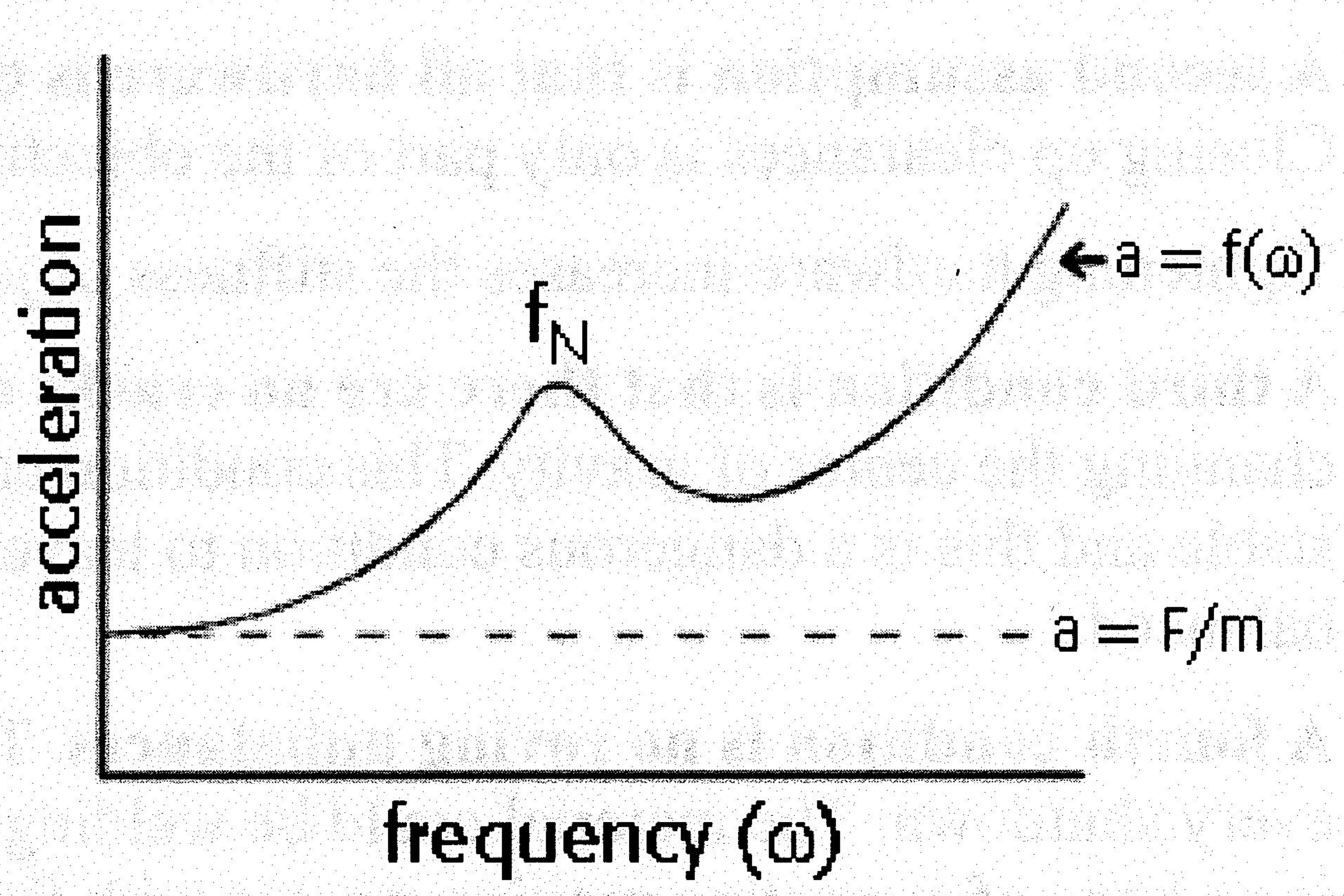

Figure 1. Newton's Second Law states that acceleration is constant with frequency. A resonance plot disproves this.

The known methods of mass balancing are listed here for completeness, and will be discussed more thoroughly later.

1. Manufacturing Control of Mass Properties, Geometry, and Fits During Assembly
2. Cleaning
3. Trial and Error
4. Four Run Without Phase
5. Seven Run Without Phase
6. Phase-Only Balancing
7. Influence-Coefficient Method for Single and Two-plan
8. Static-Couple Method for 3 Planes
9. Modal Balancing
10. Influence Coefficient Method Extended to Multi-Speed and Multi-Plane Balancing
11. Mass Centering
12. Index Reorientation of Coupled Machines

Some of the above methods are not applicable to the present discussion, and will be dropped shortly. Number 6, "Phase-Only Balancing," is of academic interest. (See bibliography item 1.)

"The Static-Couple Method" is of historic significance where single-plane methods were used to balance in three planes. It did not compensate fully for cross-effect. It is available in some program routines after completing the influence-coefficient procedure, and it is desired to place the static correction weight in the center and the couple correction weights at end planes.

"Modal Balancing" and "Multi-Speed, Multi-Plane Balancing" are certainly applicable to flexible rotors which abound in high-speed power plant type of equipment. The key to success with these methods is knowing something about how the rotor bends to make intelligent selections for the correction planes. These methods are similar to the influence coefficient method for rigid rotors and employ methods to overcome some of its problems. They work well in converging to the "best" solution using least squares techniques. A full discussion of flexible-rotor balancing is beyond the scope of this article, but the methods can be plagued with the phase-measurement errors, poor-test-weight location, and non-independent equations in the matrix.[2,3]

"Mass Centering" is applicable in production of castings where the center of rotation is repositioned in a machining fixture. Rather than adding correction weights to relocate the center of gravity, the procedure relocates the rotating center to coincide with the existing center of gravity. It is never used for field balancing or even shop balancing.

"Index Re-Orientation of Coupled Machines" takes advantage of the existing unbalances in two machines by indexing the angular orientation of one machine relative to the other to achieve some phase cancellation. It is usually a trial-and-error process.

Prerequisites

Before mass balancing can have any hope of being successful, some conditions must be satisfied. First, **we must assume that the instrument that measures and calculates is properly set up and is functioning okay.** Phase is the most difficult parameter to measure consistently and stable phase measurements are necessary for the influence-coefficient method to work.

A second assumption is that all hardware is tight that needs to be tight. Looseness cannot be balanced out. Closing up clearances is only part of the objective in checking hardware.

Tightening hardware increases the stiffness of joints which raises the natural frequency of the system.

A third condition is that there are no cracks on the rotor. Centrifugal force will cause cracks to open up, changing the center of gravity. This condition can be corrected with weight placement, but it will not remain stable and this is a dangerous condition to leave the rotor in. Cracks should be repaired, and then the rotor balanced.

A fourth condition is no roving unbalances. This is material captured in hollow cavities that repositions at every shutdown. The material could be welding slag, water, insects, rodents, bird nests, or process debris. There is no hope of correcting moving masses with fixed weight placement. The way to detect for roving unbalances is to repeat the initial conditions. The vibration amplitude and phase should repeat.

The fifth condition is that the rotor is clean. Dirt buildup on a rotor can be corrected with weight, but it will not remain so over time. As some debris sloughs off, or more debris accumulates, the balance condition will change. It is best to mass balance a clean rotor, then this smooth operating condition can be restored in the future with another cleaning. In fact, after a visual inspection, about 1/3 of all my balance jobs turn out to be cleaning jobs. Cleaning is a method of balancing (Number 2) and is always recommended as a starting condition.

A sixth condition is that there are no other root causes of 1x-rpm vibration, such as bent shafts, misalignments, eccentricity, or instabilities. Figure 2 shows a hypothetical spectrum where part of the 1x-rpm amplitude is not caused by unbalance. This immediately makes the measurement of balance in error. No vibration instrument can segregate out the relative contributions, so the full amount of the filtered measurement is entered into the calculation algorithm and this makes the mathematics come out wrong. The influence-coefficient method assumes that the full amplitude of the 1x-rpm vibration is due to unbalance and nothing else. Bent shafts and eccentricities can be detected with a runout measurement using a dial indicator. So a dial indicator with a magnetic base should be available to the balancer as a diagnostic tool. Shaft misalignment can only be verified with a measurement fixture mounted on the shafts to check coaxiality of the centerlines. Vibration analysis cannot positively separate unbalance from misalignment in all cases, so it is best to verify shaft alignment prior to balancing whenever possible. Bearing misalignments are more difficult to detect. The vibration symptoms are identical to shaft misalignment. The diagnostic procedure is usually a process of elimination. If the shaft is straight with less than .001" total indicator reading and a shaft-to-shaft alignment fixture verifies the shafts to be coaxial, then we are left with bearing misalignment as a strong possibility. Mass balancing will not work well until this condition is corrected. Instabilities manifest themselves as a step change in vibration. The rotor has more than one stable position for the center of rotation to settle at, and it can flip-flop from one to the other. It is time to examine the bearing support system for flatness, clearances, or distortion. Instabilities are the most perplexing of difficulties with balancing because of the non-obvious symptoms.

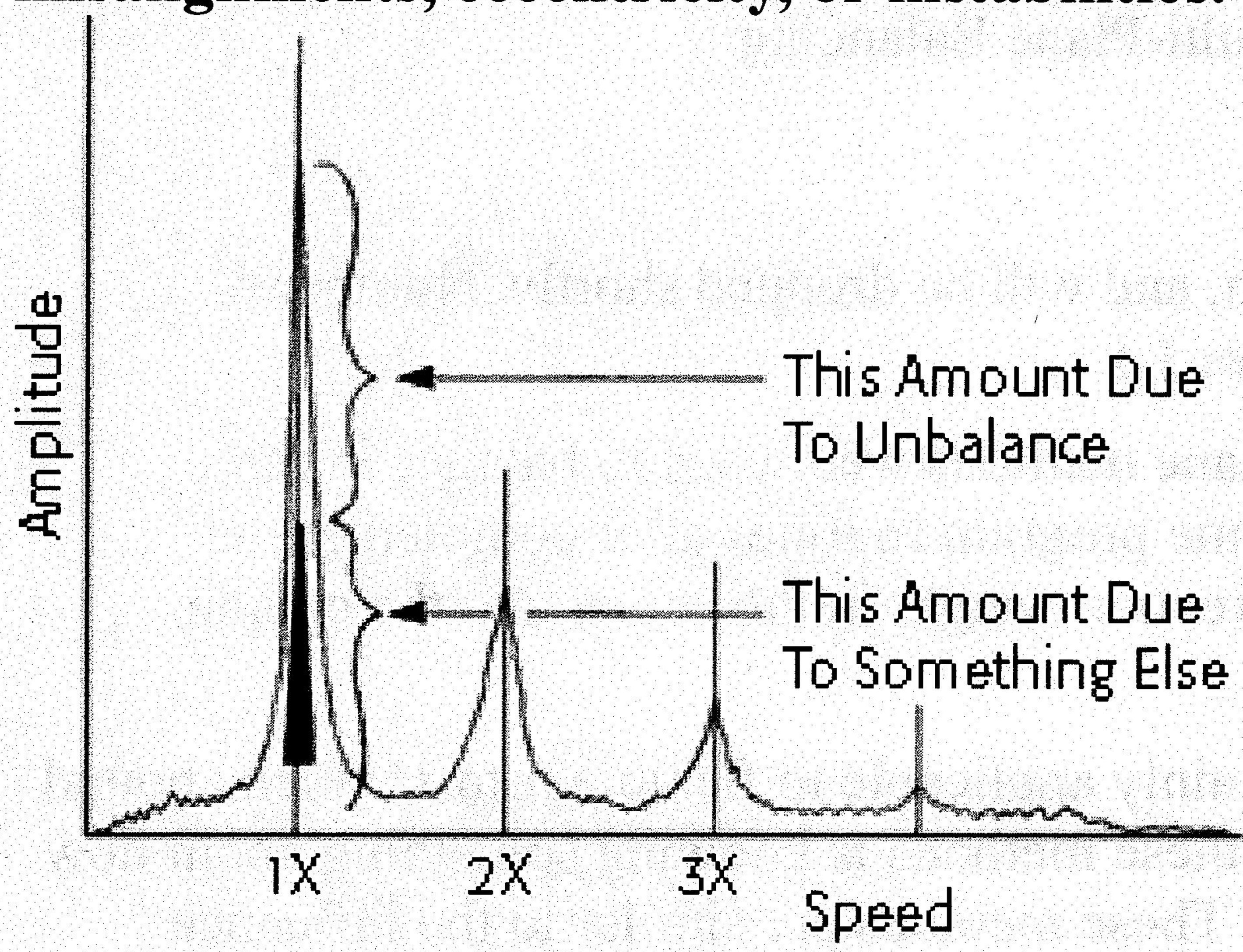

Figure 2. Hypothetical spectrum with two sources contributing to the filtered measurement at 1x rpm

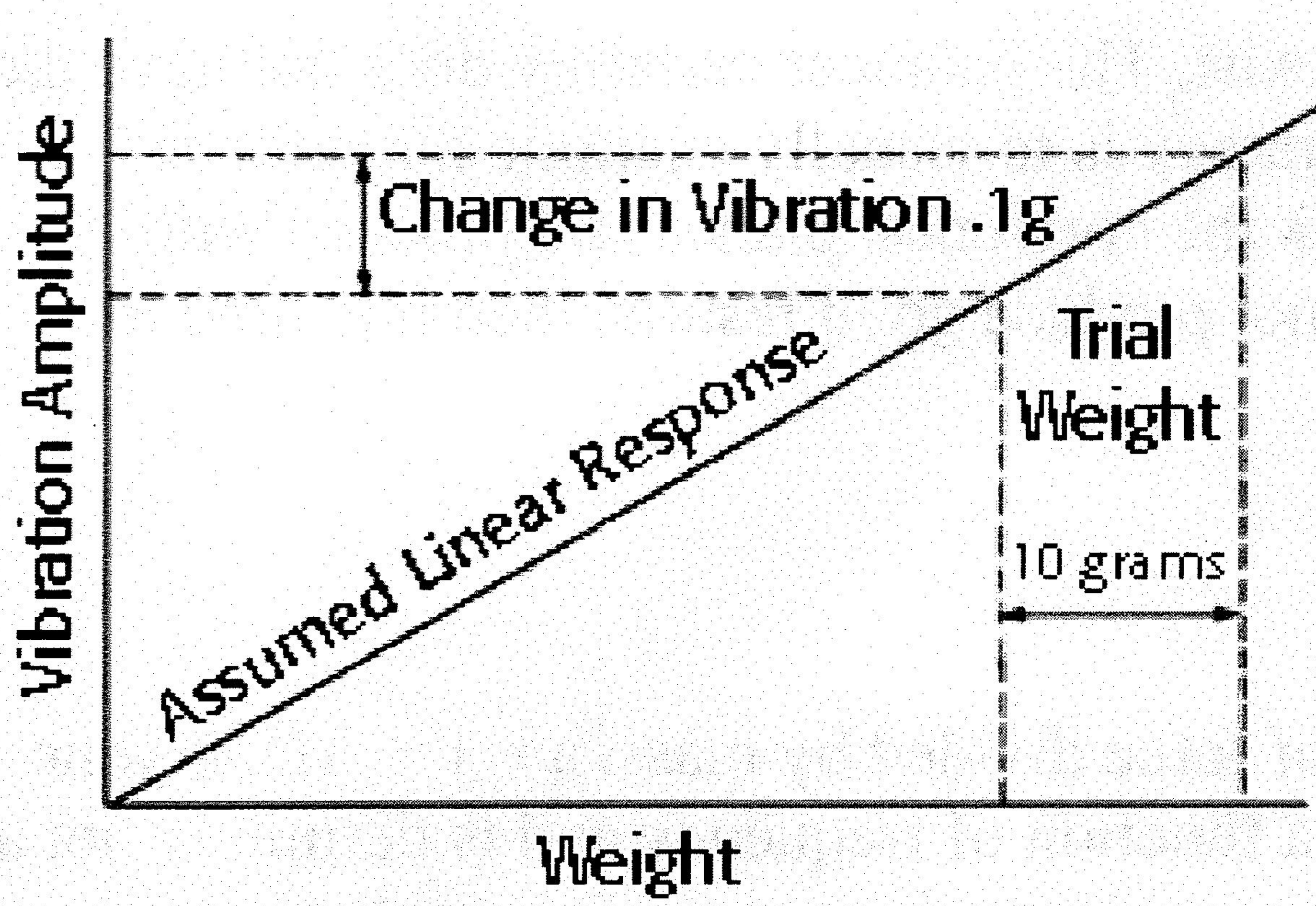

Figure 3. The Influence Coeffecient method assumes linearity response of the physical system.

A seventh condition is that the physical system behaves in a linear manner where a test weight produces a directly proportional change in vibration (Figure 3). This is a requirement of the influence-coefficient algorithm. Two points are measured, one in the original condition and another with a test weight on board. The two points are connected with a straight line and extended to zero vibration amplitude. The proportional change in vibration for a known test weight is used as the sensitivity constant to calculate a correction weight that should drive the vibration to zero. If the physical system does not behave in a linear manner, which it will not near a resonance, then the true shape of the curve to zero vibration may follow a different path than a straight line connecting two points. The balancer has no previous knowledge of the system response curve and his/her first indication is that the first correction weight does not produce at least a 50-percent reduction in vibration. Amplitude linearity was described here, but phase linearity is also assumed.

An eighth condition is the principle of reciprocity, which states that the input and output of a physical system can be interchanged without altering the response of the system to a given excitation. Mathematically, Fa Vb = Va Fb , a force at position a creating a velocity motion at b is equivalent to a force at b producing a velocity motion at position a. This is a requirement for two-plane balancing to compensate for cross effect. It is also a requirement for some balancing machines to do plane transposition. There is no known way for the balancer to check the validity of the principle beforehand. It is always assumed to be true unless balancing doesn't work. In that case, the methods that rely on this principle, and the assumption of linearity, must be abandoned, and another balancing procedure tried. Specifically, the four-run method, the seven-run method, and trial-and-error balancing can find a solution without linearity or reciprocity.

A ninth condition is the absence of resonance on the rotor (called critical speeds) and on the structural supports. When the residual unbalance activates a resonance because the operating speed is close to a natural frequency, the amplitude and phase response become highly non-linear and sensitive to small speed changes. The influence-coefficient method quickly falls apart, but the four-run method shines. In fact, the finest balancing is done at resonance, taking advantage of the structural amplification. The balance condition is then really smooth at all other speeds. The old Stewart-Warner cantilever beam resonance balancing machines produced the finest balanced rotors. Field balancing on a resonance, using appropriate methods also has the potential to produce the finest levels of balance.

The final prerequisite condition is one that the balancer has control over, but no previous knowledge to make the right choice, which is selecting the planes for correction weights. This is usually done when the first test weight is placed. The balancer selects this plane based on accessibility and after a visual examination of the rotor and its shape. This selection is not done based on where the true unbalance is, but it would be nice if it were. The imaginary true heavy spot location is unknown and will remain so until the completion of the balance job. It would be nice to have an imaging system that can view the rotor and detect the plane of the true heavy spot, but until such a device is invented, the balancer makes his choice in the dark. All balancing methods assume that the planes for heavy spot and correction weight coincide. This is a fair assumption for single-plane balancing of a thin disk. For long rotors, it is a wise strategy to distribute the correction weights in more planes along the length of the rotor. This strategy becomes especially advantageous for flexible rotors.

These prerequisite conditions for good balancing may or may not exist. The balancer certainly does not have the benefit of knowing their status at the start. So the recommended strategy is to visually examine the rotor for cleanliness, cracks, and hardware condition, then go ahead and pick the correction planes and proceed with balancing. When the results of the first few correction weights do not produce good results, then it is time to revisit these assumed conditions with some investigative steps.

SIDEBAR:

Newton's Laws

Newton's Second Law states that the acceleration equals the ratio of force divided by mass, $a = F \div M$. Anyone who has ever done a resonance plot knows that the acceleration is a function of frequency and is definitely not a linear relationship to the force. To preserve the linearity relationship, engineers have dreamed up abstract concepts of dynamic mass and dynamic stiffness, which change with frequency. Newton's Second Law for static force is analogous to Ohm's law for DC circuits. The dynamic equivalent of the Second Law will take on a form similar to AC circuit formulas with impedances rather than resistances. Impedances are a function of frequency.

In addition, the Second Law assumes rigid body motion, which only occurs at zero frequency, and a non-accelerating frame of reference. A rotating body is accelerating, and the direction of acceleration is always changing. To continue to preserve the linearity of the Second Law, we must imagine fictitious inertia forces and Coriolis accelerations.

Newton's Third Law of action-reaction, only applies at the point of contact for massless ideal springs. For real objects that have distributed mass, some of the energy of contact is consumed in compressing and distorting the object's shape, and in heating it up. The forces travel through the object as a stress wave at the speed of sound in the material. So there is a time delay for it to be felt somewhere else. The amplitude will also be modified by the geometry and stiffness of the object. It could be attenuated or amplified.

Check out the August issue of **Energy-Tech** ***for Part II to this article where you'll read about What Can Be Done When Balancing Is Not Working.***

Bibliography

1 W.C. Foiles and D.E. Bentley. "Balancing with Phase Only (Single Plane and Multi Plane)," Transactions of the ASME Journal of Vibration, Acoustics, Stress, and Reliability in Design, Vol. 110, April 1988, pp. 151-157.

2 Victor Wowk. Chapter 13, "Flexible-Rotor Balancing," Machinery Vibration: Balancing, McGraw-Hill 1995.

3 Mark S. Darlow. Balancing of High-Speed Machinery, Sprinter-Verlag, 1989.