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## **UPDATE YOUR SHAFT-ALIGNMENT KNOWLEDGE**

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When the shaft of two rotator machines are directly coupled via a flexible coupling, any misalignment between their centerlines of rotation can result in vibration and additional loads which, depending on their severity, can produce premature wear, or even catastrophic failure of bearings, seals, the coupling itself, and other machine components. Such misalignment has long been recognized as one of the leading causes of machinery damage, and has been responsible for huge economic losses. The more misalignment, the greater the rate of wear, likelihood of premature failure, and loss of efficiency of the machine. Moreover, misaligned machines absorb more energy and consume more power.

And yet, even excellent alignment of the shaft centers of rotation does not in itself guarantee the absence of vibration. This is because there is still the possibility of imbalance of rotating components. There could also exist 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.

### **A) Don't Overlook Your Basics**

If your plant is like the majority of manufacturing facilities in the industrialized world, your engineering and technician staffs and 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 the importance of back-to-basics thinking and solutions should not be overlooked.

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

If, on the other hand, you update your knowledge of shaft-alignment techniques and acceptable alignment tolerances, you might be on the way to becoming more reliability-focused.

For example, 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 define alignment tolerances in terms of correction values at the machine feet.

The reason why simple “foot alignment” is not enough is because absolute perfection in the alignment of shafts is not realistically possible, nor is it even needed. An analogy might be found in the polishing of a piece of metal: No matter how long the piece is being polished and how fine the different polishing media, a powerful microscope will detect a surface composed of peaks and valleys. The issue is quantification of the alignment quality and determination of allowable deviation- the so-called alignment tolerance.

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.

We define misalignment by visualizing the shaft centerlines of rotation as two straight lines in space. The trick is to get them to coincide so as 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.

Moreover, since the shafts exist in three-dimensional space, these misalignments can exist in any direction. Therefore, it is most convenient for 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 it is here that the harmful machinery vibration and forces are created whenever misalignment exists.

The magnitude of an alignment tolerance (in other words, the description of desired alignment quality), must therefore be expressed in terms of these offsets and angularities, or the sliding velocities resulting from them. Any attempt to correct shaft misalignment by simply selecting standard shaft parallelism tolerances will not yield acceptable results. It is always necessary to take into account the size, geometry, or operating temperature of a given machine.

How much vibration and efficiency loss will result from the misalignment of shaft centers depends on shaft speed and coupling type. Acceptable alignment tolerances are thus a function of shaft speed and coupling geometry.

It should be noted that today's high-quality flexible couplings are designed to tolerate more misalignment than what is good for the machines involved. The bearing load increases with misalignment, and bearing life decreases as the cube of the load increases (so doubling the load, for instance, will shorten bearing life by a factor of eight).

The reason for this current practice in coupling design is that a large percentage of machines must be deliberately misaligned- sometimes significantly so- in the “cold,” stopped condition. By design, it is assumed that as they reach their operating speeds and temperatures, thermal growth or expansion will bring the two shafts into alignment. The following example 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 445°F, and the distance from the centerline to the bottom of the feet is 18 in. The turbine drives an ANSI pump with a

casings temperature of 85°F; its centerline-to-bottom-of-feet distance is also 18 in. Both casings start up at the same ambient temperature. Applying the centuries-old equation for thermal expansion, we learn that the hotter of the two casings will experience differential growth relative to the cooler one, amounting to  $[0.0000065 \text{ in.}/(\text{in.})(^\circ\text{F})] [18\text{in.}] [445-85^\circ\text{F}]$  or 0.043 in.

If, while cold, these two machines had their shafts aligned center-to-center, this amount of offset would be certain to cast the equipment train into frequent-failure category. From the “80/20” rule, it is safe to assume that 20% of the machinery population accounts for 80% of maintenance expenditures. This pump train would be considered a “problem child” even among the troubled 20% group.

In short, 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 thermal growth into account correctly is a mandatory requirement for reliability-focused companies.

### **B) Describing Tolerances**

There are a number of acceptable ways to describe misalignment at the coupling, and to define permissible misalignment tolerances. Let’s take a closer look at the most prominent of the various approaches, and 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 to of correctly defining alignment tolerances. The offset tolerance simply describes the maximum separation that can exist between two machines 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.

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

This approach can have two different interpretations. If we describe a permissible offset between the driver and driven shafts as being equal to X amount, does this mean the magnitude of X in any direction, or the magnitude of X individually in both the horizontal and vertical planes? These two alternatives are not the same!

The first example, X in any direction, is called “vector tolerance” and is more conservative. The second approaches, X individually in both the horizontal and vertical planes, is called “standard tolerance” and is the more commonly used approach.

By way of analogy, the feasible travel from the southwest corner of one mile square to the opposite corner is one mile north and one mile east, but the most direct distance is the diagonal, or vector, distance of 1.41 miles. In alignment operations, not taking into account the vector tolerance may, in some circumstances, lead to greater-than-intended offsets. Figures 5 and 6 illustrate this point.

Figure 5 illustrates 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 limits for either of these offsets individually is 3.0 mils or  $[(2.5)^2+(2.7)^2]^{1/2}$ , 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-based shaft-alignment system will allow the user to make this distinction and to specify exactly which type of tolerance is desired.

As regards to tolerances, the two tolerance tables in this article, present the values most widely accepted as the standard industry norm. For short couplings, these values are presented in 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 not exist any offset between the spacer and each of the machine shafts. Therefore, 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 in. (or milliradians), or in terms of the offset that each individual angle between spacer and machine shaft 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 way is called the “offset-offset” (or “offset A-offset B”) method. Figure 7 illustrates this.

Since most flexible couplings have two flex planes (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 for distinguishing between spacer and short-flex couplings is the relation between the diameters of the components where flexing takes place (i.e. at the periphery of a disc, or at the pitch diameters of mating gear teeth), and the distance between them (span). Whenever the distance between flexing components is greater than the diameter, reliability professionals call the device a spacer coupling. Compared to short flex couplings, spacer couplings will make achieving tolerance easier when performing alignment corrections in the field.

Table 2 presents the values that are most widely accepted as the standard industry norm for spacer couplings.

*Sliding velocity tolerances.* Another approach for specifying a tolerance is to describe the permissible limit of 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 =  $2 \times d \times r \times a \times p$  where:

d= coupling diameter, in.

r= revolutions/second

a= angle, radians

p= Pi

If, as in an example, d= 5 inches; f= 30 revs/sec.; and a= 0.002 radians, then:

Sliding Velocity =  $(2)(5)(0.002)(3.14)$  which equals 1.88 in./sec.

When either offset or angularity exists, there will be a back-and-forth movement for each revolution. In other words, flexible or moving coupling elements are actually traveling either to and fro during each half-revolution, and twice the amount of shaft offset per full revolution. 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 in./s. for excellent alignment, and 1.89 in./s for acceptable alignment. Note that in the above example, the sliding velocity is would be just within the acceptable range. A good laser-based shaft-alignment system will let users easily apply this approach.

### **Expressing Tolerances**

*Expressing tolerances as corrections at the machine feet is not acceptable.* Again, and for emphasis this approach- expressing tolerances as corrections at the machine feet- 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.

Alignment “tolerances” specified generically in terms of foot corrections can have two equally bad consequences: the values may be satisfied at the feet, yet allow poor

alignment to exist between the shafts; or, these values may be greatly exceeded while alignment between the shafts is nevertheless excellent. This means that, in the first scenario, the aligner may stop correcting his or her alignment before the machines are properly aligned, whereas in the second case, he or she may be misled into continuing to move machines long after they have been brought into tolerance.

Let's take a look at a couple of examples that illustrate the fallacy of the generic foot correction approach. 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 at the back feet: therefore, it is deemed by this method to be in tolerance. If the distance between the feet is 8 in., it would imply an existing angular misalignment of 0.5 mils/in. If the distance from the front bearing to the coupling center is 10 in, 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 illustrates this point.

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 in., and assume further a distance of 30 in. from the front feet to the coupling. Let's say that a plotting board or simple calculation making use of anticipated operating temperatures and engine dimensions demonstrates a 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 couplings is only 1.25 mils of the offset and only 0.225 mils/in. of angularity. Referred to the coupling, both of these alignment conditions are already much better than those conditions required by the standard industry norms for 1,200 RPM, as given in Table 1. 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.

## FIGURES

Figure 1: Simply parallel offset is rarely produced by itself

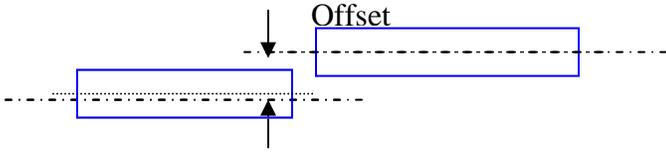


Figure 2: Angular offset, as shown here, of couples shafts often accompanies parallel offset.

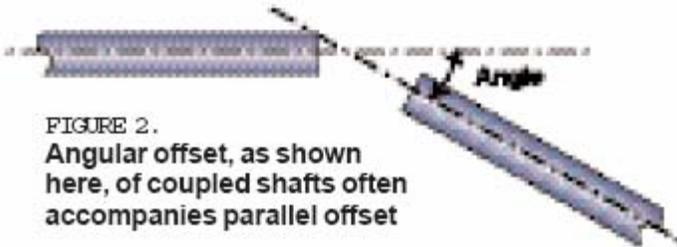
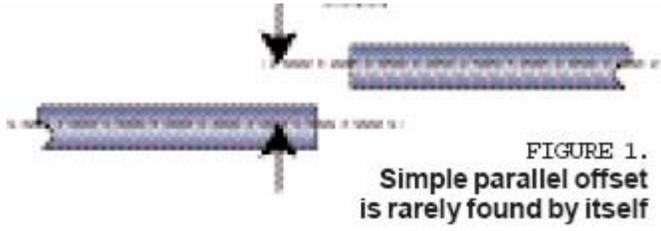
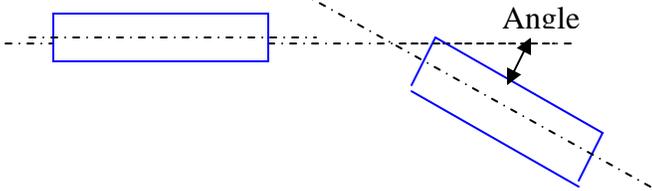


Figure 3: The tolerance envelope shows how the “rise” of a machine (on vertical scale), over its “run” (the distance from the coupling to the support feet of a machine on the horizontal scale), determines the allowable total misalignment angle.



Figure 4: The angle formed by the gap between coupling faces equals the misalignment angle.

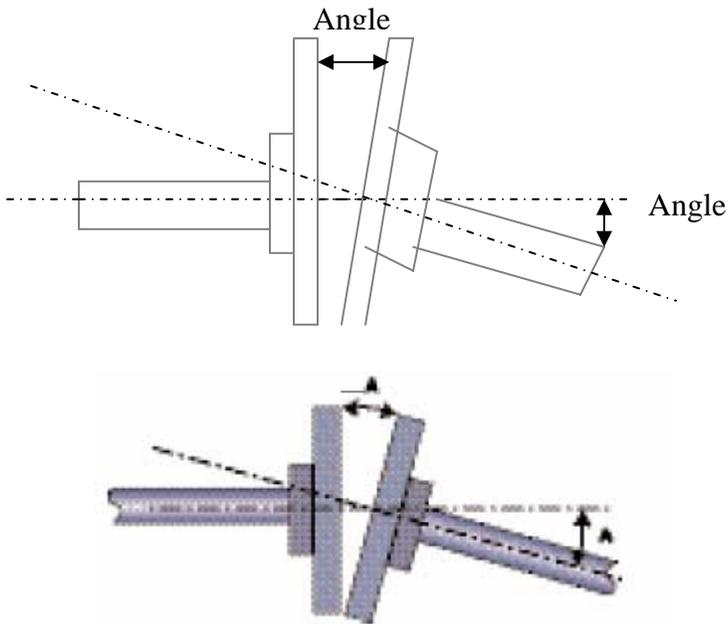


Figure 5: This modern display shows coupling gaps (left) and shaft centerline offsets (right) that are within the acceptable limits of the color-coded tolerance envelope.

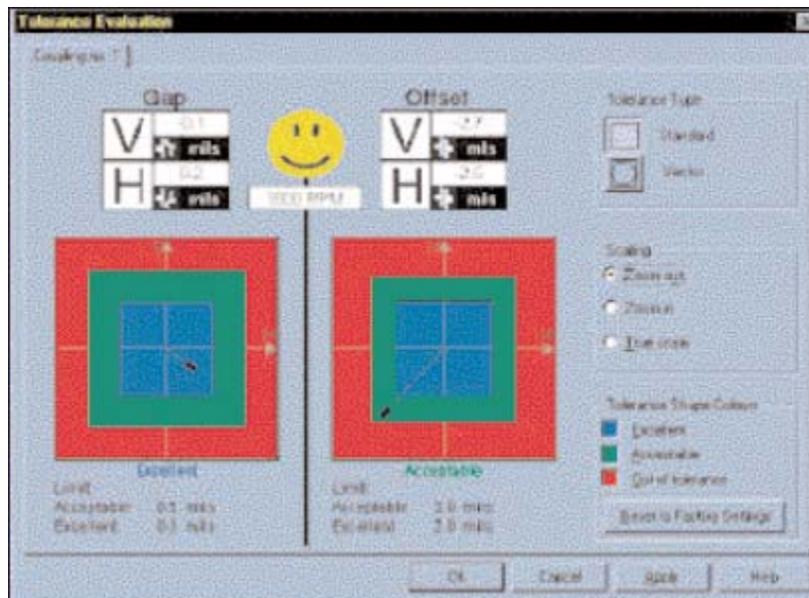


Figure 6: This modern alignment display uses the same V-H gaps and shaft offsets as Figure 5, but shows true misalignment values calculated vectorially. Note that the resulting maximum offsets are now out of tolerance.

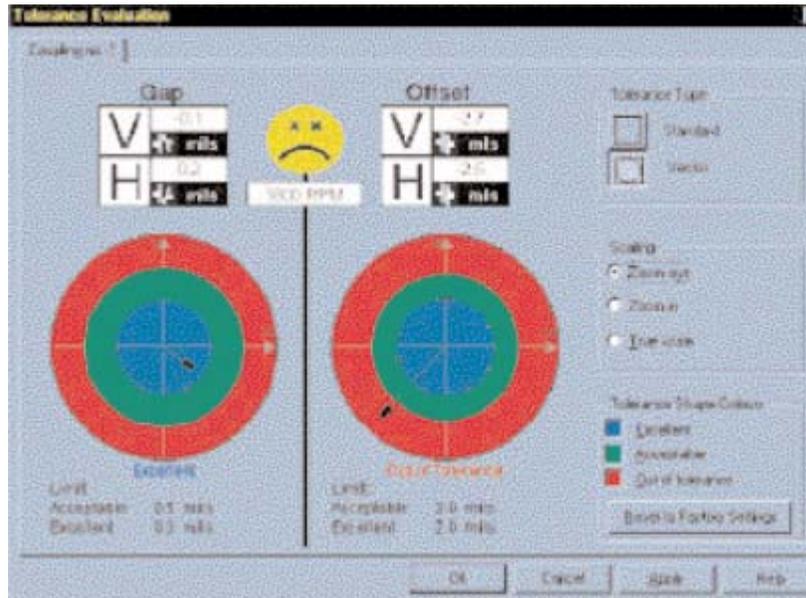


Figure 7: Where a single flexplane (“short”) coupling would have resulted in offset “B” use of the suitable spacer coupling reduced the offset to “A.”

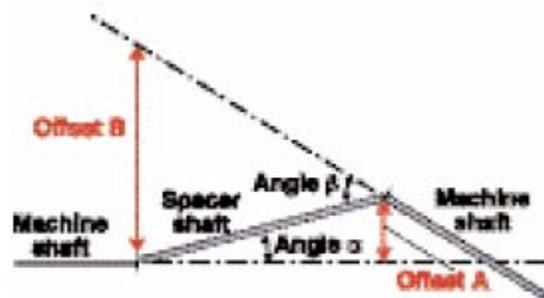
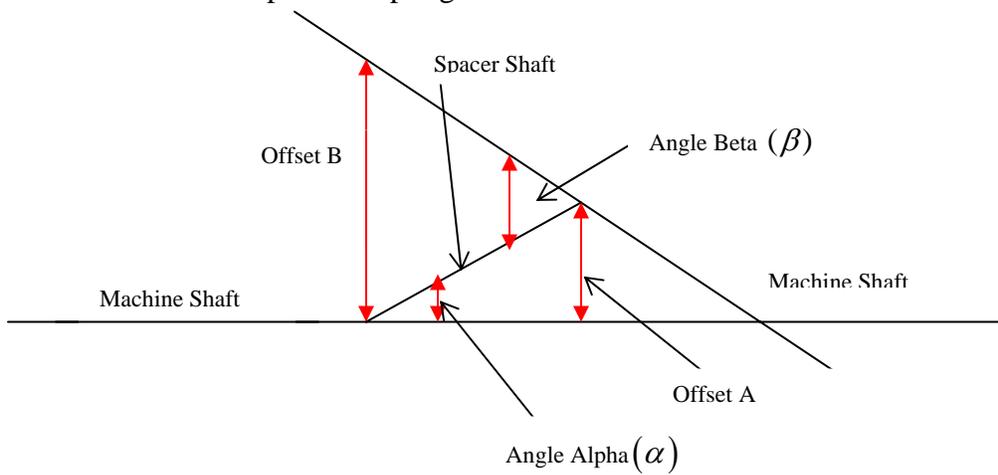


Figure 8: Correction on accuracies of 0.002 in. at the feet would yield unacceptable offsets at the coupling for this example.

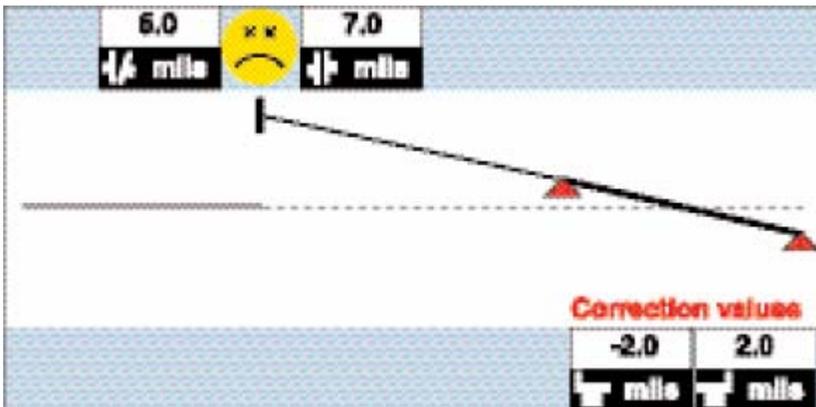
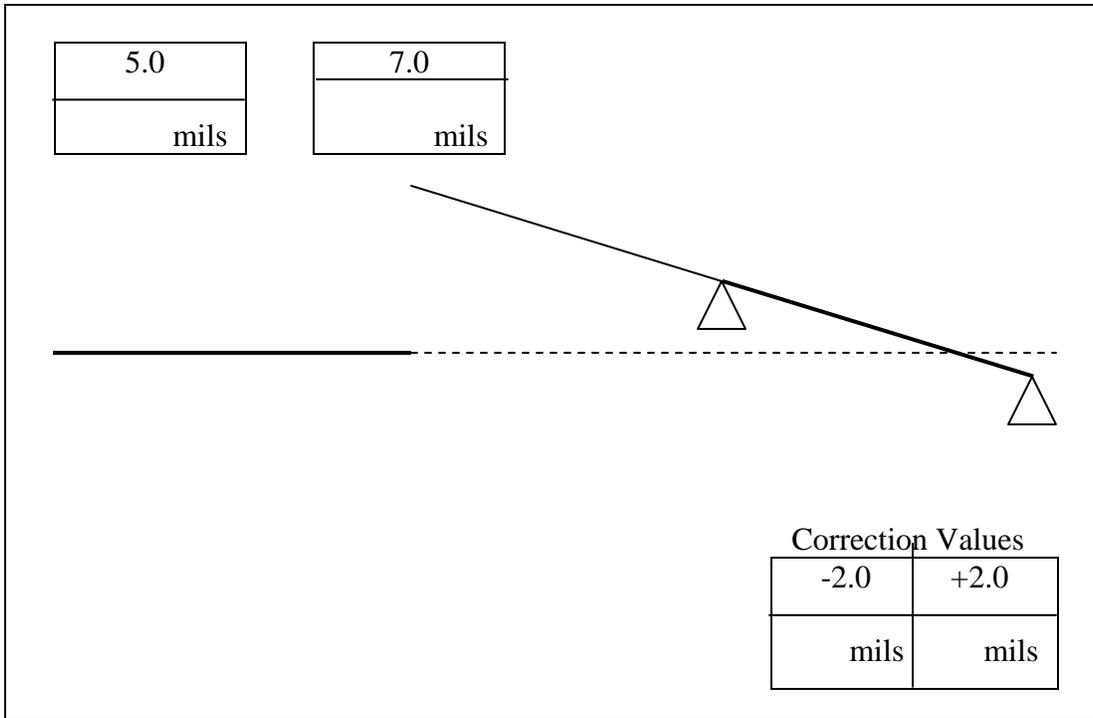
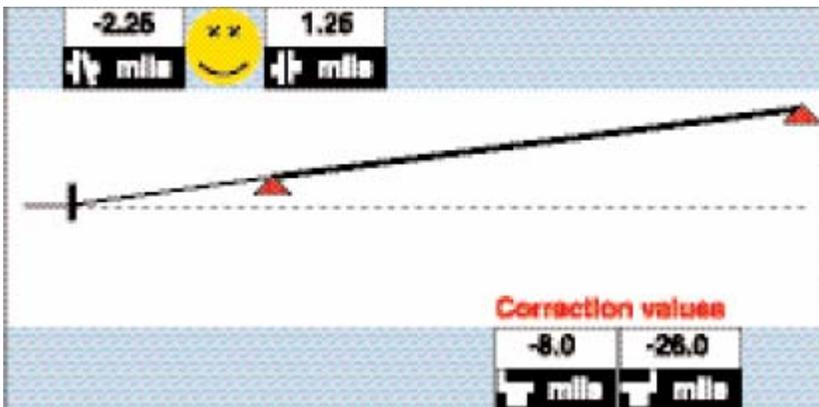
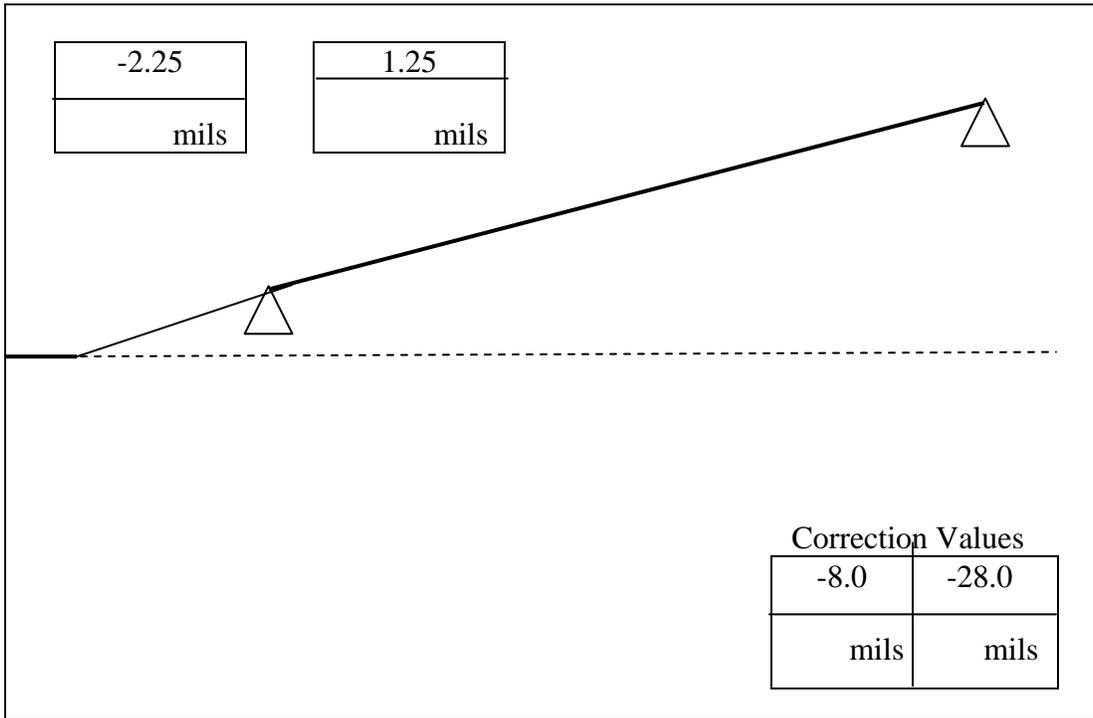


Figure 9: Although asking for corrective values as high as 0.026 in., this diesel engine is already in full compliance with applicable industry norms and requires no further correction.



## TABLES

<b>TABLE 1 SHAFT-ALIGNMENT TOLERANCES (SHORT COUPLINGS)</b>				
RPM	EXCELLENT		ACCEPTABLE	
	Offset, mils	Angularity mils/in	Offset, mils	Angularity mils/in
600	5.0	1.0	9.0	1.5
900	3.0	0.7	6.0	1.0
1,200	2.5	.05	4.0	0.8
1,800	2.0	0.3	3.0	0.5
3,600	1.0	.02	1.5	0.3
7,200	0.5	0.1	1.0	0.2

<b>TABLE 2</b>		
SHAFT-ALIGNMENT TOLERANCES (SPACER COUPLINGS)		
Angularity (Angles a and b), or projected offset (Offset A) , mils/in.		
RPM	Excellent	Acceptable
600	1.8	3.0
900	1.2	2.0
1,200	0.9	1.5
1,800	0.6	1.0
3,600	0.3	0.5
7,200	0.15	0.25

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