# Field Balancing Revealed

## Sometimes it's Simply a Game of Chance

*by Victor Wowk, P.E.*

As I teach balancing seminars, I notice one common problem that balancers with some experience have. The problem is, and I quote, "balancing doesn't always work." A more accurate description of their grievance is that the results are variable. There may be a partial improvement, but insufficient to be acceptable. In extreme cases, there is no convergence to a smooth-running machine, and the vibration at operating speed becomes worse with each weight placement. If this sounds familiar, read on. There are valid reasons for variable balance results which have nothing to do with faulty instruments or inexperience. The reasons can be classified into two broad categories:

1. The Physical System
2. Chosen Methods

I will assume, for the remainder of this discussion, that the balancing instruments are functioning well, and the balancer is following procedures properly.

### The Physical System

Shop balancing on a balancing machine usually has more consistent results. This is because it is a much more controlled situation. The rotor alone produces the measured motion, unmodified by the surrounding structure (i.e., housing and bearing supports), and the response of the balancing machine to unbalances on the rotor are known and calibrated. These stable and known conditions do not exist in field balancing, where each rotating machine is a different configuration, with different bearings, different support stiffness, and perhaps, some pre-existing defects in the rest of the machine. These pre-existing defects could be any number of the following:

- Worn bearing
- Bent shaft, or some other distortion
- Looseness
- Misalignment
- Eccentricity
- Reciprocating forces
- Resonance
- Instability

Worn rolling-element bearings are characterized by unstable amplitude and phase readings, because the bearings cannot repeat the shaft position each rotation. In addition, the bearings will be noisy. Gross unbalances can be corrected to some extent, but we cannot balance any finer than the precision in the bearings. A lift test with a dial indicator can measure the radial clearance and gauge the amount of wear. The way to deal with worn bearings is to advise the machine owner, letting him/her decide how to proceed, which could be:

1. Do the best you can today.
2. Replace the bearings and call you back when ready to continue balancing.
3. Ask you to replace the bearings.

Be advised that the 1x-rpm amplitude and phase can also wander if the inner ring has non-uniform thickness, the outer ring rotates in the housing, or a nearby machine at close to the same speed contributes some vibratory energy.

A bent shaft looks just like an unbalance with a high 1x-rpm amplitude of vibration. Unfortunately, this condition usually cannot be corrected by weights. It can be corrected if weights are placed in the axial plane where the bend is the largest. This usually requires an unusually large correction weight and it works well only at a single speed. The way to deal with a bent shaft is first to recognize that balancing is not producing good results fast enough. Then stop the machine and measure the shaft runout with a dial indicator at several places. Acceptable runout is less than 0.001 inch total indicator reading (T.I.R.). When a bent shaft is confirmed, then advise the owner so they can order a new shaft. In the meantime, the bent shaft will beat up the bearings.

Looseness is assumed to be nonexistent at the start of the balancing procedures. Everything should be tight that is supposed to be tight. If not, get out the wrenches. Looseness will manifest itself as high amplitude vibration at ½ rpm, 1x rpm, 1½ rpm, 2x rpm, 2½ rpm, etc. It is a string of harmonics and half harmonics that is easily discernible with a spectrum analyzer. Misalignment of shafts creates a string of harmonics, in addition to some component amplitude at 1x rpm.

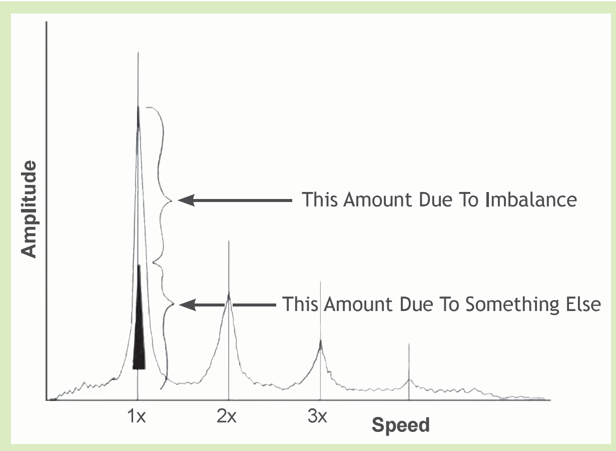



Figure 1 - All vector balancing algorithms assume that the full amplitude of the 1x-rpm vibration is due to unbalance and nothing else. The full amplitude is used in calculations.

This component due to misalignment cannot be removed by mass balancing, but even worse, it will make the 1x-rpm amplitude measurement inaccurate. This will make the calculation of the balancing weights incorrect because the input readings are wrong. We are off to a bad start, and it gets worse from there. The only way to verify good shaft alignment is to swing a set of readings with a fixture attached to the shafts. If balancing is not working well, and the shaft alignment is questionable, then the proper thing to do is to stop the futile balancing, align the shafts, then proceed with balancing afterwards.

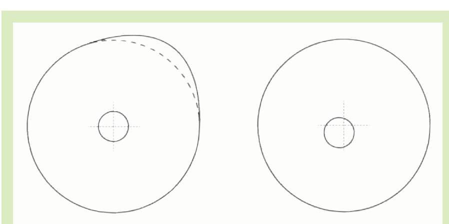


Figure 2 - Electronic Pulleys

Eccentric pulleys, gears, sprockets, and wheels will produce a strong 1x-rpm pulse every time torque is transmitted over the high spot. This looks just like unbalance, but has no hope of correction with weights. This again, is best diagnosed with dial indicators and corrected by replacing the defective part. For the balancer, his/her responsibility ends when this condition is recognized and reported to the machine owner. We should stop burning up balancing dollars as soon as we smell the smoke.

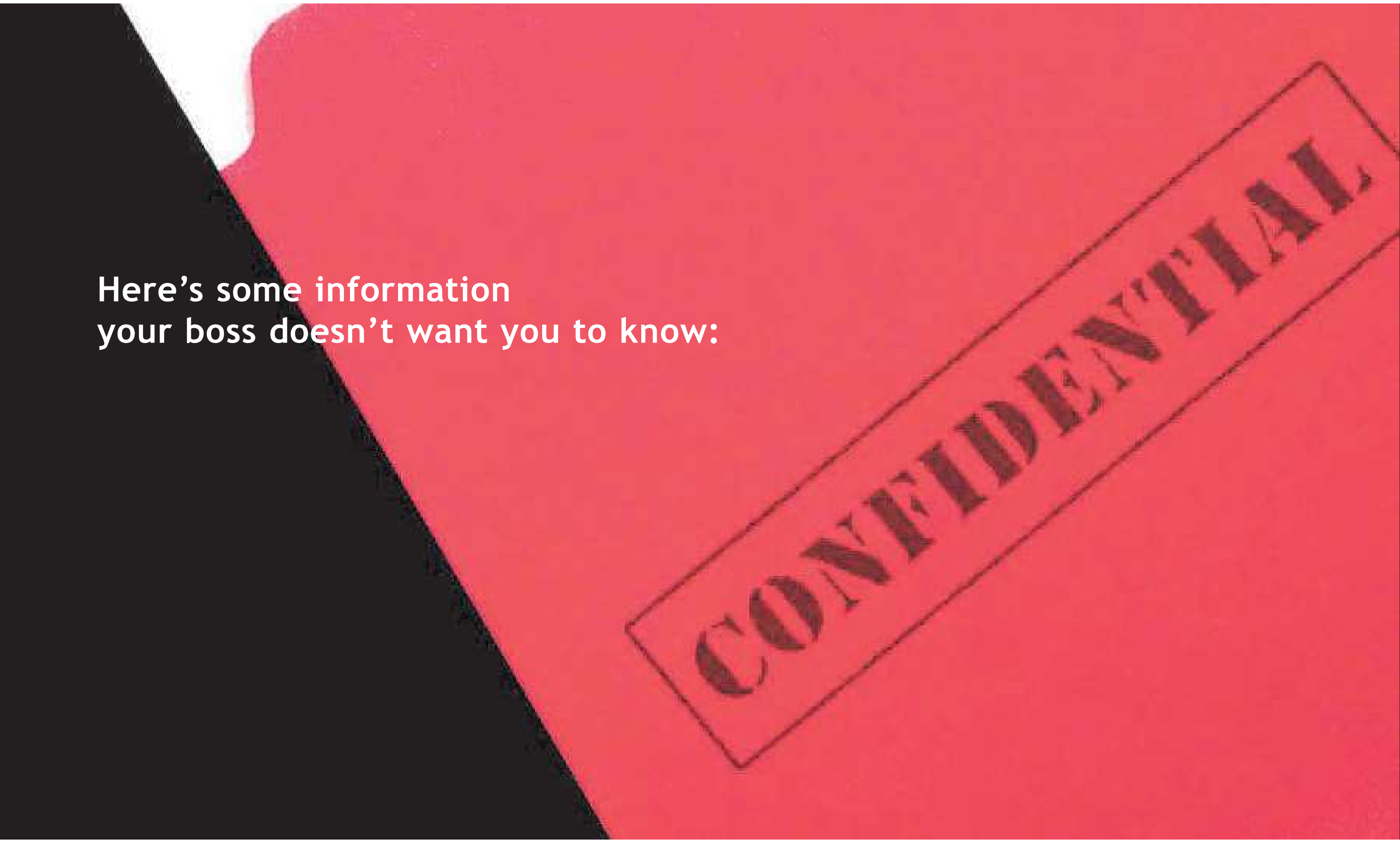
All reciprocating machines produce a strong pulse at 1x rpm from gas pressure forces, that can only be partially compensated by crankshaft balance weights. Even so, multi-cylinder reciprocating machines (engines, compressors, and pumps) can be mass balanced to less than 1.0-mil peak-to-peak amplitude if no other operational defects exist, like poor fuel flow, sticky valves, incomplete combustion, or leaking rings. Since reciprocating machines do not always maintain a constant speed, a tracking filter must be used to acquire valid phase data, or use the no-phase balancing method.

Resonance is a dynamic structural weakness that really confounds the balancing task for two reasons—the amplitude response is nonlinear and the phase response is highly variable near a natural frequency. It is a structural defect and the appropriate fix is to modify the structure. Any nearby part can be resonating—the rotor itself (it is then called a critical speed), the bearing supports, a panel, the base or foundation, the housing, or an attached pipe. It is a cloaked defect that the balancer is unaware of until the balancing process goes amiss—it diverges with many weights all around the wheel with little improvement. The subject of resonance could consume an entire book itself, and how it affects balance could be a 50-page chapter. The fixes for resonance could be another book. For the balancer, it is important to recognize that resonance is speed sensitive, so one strategy is to pick another speed at which to balance. Another strategy is to use the 4-run method without phase, which is one of the few methods that works at or near a resonance. Another strategy is to stop the machine and do some impact testing to find the resonating part and stiffen it. Most balancing instruments can be used for this natural-frequency testing, if configured properly and a systematic test method is used. The important point is to recognize the creature, stop balancing as usual, and do something different.

| Method | Instrument Requirements | Planes | Advantages | Disadvantages |
|---|---|---|---|---|
| Trial & Error | Minimum, None or Simple Filtered Amplitude | As many as Desired | Potential for Producing Fine Levels of Balance | Time Consuming |
| Four Run without Phase | Filtured Amplitude | One at a Time | Quickly Converges, Always Works, Simple Graphical Calculations | Requires Four Starts and Stops |
| Seven Run without Phase | Filtered Amplitude, Computer | Two | Compensates for Cross-effect, Quickly Converges | Requires Seven Starts and Stops |
| Single Plane | Filtered Amplitude plus Phase | One at a Time | Fast Balancing when it works, Best for Thin Disks, Applicable when Phase is Nearly the Same at Both Bearings, Graphical Calculations | Cannot Compensate for Cross-effect, Doesn't Work Well Near Resonances, Faulty Foundations, Instabilities, Non-linearities or when Other Root Cause Exists |
| Two Plane | Filtered Amplitude plus Phase, Computer | Two | Compensates for Cross-effect, Applicable when Phase is more than 30° Different at Both Bearings | Same Difficulties as Above for Single Plane when Phase is not Reliable, Requires Computer to Calculate Correction Weights |
| Static Couple | Filtered Amplitude plus Phase | Three | Graphical Calculations using Single-Plane Vectors, Useful when Three Planes are Available | Requires More Runs than Two-Plane Method on Rigid Rotors, Limited to 1st and 2nd Flexural Modes on Flexible Rotors |
| Modal | Filtered Amplitude plus Phase | As Many as Necessary | For Flexible Rotors | Requires Knowledge of Bending Modes |
| Multiplane Multispeed | Filtered Amplitude plus Phase, Computer | Many | For Flexible Rotors, No Previous Knowledge of Bending Modes Needed | Many Runs, It's a Mechanical Method - Physical Insight is Lost |
| Manufacturing Tolerance Control | Precision Metrology Tools | Many | Potentially Smoothest Machines, Can Make Field Balancing Unnecessary | Most Costly |
| Cleaning | None | All | Fast, Cheap | None - A Cleaning is Always Recommended for a Dirty Rotor |

Table 1 - Pros and Cons of Balancing Methods

Instability is almost as elusive as resonance, and like resonance, makes for a bad balancing day. It displays itself as a step change in vibration. The machine could be balanced down to 1.5 mils, then abruptly jump up to 7.5 mils as you drive away. The instability could be a cracked foundation, a crowned bearing pedestal, or a thermal-induced expansion. The machine has two stable conditions that it can flip-flop between, usually against some physical stops, and insufficient friction to hold it in any intermediate position. The only solution is to find the defect and fix it. One clue though; it usually has something to do with the bearings, and



a bearing inspection, or bearing replacement may be in order.

## Chosen Methods

There are ten known methods of mass balancing, listed in Table 1. Of those, the single-plane and two-plane influence coefficient methods are the ones that most all balancing instruments come pre-programmed with. Balancing classes focus on these two methods, which are actually only one method with different-size matrices in the calculation. The balancer is prompted at the start, on which subroutine to follow—single- or two-plane. The balancing world is really bigger and there are more choices, with others that work better than the influence coefficient method (I.C.M.).

The influence coefficient method is flawed in several respects when applied to field-balance situations. It works best on a balancing machine with soft supports, a rigid rotor, and at slow speeds, i.e., less than 1,000 rpm. In the field, with a machine in its own bearings and powered by a driver machine, on a foundation and support system of unknown response, and at high speed, anything goes. The first flaw with the influence coefficient method is that it assumes linearity, for both amplitude and phase response. This is implied by the matrix calculation and is a required precondition for the math to produce a valid correction weight. We know that the world is not linear near natural frequencies, and phase response goes wild. There can also be subtle situations with a hardening or softening bearing support system that make the system respond in a non-linear manner. There is no big flashing neon sign on the machine that says, "Balancers beware of non-linear dog." What to do about this? Leave the balance weights on that made an improvement and start over with new original data.

The second major flaw with the influence coefficient method is that it is sensitive to the test-weight location. It shouldn't be, theoretically, but it is (See Ref. 1). Placing the test weight is like placing a bet on a roulette wheel. Any place is just as good as any other, supposedly. It is a guess on where to place the test weight (without prior knowledge of the machine behavior), so you may as well satisfy your gambling urge at this balance step. You will come out richer in the end than if

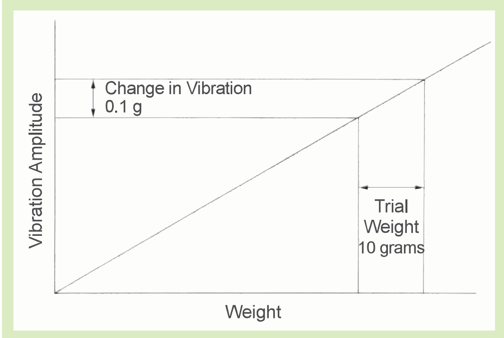



Figure 3 - Linear relationship between weight and vibration.

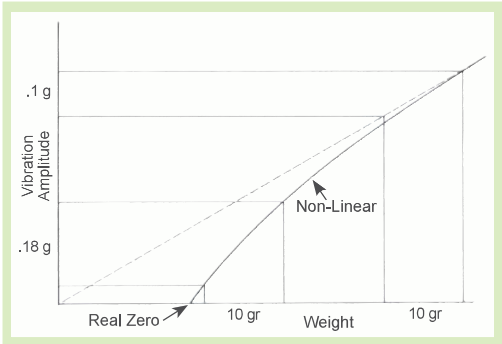



Figure 4 - Nonlinear relationship between weight and vibration.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you sought your satisfaction at a casino. If your bet was a bad one, you can always remove it, put it somewhere else, and spin the wheel again, with the loss of only a little time.

The third major flaw with the influence coefficient method is that it requires expensive phase-measurement instruments and a computer to do the matrix math. Single-plane calculations can still be done with vector graphs, but few people are still alive that can do two-plane vector calculations on graph paper. Who would want to anyway, when laptops and programmable calculators do it so easily? The main point here is that all of these instruments must be properly connected, configured, and working as intended. The balancing process moves to the inside of an electronic box, and the balancer loses "touch" with what is happening. He/she is flying (balancing) on instruments and their heads are in the clouds. The vibration measuring instrument will take the readings, good or bad, and the computer will spit out a correction-weight calculation based on the input data. Whether it is a good weight that reduces vibration depends on the input data, various assumptions made earlier, and the condition of the matrix. An ill-conditioned matrix is more likely in field balancing with non-linearities, than on a balancing machine in the shop.

What to do about this? There are eight other methods of balancing. Give another method a try. In my business we are quick to abandon the influence coefficient method if it does not produce at least 50-percent reduction in vibration with the first correction weight. In fact, my balancers prefer to just start with the 4-run method without phase on all balance jobs, and transition to the 2-plane influence coefficient method if it looks like there is a strong couple unbalance with cross-effect. The reason, we have found from experience, is that the 4-run method without phase works consistently more often than any other method in the field, besides cleaning. If any reduction in vibration can be had with weight placement, the 4-run method will find it.

An added benefit with the 4-run method is that it requires minimal instrumentation—just a filtered amplitude reading (no phase required). In fact, in the classroom in Dallas, students have balanced very successfully with an AC voltmeter and a velocity sensor. A student pulled out a multi meter from his tool bag, we made up a quick cable to connect an IRD 544 velocity sensor, and with the 4-run method, he had the machine balanced down just as well as an IRD 245, and just as fast. Granted, this is an overall AC RMS voltage, not a filtered reading at 1x rpm, but the method works.

So why don't balancing courses taught by instrument manufacturers show these simple methods with simple instruments? The answer is obvious. You would not purchase their expensive instruments with phase measurements that do the influence coefficient calculations. We have those expensive instruments, and use them when necessary, which is about 15-percent of the time. The remaining 85-percent of balance jobs can be done single plane. In the field, the 4-run method without phase has the highest success rate.

A common problem in all balancing is not knowing, in advance, where to place weights along the axial length of the machine. This is illustrated in Fig 5. It would be nice to have an imaging system, like a mass density camera (which does not exist), that we could view the rotating assembly with and see the location of the imaginary heavy spot. Then, at least, we could place the test weight in the correct axial plane. As it stands, we usually place the test weight where it is most convenient to place it. This, then, also defines the correction plane (unless I do plane transposition, which is never done in field work). If it was a good day, and the test weight was not too far away from the heavy-spot plane, then balancing should work O.K. If my test weight missed the heavy-spot plane by a significant amount, then the test weight and heavy spot create a couple unbalance. This would require two-plane balancing to fix on a rigid rotor. On a flexible rotor, the world gets considerably more complicated. The point with this discussion is that we have no way of knowing where the true heavy spot is on a long rotor. We can guess, based on what we see, that it is probably in a plane that has more mass, but if there are several mass-loaded planes, then which one? After balancing is complete, and it being successful, then we can look at the correction weights and see where they are distributed to come up with an "after the fact" estimate of the heavy-spot plane. The problem with all balancing is that we are not entitled to this knowledge beforehand, and this could be a reason why balancing is not working.

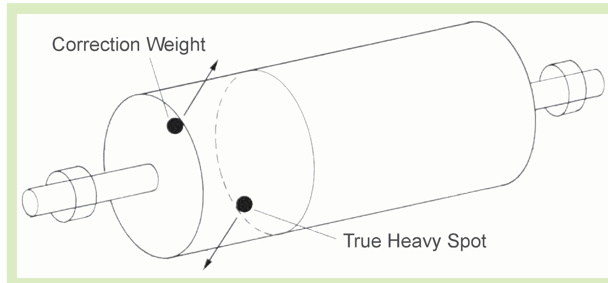

Figure 5 - A couple is created when the correction weight misses the plane of the true heavy spot.

So with all of these obstacles to field balancing, why do we even try? The answer is that it does work well more than 50-percent of the time. Balancing is a joy for me because it provides immediate job satisfaction. I enjoy balancing more than any other process in correcting vibration on machines. It takes me to interesting places. This article is intended to provide you with knowledge to handle those occasional troublemakers. Keep an array of balance methods in your toolbox and pick an appropriate method based on what you observe. Don't be afraid to admit failure with one initial method, abandon it, and proceed on with another approach. We sometimes need to switch 2 or 3 times until we find a method that works on that day, on that machine.

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Ref. 1 "What's Wrong With My Balancing Instrument?" by Victor Wowk, P/PM Technology, August 1997

Ref. 2 "A Management Guild to Balancing," by Victor Wowk, Maintenance Technology, June 1998

Ref. 3 "The Trouble With Balancing," by Victor Wowk, Energy Tech, June 2005 (Part 1) and August 2005 (Part 2)