PLANT/EQUIPMENT RELIABILITY:

# Vibration Measurement On Turbomachinery

**Since most equipment failures occur on startup, provisions should be made to insure that vibration monitoring systems function during this critical period.**

**C. Jackson,** Monsanto Co., Texas City, Tex.

ONE OF THE IMPORTANT SYMPTOMS OF A healthy centrifugal compressor train is the vibration of the main rotors. This vibration can be measured by many different sensors using several different guidelines for severity, but for workability, sensitivity, and overall effectiveness, Monsanto uses the bearing-mounted proximity probe more than any other protection monitoring sensor. Probes might be described as electronic eyeballs which observe the movement of the rotor shaft with respect to its own journal bearings. The probes take many shapes, as noted in Figure 1. Generally, they are either 190- or 300 mils in dia. The manner of holding varies from machine to machine. The clamped holder is preferred by Monsanto's Mechanical Technology Department rather than screw thread locking nuts which can damage the sensor if they are over-stretched by an installer.

There are several different kinds of probes; the magnetic, the capacitive, the photo or light sensing, and the inductive or eddy current types. For the most part, we rely upon the eddy current type (*1*) since it is least affected by shaft material or the environment around the probe such as oil, gases, or other liquids. The eddy current probe uses a pancake type coil imbedded in the tip. A high frequency field is excited in this coil which has a negative voltage bias, Figure 2. As the distance from the probe coil to the conductive shaft is changed, the output voltage from the detector-oscillator circuit is also changed, as illustrated in Figure 3. This figure also illustrates how the sinusoidal motion of a shaft in its synchronous whirl would be projected through the calibration curve to a read-out generally mounted several hundred feet away in a control house.

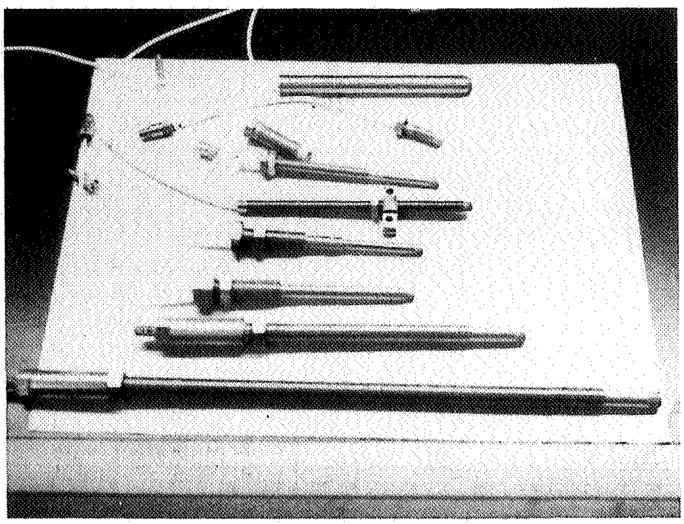


**Figure 1. Proximity probes of various sizes.**

## Action points

In my opinion, it is important to have two relays or action points in vibration measurement. One should be a warning and the other a shutdown point. If a rapid failure is imminent, the warning would allow a few precious seconds for operators to get to the board for a crash shutdown. Normally, the warning means that a level above the acceptable limit has been reached and the proper personnel with the proper diagnostic equipment can be called in to carefully scrutinize the information. In processing the information, extra tools such as oscilloscopes, filters, spectrum analyzers, tapes, and real-time analyzers can be used to develop a complete array of symptoms vs. causes. Sohre (*2*) has compiled the best single breakdown of the many vibration responses and their possible sources, some of which are listed in Table 1.

For every symptom of a problem, there are probably three exceptions to the rule. The more information and time one can get to observe a malfunction, the better the diagnosis. Often the machinery will not be patient and a forced shutdown by the second alarm closes out the examination, and rightfully so, if that limit has been properly selected. *Monitoring is not analyzing*. Good monitoring will provide good points and data for analyzing.

Some machines can be successfully analyzed at the bearing caps. Blake (*3*), Maten (*4*), and IRD (*5*) have published some good guidelines. These are for bearing cap readings and are not to be confused with shaft-to-bearing readings.

Further, some machines are more sensitive than others, have more mass, and operate at many different speeds with different damping (*6*). Maten suggests a velocity limit in in./sec. peak (o/p) for the equipment that he measures at Gulf in Canada. I find 0.2- to 0.4 in./sec. (o/p) a practical value before forcing most units off the line. These measures and values are not used when dealing with the large centrifugal compressor trains as used in ammonia, methanol, ethylene, or other hydrocarbon plants. Other operational guidelines must be used for such equipment.

## Pressure vibration curves

Vibration is generally measured in terms of displacement, velocity, or acceleration. In Figure 4 these terms are indicated in a sine wave of motion with respect to time. Note that the movement (shaft) at any time, $t$, and frequency, $\omega$ (radian sec.) is, disregarding any phase lag, $x = A \sin wt$. Also, from any calculus or physics text the velocity would be the first derivative, or $\dot{x} = A\omega \cos wt$., or at the peak $\dot{x} = A\omega$. The acceleration would be the second derivative, or $\ddot{x} = -A\omega^2 \sin wt$., or $\ddot{x} - \omega^2 x$. Most reputable instrument builders can convert any vibration signal whether it be displacement, velocity, or acceleration, into any other term by using either differentiating or integrating circuits. I have not mentioned that there is a velocity (seismic geophone) type sensor that gives velocity directly, plus the piezoelectric crystal (or strain gauge) accelerometers which give "g's" direct. These two types have had difficulty in the practical measurement of high speed centrifugal machinery or its drivers because of size, temperature, environment, or other factors. A nomograph, as shown in Figure 5, can be easily used to convert terms.

Sticking with the proximity probes and its circuits, we find that there are several things to understand about them. First, a material that varies greatly in its electrical properties will cause an electrical run-out or variable signal output as a shaft is rotated about a true center with the probe in position sensing this shaft. A material that has caused some concern is the precipitating hardening stainless steels, e.g., 17-4 Ph. However, electrical run-out is generally negligible.

Since the surface of the shaft is being observed by the probe which has a high frequency eddy field actually penetrating the shaft's surface, the smoothness of the shaft's surface and the mechanical run-out of the surface have a bearing on the accuracy of measurement. The probe will see discontinuities in the surface and will also read any surface run-out. The rotor has been balanced with the mechanical run-out present and, thus, this amount is really not the vibration signal to be concerned with when measuring a shaft in operation in its synchronous whirl within the bearing limits. The mechanical run-out of most centrifugal compressors and drivers will fall between

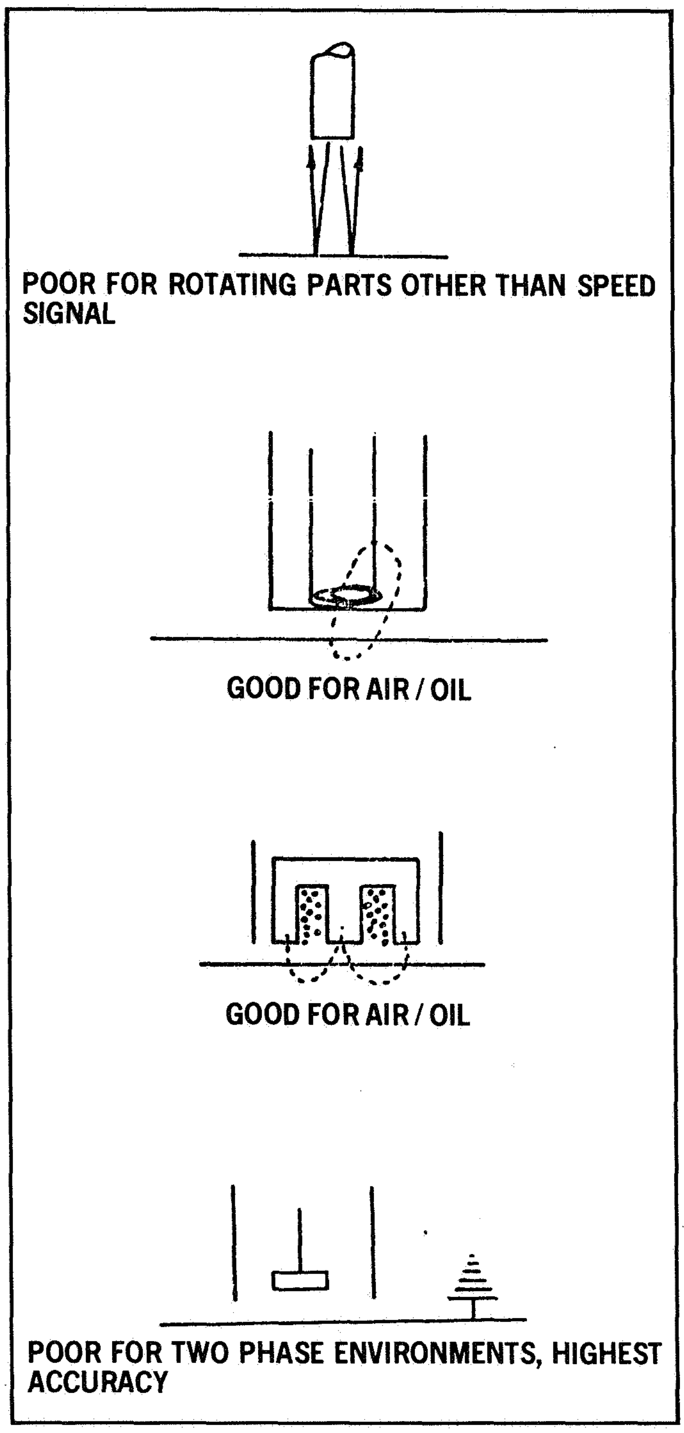
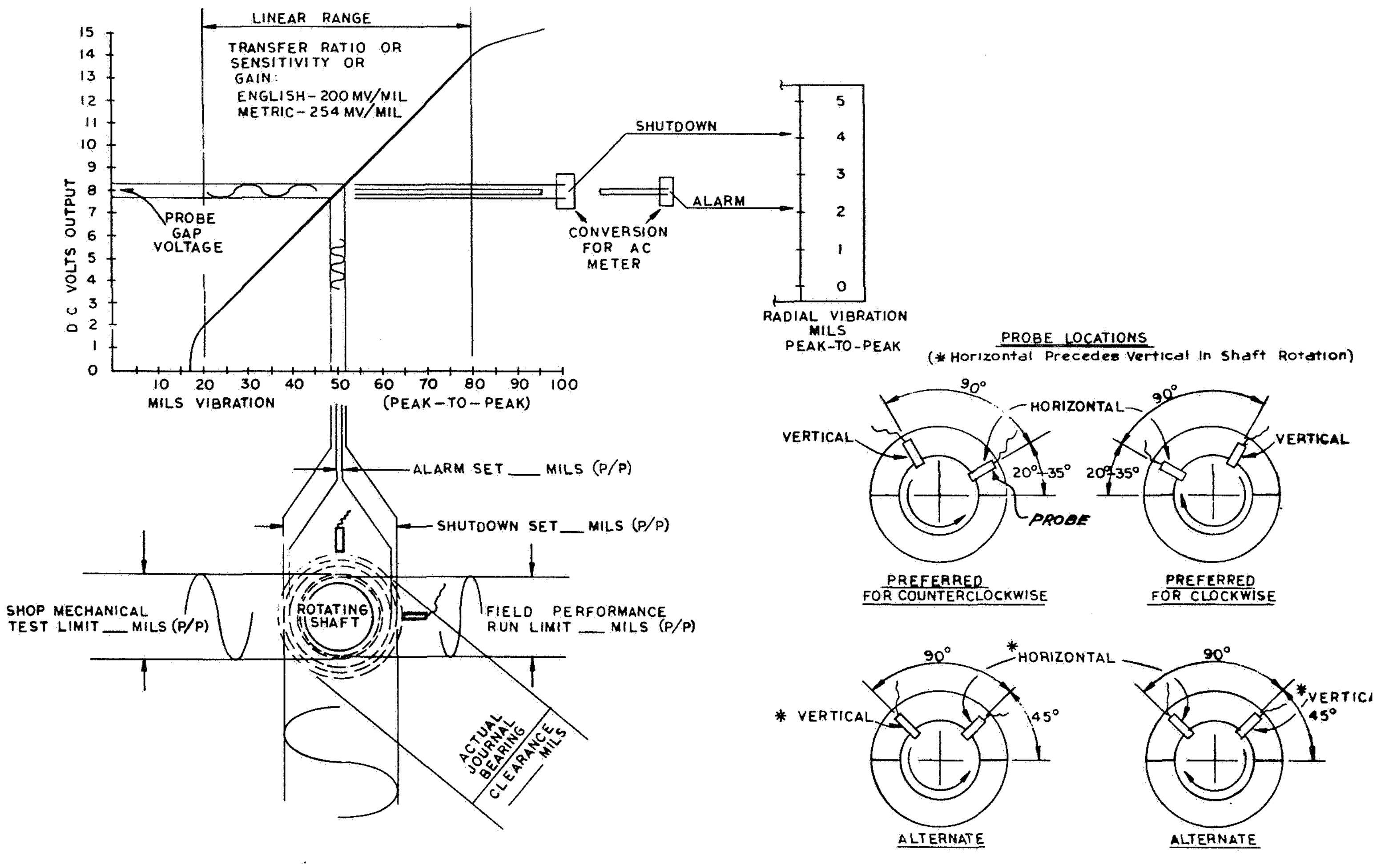


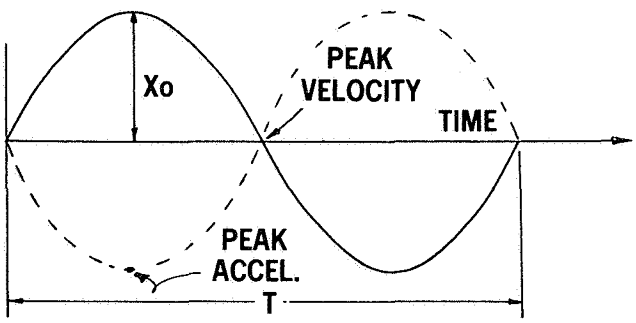

Figure 2. Displacement proximeters.

**Table 1. Symptoms of, and possible contributors to vibration.**

| Frequency | Other Indications | Possible Contributors |
|---|---|---|
| 1 × speed (synch) | Phase is steady, increases with speed, clean signal; no other multiple frequencies indicated. | Rotor unbalance |
| 2 × speed | Changes throughout the day and night, changes in magnitude; correlates with high axial motion and thrust load or unload; phase fixed or flipping 180°; orbit is elliptical with loop in or out. | Misalignment or sticking coupling |
| 0.4 to 0.45 × speed | Very violent, but also sporatic and unstable, phase rolls, changes with oil temperature (maybe); changes with load, dies and comes back; orbit makes loops which disappear and return. | Oil whirl |
| 1 / 2 speed | Steady phase angle, verticle is more sensitive, case and foot frequency is in phase (seismic measure); both bearing points increase. | Baseplate resonance (could be loose bearings) (oil frequency and first critical could have joined hands) |
| 2 × speed | Phase is unsteady, jerky but in same neighborhood; values jump high and low, axial motion low, shifts with load changes, often reduces or stops on slowdown; scratches on orbit and orbit appears to beat like a heart. | Looseness |




▲ Figure 3. Calibration curve for radial vibration measure.




X = AMPLITUDE AT ANY TIME "t".

Xo = MAX. AMPLITUDE OF THE VIBRATION.

$\omega$ = CIRCULAR FREQUENCY = $2\pi f$.

f = FREQUENCY OF THE VIBRATION.

T = THE PERIOD OF THE VIBRATION.

F = FREQUENCY METER READING, CPM.

D = DISPLACEMENT METER READING, MILS.

Vmax. = PEAK VELOCITY IN INCHES / SEC.

$X = X_o \sin \omega t$
For sinusoidal only

$$\text{Velocity} = \frac{dx}{dt} = X_o \omega \cos \omega t$$

$$\text{Acceleration} = \frac{d^2x}{dt^2} = -X_o \omega^2$$

$$\text{Vmax.} = \text{Peak Velocity} = (X_o)(\omega) = \left(\frac{D}{2}\right)(2\pi f) = 52.3\,D\left(\frac{F}{1000}\right) \times 10^{-3}$$

Max. Acceleration = $X_o \omega^2$ or in meter readings

$$G\ \text{max.} = 14.1\,D\left(\frac{F}{1000}\right)^2 \times 10^{-3} = \text{gravity units or (g's)}$$

Figure 4. Simple sinusoidal vibration motion.

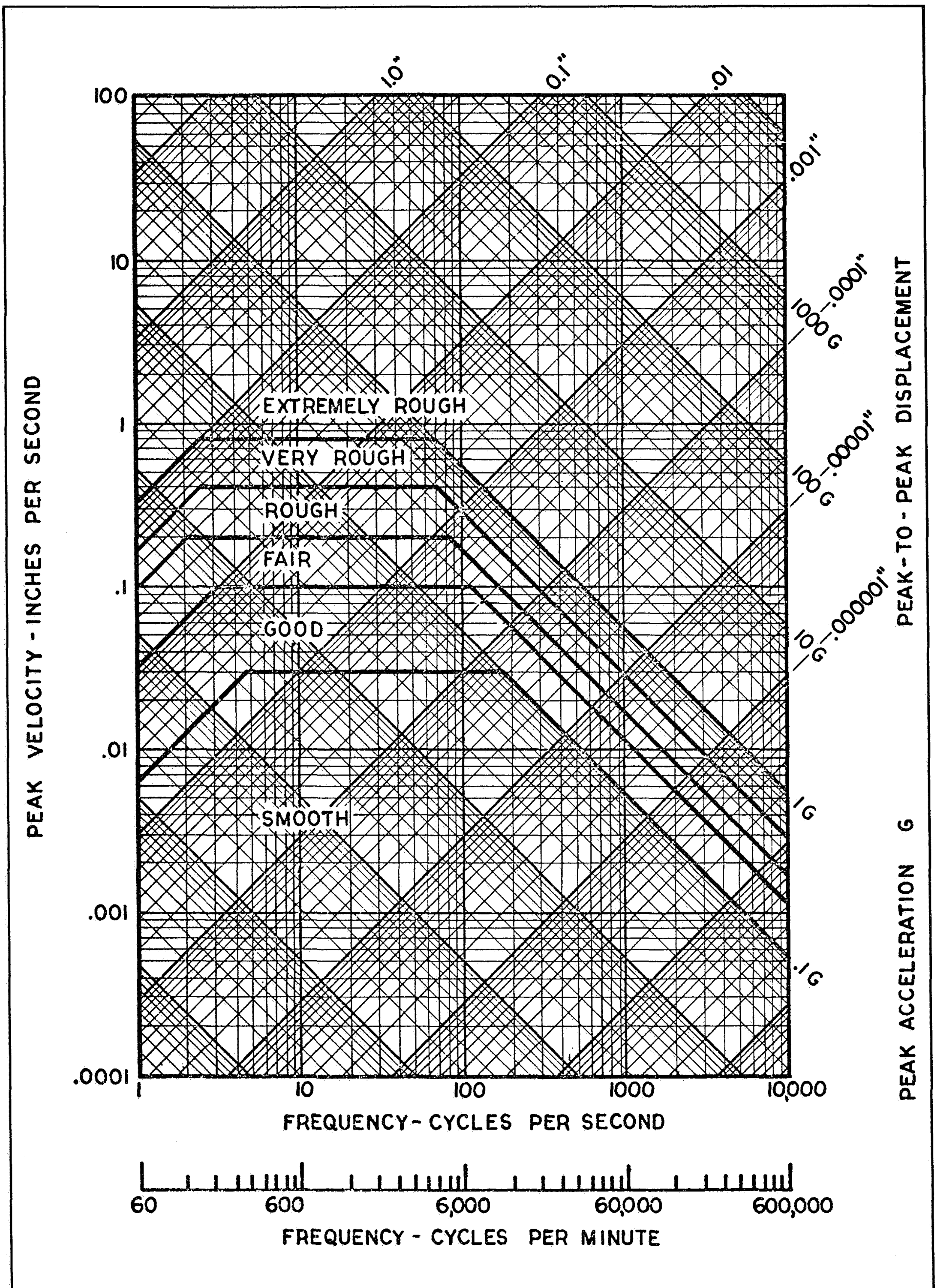



Figure 5. Displacement, velocity, acceleration conversion nomograph.

0.2- to 0.3 mils TIR (total indicator reading).

## Design and maintenance

Good monitoring systems should allow the following provisions:

1. Check for proper probe-to-shaft gap while operating.
2. Monitoring by oscilloscope or other device including tapes, recorders, filters, and frequency vs. vibration processing analyzers.
3. Continuous monitoring of all points with *alarm* or *trip* relays in service at all times.
4. Display of all points, or frequent scan or review by operations.
5. Ability to check calibration by introducing a reliable simulation signal at the machine to check questionable response in the instrumentation and leads.
6. Freedom from false signals introduced from other sources such as RF in the plant or induced electrical pulses from switching circuits.
7. A grounded system at low voltage for intrinsic safety against ignition of gases.
8. Maximum resistance to mechanical damage by people and weather, particularly rain and steam leakage.
9. Good service by the builder and adequate training/instructions for the user's plant personnel.
10. High reliability from plant electrical interruptions such as power flickers.
11. Provisions to startup equipment and pass through criticals without disarming the protection, since most failures occur on startups.
12. Provisions for disarming the system safely for planned troubleshooting of the protection (instrumentation) circuits.
13. Quick identification of an *alarmed* or *tripped* channel for response by the operating unit.
14. Back-up spares, standard connections, basic tooling, and adequate schematics.

Figure 6 shows a circulator compressor which is protected by probes mounted internally to the bearings observing the shaft motion. This compressor weighs 74,000 lb. The rotor weighs about 500 lb. With this weight ratio of 147:1, it is conceivable that a high isolation ratio (isolation ratio is the ratio of the generated signal or forcing function to the received or measured signal) of 75:1 or 50:1 could be expected. I do not think that case measuring is practical here.

Figure 7 shows a typical barrel compressor protected by bearing-mounted probes. While the weight ratios might narrow down to 20:1, the more sensitive measure is at the shaft where the trouble generally starts in the first place. More normally, a vibration forcing function will originate to, or on, the rotating shaft. Why not start the measure at that point? If a 4 in. dia. shaft or journal is whirling about at a 4 mil peak-to-peak displacement relative to its bearing (4 mil dia. orbit), then this bearing, which may have a total dry clearance of 5- to 6 mils over the journal at ambient temperature, sets some pretty exacting limits on how far one should allow the vibration to go. A limit of 8 mils peak-to-peak (p/p), for example, would be pretty ridiculous it would seem, i.e., 1- to 2 mils of babbitt may have to be wiped off to allow the vibration.

If this were a turbine rotor at 5,000 lb. weight equally supported with 2,500 lb. of static weight supported at each bearing, then API 612 would have called for a balance, minimum, to have residuals not exceeding 10% of the journal static weight, or in this case 250 lb., and calculated at maximum continuous operating speed which we could say, for illustration, is 10,000 rev./min. Newton says that force equals mass times acceleration, $F = MA$ or $F = (W/g)\omega^2 r$ or converting from pound mass, in./sec./sec., radian/sec., inch units to pounds, rev./min., oz-in. units we have the force, in pounds, equal to 1.77 times the (speed/1,000)$^2$ times the oz.-in. unbalance. Or 250 lb. = 1.77 (10,000/1,000)$^2$ (oz.-in.), then the oz.-in. residual unbalance limit would be 1.41 oz.-in. The peak-to-peak measure of this motion would be

$$\frac{1.41 \text{ oz.-in.}}{(2{,}500 \text{ lb.})\,(16 \text{ oz./lb.})} \times 2 = 70\mu \text{ in.}$$

or $70 \times 10^{6}$ in., or 0.07 mils peak-to-peak. The 4 mils measured from above would represent 57 times as much as was accepted on the balance stand. From another standpoint, and really the one of most interest, the force on the bearings (assuming zero damping) from the vibrations measured represents a cyclic force of 57 × 250 lb., or 14,250 lb. Almost 5 g's at 167 cycles/sec. which is very damaging, in my opinion.

This illustration is offered only to attempt a correlation to the work in diagnostics which can be performed with data, and to show how vibration measure is an important parameter in the health of a critical large high speed, single line compressor train. The first measure would hopefully start at the time the equipment is given the mechanical or performance run test at the manufacturer's plant.

**Figure 6.
Pipeline type booster centrifugal compressor.**

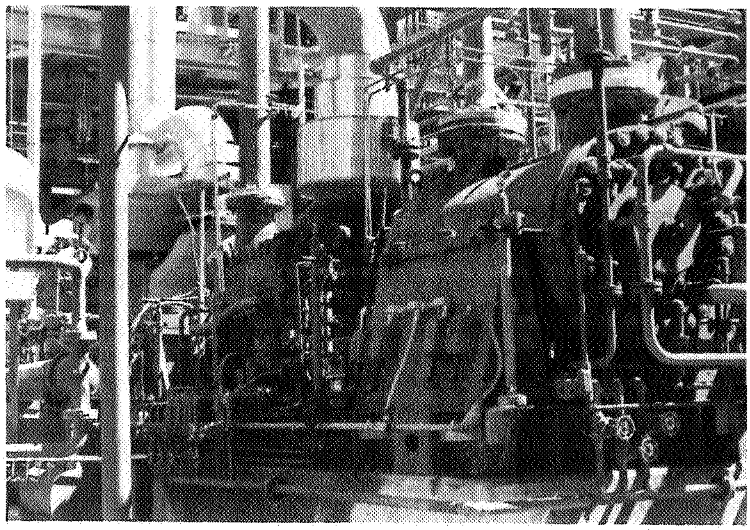


**Figure 7. Barrel centrifugal compressor plus horizontal split case compressor.**

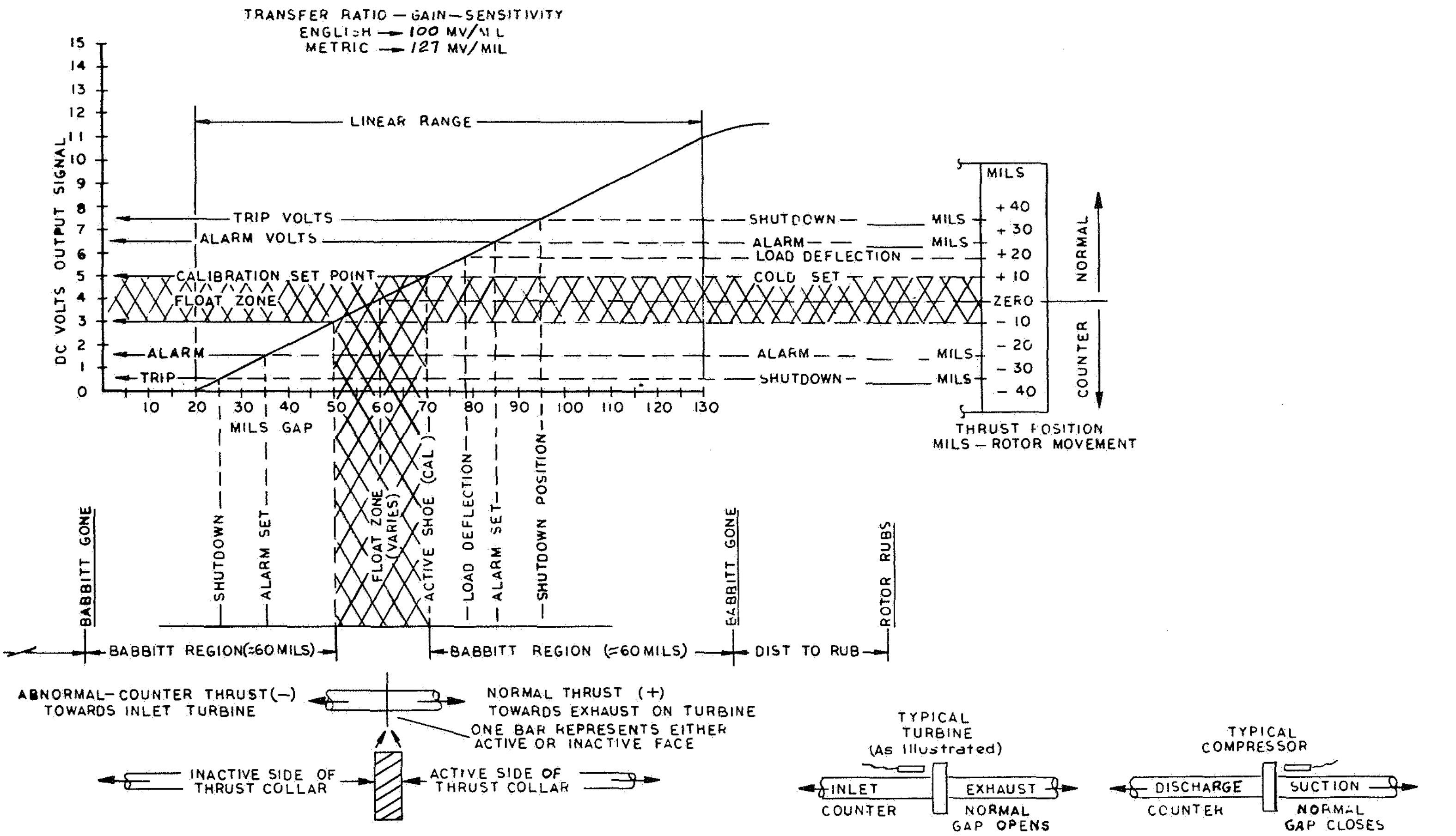



**Figure 8. Thrust displacement calibration curve.**

The same measuring facilities used for vibration measure can be used for thrust displacement which would indicate thrust bearing failures. Figure 8 is a typical calibration curve for such a system. Our Mechanical Technology Department is presently asking for an 80 mil range for such a system. Our past practice called for 60 mils of measuring range.

## In conclusion

I hope that this article has given some insight into the issue of measuring vibration with proximity probes. There are many pitfalls to such a system, e.g., chrome plating or metalizing a shaft in the region where the probe observes the shaft. This article does not allow time to discuss these factors. Monsanto has over 150 channels at the Texas City plant alone. We plan to install more. We have started up over 40,000 h.p. of turbomachinery since last winter with no loss production attributed to machinery, yet we have corrected two rubs, several misalignments, several oil whirls, one anti-surge valve spring failure, two sludged couplings, two pump vane failures, and one bearing failure, in addition to three miscalculated criticals in the manufacturer's shop during 1970. We use and strongly support (7) two probes mounted per bearing, 90° circumferentially apart to sense the weakest plane motion. #

## Nomenclature

$x$ = amplitude at phase wt.
$\omega$ = angular velocity, radians/sec.
$t$ = time in sec.
$\dot{x}$ = velocity, in./sec.
$\ddot{x}$ = Acceleration, in./sec.$^2$
$g$ = gravitational acceleration, 386 in./sec.$^2$
$o/p$ = zero to peak amplitude
$p/p$ = displacement, peak-to-peak, double amplitude

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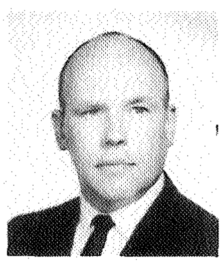

**C. Jackson** is a senior engineering specialist in a Mechanical Technology Section working on mechanical and method-oriented problems at the Monsanto Co. plant, Texas City, Tex. He has been with the company for 21 years working in division engineering, production, maintenance and plant design, before setting up the present department in 1960. Jackson received a B.S. Degree in mechanical engineering from Texas A&M University. He is a registered professional engineer.

# DISCUSSION

**JOHN LIVINGSTONE,** ICI, Billingham, England: Welcome to the carbamate failure club, Charlie, but we're still the champions. I would just like to know if it's possible to know what sort of conditions you were getting when you started to have the failure. Presumably it happened in a very short time. The usual thing is either the $CO_2$ concentration went up or the temperatures went down. I wonder, have you any information on this at all?
**JACKSON:** I can answer the process questions really easy. I don't know anything about them. I'm sure the buildup occurred for some time. Actually, we do have a $CO_2$ feed into the inner stage of that second case, coming from elsewhere. So it is complicated by that one point. Now as far as when it started, we really don't know. Of course, the unbalance can jump from 1/2 mil to 2 1/2 mil and when this happens some of the material was thrown off and probably caused the jump in vibration.

We're interested in the fact that location of probes is being investigated by turbine manufacturers. We don't want to locate on a nodal point for if you do, the rotor shaft can go to hell and back but it won't show anything. You're only seeing the movement at the node, and that's very small.

We just brought down a turbine in ethylene service which was alarming. We were seeing what we would call a coupling critical, but couldn't prove it. We went up to 4 mils and shut down at 4.2. We ran somewhere between three and four mils for about three days to try to locate our repair sets of blading. Then we brought it down two weeks ago and we had a 70% crack through the shaft at the coupling hub through the keyway. We don't know how much the fatigue was going on, but we do feel there was a change in section modulus.
**ANON:** After we repaired a unit we noticed big vibrations proportional to the frequency of the compressor when starting, and it looked like a severely unbalanced system. We investigated very thoroughly and finally we found it was a misreading due to the rotor shaft having some residual magnetism. We later found the magnetism was caused by a workman who had put a magnetic device on the shaft
**JACKSON:** This is a very good point. We had this happen on one test stand. The manufacturer did a magnetic particle check and they did not de-gausse the shaft. You get a square wave and this is really weird. The unit went on the balance stand three times before the vendor located the problem.