Field Balancing Standards

By Victor Wowk, P.E

Historically, 1.0-mil peak-to-peak displacement has been quoted by many field balancers as the desired goal. This is a good number to strive for, but may be overbalancing in some cases. The 1.0 mil came from balancing rebuilt motors on a soft-bearing balancing machine with velocity transducers integrated to displacement. The 1.0 was an easy number to remember and the vibration was barely perceptible. The shop balancer then applied this same 1.0 mil when taking his instruments out to a field balance job.

Perfect balance is unachievable. Rotating machines vibrate. The only time when no vibration is perceptible is when the machines are stopped, so it is impractical to expect to find a non-vibrating machine when in operation. The purpose for balancing standards is to prevent a balancer from attempting to achieve an unattainable goal, and to protect the machine from beating up its bearings. So there is a level of vibration that is economically achievable with a reasonable amount of effort and will not cause damage or premature wear to the machine.

Balancing standards are formally defined for shop balancing, but in field balancing we have no standards, only industry guidelines. The purpose of this article is to clarify some of those guidelines for field balancing, but lets review shop balancing first.

Shop Balancing

The purpose for shop balancing is to correct for less-than-perfect manufacturing. Today it is usually more cost effective to fabricate parts and assemble them quickly, then place the assembled rotor on a balancing machine and add weights to correct for non-uniform mass density, eccentricities, poor fits, and hardware variables. The universal balance standard, worldwide, is ISO 1940. This is translated into ANSI S2.19 and ASA STD 2.

These shop balance standards are actually long overdue for overhaul and there are finer shop balance standards. The American Petroleum Institute uses the 4W/N criteria for residual unbalanced, which is an adaptation from a U.S. Navy Balance Standards for Submarines. This result is a residual unbalance that is less than 10 percent of the ISO 1940 lower unit. It is achievable with some effort, and perceived to be necessary for some operations

Another empirical standard is that residual unbalance should produce a centrifugal force no greater than 0.1 times the weight of the rotor—called the 10-percent rule. This rule is easy to apply in field balancing, because if vibration is measured in acceleration, then the goal is 0.1 *g* or less.

Unbalance is defined as a mass times a radius. Shop balancing machines measure vibration, but the readout instruments are calibrated to correlate measured vibration to unbalance weight.

Field Balancing

There is no universal balance standard for all machines in the field. The reason is that the vibration measured at a point depends on the mass in motion and the mass and stiffness of the supports where vibration is being measured. The structure modifies the centrifugal force that is generated at the unbalance heavy spot. The structure can attenuate or amplify the oscillating force, depending on the natural frequencies of the components in the force transmission path. This is the reason for using a test weight in field balancing. The test weight is an attempt to calibrate the instruments for amplitude and phase response and correlate the measured vibration to the unbalanced amount. For field balancing, we must rely solely on vibration measurements to be the judge. As a first pass, we could use Table 1 as a guideline.

Table 1 Field Balance Guidelines

	Displacements, mils, Peak-to-Peak	Velocity in/sec, Peak	Acceleration g, Peak
Good	1.0	0.1	0.1
Best	0.1	0.01	0.01

This guideline is easy to remember because the numbers are all one with the decimal point in a different place. If measuring in displacement, then 1.0 mil peak-to-peak is a good balance level to strive for, and it is repeating the decades-old historical level that has worked well. The best that can be achieved, when bearings are new, shafts straight, pulleys round, no looseness, small misalignments, and no resonance, is 0.1 mil.

If measuring in acceleration, then 0.1 g is an acceptable balance level, and is equivalent to the 10-percent rule. The best that is normally achievable is 0.01 *g* acceleration.

When measuring vibration in velocity units, then 0.1 in/sec is a good balance, with 0.01 in/sec the best achievable. In velocity, the 1x-rpm amplitude should not be the dominate peak in the spectrum out to 10 times rotating speed. All of these numbers are filtered amplitude readings at rotating speed, and apply at the bearings for all three orthogonal directions—horizontal, vertical, and axial.

In field balancing, the correction is not for less-than-perfect manufacturing, but for other reasons. The rotor can have material buildup (dirt or grease on fans) or material loss (erosion on coal-ash fans), bearing changes that affect the center of rotation, shaft distortion due to residual stress relaxation, structural changes, or speed increases. The machine is usually in a service posture and there is a narrow window of time opportunity to do balancing. The goal in field balancing is to reduce the 1x-rpm vibration amplitude to be as small as reasonably achievable in the alloted time. Time sometimes becomes the balance guideline in field work, because when the allocated time is exhausted, then that defines the end of the job. The numbers shown in Table 1 should not be considered as hard limits. Remember, the purpose for mass balancing is to reduce the centrifugal forces that beat up bearings, and also to reduce the transmitted forces causing collateral damage or discomfort. If the machine only needs to operate for six more months until a planned replacement, then there is no need to fine balance to extend the bearing life. I might accept 0.2 *g* as acceptable if further reduction proved difficult in the allotted time. Many machines have operated at 5 to 10 times the levels in Table 1 for short periods, like several weeks, with some wear, but they survived.

Some industries have developed clear balance guidelines for their class of machines. We spend more time balancing fans in the field than any other machine. The American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) has long used the balance limits in Table 2.

Table 2. ASHRAE Balance Limits For Fans

Speed	Displacement, mils, Peak-to-Peak
Over 2,000 rpm	1.0
1,000 - 2,000 rpm	2.0
600 - 1,000 rpm	3.0
Less than 600 rpm	4.0

Cooling-tower manufacturers have used 5.0 mils measured horizontally at the base of the gearbox as their field balance standard. I have been successful at balancing about 50 cooling-tower propellers to less than 2.0 mils when measuring at the top of the gearbox.

Diesel-engine manufacturers have also used 5.0 mils measured horizontally on the casting at the centerline of the crankshaft as the definition of an acceptable engine. Good ones typically measure 1.0 to 2.0 mils when everything is firing properly. A recent international standard, ISO 8528-9, defines acceptable vibration on engine-driven generator sets in terms of overall RMS vibration in displacement, velocity, or acceleration in the frequency range of 10 to 1,000 Hz

Indeed, this has been the trend among trade associations, to depart from filtered amplitude readings in displacement to judge balance, and replace it with overall velocity to judge acceptance in the field at the time of startup. This encompasses other vibrations in addition to unbalance, like vane passing frequencies, misalignment, bearings, and especially structural resonances. The Hydraulic Institute (for pumps) and the Air Movement and Control Association (for fans) have revised their balance and vibration standards to be in overall velocity. This happened in the mid-1990's.

Proving Balance

Balance level is frequently the acceptance criteria for new machines. To prove good balance in the field requires some interpretation and judgment because other vibration sources can get counted in the mix.

First, when testing any machine with a keyway, like motors, we must install a half-key in the slot as defined by ISO 8821, "Balancing—Shaft and Fitment Key Convention."

Second, we must have confidence that what we measure is purely unbalance, and not something else being added in. This could mean verifying good alignment by swinging some readings. It could mean doing resonance bump testing to find the natural frequencies of the structures to be sure that they are not amplifying the motion. It could mean measuring runouts with a dial indicator to make sure components are round and straight.

Third, what if a nearby machine is thumping the foundation at the same frequency? We may need to turn it off to measure balance accurately on the machine under test. But what if the driver machine is a diesel engine driving a pump, and the pump balance limit is 1.0 mils? The diesel engine limit is 5.0 mils, and we measure 2.0 mils on the pump. Does the pump fail? You may need to convince the accepting authority, with data, that a portion of the measured 2.0 mils is being transmitted from the engine and the pump balance level is really O.K.

There is a method of actually measuring the residual unbalance in an operating machine. It is described in ANSI S2.19, Fig 8. It requires a minimum of eight runs with a test weight and developing a sine curve where the test weight adds to and subtracts from the on-board residual unbalance. Once this residual is determined from the zero to peak value of the plotted sine curve, then it can be compared to the balance standards in ISO 1940.

Trends

One trend has already been mentioned, which was the conversion to overall velocity for vibration acceptance during startup. This de-emphasizes the balance condition. There have been two other trends, one among equipment manufacturers and the other from users.

Some equipment manufacturers have ignored or deleted the balance requirement, and have used ISO 1940 as the justification. After statistical analysis, they have determined that their manufacturing process can achieve ISO 1940 levels 68 percent of the time, and have relied on the assembly tolerance and field balancing to take care of the rest. They could then retire the shop balancing at the factory, and focus on build and ship. I see new equipment today that is poorly balanced on startup.

The second trend, among users, is to distrust the balance level from manufacturers. The petroleum industry, power utilities, and the U.S. Navy have developed their own finer levels of acceptable balance. This is evidence that ISO 1940 is out of date. Other end users have purchased balancing machines and disassemble newly purchased machines, like motors and pumps, re-balanced them to their own standards, then re-assemble them before placing in inventory, or placing them into service.

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