By Heinz P. Bloch, P.E. Consulting Engineer

UPDATE YOUR SHAFT ALIGNMENT KNOWLEDGE

It's now or never· · · only reliability-focused facilities will survive.

If you are like most industrial facilities in the industrialized world, your worker and technician resources are probably stretched to the limit. Understandably, you might be looking for ways to simplify some of your traditional work processes and procedures. You may even have had an experience that reinforces the contention that high-tech tools are not always the answer, and that back-to-basics thinking has considerable merit.

While no reasonable and experienced reliability professional will take issue with the above findings, industry must be cautioned against drawing the wrong conclusions. A recent example of "wrong conclusions" involves claims that rotating equipment alignment is sufficiently accurate as long as the shaft centerlines in their standstill, or cold, condition are within 0.002 inches (0.05 mm) of each other. If you blindly follow this questionable advice, you may soon find yourself among the repair-focused dinosaurs that are struggling to survive.

But, if you *update* your knowledge of shaft alignment and alignment tolerances, you might be on the way to becoming *reliability-focused.* Indications are that only reliability-focused facilities will be around a few years from now!

How Shaft Alignment Tolerances Must Be Expressed

The only correct way to express shaft alignment tolerances is in terms of alignment conditions at the coupling, and we will describe several ways to do this. It is incorrect to describe them in terms of correction values at the machine feet, and we will examine why. Before we get into too much detail, however, let's first examine our alignment objectives.

When two machines are directly coupled via a flexible coupling, any misalignment between their centerlines of rotation can result in vibration, which, depending on its severity, can produce premature wear, or even catastrophic failure of bearings, seals, the coupling itself and other rotating components. Indeed, misalignment of the centerlines of rotation has long been recognized as one of the leading causes of machinery damage.

Decades of well-documented observation attest to the fact that misalignment has been responsible for huge economic losses. The more misalignment, the greater the rate of wear, likelihood of premature failure and loss of machine efficiency. Moreover, misaligned machines absorb more energy—they consume more power. Yet, even excellent alignment of the shaft centers of rotation does not in itself guarantee absence of vibration. This is because there is still the possibility of imbalance of rotating components. Structural resonance, fluid flow turbulence and cavitation, or even vibration from nearby running machines that is transmitted to adjacent machines through either foundation or piping also could be present.

Why Reliability-Focused Companies Will Not Use Simple Foot Alignment

Absolute perfection in the alignment of shafts is not realistically attainable, nor is it needed. A good analogy is the polishing metal: No matter how long it is polished and how fine the different polishing media, a powerful microscope will detect a surface composed of peaks and valleys. The rather obvious issue is quantification of alignment quality and allowable deviation—the alignment tolerance.

We define misalignment by visualizing the shaft centerlines of rotation as two straight lines in space. The trick is to get them to coincide to form *one* straight line. If they don't, then there must exist either *offset* misalignment (Figure 1) or *angular* misalignment (Figure 2), or a combination of both.

Furthermore, since the shafts exist in three-dimensional space, these misalignments can exist in any direction. It is most convenient, therefore, for the purpose of description to break-up this three-dimensional space into two planes, the vertical and the horizontal, and to describe the specific amount of offset and angularity that exists in each of these planes simultaneously, *at the location of the coupling*. Thus, we end up with four specific conditions of misalignment, traditionally called *Vertical Offset (VO), Vertical Angularity (VA), Horizontal Offset (HO)*, and *Horizontal Angularity (HA)*. These conditions are described at the location of the coupling, because that is where harmful machinery vibration is created whenever misalignment exists.

The magnitude of an alignment tolerance (i.e. the description of desired alignment quality), must therefore be expressed in terms of these offsets and angularities, or the sliding velocities resulting from them. Attempts to describe misalignment in terms of foot corrections alone do not take into account the size, geometry or operating temperature of a given machine. It can be shown that accepting the simple "foot corrections approach" can seriously compromise equipment life—and has no place in a reliability-focused facility. An illustration of the fallacy of the "foot correction approach" will be given later.

How much vibration and efficiency loss will result from the misalignment shaft centers depends on shaft speed and coupling type. Acceptable alignment tolerances are thus functions of shaft speed and coupling geometry. It should be noted that high-quality flexible couplings are designed to tolerate more misalignment than what is good for the machines involved. Bearing load increases with misalignment, and bearing life decreases as the cube of the load increase (i.e. doubling the load will shorten bearing life by a factor of eight).

Why, then, would high-quality flexible couplings

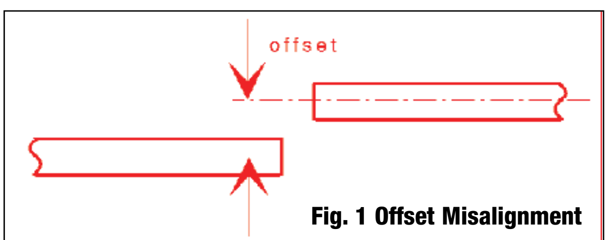

Fig. 1 Offset Misalignment

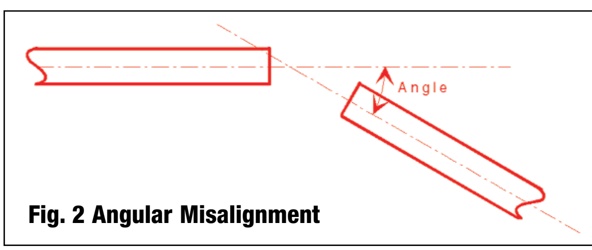

Fig. 2 Angular Misalignment

generally be able to accommodate greater misalignment than what is good for the connected machines? Well, a large percentage of machines *must be deliberately misaligned*—sometimes significantly so—in the "cold" and stopped condition. As they reach operating speeds and temperatures, thermal growth is anticipated to bring the two shafts into alignment. The following case history illustrates the point.

A refinery has a small steam turbine, foot-mounted and enveloped in insulating blankets. The operating temperature of the steel casing is 455° F, and the distance from centerline to the bottom of the feet is 18 inches. The turbine drives an ANSI pump with a casing temperature of 85°F; its centerline-to-bottom-of-feet distance is also 18 inches. Both initially started up at the same ambient temperature. The differential in their growth is

(0.0000065 inches per inch per deg. F) x 18 inches x (455 – 85) deg. F = 0.043 inches.

If these two machines had their shafts aligned center-to-center, this amount of offset would be certain to cast the equipment train into the frequent failure category. Using the "80/20" rule, it would be safe to assume that 20 percent of our machinery population eats up 80 percent of our maintenance money. *This* pump train would be a "problem child" in the 20 percent group.

As we mentioned earlier, aligning center-to-center without paying attention to thermal growth is surely one of the factors that keeps its practitioners in the repair-focused category. Using alignment tools and procedures that take into account all of the above is mandatory for reliability-focused companies.

Describing Permissible Tolerances

There are a number of acceptable ways to describe misalignment at the coupling and to define permissible misalignment tolerances. While many are of equal merit, describing alignment tolerances in terms of foot corrections is unacceptable in a reliability-focused environment. Let's look at the most prominent of the various approaches we might use, then summarize what *not* to do.

Offset and angularity at the coupling (for short couplings)

"Offset and angularity at the coupling" is one of the most common ways of correctly defining alignment tolerances. The offset tolerance simply describes the maximum separation that can exist between two machine shafts at a specific location along their shaft axes, usually the coupling center. The angularity describes the rate at which the offset between the shaft centerlines may change as we travel along the axes of the shafts. Figure 3 serves as a typical illustration of such a tolerance envelope.

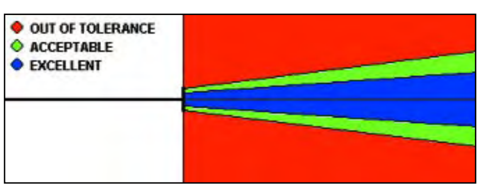

Figure 3. Typical tolerance envelope

The angularity may be described either directly, as an angle in terms of mils per inch (or milliradians), or as a gap difference at a particular coupling diameter. The latter method is popular because it relates directly to what a mechanic can detect with his feeler gages between the coupling faces. A modern laser shaft alignment system measures the angle between shaft centerlines; such a system also can be set to describe this angle as a gap difference at any desired diameter (Fig. 4).

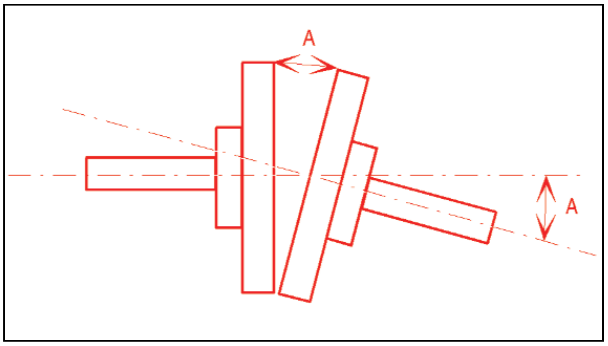

Figure 4. Measuring the angle between shaft centerlines

This approach can have two different interpretations, however. If we describe a permissible offset between the driver and driven shafts as being ***x*** amount in any direction, or ***x*** amount individually in both the horizontal and vertical planes? These two alternatives are *not* the same! The first example is called "vector tolerance" and is more conservative. The second approach, called "standard tolerance," is more commonly used. If you don't want more than ***x*** amount of offset to exist between the shafts in any direction, "standard" tolerances should not be used. Doing so would, in some circumstances, lead to greater-than-intended offsets. Figures 5 and 6 illustrate this point.

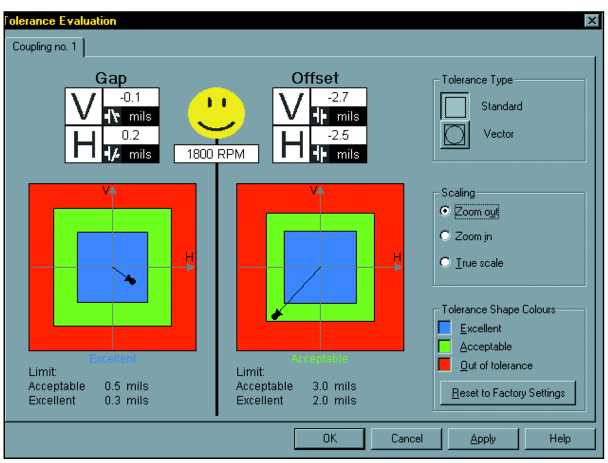

Figure 5. Results of using standard tolerances

Figure 5 depicts a case where applying "standard" tolerances results in an offset of 2.5 mils horizontally and 2.7 mils vertically being deemed acceptable, because the permissible limit for either of these offsets individually is 3.0 mils. However, the actual offset between the shafts is 3.7 mils ($\sqrt{2.5^2 + 2.7^2}$), which is unacceptable if your absolute limit is 3.0 mils. This result can be seen as a "vector" tolerance, shown in Figure 6. A good laser shaft alignment system will allow the user to make this distinction and to specify exactly which type of tolerance is desired.

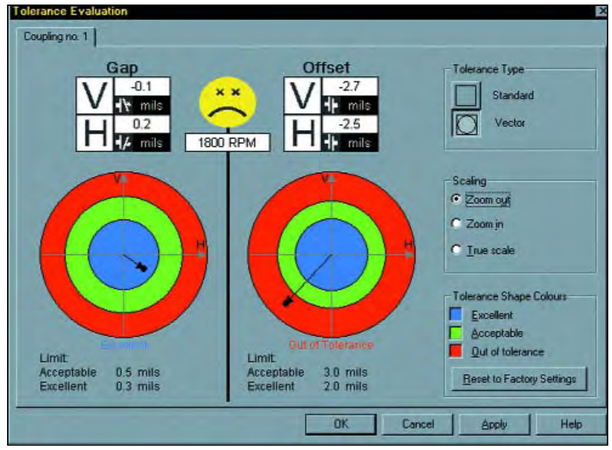

Figure 6. Results of using vector tolerances

As regards tolerances, Table 1 presents the values most widely accepted as the standard industry norm for short couplings:

SHAFT ALIGNMENT TOLERANCES (SHORT COUPLINGS)				
	EXCELLENT		ACCEPTABLE	
RPM	Offset (mils)	Angularity (mils/inch)	Offset (mils)	Angularity (mils/inch)
600	5.0	1.0	9.0	1.5
900	3.0	0.7	6.0	1.0
1200	2.5	0.5	4.0	0.8
1800	2.0	0.3	3.0	0.5
3600	1.0	0.2	1.5	0.3
7200	0.5	0.1	1.0	0.2

Table 1

Spacer Coupling Tolerances

Spacer coupling tolerances are generally expressed as limits to the angle that may exist between each machine shaft, and the spacer piece between them. Since the spacer piece (or spool piece) connects to each of the machine shafts at either end, by definition, there should be no offset between the spacer and each of the machine shafts. Consequently, all that needs to be specified is the maximum angle allowed between the spacer shaft and each of the connected machine shafts. This angle may be specified directly in mils per inch (or milliradians), or in terms of the offset that each individual angle between spacer and machine shaft

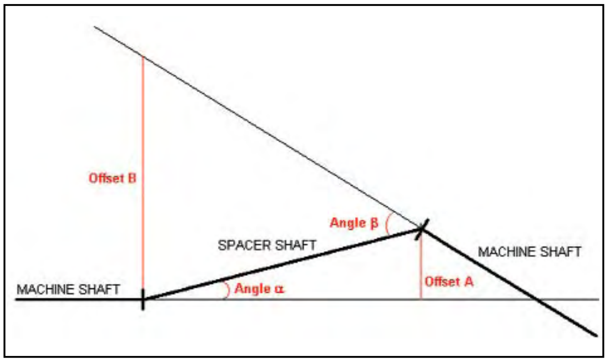

Figure 7. Methods for specifying maximum angle allowed between spacer shaft and connected machine shafts

projects to the opposite end of the spacer. The first way is called the "angle-angle" method (also sometimes called the "alpha-beta" method), and the second is called the "offset-offset" (or "offset A-offset-B") method. Figure 7 illustrates this rather well.

Since most flexible couplings have two flex planes (or points of articulation), the spacer coupling tolerances may safely be used with all such couplings, even the ones usually considered "short flex" couplings. The best criterion to make the distinction is the relation between the diameter of the flex planes and the distance between them. Whenever the distance between flex planes (span) is greater than the diameter, reliability professionals call it a spacer coupling. This will make achieving tolerances easier when performing alignment corrections in the field. It in no way diminishes the conservatism of these values. Table 2 presents the values most widely accepted as the standard industry norm for spacer couplings.

SHAFT ALIGNMENT TOLERANCES (SPACER COUPLINGS)		
Angularity (Angles α and β), or Projected Offset (Offset A, Offset B) (mils per inch)		
RPM	EXCELLENT	ACCEPTABLE
600	1.8	3.0
900	1.2	2.0
1200	0.9	1.5
1800	0.6	1.0
3600	0.3	0.5
7200	0.15	0.25

Table 2

Sliding velocity tolerances

Another approach for specifying tolerances is to describe the permissible limit of the velocity that the moving elements in a flexible coupling may attain during operation. This can be easily related to the maximum permissible offset and angularity through the formula:

Maximum allowable component sliding velocity = $2 \times d \times r \times a \times \pi$

where: d = coupling diameter, r = revolutions/time, and a = angle in radians.

When offset and angularity exist, the flexible or moving coupling element must travel by double the amount of the offset and angularity, every half-rotation. Since the speed of rotation is defined, so must be the velocity that is achieved by the moving element in accommodating this misalignment as the shaft turns. In essence, when we limit the permissible sliding velocity, we have—by definition—also limited the offset and angularity (in any combination) that can exist between the coupled shafts as these turn. For 1,800 RPM, typically this limit is about 1.13 inches per second for excellent alignment, and 1.89 inches per second for acceptable alignment. A good laser alignment system will let reliability-focused users apply this approach as well.

Not Acceptable: Tolerances Expressed as Corrections at the Machine feet

Again, and for emphasis, this approach is wrong. It is impossible to define the quality of the alignment between rotating shaft centerlines in terms of correction values at the feet alone, unless one also specifies the exact dimensions related to these specific correction values, and does so each time. This approach is too cumbersome and error-prone, since two machines will rarely share the same dimensions between the coupling and the feet, and between the feet themselves.

A tolerance that only describes a maximum permissible correction value at the feet without references to the operative dimensions involved makes no sense. This is because the same correction values can yield vastly different alignment conditions between the machine shafts with different dimensions. Such a tolerance simply ignores the effects of RISE over RUN, which is essentially what shaft alignment is all about. Furthermore, such a tolerance does not take into account the type of coupling or the rotating speed of the machines.

These alignment "tolerances" specified *generically* in terms of foot corrections can have two equally bad consequences: the values may be met at the feet, yet allow poor alignment to exist between the shafts; or, these values may be greatly exceeded, while representing excellent alignment between the shafts! This means that, in the first scenario, the aligner may stop correcting his alignment before the machines are properly aligned, and in the second case may be misled into continuing to move machines long after they *have already arrived* in tolerance.

Let's review a couple of examples that illustrate the fallacy of the generic foot correction approach and assume that the specified alignment tolerance for an 1,800 RPM machine is defined as a maximum correction value for the machine feet, ±2 mils. A machine is found to require a correction of –2 mils at the front feet and +2 mils at the back feet, therefore it is deemed by this method to be in tolerance. If the distance between the feet is 8 inches, this would imply an existing angular misalignment of 0.5 mils per inch. If the distance from the front bearing to the coupling center is 10 inches, the resulting offset between the machine shafts at the coupling would be +7.0 mils! This offset is considerably in excess of the ±3 mils of offset (either standard or vector) that is considered the maximum acceptable for an 1,800 RPM machine at the coupling. Yet, with the improperly specified foot correction tolerances, this alignment would be—*erroneously*—considered to be in tolerance! This is a classic example of where small correction values at the feet *do not* necessarily reflect good alignment at the coupling. Figure 8 explains this quite well.

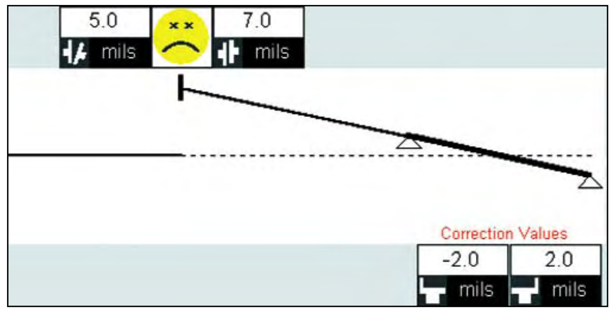

Figure 8. Small correction values at the feet do not necessarily reflect good alignment at the coupling.

Figure 9. Using improperly specified tolerance values at the feet

An equally bad consequence of this approach is that the opposite scenario is just as likely to occur. Assume you have a large machine (such as a diesel engine), running at 1,200 RPM, whose distance between the feet is 80 inches. Further, the distance from front feet to the coupling is 30 inches. Such a machine is found to have misalignment requiring foot corrections of –8 mils at the front feet and –26 mils at the back feet. In this case, the resulting misalignment at the coupling is only 1.25 mils of offset and only 0.225 mils per inch of angularity. Referring to the coupling, both of these alignment conditions are already much better than required by the standard industry norms for 1,200 RPM. However, using the improperly specified tolerance values of ±2 mils at the feet, the aligner would be misled into working much harder and longer than necessary to bring the machines to these values. This can be observed in Figure 9. **P&S**

Figures 5, 6, 8 and 9 created with the assistance of Alignment Explorer™ software by Prüftechnik, Ismaning/Germany, and input from Ludeca, Miami, FL.

Heinz P. Bloch, P.E. is a consulting engineer with over 40 years of experience in chemical process plants and oil refineries. He advises industry on reliability improvement and maintenance cost reduction issues and is the author or co-author of 13 comprehensive texts and over 280 papers or articles on reliability-related subjects. His "Pump Life Extension Handbook" will be available in 2004.