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2

Vibration Monitoring for Predictive Maintenance in Central Energy Plants

by
Ralph Moshage
Lyndell Bowman
Clay Bethea
Robert Chesser
David Burch
David Dyer

Equipment maintenance and unexpected downtime resulting from equipment failure can make up a significant part of the cost of running a central heating plant (CHP). As with most rotating equipment, the machinery in CHPs often suffers from the effects of vibration. A predictive maintenance program uses vibrational analysis to deal with potential vibration problems by monitoring vibration electronically, and by using regular measurements to distinguish between normal and exceptional vibration signals. Since vibrational analysis can help predict component failure, those parts identified as defective can be scheduled for repair or replacement during planned equipment shutdowns, rather than during costly emergency downtime due to equipment failure. Vibrational analysis can also be used as a tuning device for rotating equipment, as a way to pinpoint and reduce machine inefficiencies, thus reducing operating costs.

This study identified specific CHP plant equipment that could benefit from a predictive maintenance program that uses vibration analysis, and outlined the steps to implement such a program for Army CHPs. This report also lists vendors and equipment specifications for predictive maintenance systems.

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FOREWORD

This study was performed for the U.S. Army Center for Public Works (USACPW), Fort Belvoir, VA, under Project 4A162784AT45, "Energy and Energy Conservation"; Work Unit EB-XW2, "Advanced Operations and Maintenance Technologies for Central Heating Plants." The technical monitor was Satish Sharma, CECPW-FU-M.

This research was conducted by the Energy and Utility Systems Division (FE), of the Infrastructure Laboratory (FL), U.S. Army Construction Engineering Research Laboratories (USACERL). The USACERL principal investigator was Ralph Moshage. Appreciation is owed to Lyndell Bowman and Christopher L. Dilks, of USACERL, for valuable technical input in the writing of this report. Clay Bethea, Robert Chesser, David Burch, and David Dyer are associated with Auburn University, AL. This research was partly supported by an appointment to the Research Participation Program at USACERL, administered by the Oak Ridge Institute for Science and Education, through an interagency agreement between the U.S. Department of Energy and USACERL. Dr. David Joncich is Chief, CECER-FE, and Dr. Michael J. O'Connor is Chief, CECER-FL. The USACERL technical editor was William J. Wolfe, Information Management Office.

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1 INTRODUCTION

Background

The Army operates about 75 central heating plants (CHPs) throughout the United States. Equipment maintenance and unexpected downtime resulting from equipment failure can make up a significant part of the cost of running a CHP. As with most rotating equipment, the machinery in CHPs often suffers from the effects of vibration, defined as "a rapid, periodic, oscillation of an elastic body." Even though some vibration is unavoidable, excessive vibration should be minimized since it can lead to premature deterioration of equipment components, and eventually result in complete failure of the unit.

A predictive maintenance program uses vibrational analysis to deal with potential vibration problems by using electronic hardware and software to monitor vibration, and to distinguish between normal vibration signals and those that indicate problems in equipment operation. Vibrational analysis is rapidly becoming standard practice in most industrial predictive maintenance programs. Many industries have found that there are significant advantages--including cost savings--in using this method to predict component failure in rotating machinery. Since vibrational analysis is a tool that predicts component failure, those parts identified as needing corrective action can be scheduled for repair or replacement during planned equipment shutdowns, rather than during costly emergency downtime due to equipment failure.

Vibrational analysis can also be used as a tuning device for rotating equipment. An analysis of different areas of a systems' components can help pinpoint and reduce inefficiencies, thereby reducing operating costs. As yet, this technology has not applied to Army CHPs. Work needs be done to identify the specific components of Army CHPs that could benefit from a vibrational analysis program, and to outline the steps to implement such a program.

Objectives

The objectives of this study were: (1) to provide an overview of vibration monitoring as a predictive maintenance tool for use in U.S. Army central heating plants, (2) to identify specific plant equipment that could benefit from a vibration analysis program, and (3) to outline the steps to implement a vibrational analysis program for central heating plants.

Approach

A literature search was done to identify sources of information and manufacturers of vibration analysis equipment. This information was then used to identify central energy plant equipment that would benefit from this technology. Finally, procedures for doing an economic analysis and implementing the technology were developed.

Mode of Technology Transfer

Information of the vibration analysis technology will be disseminated through field demonstrations at Army facilities. It is anticipated that the results of this study will be incorporated into an Army Engineer Technical Note (ETN), and an article to be published in the *DEH Digest*.

2 PREDICTIVE MAINTENANCE USING VIBRATIONAL ANALYSIS

An effective predictive maintenance program will depend on several factors, the largest of which is program management. If a facility already has a working preventative maintenance program, then the addition of a predictive maintenance program should be an almost effortless change.

Two major factors in creating a successful program are to acquire the correct equipment and to train the personnel that will administer the program. Fortunately most companies that sell vibration monitoring equipment offer free seminars on the use of their equipment.

Starting a predictive maintenance program is a large task that demands a structured approach. Predictive maintenance should be gradually integrated into the current maintenance program in five steps:

1. Choosing personnel
2. Selecting a program type
3. Purchasing the equipment
4. Providing office space
5. Choosing a route.

Choosing Personnel

The first step in beginning the program is to choose one maintenance person to help start the program. Ideally, this employee should have:

1. An interest in starting a vibrational program

2. Excellent knowledge of the plant equipment and should have performed many hours of maintenance on the equipment (the candidate should be a seasoned veteran.)

3. Extensive experience in troubleshooting the equipment located in the facility

4. The ability to comprehend technical data.

Selecting a Program Type

The second step in beginning the program is to decide what path the program should take. Four methods of running a predictive maintenance program are suggested:

1. The first option for setting up a predictive maintenance program includes the use of a vibration meter. These devices are much less expensive than vibration analyzers, and also provide much less information. Vibration meters alarm the user of worsening vibration before failure, without actually identifying the cause of the vibration. It is left up to the seasoned veteran to diagnose the actual problem, and propose the appropriate solution.

An additional advantage of using a vibration meter in plants where equipment is maintained contractually is one of quality control. By taking a few simple vibration readings, proper equipment installation and maintenance may be verified. If a machine vibrates excessively, the contractor can be notified and requested to rectify the problem.

2. The second option in setting up a predictive maintenance program includes the use of a vibration analyzer in conjunction with appropriate training for the personnel administering the program. This training, which is offered by several of the analyzer manufacturers, teaches the employee to read the plots generated from the data collected with the analyzer to enable the personnel to predict possible failures, and then to correct the problem.

Available software can help the technician find problems so data plots can be analyzed to predict equipment failure. One way to do this is by using broad band analysis. In this method, alarms are set on each individual machine to detect an appropriate vibration threshold. When this level is exceeded, the software produces a report that informs the technician of the problem. The technician can then retrieve the graphs for that machine, predict the probable cause of the vibration, and schedule the proper corrective actions. With the correct training and a little experience, the personnel will excel at predicting the trouble areas. Appendix A lists several institutions that offer training in data analysis.

This approach is recommended because it requires training that will give the technician an understanding of the process. This gives the technician valuable experience in predictive maintenance, which in turn, will help in later decisions regarding when and how to upgrade the system. A technician who understands the system's nuances can provide valuable feedback.

3. The third option for setting up a predictive maintenance program includes the use of a vibration analyzer, in conjunction with an expert system computer program. With this system, the technician must still take data readings from the equipment, but when the readings are downloaded into the computer, the software automatically analyzes the data and produces reports on different pieces of the equipment in the plant.

This approach is the best that present industry has to offer, but it does not require the technician to understand vibration analysis technology; the technician must depend on someone else to solve the problems that arise. With the second approach, a facility can start a predictive maintenance program and later expand into the third approach, without initially investing in an expensive expert system.

4. A fourth option for setting up predictive maintenance program is to contract with a qualified company to provide all predictive maintenance services. Despite savings in the initial start up costs, most CHPs would find this option considerably more costly to maintain throughout the life of the plant than any of the other three previous options; this option is not recommended.

Because option number two is recommended as the best program type for CHP's, the remainder of this report will deal primarily with expanding upon topics associated with this option—setting up a preventive maintenance program that uses a vibration analyzer in conjunction with a personnel training program.

Purchasing the Equipment

The third step in beginning the program is to purchase the equipment. Before purchasing the equipment, the maintenance team using the equipment should decide what brand will best suit their needs. As in purchasing any item, the users are more likely to be satisfied with the product if they were part of the decision making process. Appendix B includes a table of vibration monitoring equipment vendors. Sample hardware/software configurations from three vendors are presented in Chapter 8, "Available Hardware" (p 29).

Providing Office Space

The fourth step in beginning a program is to provide sufficient office space for the predictive maintenance program. The amount of space needed will vary according to the size of the program. The space can be determined by each facility, but a few essentials are:

1. If a computer is to be used in the data storage/analysis process, the office containing the computer will need to be air-conditioned.

2. The office should be adequate to accommodate storage of hard copies of generated reports for future reference.

3. There must be an area to store and charge the data collectors while they are not in use.

Choosing a Route

The fifth step is to decide which machines should be included in the program. At the beginning of the program, only a few machines should be selected for analysis. Prematurely monitoring too many machines can actually foster mistakes and damage the program's credibility. Once the technician is comfortable with the program, it should expand to encompass all the critical or expensive rotating machinery in the facility.

Since the vibration level in equipment is proportional to the life of the equipment, it is important to minimize vibration levels. Mechanics have used this logic for years to increase the performance life of mechanical equipment. For example, if a newly installed piece of equipment obviously vibrated erratically, the mechanic assumed that, without correction, the machine would likely fail in a relatively short period of time. The mechanic would adjust the various mechanical parameters, to "tune" the equipment to run more smoothly.

Sometimes vibration can be subtle. A mechanic may visually inspect some running machinery and see no real significant vibration, but a hand on the equipment can detect a definite periodic vibration. Better yet, a screwdriver placed on the machine and held to the ear, can transmit even slighter vibrations. By using extra resources, a mechanic can locate less obvious sources of vibration that could limit the life of the equipment.

Modern electronic technology can improve maintenance personnel's ability to prevent machinery breakdown. Just as the mechanic used a screwdriver to reveal vibrations invisible to the visual inspection, modern technicians can use improved resources in data collection and analysis to better estimate the condition of the equipment. Gathering the vibration signatures from the plant equipment is a vital step. By using a portable vibration analyzer, maintenance personnel can monitor many types of equipment in a relatively short period of time.

The fundamental element of a vibration analyzer is the probe, which is an instrument containing an accelerometer, which converts vibration into an electronic signal. This transducer consists of a spring-mass system (with a damping factor of virtually zero) which, under the influence of acceleration, generates a force that presses against a piezoelectric crystal. These crystals (such as quartz or barium titanate) generate an electric charge proportional to the applied pressure. The natural frequency of the transducer (which is equal to the square root of the spring constant divided by the mass) is, as an example, of the order of 50,000 Hz. This will allow accurate measurement of vibration frequencies of up to 3000 Hz (180,000 CPM).

To further ensure accuracy in the measured vibration data, readings should not be taken while a machine is operating at or near resonance. Resonance is a condition that arises when a machine is being driven (operated) at or near its natural frequency. Any measurements taken while a machine operates at or near resonance will contain large errors in the vibration amplitude data. A solution to resonance problems is proposed later in Chapter 4, "Data Interpretation" (p 18).

It is important that the measured frequency and the natural frequency do not coincide to maintain accuracy in measurement. As the measured frequency approaches the natural frequency of the system, resonance can occur, which causes tremendous errors in measurement. To alter the natural frequency, the mass or stiffness of the body could be changed.

To collect the data from the machinery, the probe is simply attached to the surface of the equipment (hand-held, glued, clamped, etc.), and the time-domain information is downloaded into the hand-held collector portion of the vibration analyzer. Note that the position of the probe relative to the equipment is important. In dealing with rotating machinery, vibration can occur in two primary directions, axial and radial; the radial direction can be further broken down into vertical and horizontal components. As a result of the multidirectional vibration possibilities, measurements should be taken in all three directions. Experience may dictate that only one or two directions be monitored due to nature of the equipment. For example, a rotary pump might produce a displacement in the horizontal direction only due to the way it is mounted to its support structure. However, monitoring an additional axis is relatively simple and should be done whenever possible to ensure a complete analysis.

When monitoring a piece of machinery for vibration, it is important to decide what points on the equipment to observe. In rotary equipment, most detectable problems occur in either the shafts or rotors; thus an obvious examination point is the bearing housing. The bearings are essential in the machine performance, especially in equipment that operates at high speed.

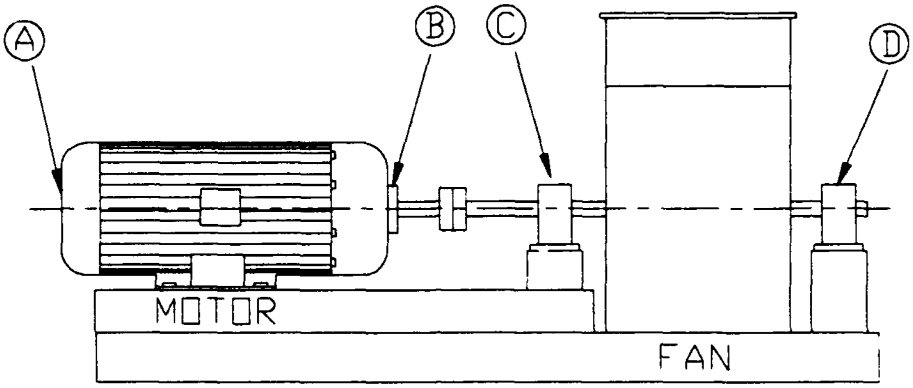
Faults in machinery detectable by routine analysis include:

1. Imbalance
2. Misalignment
3. Bent or bowed shafts
4. Bearing faults of any nature
5. Structural degradation
6. Aerodynamic-related effects
7. Various coupling problems
8. Drive-related problems.

All of these faults can be identified with properly collected data. Routine vibration data should be collected periodically from each bearing in the drive train, with measurements made in both the horizontal and vertical planes at each bearing. It is also recommended that at least one axial measurement per shaft be taken to provide pertinent information on the vibration status transmitted axially along the shaft.

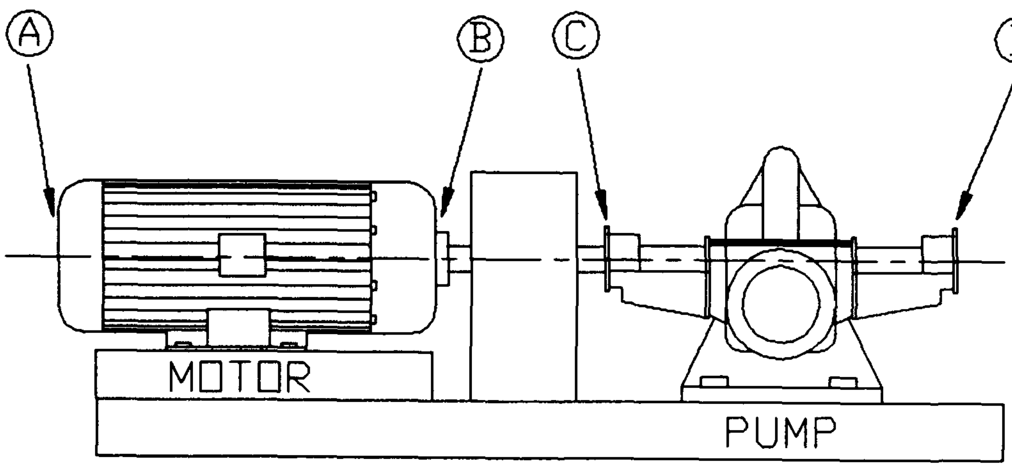
Figures 1 and 2 show typical monitoring points for the centrifugal pump and the centrifugal fan, respectively. Note that the particular points of interest are the bearings, and the location names assigned to these points. To ensure consistency in data and to improve the effectiveness of the analysis, vibration data must always be collected from the machinery at or near the same points.

Because of the large amount of data to be gathered, it is extremely important to take a systematic approach to data collection. For example, when monitoring a centrifugal pump, readings must be gathered at several locations on the pump. Always begin at a specific point on the equipment, such as the outboard



Ⓐ MOTOR OUTBOARD (Horizontal, Vertical, Axial)
Ⓑ MOTOR INBOARD (Horizontal, Vertical)
Ⓒ PUMP/FAN INBOARD (Horizontal, Vertical, Axial)
Ⓓ PUMP/FAN OUTBOARD (Horizontal, Vertical)

Figure 1. Typical Monitoring Points for the Centrifugal Pump.



Ⓐ MOTOR OUTBOARD (Horizontal, Vertical, and Axial)
Ⓑ MOTOR INBOARD (Horizontal, Vertical)
Ⓒ PUMP/FAN INBOARD (Horizontal, Vertical, and Axial)
Ⓓ PUMP/FAN OUTBOARD (Horizontal, Vertical)

Figure 2. Typical Monitoring Points for the Centrifugal Fan.

shaft bearing on the motor. Once the data is gathered, proceed to the next collection point. If monitoring the centrifugal pump, the next point should be the inboard shaft bearing of the motor. These collection points will be stored in the hand-held collector that will prompt for specific probe positions.

Labeling collection points is arbitrary, but should always help the user to easily identify relevant points on the equipment. For example, the user could always begin the data collecting at the motor end of the machinery. From there, points could be labeled along the machine using a consistent terminology such as inboard/outboard, or simply by using the alphabet or numbers. Labels should always follow a logical and systematic format.

Maintenance personnel generally follow a predetermined route each time data is collected from the plant equipment. This should correspond to the route information to be downloaded into the hand-held collector. To ensure consistent data, it is also a good idea to use the same vibration analyzer on the same route throughout each inspection (assuming that there are at least two units available for use).

Once the data has been collected, it may be analyzed. First, the data must be downloaded to a personal computer. This is done by simply linking the hand-held collector with the computer and running the appropriate download program that comes with the software package.

After the data has been transferred to the PC, it is transformed into the desired graphical option using diagnostic software. Generally, a velocity-versus-frequency graph is used for machine performance analysis.

Analyzing the data correctly is a critical step for ultimate success of the program. It doesn't take too many false calls to lose credibility in predicting failure. It is important for maintenance personnel to have confidence in the role of vibrational analysis in the predictive maintenance program. Inevitably, there will be some mistakes. For example, there could be an alarm signal that will lead to the equipment being disassembled, the part being replaced, and upon examination finding that there was no problem after all. Fortunately, as the program matures and maintenance personnel become more experienced, this will occur only rarely.

At the initial stage of the program, only a few machines should be analyzed. Machines that are critical to the plant's operation, expensive to repair, or have a poor maintenance history are typically good starting points. A good rule is not to start "too big" to avoid becoming buried in the vast amount of new (and, as yet unfamiliar) data collected from each machine. This will also help retain the credibility of the program.

Additional detailed information on data retrieval procedures is found in the literature obtained from the system manufacturers.

It should be noted that the vibration analysis equipment can also be used to detect damage in other parts of the machine train, for example, in the fluid transportation ducts, because many of the ducts in service today were installed without noise and vibration consideration. Appendix C gives a basic outline for reducing vibration levels in ducts. The user may find many applications for vibration testing and analysis.

3 SOFTWARE

The goal of a vibration software system is to process, trend, and analyze vibrational data, which are tasks that cannot otherwise be done quickly or conveniently, if at all. The primary functions of a software system are to:

- provide rapid transfer of data to long-term storage media
- download monitoring route and procedures for data collection
- allow comparison of data against some appropriate guideline
- possibly provide analysis using an expert system.

Most software analysis will use any of three methods of problem detection: overall analysis, broadband analysis, and narrowband analysis. With overall analysis, an alarm condition is indicated if the total sum of all the vibration peaks in the spectrum exceeds a preset value. An alarm condition is indicated in broadband analysis if the amplitude of any vibration peak within a prescribed frequency band exceeds a specific value. Narrow band analysis uses an "envelope" over the entire spectrum. This envelope typically is set at some marginal percentage higher than the local normal vibration amplitude, with a given frequency width around normal peaks. Any vibration signal that crosses this envelope, either on a horizontal boundary (increasing amplitude of a peak, or a vertical boundary (increasing frequency bandwidth of a peak), indicates an alarm condition.

There are two possible guidelines for the analysis process, those of "Industrial Standards" and "Baseline Data." Industrial standards are most often used in broadband analysis and overall analysis, while baseline data is most often used for purposes of narrowband analysis. Both methods can be used in any successful predictive maintenance program.

The industrial standards now in use result mainly from empirical data and years of experience with rotating equipment. This data has been compiled into severity charts (Figure 3), which relate the intensity of a vibration peak with the frequency at which the peak occurs. The location on the chart will indicate the severity of the vibration, from "very rough" to "very smooth." For example, if you measure a displacement amplitude of 0.2 mils* peak-to-peak at a frequency of 1200 CPM, by cross-referencing these two values on the severity chart, you will find that the machine is operating in the "smooth" range. The chart shows that the severity of vibration depends on the amount of displacement and the frequency of vibration. If a velocity reading showed the value 0.04 in./sec., the machine would be running in the "good" range, meaning that velocity is proportional to the severity, regardless of the displacement and frequency values. These can be used as an indicator of the condition of the machine. Note that applicability of any one of these charts is often limited to a particular machine or type of machine.

Baseline data is the vibration signature of a particular machine in good mechanical condition and under normal operation conditions. Future vibration measurements are then compared to this baseline data to determine if the condition of the machine is deteriorating. For the baseline data to be valid, the machine must be operating at the same load condition at the time of both data collections. This comparison, if correctly done, will often indicate potential problems early enough for repairs to be scheduled before further deterioration occurs.

Finally, some software packages have "expert" or "artificial intelligence" (AI) capability. These can combine analysis from broadband, overall, and even narrowband analysis techniques to determine the

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*1 mil = 0.001 in. = 0.0254 mm.

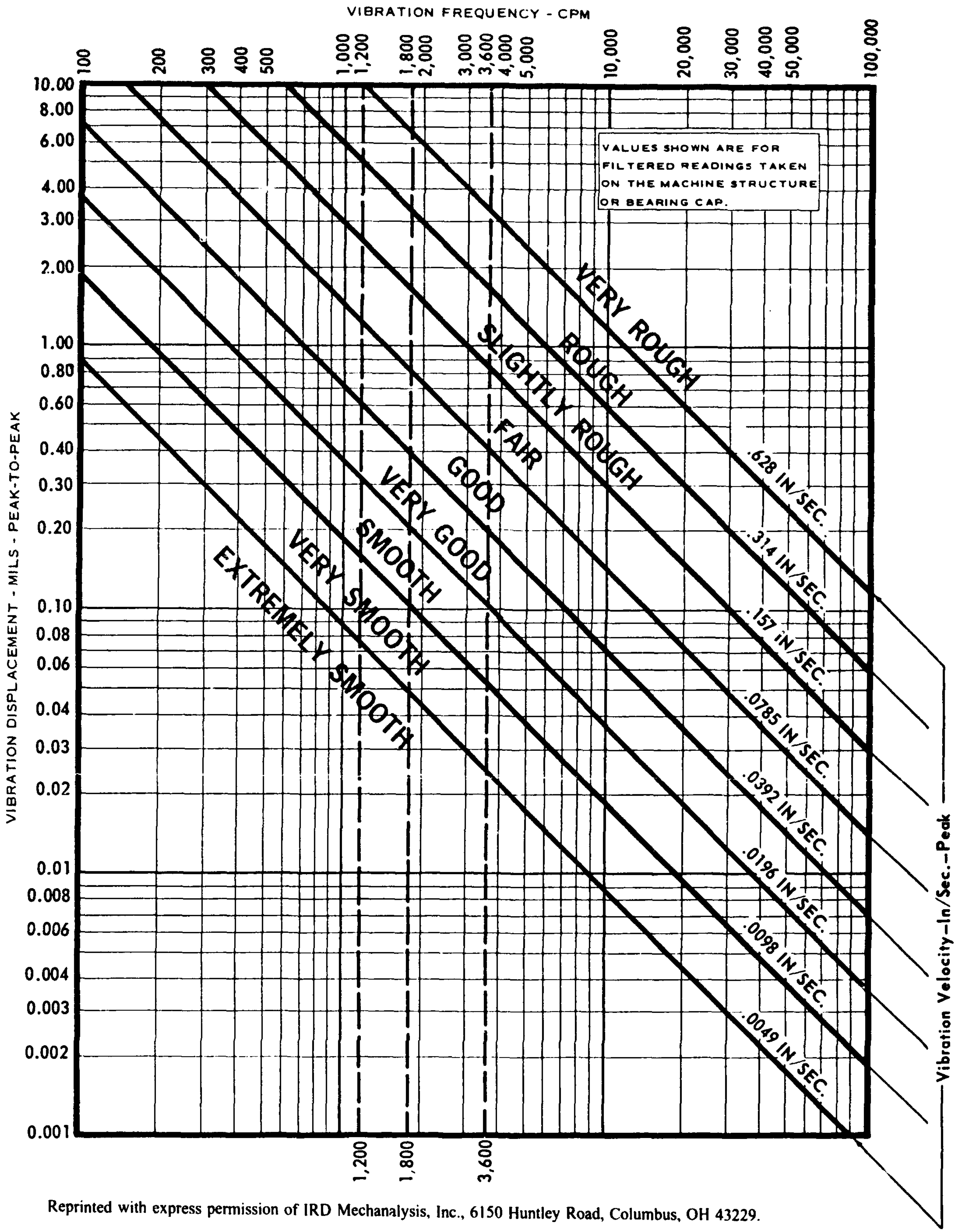

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Figure 3. Vibration Severity Chart.

condition of the machine and to report problems, their location, the probable cause, and the most likely solution. Expert systems may also serve as a tool to train employees on vibration analysis, since most systems explain why particular predictions were made. Principal drawbacks to expert software systems are that they tend to be much more expensive than the more basic versions, and that they may result in a reliance on the software rather than on human expertise and a good understanding of central energy plant operation.

4 DATA INTERPRETATION

Once a problem has been detected and vibration data has been collected, an analysis will reveal the problem's cause and solution. Since the highest amplitude vibration is normally associated with the trouble source in the machine, the cause of excessive vibration can be located by a process of elimination. Occasionally, however, a detected vibration may be transmitted from another machine via vibration coupling. What follows is a discussion of several common vibration sources and their particular characteristics. Appendix D contains further discussion on vibration coupling problems.

Unbalance

Unbalance is one of the most common sources of excessive vibration. Vibration caused by an unbalanced component (Figure 4) will occur at a component's running speed. The amplitude of the vibration is proportional to the degree of unbalance, and the direction is radial. Appendix E outlines how to correct balancing problems.

Misalignment

Misalignment (Figure 5) of a shaft coupling may be offset, angular, or a combination of both. Offset misalignment generates a vibration in the radial direction at a frequency equal to twice the shaft running speed. Angular misalignment generates an axial vibration at the shaft running speed. The amplitude of vibration is proportional to the degree of misalignment. Caution: misalignment-generated vibration patterns are often mistaken as symptoms of unbalance. Appendix F details specific techniques for aligning machines. Some common techniques for correcting misalignment are:

1. Laser-optic alignment
2. Strain gage alignment
3. Dial indicator alignment.

Eccentricity

Eccentricity refers to the condition where the center line of a rotor does not coincide with the center line of its shaft, or the center of rotation of a body does not coincide with its geometric center. Eccentricity generates a vibration at the component's running speed, and may be mistakenly diagnosed as unbalance. Figure 6 shows examples of eccentricity that can often cause vibrational problems.

Faulty Rolling Element Bearings

Flaws, defects, or damaged areas on raceways or rolling elements can cause high-frequency vibrations. This does not necessarily occur at multiples of the shaft speed, but may be similar to vibration induced by impact or by rubbing. Bearing faults may excite natural frequency vibrations in the bearing itself, the shaft, or the bearing housing. These vibrations usually occur at 10,000 to 100,000 cycles per minute (CPM). Figure 7 below shows a typical frequency plot that exhibits bad bearings.

Vibration analysis can also help determine the type of fault within the bearing itself. Appendix G briefly outlines how to determine the exact type of bearing flaw from vibration data.

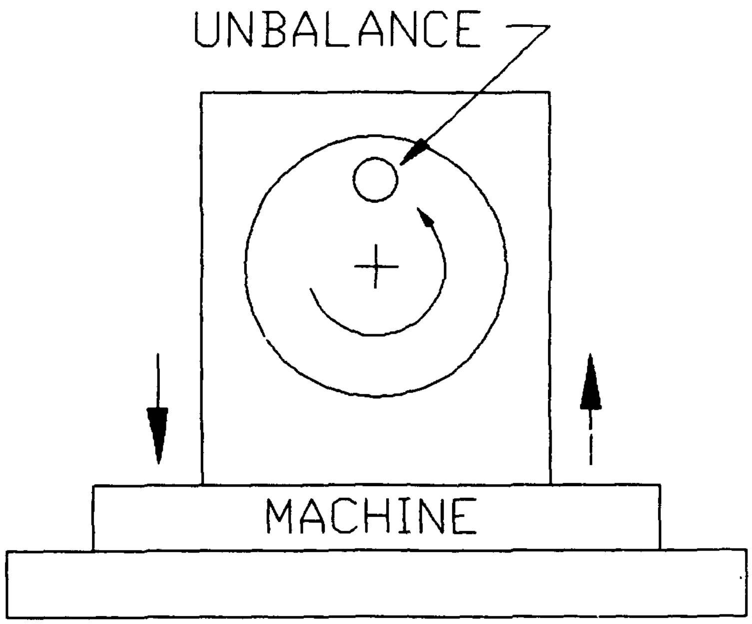

Figure 4. Examples of Bad Bearings.

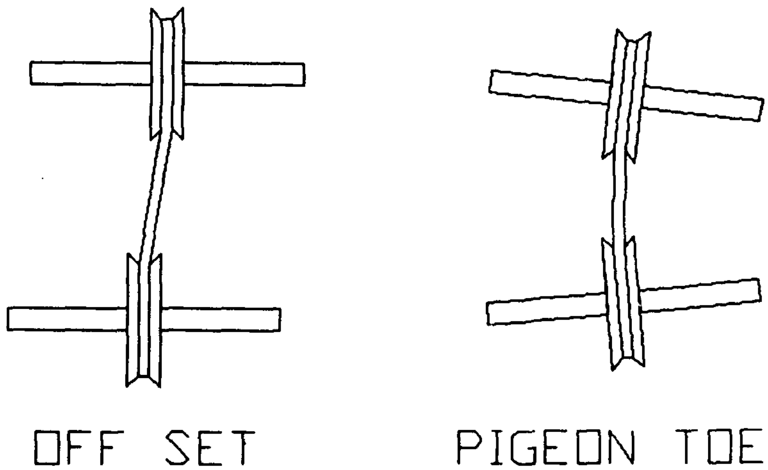

Figure 5. Examples of Eccentricity.

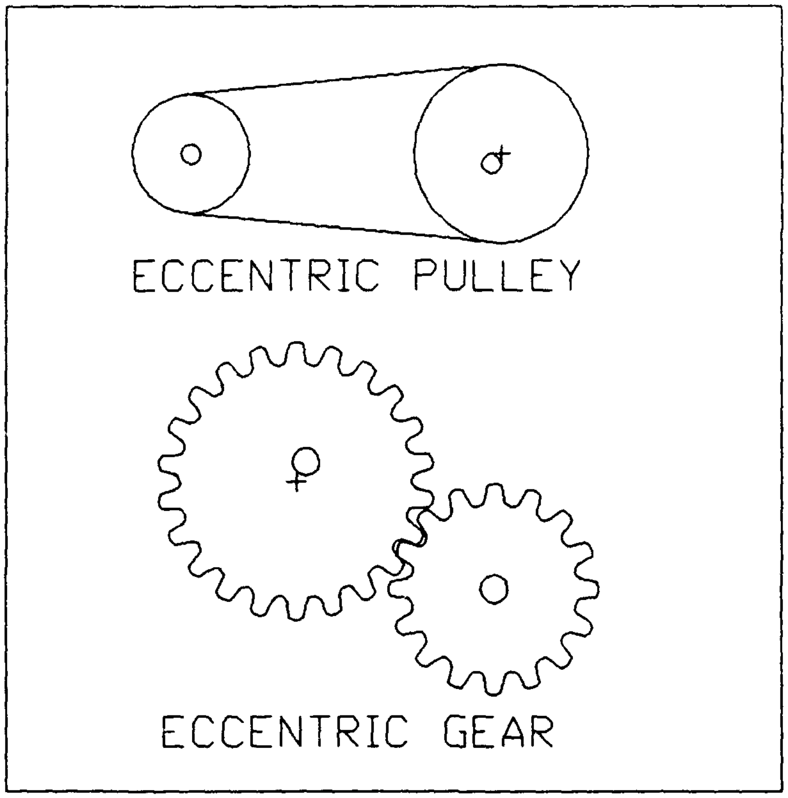

Figure 6. Examples of Misalignment.

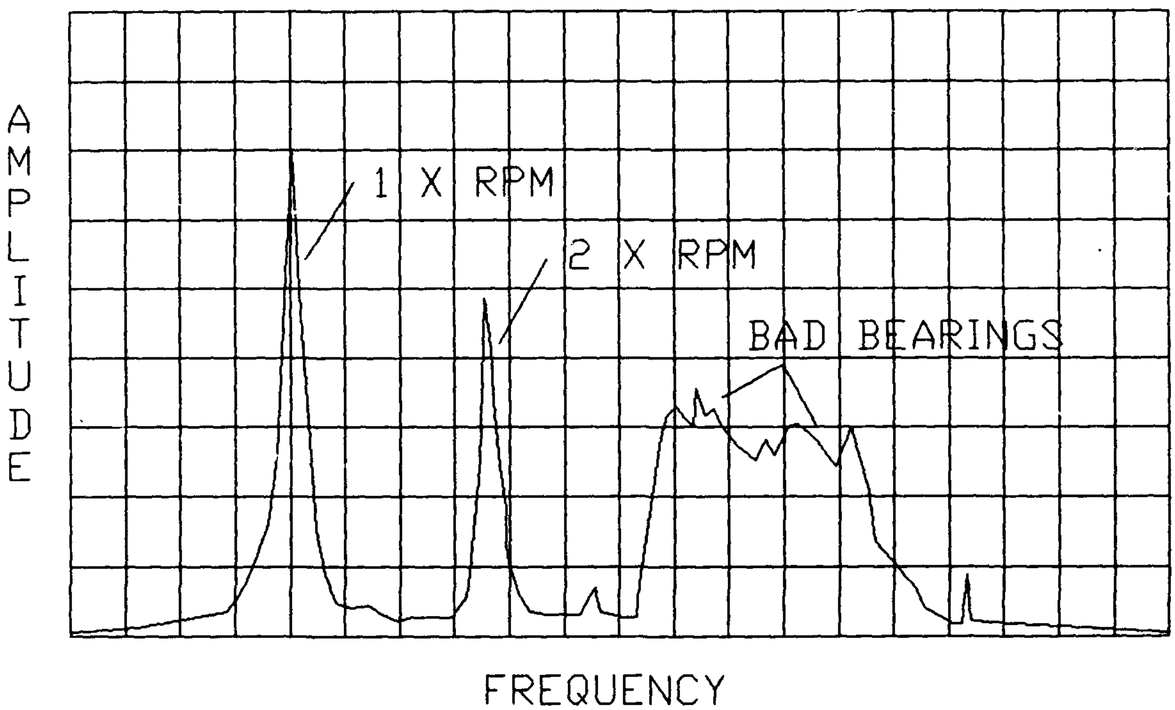

Figure 7. Examples of Unbalance.

Faulty Sleeve Bearings

Excessive vibration may be generated in sleeve bearings due to excessive bearing clearance, mechanical looseness or lubrication problems. Sleeve bearings with excessive clearance may allow minor unbalance or misalignment-induced vibrations to cause an impact type vibration at the bearing. The amplitude of this impact is proportional to the amount of clearance in the bearing, and therefore the vibration would be much less severe if bearing clearances were correct. Lubrication problems may include improper or insufficient lubricant, oil whirl, or hysteresis whirl. If insufficient oil or oil of the wrong viscosity is supplied to the bearing, then excessive friction may result in rubbing type vibration signals, and excessive wear of the bearing and shaft. Oil whirl may induce severe vibration in the machine, and occurs at just less than one-half the shaft speed (46 to 48 percent of shaft revolution per minute [rpm]). Hysteresis whirl occurs when a rotor is running between its first and second critical speeds, and the whirl is at the first critical speed of the rotor.

Mechanical Looseness

The pounding vibration associated with mechanical looseness typically occurs at a frequency two times the running speed of the component, but has also been observed at higher harmonics as well. This may result from loose bolts, worn bearings, or a cracked or broken mounting structure. The source of the vibration may be an imbalance, misalignment, or other vibration source, but in all such cases, looseness allows a small exciting force to result in a large amplitude vibration.

Drive Belts

Drive belt vibration may result from problems with the belts or from the reaction of the belts to other disturbing forces. Belt vibration may be caused by eccentric pulleys, mechanical looseness, misalignment or unbalance. All these factors will cause vibration in the belts, although the belts themselves are not the source of vibration. In these cases, the frequency of belt vibration is typically the same as the frequency of the disturbing force. Belt defects may include cracks, hard spots, soft spots, or areas of missing belt material. Vibration due to belt defects occur at integer multiples of the belt rpm.

Gear Problems

Many gear problems result in vibrations at the gear mesh frequency (the shaft speed times the number of teeth on the gear). These problems may include excessive gear wear, faulty lubrication, foreign materials in the gears, or tooth inaccuracies. Gear mesh frequency vibrations may also be caused by a misaligned or bent shaft. Eccentricity and unbalance cause vibrations at submultiples of the gear mesh frequencies. A problem such as one broken or deformed tooth, however, will result in a vibration at the shaft running speed.

Electrical Faults

Vibration from electrical problems result from unequal electromagnetic forces acting on the rotor. Unequal forces may be generated by any one or more of the following factors:

- eccentric armature journals
- open or shorted windings
- rotor not round
- shorted rotor iron
- unequal air gap (rotor and stator not concentric).

Electrical fault vibrations usually occur at the shaft running speed. Vibrations caused by electrical fault can be distinguished from those caused by unbalance by shutting off electrical power and monitoring vibration as the machine speed drops. Vibration due to unbalance will taper off as speed falls, but vibration due to electrical faults will disappear as soon as electrical power is interrupted.

Resonance

Resonance vibration occurs when a body is excited at its natural frequency. It is often impractical to eliminate the driving force or change its frequency. In these cases it may be necessary to change the natural frequency of the structure. This may be accomplished by stiffening (which raises the natural frequency) or by adding mass (which lowers the natural frequency).

Aerodynamic and Hydrodynamic Forces

Pumps, fans, and blowers are subject to the forces generated by the fluids that they move. These forces generate vibration at a frequency equal to the rpm times the number of blades or vanes on the impeller. This vibration is usually not severe, but may coincide with the natural frequency of the machine. This may result in excessive vibration and severe damage. In this instance, as with resonance vibration (above), the natural frequency of the machine should be changed.

Rubbing

Rubbing between a rotating and a stationary part generally results in vibration at integer multiples of the running speed. Continuous rubbing friction may also excite high frequency natural frequency vibration in other parts of the machine. Rubbing almost always results from a bent shaft or a broken or damaged component in the machine. These situations should be detectable as already described.

Beat

Beat is a pulsing vibration that can occur when two or more machines are close to each other and operate at slightly different running speeds. Typically, beat occurs at the difference in the running speeds (speed 1 less speed 2), but may also be detectable at the sum of the running speeds. The higher frequency is not usually detectable unless it coincides with a natural frequency of the mounting structure or machine.

Source Guide

Table 1 summarizes possible sources of vibration based on frequency only. It is not a complete listing, but does sufficiently illustrate the wide range of faults that can be detected with vibrational analysis. The guide is taken from *Practical Solutions to Machinery and Maintenance Vibration Problems* (course taught by Update International, Inc., 1979).

Table 1

Possible Frequency-Based Sources of Vibration

Cause*	Symptom/Description
Less than 1 x rpm	
Looseness such as bearing "wiggle" looseness in housing; other loose parts	Most often at exactly 1/2 x rpm. To be stimulated, it usually has to have enough vibratory force from another source such as from unbalance. Sometimes accompanied by harmonics above 1 x rpm such as 1/2 x; 2 x; 2 1/2 x, etc. Can also be seen as higher than usual harmonic amplitudes up to and beyond 5, 6, and 7 x rpm
Rub (with no resonance)	Exact fractions of 1 x rpm. Most usually 1/2 x rpm. Often accompanied by harmonics above 1 x rpm such as 1-1/2 x, 2 x 2-1/2 x, etc. Often shows larger than usual harmonics at 3 x, 4 x, 5 x—up to approximately 6 to 8 x rpm (depending on the intensity of the rub).
Oil whirl (journal bearing)	Slightly less than 1/2 x rpm such as 43 to 46 percent of 1 x rpm.
Rub, exciting a subharmonic resonance of shaft or motor	Exact fractions of 1 x rpm, such as 1/4 x rpm, 1/3 x rpm, 1/2 x rpm, 2/3 x rpm, 3/4 x rpm, etc. Seems to show up only when rotor resonance is near an exact fraction.
Rolling element bearing cage defect	Sometimes exhibited at less than 1 x rpm. Varies with the bearing design. Most cage defect-calculated frequencies in range of 35 to 46 percent of rpm. This nonsynchronous vibration could easily be confused with oil whirl frequencies (journal bearings). Often does not show at calculated frequency but instead at harmonics of that frequency.
Defective belts	If unbalanced belt, frequency would be at rotational frequency of the belt.
1 x rpm rotor speed	
Rotor unbalance	Large 1 x rpm peak with relatively low 2 x rpm and negligible other harmonic amplitudes.
Shaft/coupling misalignment	Check for other frequency symptoms such as higher than usual peaks at 2 x rpm and other lower harmonics. Check phase symptoms as well to more surely distinguish from unbalance.
Sheave misalignment	Often looks like unbalance symptoms of either the sheave or the rotor to which it is attached except that the axial amplitudes are usually much larger than for unbalance. Sometimes accompanied by larger than usual amplitude at 2 x rpm. May give symptoms similar to those of shaft/coupling misalignment.
Mismatched belts	Often accompanied by higher than usual axial vibration.

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Table 1 (Cont'd)

Cause*	Symptom/Description
Eccentric sheave	Large amplitude is usually unidirectional in line connecting center with other sheave. Approximately zero or 180 degrees out-of-phase relationship between vertical and horizontal readings at the same bearing.
Bent shaft	Similar to static unbalance except for 180 degrees out-of-phase axial readings. Could be resonant rotor (resonant whirl).
Eccentric armature or armature "running eccentrically"	Either machined eccentrically or made to "run eccentrically" due to shaft centerline orbits originating from unbalance or misalignment.
Eccentric gear (very rare) or gear "running eccentrically"	Rarely due to machining errors. Most often made to "run eccentrically" due to centerline orbits originating from unbalance or misalignment. Could be caused by assembly error. (See section on assembly errors.)
Motor with loose or otherwise defective rotor bars	Primary frequency is 1 x rpm. (With one or more sidebands on each side). the different frequency between a sideband and the primary frequency is equal to the rotor slip times the number of poles. Normally. its amplitude and amplitudes of sidebands are proportional to load. The symptom for this defect at times produces excessive vibration at relatively high frequency. Instead of a sharp peak at rpm x number of rotor bars, there are many sideband peaks, producing what is referred to as a "haystack." the different frequency between sidebands is usually equal to the electrical hum frequency
Lower harmonics	Up to approximately 10 x rpm or number of fan blades x rpm (whichever greater). Does <u>not</u> include "Higher Frequencies" described later. Some frequencies in this range <u>not</u> true harmonics, as they are nonsynchronous with 1 x rpm (such as electrical hum and bearing frequency). Usually not high amplitudes in absolute terms. Important to review section "Evaluating harmonics for complete analysis to determine vibration source."
Shaft/coupling misalignment	Primary vibration is still most often (but not always) 1 x rpm. Amplitude at 2 x rpm is larger than usual. Misalignment suspected when 2 x rpm is above 1/3 the 1 x rpm amplitude. When higher 2 x rpm amplitude exceeds 1/2 the amplitude of 1 x rpm, the symptom is much stronger. Symptoms are very strong when other lower harmonic amplitudes are also "higher than usual." If coupling has segment amplitudes at rpm x # of segments in one coupling, half would also be much larger than usual. Always compare with phase symptoms.
Coupling binding	Misalignment symptoms increase even though the misalignment itself may not have changed. Instead, the so-called flexible coupling becomes less effective due to excessive wear or improper lubrication.
Number of pump vanes x rpm	Very often accompanied by resonance. If not resonance, check for pump with too a low flow rate. Sometime new product design problem.
Number of blades x rpm	Most often a resonance problem.
Loose base, "wiggle," loose bearings, loose part	2 x rpm, but sometimes shows up at 1/2 x rpm bearings instead. Also see looseness at "less than 1 x rpm."

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Table 1 (Cont'd)

Cause*	Symptom/Description
Rolling element bearing with too much clearance between inner race and outer race	Symptoms similar to those of looseness. Most often exact harmonic frequencies. Rollers sliding too much. See section on bearing vibrations.
Unbalanced reciprocating parts, such as pistons, connecting rods, vibrating screens	Replacement parts should be weight-balanced to previous weight. All like parts are to weigh the same. Reciprocating screens require proper counter-weighting.
Electrical hum	Often looks like harmonic, but is nonsynchronous and, therefore, not a true harmonic. For 60 Hz current, frequency is 120 Hz (7200 cpm). For 50 Hz current, frequency is 100 Hz (6000 cpm).
Rub or looseness	Could produce higher amplitudes than usual fr several lower harmonics up to and beyond 5, 6, and 7 x rpm. Sometimes reported at frequencies of 1-1/2 x rpm, 2-1/2 x; 3-1/2 x; etc. Sometimes in the series a harmonic or two are skipped. Orbit or displacement ot time waveform analysis very helpful to distinguish between rub and other sources.
Misaligned rolls	Paper machine rolls may be aligned within tolerance and yet exhibit high 2 x rpm amplitude due to the magnification of resonance. compare with time waveform.
Higher frequencies	
Defective or worn gear teeth	Gearmesh of number of teeth x rpm and/or harmonics of that frequency. Often there is no defect or excessive wear, but instead vibration at that frequency could be grossly magnified by resonance. Best seen via time waveform.
Gear eccentricity and gear teeth are OK	High gearmesh amplitude and its harmonics. Center line has large orbit due to unbalance or misalignment or other lower frequency-related error, causing teeth to mesh improperly. Also check for resonance.
Defective roller element bearings	Symptoms are in stages depending on how close to failure. Fist stage is at relatively high frequencies, nonsynchronous. Frequencies spread out and create more peaks as bearing deteriorates. Later stage shows frequencies as calculated. Number of sidebands of 1 x rpm increases. In all situations, amplitudes may or may not increase.
Turbine blades	Frequency of number of turbine blades x rpm. This one usually requires a turbine specialist. Sometimes due to other turbine problems not readily put into simple chart forms. If in doubt, contact turbine manufacturer.
Rub	Similar to previous rub symptoms, but producing higher frequencies. (See rub symptoms at lower frequencies.)
Motor with loose or otherwise defective rotor bars	Sometimes the symptom for this defect is excessive vibration at relatively high frequency. Instead of a sharp peak at rpm x number of rotor bars, there are many sideband peaks, producing what is usually referred to as a "haystack." The difference frequency between sidebands is usually equal to the electrical hum frequency. Also check for primary frequency that is often at 1 x rpm (with a sideband on either side of it). The difference frequency between a sideband and a primary frequency is equal to the rotor slip times the number of poles. Normally, sideband amplitudes are proportional to the load.

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Table 1 (Cont'd)

Cause*	Symptom/Description
Miscellaneous	
Beats	Beat frequencies, which are usually equal to the difference between two nearly equal frequencies, could cause very low frequency vibrations. Often the frequencies are so low that they are below the ability of some pickups and instruments. The source frequencies will show periodic, rhythmically timed increases and decreases in amplitude. May be helpful to use time waveform analysis or synchronous-time-averaging. A symptom that may look like a beat but originates from only one source is gearmesh vibration, resulting from a gear that has been machined eccentrically or that is "running eccentrically" due to the orbit of its centerline, due to unbalance or misalignment.
Cavitation	Cavitation produces much higher than usual amplitudes at almost all frequencies throughout the whole spectrum. Although it is usually the result of imploding condensing steam bubbles, visualize the type of frequencies that would be created by very rapidly boiling water. Usually amplitudes are relatively small, but crackling noise is high.

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5 VIBRATION DATA

Vibration Data From Centrifugal Fans and Pumps

This section groups the analysis of centrifugal fans and pumps together. Since from a vibrational standpoint, the operating parameters of the fan and pump are very similar, their data can be analyzed in much the same way.

Total unit failure in centrifugal fans and pumps usually occurs from operating under faulty conditions for several months. If the fault condition is severe, however, failure can happen in less than a month. Therefore, it is important to frequently monitor equipment that is critical to the operation of the facility. It may become necessary to survey equipment that exhibits vibration at more frequent intervals. Such decisions are usually based on safety and economic ramifications.

Analyzing the vibration data taken from a centrifugal fan or pump will show many characteristics similar to those exhibited in almost any form of rotating equipment. Oil whirl, for example, is common in pressure-fed bearings, and will usually appear at approximately 44 percent of the shaft rpm, and will decrease as the speed of the shafts slows down.

Imbalance and misalignment will typically appear at the running speed frequency. For example, consider the imbalance problem that can occur in the coupling of two shafts. If the coupling is keyed to the shaft, it is required that some of the shaft material be slotted to accept the key. If the amount of shaft material removed is not equal to the amount contained in the key stock, there will be a balance problem since one side of the shaft will be weighted. This will cause a force that will be generated with each rotation of the shaft. Therefore, a vibration frequency will appear at the running speed of the shaft.

Blade pass vibration is an aerodynamically induced effect. As the blades rotate, they produce a force on the surrounding fluid that is transmitted through the volute. Since the rotating shaft has blades extending from its center, these blades will rotate at a frequency equal to the number of blades multiplied by the shaft rpm. Therefore, a spike will appear at this frequency. This is considered normal and, therefore, shouldn't alarm the user, unless something such as structural changes or obstructions compound the phenomenon.

Vibration Data From Steam Turbines

Many factors can cause abnormal vibration in a steam turbine. Misalignment of the turbine output shaft coupling is often a source of excessive vibration, as is imbalance in the turbine rotor. Bearing problems, including oil whirl, are also possible sources of vibration. Vibration can even result from misalignment when the coupling is aligned "within the manufacturers specifications." Misalignment of the shafts may be offset, angular, or both. Angular misalignment (when the two shafts meet at an angle, rather than parallel) will usually result in a vibration peak in the axial direction at a frequency corresponding to the turbine rpm. This may be accompanied by peaks at multiples of the turbine speed (two or three times the rpm). Offset misalignment (when the shafts are parallel but their center lines do not coincide) produces a characteristic vibration at twice the operating speed. Also, the highest vibration will usually occur in the direction of the offset (i.e., if the shafts are vertically offset, the vibration will be primarily in the vertical direction).

Imbalance of a turbine rotor means that the center of mass of the rotor is not exactly at the center of rotation of the rotor. Imbalance will result in vibration in the direction perpendicular to the shaft, occurring only at a frequency equal to the shaft rpm. Imbalance will typically not produce the axial direction vibration characteristic of misaligned shaft couplings. Problems with anti-friction (ball or roller) bearings can result from misalignment of the bearings, or from mechanical problems with the bearings themselves. Vibration from misaligned anti-friction bearings is usually detected in the axial direction, and at a frequency corresponding to 1, 2, 3, or the number of balls or rollers, times the rpm of the shaft. Physical damage to a bearing will result in vibration signatures that depend on the bearing size and configuration, and on the shaft speed.

Journal bearing vibration is usually associated with oil whirl, or excessive clearance or wear. Oil whirl occurs when viscous forces in the bearing push the shaft around within the bearing. Oil whirl vibration usually occurs at less than one-half the rpm and only in pressure-lubricated type bearings. Only bearings in which the shaft is not perfectly centered in the bearing are susceptible to oil whirl. Several journal bearing designs exist that counter these effects: the axial grooved bearing; the lobed bearing; and the tilted pad bearing. Excessive wear will usually result in vibration at multiples (one, two or higher) of the rpm. Tilted pad bearings, however, will vibrate at the number of pads multiplied by the rpm.

Period of Vibration Surveillance

Several important factors are involved in deciding how frequently equipment should be monitored. The most important factor is the rpm at which the machine operates:

- If the machine operates under 500 rpm (slow speed), analysis should be performed every 4 to 6 months.
- If the machine operates between 500 and 3000 rpm, analysis should be performed every month.
- If the machine operates above 3000 rpm, analysis should be performed every 2 weeks.

Several factors will alter the monitoring period. For example, if the machine is critical to the operation of the facility, then the machine should probably be monitored more frequently than less critical machinery. Keeping vital equipment on-line will ensure the success of the program.

The monitoring period should be increased for machines with a history of frequent failures. Ideally, the problem of frequent failures should be amended by correcting the design flaws in the equipment. Until this is accomplished, the program should target poorly performing equipment.

Another consideration should be safety. If degradation to certain types of equipment threatens the plant safety, frequent analysis of the performance of that type of equipment is indicated. Operating experience will most likely dictate how often the analysis should be performed.

8 AVAILABLE HARDWARE

After reviewing various programs and the different available hardware, three generic vendors of vibration equipment and services were selected. Other vendors supply this type of equipment in the industry, but these were selected for their good service reputation. These three representative systems have similarities, but typify different approaches to setting up a predictive maintenance program. A description of these systems will help different facilities choose an appropriate predictive maintenance plan for their individual needs. Appendix B lists vibration analysis hardware and software vendors.

Company One

Company one sells portable vibration data collectors and the software needed to set up a predictive maintenance program. This company uses a bar code system to identify each data point throughout the data collecting route. When the bar code is scanned, then the data collector automatically knows what information should be collected at this point. Also, the portable collector retrieves data with a triaxial probe fastened into a base that has been previously attached to the machine with an industrial adhesive.

The program has been set up to take the best repeatable data possible without needing a trained technician. The collector is not designed for field analysis. (The screen only prompts the technician to move on to the next point.) This collector does not have any type of controls to prevent the operator from making mistakes. Some of the advantages of this system are:

- It generates repeatable data.
- It does not require a trained technician to collect the data.
- It takes less time at each machine with triaxial pickup than other methods.
- It reduces operator error since there are no controls on the hand-held data collector.
- Its software is compatible with other companies' hardware.
- Its software is IBM-compatible.

Some of the disadvantages of the system are:

- Bar code can eventually become illegible (from equipment repair, paint, soil, etc.).
- Pads and bar codes need to be replaced each time a machine is replaced.
- The system cannot perform field analysis.
- The system cannot be used for balancing.
- The system does not come with free setup instructions to assist each individual facility.

For this company it is suggested that two different avenues for building a predictive maintenance program could be taken. These two approaches incorporate the same data collector, but use different software packages to evaluate the data. The first package will consist of the basic alarms that will notify the technician when the machine exceeds the set alarm levels. This software will need the technician to analyze the data and make the correct diagnosis. The second package will be an expert system that will analyze the data and predict the machine faults. Tables 2 and 3 itemize the costs for basic and expert systems setup, respectively, for company one.

The price listing for the listed equipment is subject to change as the equipment is upgraded. Also, a facility that would like to go from the basic system to the expert system will only have to pay the approximate difference in the software price to upgrade. Therefore it would be advisable to start with the basic system and then upgrade to the expert system if the facility chooses to do so.

Table 2

Basic System Setup Costs (Company One)

System Elements	Cost
Data collector analysis system	
Tri-Axial System: includes Data Collector, 50 Barcode Labels, 50 Attachment Pads, and Basic Software.	$9,950.00
Computer hardware and software	
Computer hardware 80386 Computer, 80 meg hard-drive, VGA monitor, and printer	$3,000.00
Basic software	$6,000.00
Technical support	
3-days of technical support ($750/day)	$2,250.00
Approximate expenses for technical support	$900.00
Contingencies	
Extra training, bar codes, pads, tachometer and etc	$2,000.00
Total	$24,100.00

Table 3

Expert System Setup Costs (Company One)

System Element	Cost
Data collector analysis system	
Tri-axial system: includes data collector, 50 barcode labe.., 50 Attachment pads, and basic software.	$9,950.00
Computer hardware and software	
Computer hardware: 80386 computer, 80 meg hard-drive, VGA monitor, and printer	$3,000.00
Expert system software	$10,000.00
Technical support	
3-days of technical support ($750/day)	$2,250.00
Expenses for travel	$900.00
Contingencies	
Extra training, bar codes, pads, tachometer, etc.	$3,000.00
Total	$29,100.00

Company Two

Company two also sells portable vibration data collectors and the software to set up a predictive maintenance program. However, company two has developed a hand-held data collector that combines the tasks of collecting and analyzing data automatically. This allows an untrained technician without any knowledge of vibration analysis to take data on a predetermined route. If problems are found in the equipment when the data is downloaded into the computer, then a trained technician can use the data collector to do a field analysis. This allows the untrained technician to collect extra data at different points on the individual pieces of equipment, which frees the trained technician to make a better diagnosis of the machine's faults.

The best feature about company two's Predictive Maintenance program is that it offers a considerable amount of training with the purchase of its hardware and software. Not only is training offered at the initial setup, but classes are also offered throughout the United States that can be attended free of charge by facility personnel as long as they own the equipment. (Note that this is only the basic setup class.) This is a significant advantage since the technology of the hardware and the software is changing rapidly.

Some advantages of this system are:

- It does not require a trained technician to collect the data.
- It collects repeatable data.
- It can perform field analysis.
- Its data collector options are expandable.
- Its software is IBM compatible.
- Its software is compatible with other companies' hardware.
- Upgrade the basic data collector, many different accessories are available.

Some of the disadvantages of this system are:

- It has no standard triaxial pickup, and thus requires a longer time to collect data than other systems. (A pickup may be purchased separately.)

- Its controls are located on the hand-held data collector, increasing the chance of operator error. (Can obtain push button transducer, thus decreasing chance of operator error.)

For this company, there are two suggested ways to create a predictive maintenance program. Both approaches use the same data collector, but each uses a different software package to evaluate the data. The first package consists of the basic alarms that will notify the technician when the machine exceeds the set alarm levels. This software will need the technician to analyze the data and make the correct diagnosis. The second package is an expert system that will analyze the data and predict the machine faults. Tables 4 and 5 itemize the costs for basic and expert systems setup, respectively, for company two.

Table 4

Basic System Setup Costs (Company Two)

System Elements	Cost
Data collector analysis system and software	
Data analyzer/collector, 3 days of on site technical support, and basic software	$13,680.00
Computer hardware	
80386 computer, 80 meg hard-drive, VGA monitor, and printer	$3,000.00
Contingencies	
Extra training, tachometer, balancing option and etc.	$2,000.00
Total	$18,680.00

Table 5

Expert System Setup Costs (Company Two)

System Elements	Cost
Data collector analysis system and software	
Data analyzer/collector, 3 days of on site technical support, and expert system software	$25,680.00
Computer hardware	
80386 computer, 80 meg hard-drive, VGA monitor, and printer	$3,000.00
Contingencies	
Extra training, tachometer, balancing option and etc.	$3,000.00
Total	$31,680.00

Company Three

Company three also sells portable vibration data collectors and the software needed to set up a predictive maintenance program. A trained technician can use company three's data analyzer to do an in-depth analysis of a machine in the field, or an untrained technician can use the same analyzer to run predetermined routes to collect readings on many machines and then to download the data into the computer for analysis. Options can be purchased with the data collector to enhance the predictive maintenance program.

Company three has taken a different approach to the software than either of the other two recommended companies. The software is developed by company three and is not compatible with any other companies data collector, but the software is comparable to the software offered by the other two companies. The only drawback to purchasing this system is that, if a facility changes to another data collector in the future, it would lose the database.

Some of the advantages of this system are:

- It does not require a trained technician to collect the data.
- It collects repeatable data.
- It can perform field analysis.
- The features of the data collector are expandable.
- Its software is IBM-compatible.
- Many different accessories are available to upgrade the basic data collector.

Some of the disadvantages are listed below:

- It has no standard triaxial pickup, and thus requires a longer time to collect data than other systems. (A pickup may be purchased separately.)

- Its controls are located on the handheld data collector, increasing the chance of operator error. (The collector may be turned into a "dumb box" by limiting operator's access to different functions or setups, thus reducing the chance of operator error.)

- Its software is not compatible with other companies' software or data collectors.

For this company, there are also two suggested ways to develop a predictive maintenance program. Both use the same data collector, but different software packages to evaluate the data. The first package will consist of the basic alarms that will notify the technician when the machine exceeds the set alarm levels. This software will need the technician to analyze the data and make the correct diagnosis. The second package will be an expert system that will analyze the data and predict the machine faults. Tables 6 and 7 itemize the costs for basic and expert systems setup, respectively, for company three.

Table 6

Basic System Setup Costs (Company Three)

System Elements	Cost
Data collector analysis system	
Data analyzer, 832K memory, and standard accessories	$8,995.00
Computer hardware and software	
80386 computer, 80 meg hard-drive, VGA monitor, and printer	$3,000.00
Basic software	$4,995.00
Technical support	
3 days of setup technical support	$2,180.00
Expenses for travel	$900.00
Contingencies	
Extra training, tachometer, balancing option, etc	$2,000.00
Total	$22,070.00

9 ECONOMIC ANALYSIS

Beginning a new predictive maintenance program can be a sizeable investment, but proper program management can offset the initial investment with continued annual savings. This has been repeatedly shown throughout industry, especially in the past 10 years.

Several previous studies illustrate the feasibility of investing in a predictive maintenance program. One of these studies (*Machinery Vibration Analysis* 1990) showed that the cost savings of a predictive maintenance program versus a run-to-failure program will save approximately $10/year/hp. For a typical plant facility, the payback period for the hardware, training, and software is approximately 1 year. (It should be noted that the success of the program will increase each year. As the technicians gain experience from the program, they can more accurately predict component failure.) Another study has (Master Trend Technology I) showed that the well-managed program will yield $5 of savings to every $1 spent on the program annually. (Note that these figures did not include the salaries of the technicians because each facility already has maintenance personnel that can head up the predictive maintenance program without extra labor cost.)

Tables 8 and 9 list the savings gained by using a maintenance program rather than a run-to-failure strategy, for typical HVAC plants (a 75,000 lb/hr boiler plant, and a 4000 ton chiller plant). Note that the horsepower values are totals, and that motors varied from 15 to 100 Hp.

Several case histories serve to show the economic impact of a good vibration program (*Vibration Technology-1*, IRD Mechanalysis, 1988, pp 1-10).

Table 7

Expert System Setup Costs (Company Three)

System Elements	Cost
Data collector analysis system	
Data analyzer, 832K memory, and standard accessories	$8,995.00
Computer hardware and software	
80386 computer, 80 meg hard-drive, VGA monitor, and printer	$3,000.00
Expert system software	$14,495.00
Technical support	
3 days of setup technical support	$2,180.00
Expenses for travel	$900.00
Contingencies	
Extra training, tachometer, balancing option, etc	$3,000.00
Total	$32,570.00

CASE 1

Routine vibration checks on critical machinery in a large gas processing plant revealed excessive vibration on a centrifugal air compressor. A follow-up analysis of the machine's vibration showed a misalignment between the compressor and speed increaser. In addition, a high frequency vibration on the gear box indicated that the bull gear of the speed increaser was damaged. Later, visual inspection revealed a fine crack in the gear.

Because a replacement gear was not available at the time, containing holes were drilled at both ends of the crack to keep it from extending farther. After a realignment of the unit, the equipment was put back into full service until a new bull gear could be fabricated.

When the replacement gear was ready, the unit was shut down just long enough to replace the gear. Although this took four days, a total stress failure could have caused a 6-month shutdown.

CASE 2

An overhaul had just been completed on three identical gear-driven centrifugal chiller compressors that were used to air-condition a large office building. Shortly after start-up of the three units, a disturbing, high-pitched noise developed. Soon the noise level became so intense it could be heard distinctly throughout the building. The building owner refused payment for the overhaul until the noise problem was corrected.

Analysis of the noise and vibration quickly traced the problem to a gear box on one of the units. Even though an earlier visual inspection revealed that the gears were satisfactory, a vibration analysis of the noise source and frequency disclosed extremely high levels of vibration at the output bearing of the gear box.

This fact, together with the noise and vibration frequency, indicated that the sleeve-type bearing was responsible for the problem. Mechanical measurements showed that the bearing clearance exceeded specifications by several mils. Replacing the bearing with one with specified clearance eliminated the disturbing vibration noise.

CASE 3

This incident involved a 7000 rpm ammonia compressor used in the production of fertilizer. Periodic checks revealed increasing vibration on the turbo-expander used to drive the compressor. A vibration analysis traced the problem to a faulty coupling between the expander and compressor.

By this time, the level of vibration was sufficiently high to warrant immediate shutdown. However, production demands were such that a shutdown was not possible. Five days after the diagnosis was made, the suspected coupling failed.

Although a replacement coupling was on-hand, the forced failure of the coupling plus continued operation at excessive levels of vibration, resulted in extensive damage to shafts and bearings. Repairs took four days, working around the clock.

Had the coupling been replaced at the time recommended, extensive damage might have been avoided, and the total downtime reduced to about 8 hours.

Table 8

Savings Gained From Predictive Maintenance
(5000 lb/hr Boiler Plant)

Machine	Total HP	Savings
Forced draft fan	50	550.00
Feed water pumps	50	660.00
Condensate return	50	220.00
Total		**$1,430.00/yr**

Note: The boiler plant consisted of three different boilers.

Table 9

Savings Gained From Predictive Maintenance
(4000 Ton Chiller Plant)

Machine	Total HP	Savings
Chillers	2587	5,700.00
Plant loop pumps	195	2,145.00
Chilled water pumps	165	1,815.00
Cooling tower pumps	300	3,300.00
Total		**$12,960.00/yr**

Note: The chiller plant consisted of: one 500-ton, two 1000-ton, and one 1500-ton chillers.

These case studies show how an established predictive maintenance program can yield many cost benefits, including:

- preventing catastrophic equipment failures
- increasing equipment on-line reliability
- effectively scheduling maintenance downtime
- performing maintenance only when required
- prioritizing and scheduling maintenance based on need
- reducing costs by eliminating the need for emergency repairs
- ensuring new equipment meets specifications at start up
- ensuring all problems are corrected during warranty period
- reducing spare parts inventory.

10 CONCLUSIONS

This study has given an overview of vibration monitoring as a predictive maintenance tool for use in U.S. Army central heating plants, and has documented the requirements for developing a predictive maintenance program for power house equipment. Three pieces of equipment (fan, pump [Chapter 5], and turbine [Chapter 6]) were specific plant equipment selected for detail application study. Three commercial systems were analyzed in detail for use in these applications (Chapter 3).

This study concludes that vibrational monitoring is a vital component of any good predictive maintenance program. Predictive maintenance should be gradually integrated into the current maintenance program in five steps:

1. Choosing personnel
2. Selecting a program type
3. Purchasing the equipment
4. Providing office space
5. Choosing a route.

Vibrational analysis provides a powerful tool to locate problems that cause abnormal machine vibration, and to prevent damage and catastrophic failure by correcting vibration-related problems before failure occurs. A predictive maintenance program that uses vibrational monitoring allows repairs to be scheduled during regular maintenance rather than during costly unscheduled breakdowns, thus yielding significant annual savings in maintenance costs.

REFERENCES

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IRD Mechanalysis, *Vibration Technology-1* (IRD Mechanalysis, Inc., 1988).

James, R. *P/PM Technology* (P/PM Technology, 1990).

Spectral Dynamics, *Machine Vibration Analysis* (1990).

Thomson, W.T. *Theory of Vibration With Applications* (Prentice-Hall, Inc., 1972).

APPENDIX A: Training Seminars

Update International, Inc.
2103 So. Wadsworth Blvd.
Lakewood, CO 80227
Phone 303/986-6761
Fax 303/985-3950

Scientific - Atlanta
Spectral Dynamics Division
P.O. Box 23575
San Diego, CA 92123-0575
Phone 619/268-7100

DLI Engineering Corporation
ATTN: Mark Libby
253 Winslow Way West
Bainbridge Island, WA 98110
Phone 206/842-7656

IRD Mechanalysis - USA
ATTN: Larry Witker
6150 Huntley Road
Columbus, OH 43229
Phone 614/885-5376
Fax 614/885-7668

CSI
835 Innovation Drive
ATTN: Robert Hodge
Knoxville, TN 37932
Phone 615/675-2110
Fax 615/675-3100

The Vibration Institute
6262 South Kingery Highway, Suite 212
Willowbrook, IL 60514
Phone 708/654-2254
Fax 708/654-2271

APPENDIX B:

Vibration Analysis Vendors

Vendor	Address	Phone	Vibration Monitoring Equipment
Beta Monitors & Controls, Ltd.	200, 1615 - 10 Ave. S.W. Calgary, Alberta, Canada T3C 0J7	403/245-5300 800/661-9160	Data-Trap 9000
Bently Nevada	Bently Nevada Corporation 1617 Water St., P.O. Box 157 Minden, NV 89423	702/782-3611	Trendmaster 2000
Bruel & Kjaer	Bruel & Kjaer Instruments, Inc., 185 Forest Street Marlborough, MA 01752	508/481-7000	Models 2513, 2515
Computational Systems, Inc. CSI)	CSI 835 Innovation Drive Knoxville, TN 379[illegible]	615/675-2110	Models 1900, 2100, 2115 and 2400
Data Signal Systems, Inc.	Data Signal Systems, Inc. P.O. Box 1608 Friendswood, TX 77546	713/482-9653	Model 9855
DLI Engineering Corporation	DLI Engineering Corp 253 Winslow Way West Bainbridge Island, WA 98110	206/842-7656	Watchman 8603, DC-7
IRD Mechanalysis	IRD Mechanalysis 6150 Huntley Road Columbus, OH 43229	614/885-5376	Model 890
Ono Sokki Technology, Inc.	Ono Sokki Technology, Inc. 2171 Executive Drive, Suite 400, Addison, IL 60101	708/627-9700	Models AF 550, CF-250 CF-350, CF-360, CF-880
PCB Piezoelectronics, Inc.	PCB Piezoelectronics, Inc. 3425 Walden Avenue Depew, NY 14043-2495	716/684-0001	Models 396B, 596A
PMC/Beta Corporation	PMC/Beta Corporation 4 Tech Circle Natick, MA 01760	617/237-6920	Models 201, 204, and 3000
Schenck Corporation	Schneck Corporation 535 Acorn Street Deer Park, NY 11729	516/242-4010	Vibroport 30 Vibrocam 1000
SKF Condition Monitoring	SKF Condition Monitoring 4141 Ruffin Road San Diego, CA 92723	619/496-3400	Microlog Models 10, 40
Technology for Energy Corporation	Technology for Energy Corporation, P.O. Box 22996 Knoxville, TN 37933-0996	615/966-5856	Model 1330 Smartmeter Plus, Sentry Series
VCI	VCI 5733 South Dale Mabry Hwy. Tampa, FL 33611	813/839-2826	Tektronix 2630 Real Time Analyzer
Vitec	Vitec 23600 Mercantile Road Cleveland, OH 44122	216/464-4670	Mark-400 FFT analyzer
Walker Associates	Walker Associates P.O. Box 58224 Webster, TX 77598	409/925-3074	Vibrasense M, Model 200

APPENDIX C: Vibration Reduction in Ducts

A large amount of vibration in fluid transportation ducts usually does not imply impending duct failure. However, if the vibration is allowed to continue at excessive levels for extended periods of time, then ducts composed of multiple sections may fail at the section fasteners (i.e., rivets, bolts, welds, etc.). Furthermore, excessive vibration can cause intolerable levels of noise in the work area. Presented below is a basic outline of steps to reduce duct vibration.

The easiest way to reduce duct vibration in rectangular ducts is to increase the stiffness of the duct. This is accomplished by adding cross-supports along the sides of the duct (Figure C1), or by connecting the duct to ground at more frequent intervals. The addition of expansion joints may also help reduce the level of vibration. Often, however, space limitations prohibit the addition of new material to the duct (no room for a size increase), and a different approach to vibration reduction must be employed.

When size limitations prohibit a change in duct stiffness, the vibration level may be reduced by altering the fluid flow inside of the duct. This is accomplished by smoothing the corners of the duct at turn locations, and by inserting fluid directional vanes in the duct at turn locations (Figure C2). This smoothing reduces vibrations induced from fluid impacts at the duct wall at the turn locations.

For non-rectangular ducts, cross-supports may still be added, but it is usually easier to apply addition ground supports to increase the duct stiffness, or to change the fluid flow inside the duct to reduce impacts. In extreme cases, it may be necessary to add additional components to the duct network to significantly reduce the level of vibration. These components can include metal springs, elastomeric mounts, resilient pads, or others. For a more detailed treatment of duct vibration, see the reference given below.

REFERENCE

Irwin, J.D., and E.R. Graf, *Industrial Noise and Vibration Control* (Prentice-Hall, Englewood Cliffs, N.J., 1978).

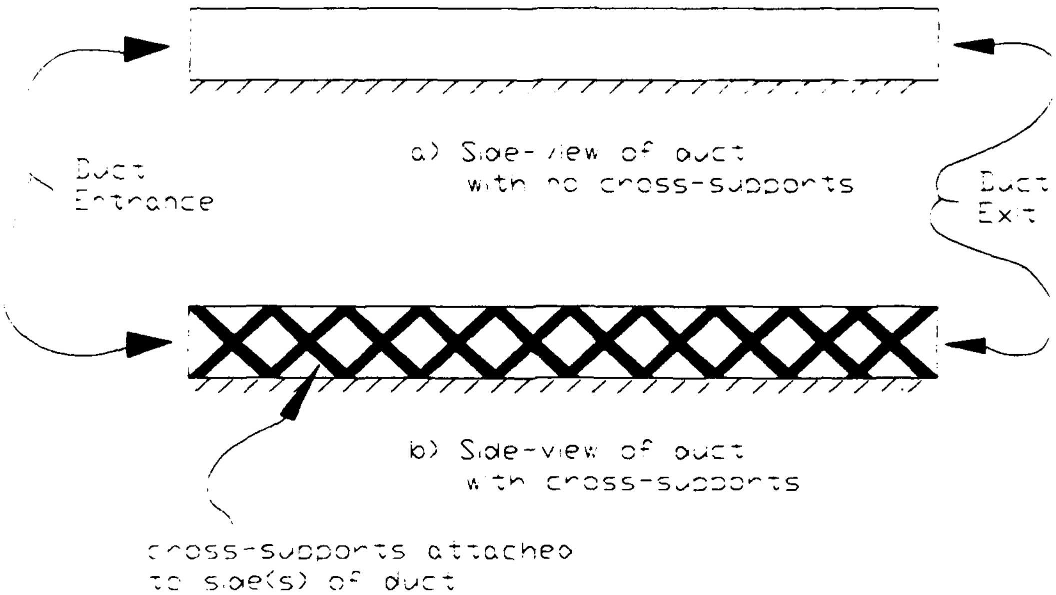

Figure C1. Vibration Reduction in Ducts Via a Change in Stiffness.

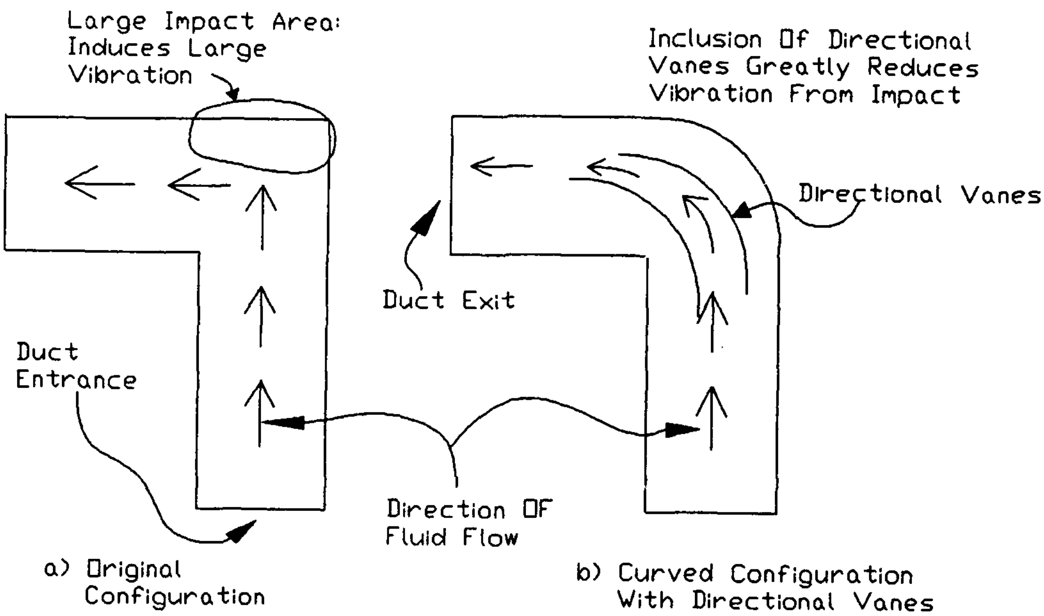

Figure C2. Vibration Reduction in Ducts Via a Change in Fluid Flow.

APPENDIX D: Vibration Coupling Between Machines (or Individual Machine Components)

A machine (or individual machine component) may be vibrating at an unacceptable level due to the vibration of some neighboring machine (or individual machine component) that is vibrating within it's acceptable level. Because the vibration in the second machine (component) is felt to be excessive by the first machine (component), and may cause damage to the first, some type of corrective action should be taken to eliminate the vibration coupling between the two machines (components). What follows is a basic list of procedures that help eliminate the vibration coupling between different machines (components).

1. When possible, the first step is to reduce or eliminate the vibration level of the second machine (component) until an acceptable level of vibration is reached in the first machine (component). This technique is called source alteration, and will work for most vibration coupling problems.

2. When source alteration is not a feasible solution, measures should be taken to isolate the excessive vibration away from the first machine (component). This can be accomplished by using metal springs, elastomeric mounts, resilient pads, or other tools. Attaching a large inertia block to the first machine (component) will also reduce the effects of vibrations from the second machine (component).

3. Another way to reduce the vibration transmitted from the second machine (component) is to attach a vibration absorption device to it. The vibration absorption device (usually a system of springs and masses) omits energy (vibrations) to the ground in a manner such that it is exactly out of phase with the second machine (component). Because the two transmitted energies (vibrations) are exactly out of phase, their effects serve to cancel each other, thus reducing the total amount of energy (vibration) transmitted by the second machine (component).

Another way to reduce the vibration in the first machine (component) is to dampen the vibrations of the second machine (component). This is done by attaching damping devices (such as shock absorbers) between the second machine (component) and ground. These damping devices dissipate the energy (vibration) away from the second machine (component) before they can reach the ground, thus reducing the vibration transmitted between the two machines (components).

For additional information pertaining to the vibration coupling of machines (components), see the reference below.

REFERENCE

Irwin, J.D. and E.R. Graf, *Industrial Noise and Vibration Control* (Prentice-Hall, Englewood Cliffs, N.J., 1978).

APPENDIX E: Balancing of Machine Components

Unbalance in machine components is one of the major causes of excessive vibration, which in turn, may result in inefficient operation, machine damage, or at worst catastrophic failure. Since this is such a common problem, included below is a simple outline for balancing rotating components, such as rotors. The procedure to be outlined is very basic and should work for the majority of balancing problems encountered in standard machine operation. For those interested in a more sophisticated approach to balancing, a list of references is given at the end of this appendix that address alternate techniques for balancing, as well as specific solutions to more complicated problems.

Some of the mechanisms that can cause unbalance are:

1. Blow holes in castings
2. Eccentricity of parts
3. Take-up of clearance tolerances
4. Corrosion
5. Deposit build-up
6. Distortion (stress relief and thermal)

A component may have any or all of the above imperfections (or others) at the same time. In general, it is not necessary to know what portion of the unbalance is attributed to which mechanism. Knowledge of the total unbalance is all that is required to balance a component.

Caution: If the unbalance of a component is caused primarily by deposit build-up (usually occurs on components involved with the transfer of materials), then any corrective balancing may be only a temporary solution. Cleaning of the component, or rebalancing may be required at frequent intervals.

To determine what is an acceptable level of vibration for a machine component, consult the manufacturer's operation specifications. For machines that do not have preset vibration tolerances, an acceptable tolerance may be chosen from the literature pertaining to the specific machine type. Several example references are included at the end of this appendix. By no means is this a complete list; it is merely a starting tool to give the user a place to begin when searching for literature.

If there is workshop with a balancing machine available, and the machine can be shut down for a substantial amount of time, then the faulty component should be sent to the workshop where a qualified technician can balance it. Often however, there is no such workshop available, and the corrective procedure must be completed by maintenance personnel. The technique outlined below is self-contained, and requires no additional training to be complete. The procedure is called Single Plane Vector For Two Plane Balancing, and is conducted in field, which saves on downtime and labor. It should be noted that, even though this procedure works for a majority of balancing problems, some problems may require a different type of balancing solution. Additional balancing techniques may be obtained from the included references.

Necessary Equipment:

1. Vibration Analyzer With Two Pickups And Tunable Filter
2. Stroboscopic Light
3. Protractor
4. Straight Edge
5. Polar Coordinate Graph Paper
6. Balancing Weights
7. Scale (for weighing balancing weights if necessary).

Procedure:

1. Place a reference mark at any arbitrary radial position on each end of the component to be balanced (Figure E1).

2. Operate the machine at balancing speed (usually at or below normal operating speed) with the vibration pickups and stroboscopic light attached to the machine at the two bearing supports nearest the component of interest (Figure E2 shows a schematic of the testing set-up); record the unbalance vibration amplitude (usually in mils) with the vibration analyzer, and the reference position (usually in degrees) with the stroboscopic light for each support bearing of the component. The tunable filter should be set at the balancing frequency (speed) to eliminate vibratory input from other operating components of the machine. The end of the component nearest the support bearing displaying the larger unbalance vibration amplitude shall be balanced first.

3. Using the Polar graph paper, a vector representation of the unbalance vibration shall be constructed. The length of the vector shall represent the amplitude of the unbalance vibration. Let one major division of the graph paper equal one unit of amplitude (i.e., one division = one mil). The angular orientation of the vector should be the same as that of the reference position. Plot this vector at the origin of the graph paper (see Figure E3 for an example plot of original vibration data).

4. Stop the machine and add an appropriate trial weight to the component at the nearest available correction plane. There are several guidelines for selecting an appropriate trial weight, and the one presented here was proposed by J.B. Wilcox.

$$\text{Trial Weight} = 16*W*A/R \text{ ounces}$$

where:

W is the weight of the component in pounds

A is the unbalance vibration amplitude in inches

R is the radial distance to the location of the trial weight in inches. Be sure to record the amount of the trial weight.

5. Operate the machine at the same balancing speed, and record the new unbalance vibration amplitude and reference position. Plot this information as a new vector on the same piece of graph paper as before using the procedure outlined in step 3 above (Figure E4 shows a continuation of the previous example including this step).

6. Draw a third vector by connecting the tip of the first vector to that of the second. The direction of the new vector should point from the first to the second (Figure E5 shows a continuation of the previous example including this step).

7. Measure the length of the third vector using the same scale as before. The actual amount of weight necessary to balance this end of the component will be determined from the measured length.

Actual Weight = Trial Weight*One/Three

where: One is the magnitude of the first vector and three is the magnitude of the third vector

This is the amount of weight to be added in the first balancing plane. (In Figure E3 of the example, the magnitude is measured to be 4.85 units.)

8. The appropriate angular location of the actual weight is determined from the included angle between the first and third vectors. Using the protractor, measure this angle, and note whether the rotation from the first vector to the third vector is clockwise or counterclockwise (in Figures E5 of the example, the included angle is measured to be 66.0 degrees, and the rotation is counterclockwise). The actual correction weight should be rotated on the component by an angle equal in magnitude to the graphically calculated angle, and in the same direction.

9. Attach the actual correction weight in the appropriate location. Run the machine at the same balancing speed, and record a new set of unbalance vibration data at each end of the component.

Due to the propensity for human error in measurements with this procedure, an unacceptable level of vibration in the first end may still exist. If this is the case, repeat steps 3 through 9 (using the initial data to draw the first vector each time) until an acceptable level is achieved in the first end.

10. Upon reaching a satisfactory level of vibration in the first end, balance corrections at the second end should be done. Due to crossover effects between the two ends, the unbalance vibration data at the second end will probably not be the same as that of the original measurement. The current data for the second end should be used to construct the initial vector in step 3. Continue to use steps 3 through 9 until the second end has reached an acceptable level of vibration.

11. Because of the mentioned cross-over effects, the first end may no longer be at an acceptable vibration level once the second end has been balanced. Repeat the entire process (steps 3 through 9) at each end until an acceptable level of vibration has been achieved at both ends.

NOTE: Each time an end is to be rebalanced, all previous balance adjustments are NOT to be disturbed. Treat the current vibration data as "original" apply an appropriate trial weight and proceed to balance as if this were a new problem.

REFERENCES

American Gear Manufacturer's Association, *Balancing Classification for Flexible Couplings*, Standard Document 151.01 (American Gear Manufacturer's Association, January 1974).

Feldman, S., *Unbalance Tolerances and Criteria*, Rpt. 586L-122, in *Proceedings of the Balancing Seminar 4* (Bureau of Ships, Washington DC, April 1958).

In-Place Balancing of Rotating Machinery, No. TP105 (IRD Mechanalysis, Columbus Oh, 1975).

International Organization for Standardization, *Balance Quality of Rotating Rigid Bodies*, Document ISO 1940-1973(E) (International Organization for Standardization, 1973).

Mechanical Vibrations of Shipboard Equipment, MIL-STD-167-2 (ships) (Naval Ship Engineering Center, U.S. Navy, May 1974).

Muster, D. and Flores, B., "Balancing Criteria and Their Relationship to Current American Practice," *Transactions of the American Society of Mechanical Engineers: Journal of Engineering for Industry*, Series B, Vol 91, No. 4 (1969), pp 1035-1046.

National Electrical Manufacturer's Association (NEMA) *Dynamic Balance of Motor*, Standard Document MG 1, par 120.5 (NEMA, 1978).

NEMA, *Method of Measuring Dynamic Balance*, Standard Document MG 1, par 12.06 (NEMA, 1978).

NEMA, *Steam Turbines for Mechanical Drive Service*, Standard Document SM 23 (NEMA, 1979).

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Rathbone, T.C., "Vibration Tolerance," *Power Plant Engineering*, No. 43 (November 1943).

Rieger, Neville F., *Balancing of Rigid and Flexible Rotors* (Naval Research Laboratory, Washington DC, 1986).

Wilcox, J.B., *Dynamic Balancing of Rotating Machinery* (Pitman & Sons Ltd., London, 1967).

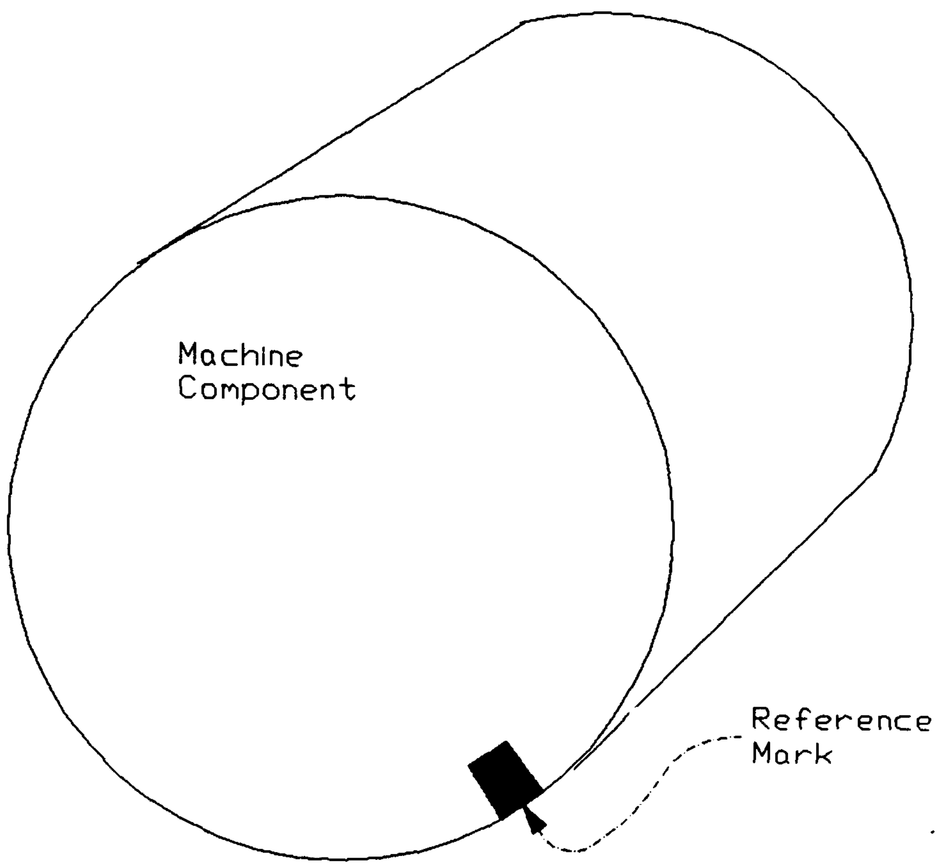

Figure E1. Placement of a Reference Mark for Balancing.

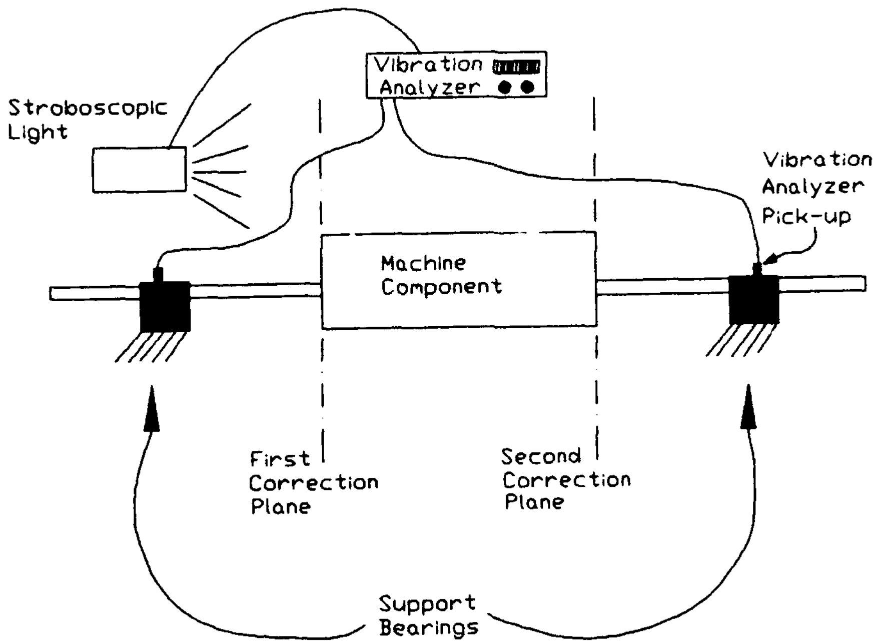

Figure E2. Schematic of Vibration Analyzer Setup for Balancing.

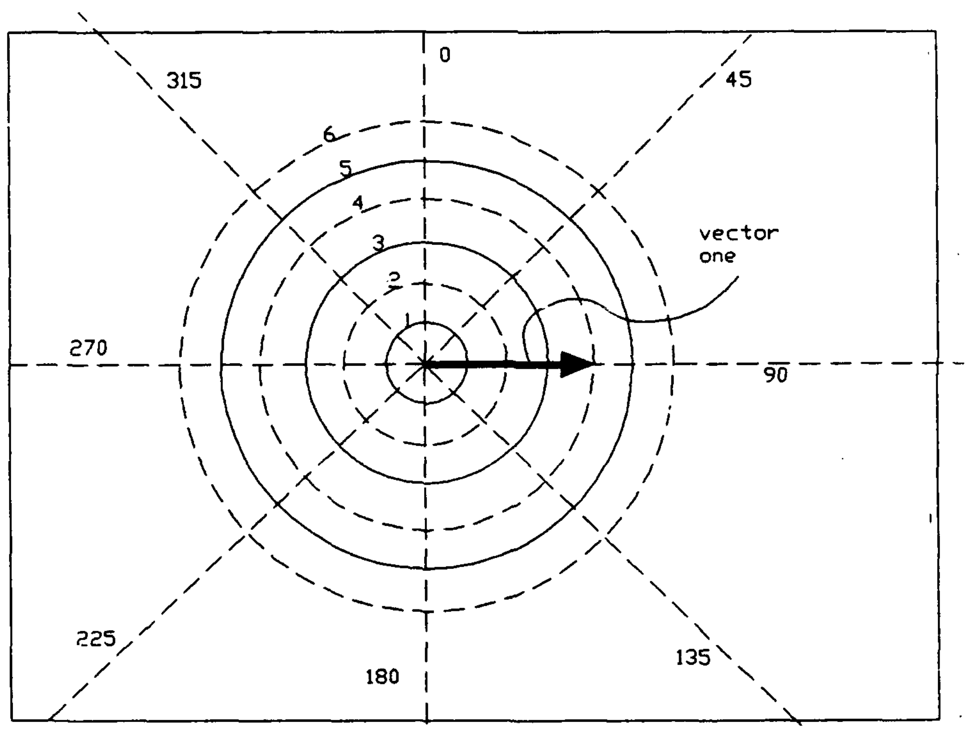

Figure E3. Plot of Original Vibration Data for Balancing.

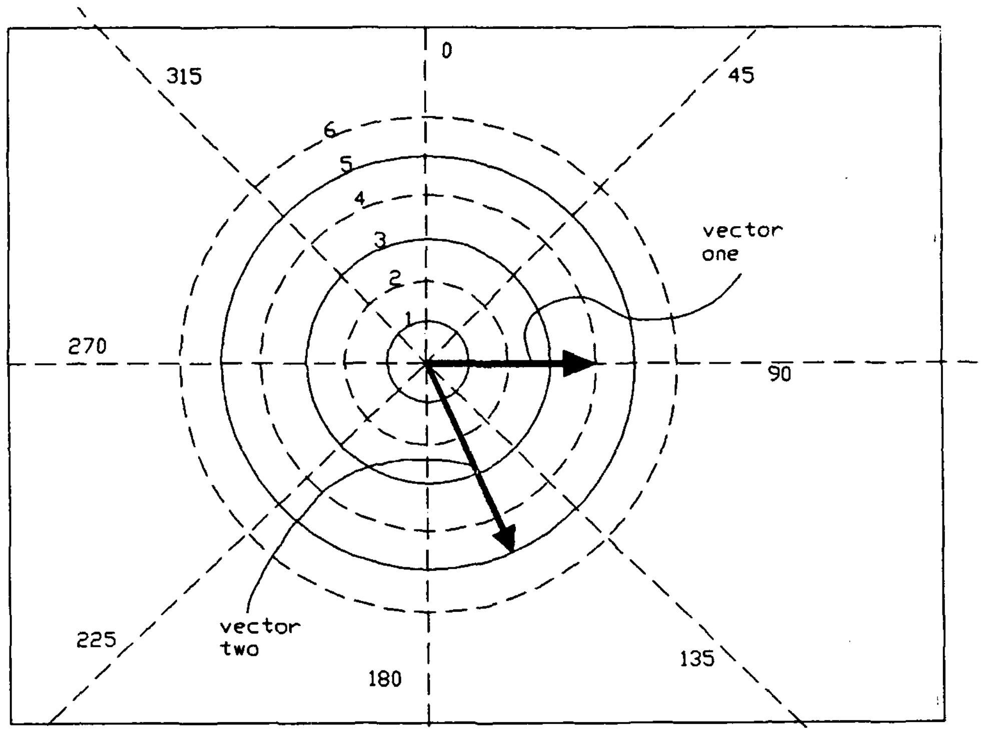

Figure E4. Plot of Trial Weight Data for Balancing.

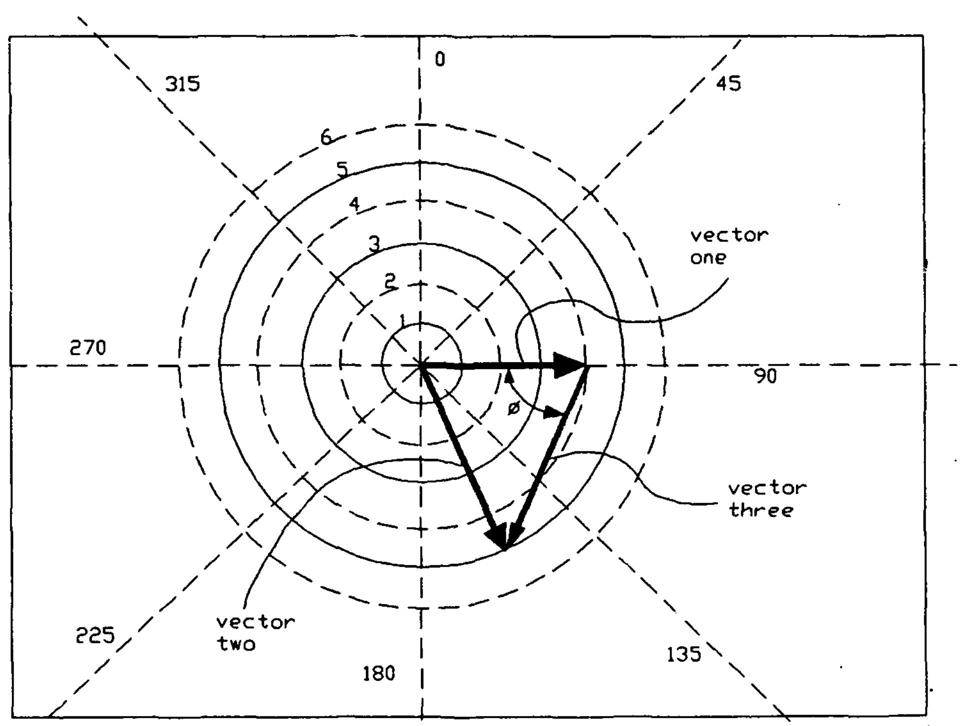

Figure E5. Measuring Unbalance Via the Vector Method.

APPENDIX F: Alignment of Rotating Shafts

Since a large portion of the vibration detected in rotary machinery is due to the misalignment of shafts, included below is a brief outline for shaft alignment. For those interested in a more sophisticated or different approach to shaft alignment, see the list of references at the end of this appendix.

Often, an unbalance in a machine component can be mistaken for shaft misalignment, since their vibration signatures are very similar. In order to prevent hours of wasted time trying to align an unbalanced machine, a simple check for unbalance is recommended before starting the alignment process. Since the machine is to be shut down for alignment, very little additional time is required to preform the balance check (for some alignments, the different machine components are disconnected, thus further simplifying the balance check).

Further ingenuity may also save some time and effort, as some alignment problems are caused by flaws in the machine platform, base, or piping. Granted, alignment of such machines may temporarily solve the problem, but until the problem source is removed, the machine will just become misaligned again after further use. To aid in identifying these types of problems, included below is a brief list of items to visually check prior to the start of the alignment process:

1. Check to see that the piping hangers responsible for carrying the weight of the piping are secure, and are located at the proper positions.

2. Look for loose piping flange bolts.

3. Look for cracked concrete bases or support columns.

4. Check for water seepage between the base-plate and concrete foundation (if this water freezes, the water expansion could cause severe structural damage).

5. Look for loose foundation bolts.

6. Check for shim packs that may have worked loose.

7. Check for rusty shims (also shims that have folds, wrinkles, burrs, hammer marks, dirt , or paint on them).

8. Check for loose or sheared dowel pins.

9. Check to see that bearings are properly installed, lubricated, and that the bearing covers are properly tightened.

One other item to check prior to the alignment process is the foundation. Due to concrete shrinkage, settling of base soils, thermal warpage, or some other mechanism, the foundation on which a machine sits may have become buckled or warped. The following fives steps outline how to check for this type of damage.

1. Using Figure F1, attach dial indicators on the foundation corners of the machine element.

2. Keeping all previous shims properly in place, tighten all foundation bolts to their required torque values.

3. Zero the dial indicators at each corner.

4. Loosen the foundation bolt at one corner, checking the dial indicator to see if that foot lifts up. Continue around the machine, loosening each corner bolt, and check each dial indicator where the foundation bolt has already been loosened.

5. If any foot has lifted more than 0.002 in., place a shim under that foot (or feet) equal to the amount of movement shown by the dial indicator.

This technique will also allow for the identification and correction of a bowed condition along one side or across adjacent corners, as shown in Figures F2 and F3.

To begin the alignment process, a dial indicator must be attached to one or both of the shafts (depending on the method of alignment). This is usually accomplished by using mounting brackets, of which the "Christmas tree" bracket is the most common. The indicators can also be attached using magnetic bases, or any one of a number of different ways (for more details concerning the mounting of dial indicators, see the references at the end of this appendix). The method chosen for each alignment should be selected on an individual basis, as the machinery type and geometry will change for each machine.

If some form of a bracket is to be used, then bracket sag during rotation of the shaft must be taken into account. Due to the weight of the dial indicator and of the bracket itself, as the shaft is rotated through the bottom position, there will be some additional deflection measured by the dial indicator due to the pull of gravity. This additional deflection will cause errors in the alignment readings, and thus, it should be measured prior to the alignment process, and then compensated for during alignment.

To measure bracket sag, attach the bracket (with the dial indicator in place) to the shaft. Start with the dial indicator in the 12 o'clock position, and zero the indicator (see Figure F4). Rotate the shaft through 360 degrees, taking readings every 90 degrees (i.e., readings should be taken at 3, 6, and 9 o'clock). Record these readings for use during alignment (compensation for bracket sag will be discussed later). If the measured sag exceeds 30 mils, a counter weight may be added to the bracket assembly. If the sag can not be reduced below 30 mils, another type of bracket or mounting procedure should be used.

The proper way to mount dial indicators is shown in Figure F5 (for the reverse indicator method of alignment). It is important to make sure the dial indicator remains in contact with the shaft (or hub) at all times. If the indicator does not remain in contact for the entire revolution of the shaft, reposition the bracket assembly and re-zero the indicator at the 12 o'clock position. Repeat this step until the indicator remains in contact for the entire revolution of the shaft. Furthermore, the dial indicator should remain perpendicular to the shaft (or hub) at all times.

There are several different means by which shaft alignment can be done. This appendix is designed for use by maintenance personnel who do not have access to expensive or sophisticated equipment. Therefor, the alignment technique outlined below is of the simplest type; that using a dial indicator. Those interested in other more sophisticated alignment techniques should see the list of references at the end of this appendix.

There is also more than one method to align shafts using a dial indicator. The method presented here is called the reverse indicator method. It tends to be one of the more accurate methods for taking shaft alignment measurements, and as such, it is becoming one of the most popular. For information concerning other dial indicator techniques, see the list of references at the end of this appendix.

Before taking readings with the dial indicators, some rough measurements and adjustments should be made. Using a straight edge, the parallel gaps can be reduced; and using an inside micrometer, the angular misalignment can be reduced. These simple steps should reduce the misalignment to level where the dial indicators will have no trouble recording the remaining misalignment.

With the dial indicators in position, rotate the shafts (on each the driving and driven parts) through 360 degrees, taking readings every 90 degrees. Repeat the readings a second time to insure accuracy, and check to see that the dial indicator has returned to the zero position. If the readings do not agree within 10 percent, take a new set of readings. If the readings do agree within 10 percent, then take the average of the readings as the measured value. One way to check to see if the readings are reasonable is to use the validity rule. This rule states that, in the ideal situation, the reading on the left, plus the reading on the right, should equal the reading on the bottom (Figure F6). However, the real world is rarely an ideal situation, and thus, it is sufficient if the measured readings approximate the validity rule. Caution, if the shaft is not rotated properly, some deflection may be induced by bowing in the shaft. Thus, it is recommended that a wrench designed for this purpose be used.

Once the measured values have been obtained, the first thing to do is to make any necessary corrections to account for sag. Note, the sag readings will always be negative; thus, any correction will involve adding the absolute numeric value of the sag to the field reading. Figure F7 shows an example of a typical correction for sag.

Now that the true misalignment readings are known, the required machine moves can be calculated to achieve alignment. The machine moves may include lateral and vertical moves on both the driven and driver units. They may also include different amounts of movement between the inboard (front) feet and outboard (back) feet of each unit. Included in Figure F8 are equations to determine the amount of movement necessary for each part of each unit.

The inexperienced person may feel that the shafts are now perfectly aligned. This may indeed be the case, but shafts in perfect alignment while the machine is not running may still be severely misaligned during machine operation. This misalignment is caused by dynamic effects, the most common of which is thermal expansion.

The effects of thermal expansion were not addressed earlier because they only cause problems for some machines, usually large or high speed ones. In general, machines that operate at high speeds generate a large amount of thermal energy (as witnessed by the amount of heat they give off). In turn, this thermal energy raises the temperature of the machine and the surrounding components. As a piece of material undergoes an increase in temperature, it will also undergo an expansion proportional to both its length, and to the amount of temperature change. Thus, any machine which has a temperature increase during operation, will undergo thermal expansion, and as a result, may no longer be in perfect alignment (as mentioned above, this usually applies to high speed machinery).

As also mentioned above, thermal expansion is proportional to the length of the item undergoing the temperature increase. Thus, even for small changes in temperature, if a machine (or its surrounding components) is large enough, a substantial amount of thermal expansion may take place.

In general, the effects of thermal expansion can be accounted for very easily. In fact, several different solution methods for this problem have been posed in the literature (see the list of references at the end of this appendix). What follows here is one of the most basic techniques for thermal expansion corrections. This technique involves obtaining temperatures for the different machine components at or near their operating temperatures, and then comparing these values to ambient (room) temperature to determine the temperature change of each part of the machine. These temperature changes are then used to calculate the amount of expansion in each part. If any part of the machine undergoes a significant amount of expansion during operation, this expansion can now be accounted for during the 'cold' alignment.

The first step is to obtain temperatures for the machine parts as close to their operating temperatures as possible. It should be noted that since shafts, hubs, and some couplings are circular (thus possessing radial symmetry), any expansion will be equal in all radial directions. The result is that the center-line (axis of rotation) will not change, and thus, there should not be any change in the alignment of these shafts (Figure F9). However, most machine platforms, stands, housings, and piping are not symmetrical. Therefor, these are the parts whose thermal expansion will play the greatest role in causing misalignment, and as such, the user should concentrate on obtaining their temperatures.

The equation for calculating thermal expansion is given below:

$$\text{Expansion} = L*A*\text{Temperature change}$$

where:

Expansion is the change in length of the machine part in inches.

L is the linear dimension of the part measured in inches (could be in the axial, horizontal, or vertical direction, depending on the direction the expansion is being calculated).

A is the coefficient of thermal expansion given in 1/°F (This is a material property unique to each material, and may readily be found in any material handbook, or any textbook on the mechanics of materials). Temperature change is the difference between the operating temperature of the machine and ambient temperature in °F.

Two examples of typical calculations are shown in Figure F10. The main drawback to this approach is that it is not always easy to obtain accurate readings for the machine operating temperatures. Furthermore, due to temperature gradients, the temperature readings obtained at one location on a machine part will not be the same as that at a different location. That is to say the temperature change is not uniform, and neither will be the thermal expansion. Despite these pitfalls, this technique usually provides a sufficiently accurate means by which to account for thermal expansion. If the user finds that the machine is still out of align at its operating temperature, a more sophisticated approach should be used.

Once the thermal expansion in the different machine parts is known, all that remains to be done is to adjust the machine using these calculations and the previous alignment calculations from Figure 8. For example, consider the following hypothetical information for vertical alignment of a shaft protruding from a motor:

OR is -12 mils

IR is + 9 mils

Thermal expansion at the outboard foot is +3 mils

Thermal expansion at the inboard foot is +7 mils

The appropriate machine movements in the vertical plane are thus:

1. Lower the outboard foot 15 mils.
2. Raise the inboard foot 2 mils.

Now that the thermally adjusted, sag-corrected, required movement is known, the machine may be moved to obtain proper alignment. It is recommended to make all horizontal corrections first, and then make the vertical corrections. This will allow for more accurate movements in the vertical direction (for reasons beyond the scope of this report). To help eliminate misalignment from shim shifting, it is recommended to use as few shims as possible (i.e., replace several thin shims with one larger shim). Once the moves have been completed, the alignment should be checked to ensure that the shafts have been aligned to within an acceptable tolerance. If the alignment is not satisfactory, repeat the above steps until an acceptable tolerance is reached. To determine what is an acceptable tolerance, always consult with the manufacturer first. If there are no manufacturer specified tolerances, consult the literature to obtain some general guidelines. Included with the references in this appendix are some typical sources containing alignment tolerance specifications.

It should be noted that other dynamic effects, such as those resulting from hydraulic forces or torque reactions, cannot be determined during cold alignment. If it is suspected that these are the cause of misalignment, other alignment techniques must be used (see the following references).

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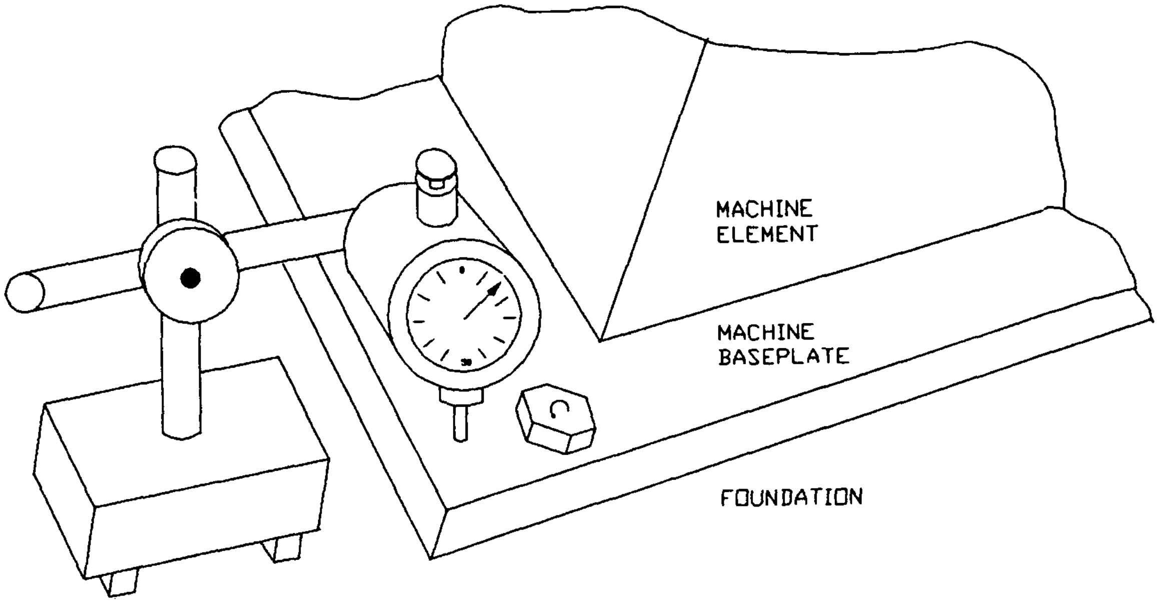


Figure F1. Placement of Dial Indicators for Foundation Measurements.

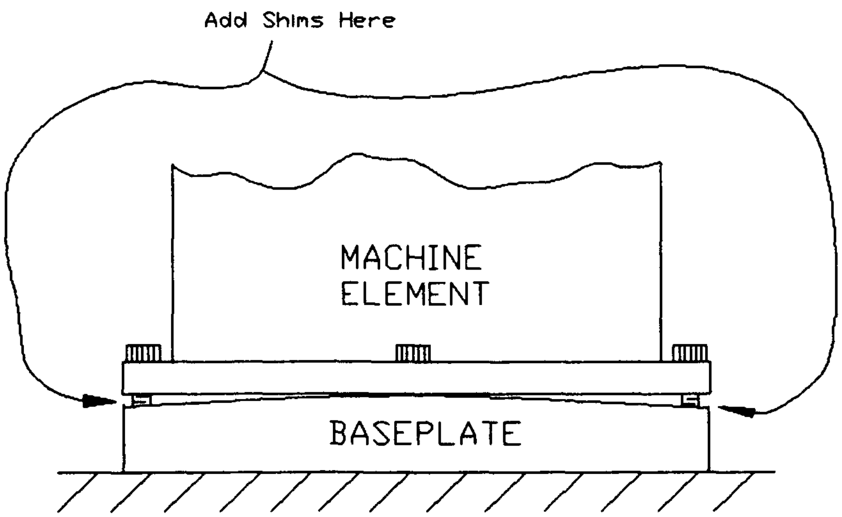


Figure F2. Bowed Baseplate Condition.

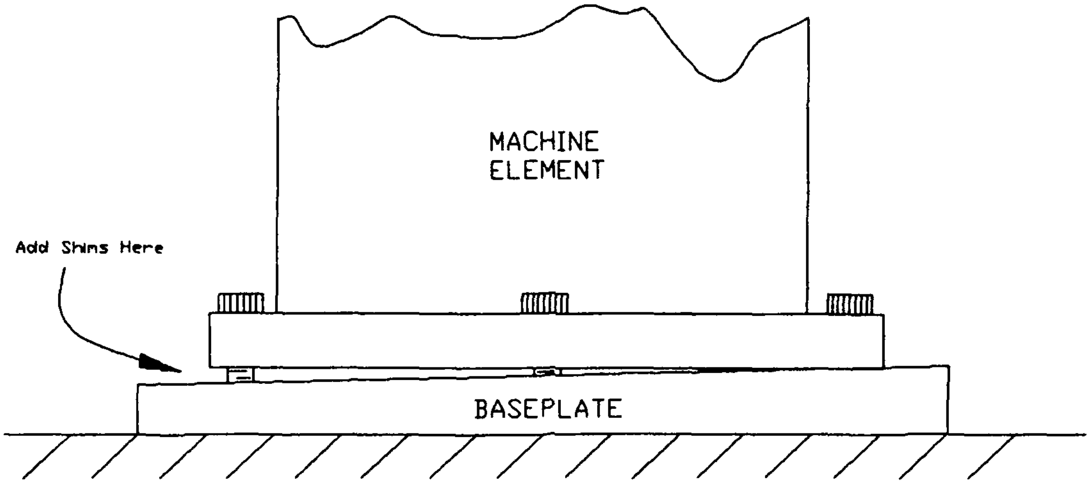


Figure F3. Skewed Baseplate Condition.

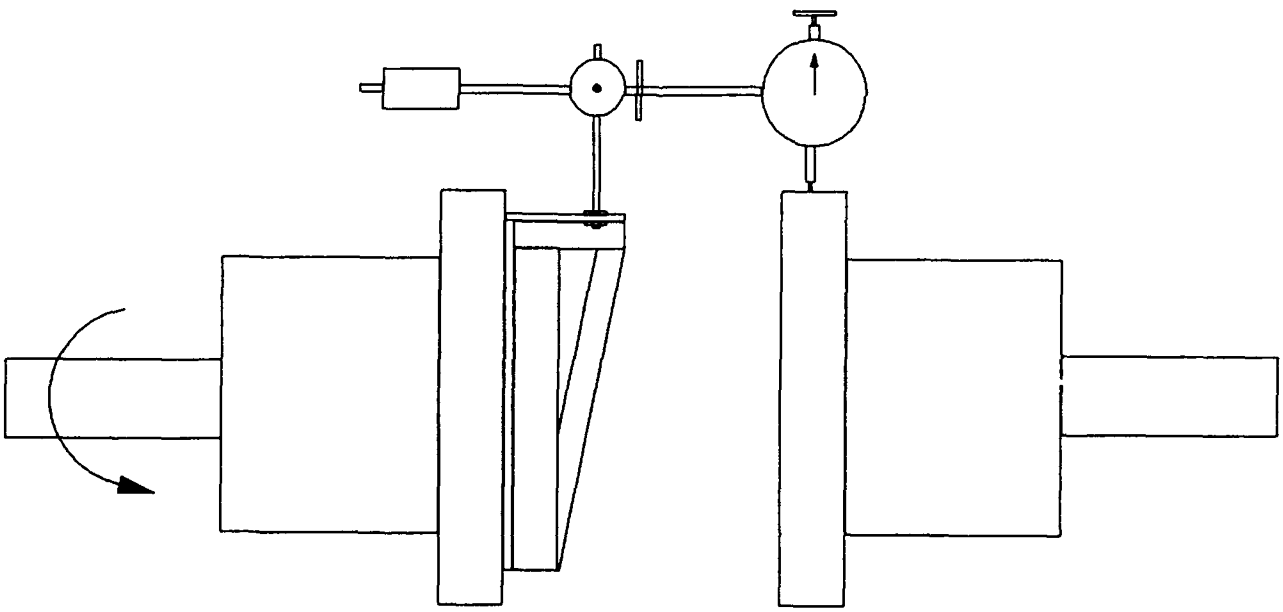


Figure F4. Bracket Placement for Measuring Sag.

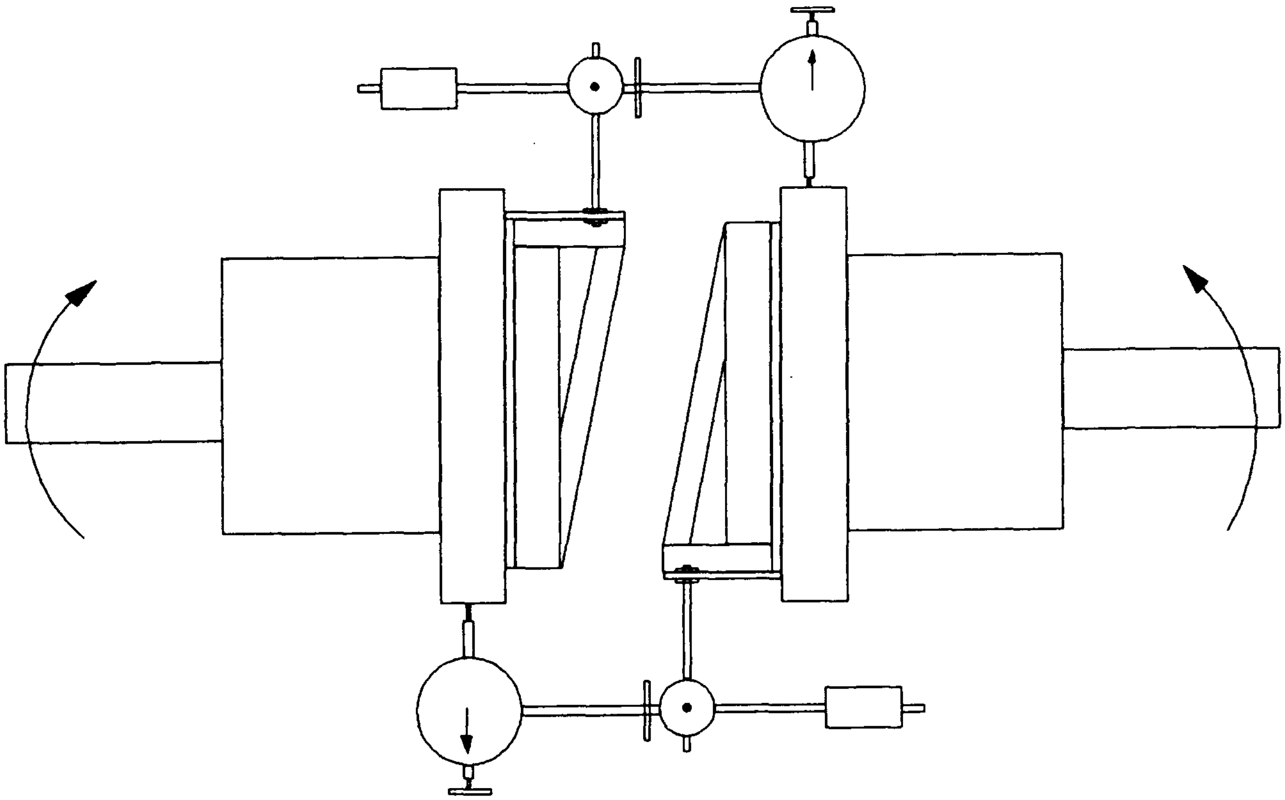


Figure F5. Proper Mounting of Dial Indicators (Reverse Indicator Method).

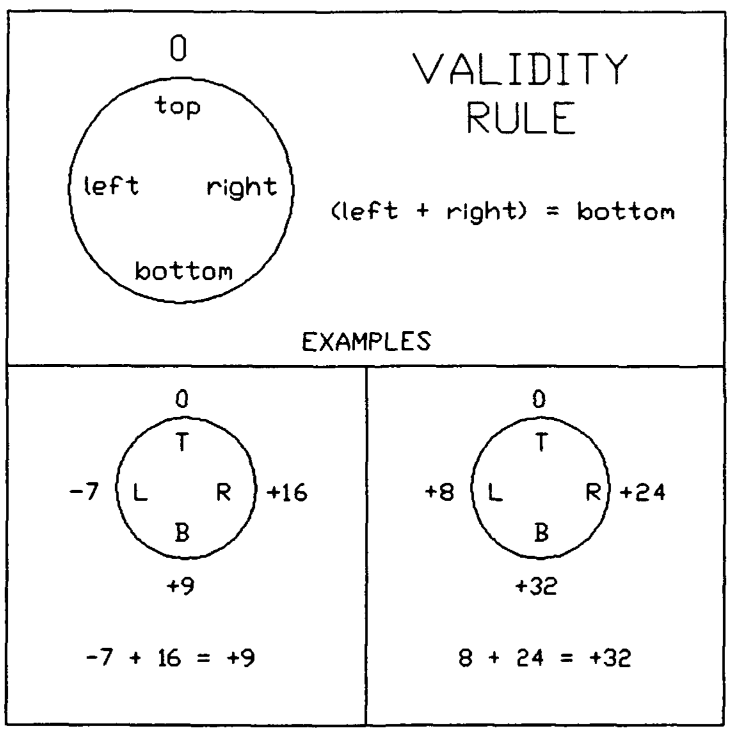


Figure F6. The Validity Rule.

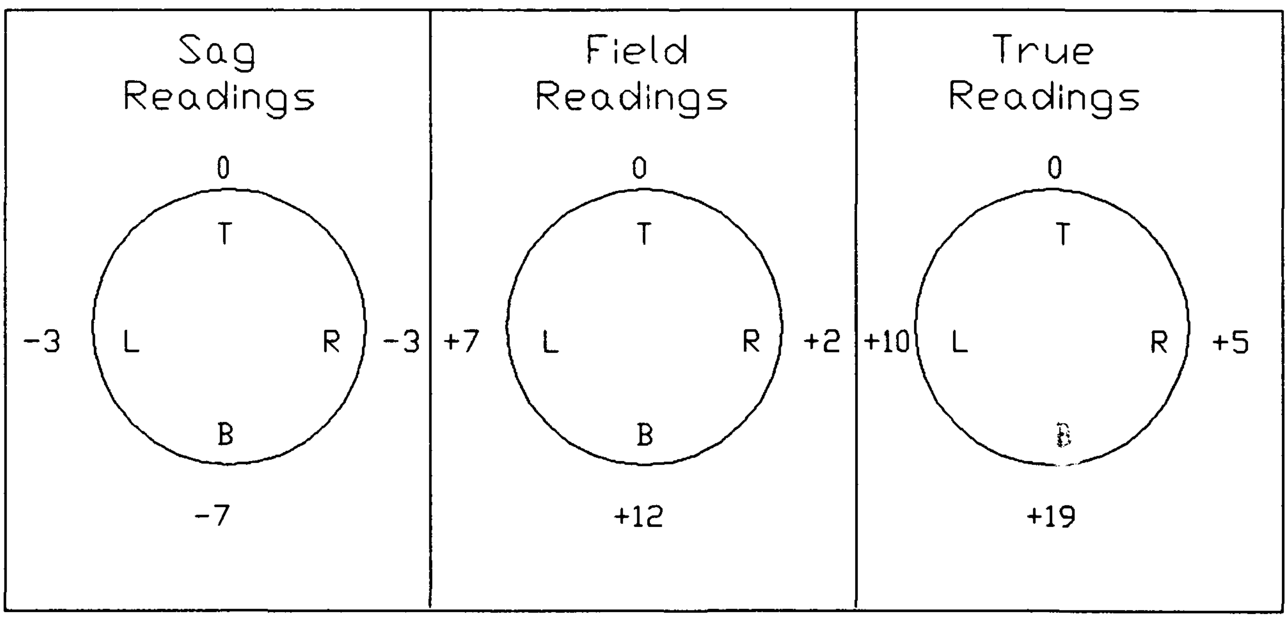
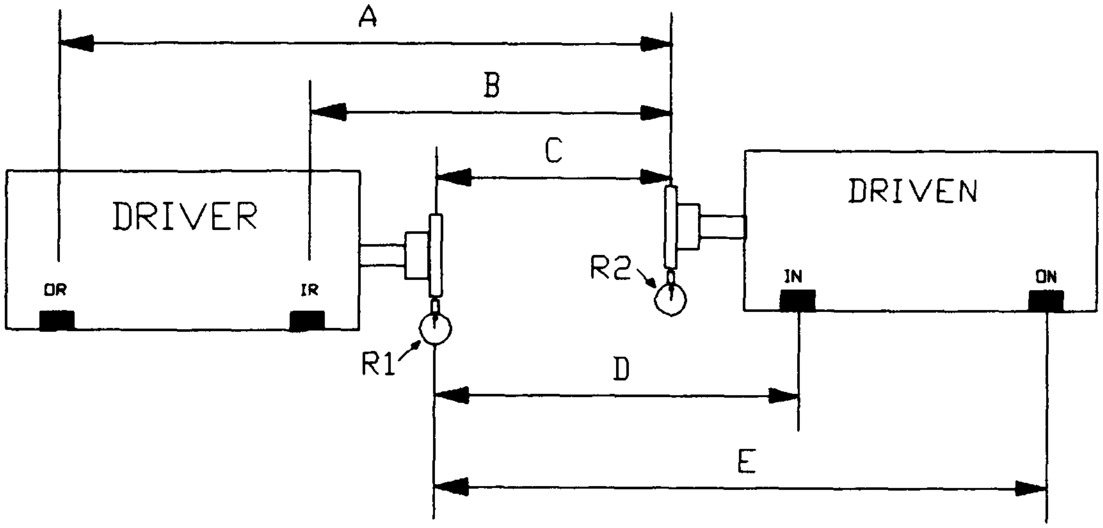


Figure F7. Reading Corrections for Sag.



A	=	is the distance from the outboard foot of the driver unit to the point of reading on the driven unit.
B	=	is the distance from the inboard foot of the driver unit to the point of reading on the driven unit.
C	=	is the distance between the dial indicator readings on both shafts.
D	=	is the distance from the inboard foot of the driven unit to the point of reading on the driver unit.
E	=	is the distance from the outboard foot of the driven unit to the point of reading on the driver unit.
R1	=	is one-half of the dial indicator reading taken on the driver unit.
R2	=	is one-half of the dial indicator reading taken on the driven unit.
OR	=	is the movement required at the outboard foot of the driver unit.
IR	=	is the movement required at the inboard foot of the driver unit.
IN	=	is the movement required at the inboard foot of the driven unit.
ON	=	is the movement required at the outboard foot of the driven unit.

Required Movement For Driver Unit	Required Movement For Driven Unit
IR = [B*(R1+R2)/C]-R2	IN = [D*(R1+R2)/C]-R1
OR = [A*(R1+R2)/C]-R2	ON = [E*(R1+R2)/C]-R1

Use these equations to determine required machine movements for the reverse indicator method. Positive values for the inboard or outboard calculations indicate that the unit has to be raised or moved to the left; negative values indicate that the unit has to be lowered or moved to the right.

NOTE: All the distances shown should be measured directly from the machines being aligned.

Figure F8. Equations To Determine Required Machine Movements.

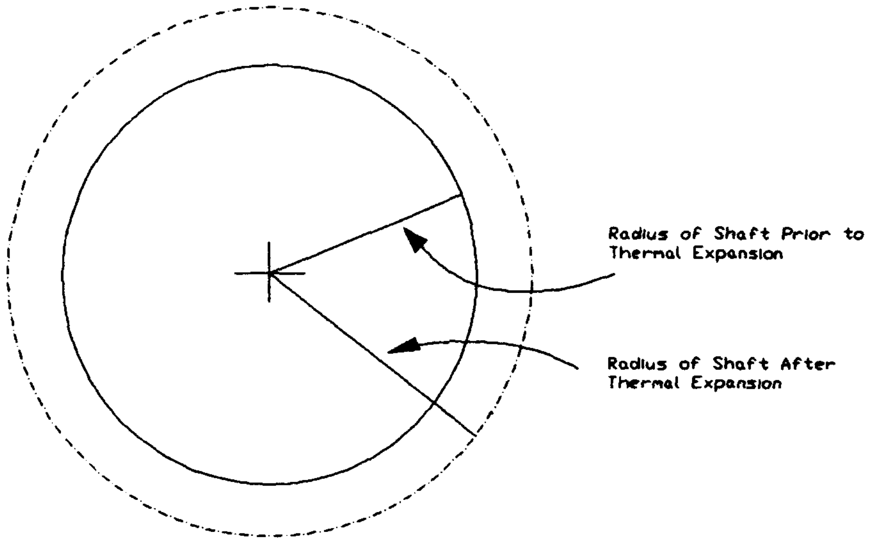
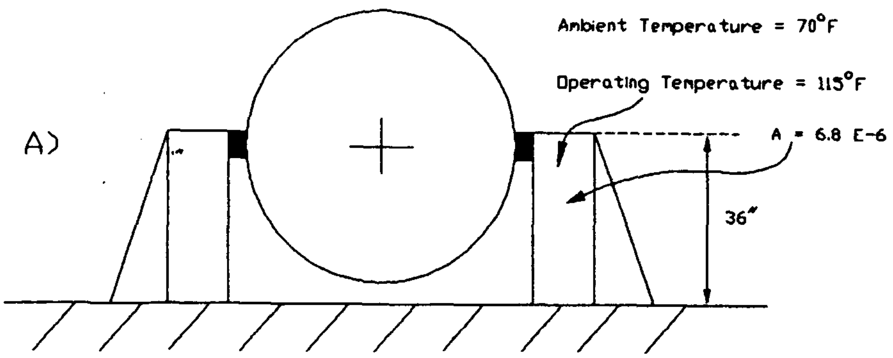
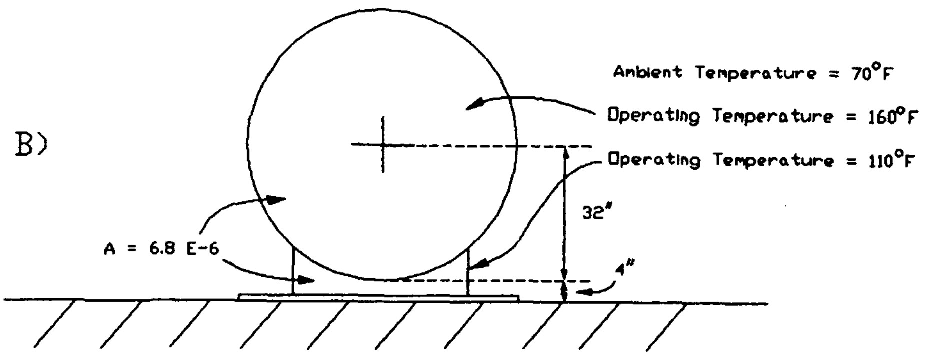


Figure F9. Effects of Thermal Expansion on a Circular Part.



Thermal expansion of legs = (36 in.)*(6.8 E-6)*(115°F − 70°F) = .01100 in. = 11 mils
(because there is no vertical constraint on the compressor case itself,
its thermal expansion may be ignored)
--this end is raised by 11 mils--



Thermal expansion of sway bar = (4 in.)*(6.8 E-6)*(110°F − 70°F) = .00110 in. ≅ 1 mil
(since the compressor case is rigidly attached to ground vertically, its thermal
expansion must now be taken into account)
Thermal expansion of compressor case = (32 in.)*(6.8 E-6)*(160°F − 70°F) = .01960 in. ≐ 20 mils
--this end is raised by 21 mils--

Figure F10. Calculations for the Thermal Expansion in a Centerline Mounted, Carbon Steel Compressor With Carbon Steel Legs and Sway Bar.

APPENDIX G: Determination of Bearing Fault Type

In general, a defective bearing may generate vibrations in four distinct frequency ranges. These frequencies can be classified as:

1. Rotational frequencies
2. Component natural frequencies
3. Sum and difference frequencies
4. Random frequencies.

Furthermore, the rotational frequencies are not, in general, integer multiples of the shaft rotating speed. In fact, they can be classified in four new frequency ranges:

1. Cage frequencies
2. Ball frequencies
3. Inner race frequencies
4. Outer race frequencies.

Thus, if the above four frequencies are known, a simple inspection of the collected vibration frequency data will tell the user which part of the bearing contains the fault (i.e., a flaw in the inner race will excite a vibration at the inner race frequency).

To determine these frequencies, the following information is needed Figure G1):

1. The shaft speed (rpm)
2. The number of balls in the bearing (n)
3. The diameter of the balls in the bearing (db)
4. The diameter of the inner race (di)
5. The diameter of the outer race (do).

The four frequencies can be calculated using the following equations:

$$\text{Cage Frequency} = rpm*Di/(Di+Do)$$

$$\text{Ball Frequency} = rpm*(Do/Db)*Di/(Di+Do)$$

$$\text{Inner Race Frequency} = rpm*N*Do/(Di+Do)$$

$$\text{Outer Race Frequency} = rpm*N*Di/(Di+Do)$$

where rpm, N, Di, Do, and Db are defined above (as shown in Figure G1)

Note: The rotational frequencies calculated with these equations may not agree exactly with the measured frequencies due to ball slippage and discrepancies between the actual ball path and the raceway diameters used in the calculations.

If the bearing component dimensions are not known, faults in the inner and outer races can still be detected by vibration frequencies which correspond to the ball passing frequencies of the inner and outer races (i.e., a flaw in the inner race will excite a vibration at the ball passing frequency of the inner race).

Estimates for the inner and outer race ball passing frequencies are given by:

Inner Race Ball Passing Frequency = 0.6*N*rpm

Outer Race Ball Passing Frequency = 0.4*N*rpm

If the rolling elements are the problem source for the bearing vibration, then it will be the rolling frequencies that show an increase in vibration amplitude, while the other frequencies remain stable. The most common causes of rolling element problems are improper lubrication, overheating, and electrical currents.

Component natural frequencies are excited by the momentary impacts between the rolling elements and the bearing raceways. Because these impacts excite the natural frequencies of the bearing components as well as the natural frequencies of other machine components, the data pertaining to natural frequency vibrations is often cluttered, and thus, is of little value in determining the cause or location of the vibration source.

Sum and difference frequencies are the result of combinations of the above frequencies, thus, knowledge of the above frequencies is required to get any information out of a sum or difference frequency. As a result, they usually are of little value in aiding in problem identification and solution process.

Random frequency vibrations (usually in the higher frequency range) indicate bearing deterioration due to surface fatigue, abrasion (friction), or similar problems. In general, these types of vibration do not exhibit distinctive frequency peaks, nor due they show any signs of periodicity.

For more information concerning bearing faults and their identification, see the reference given below.

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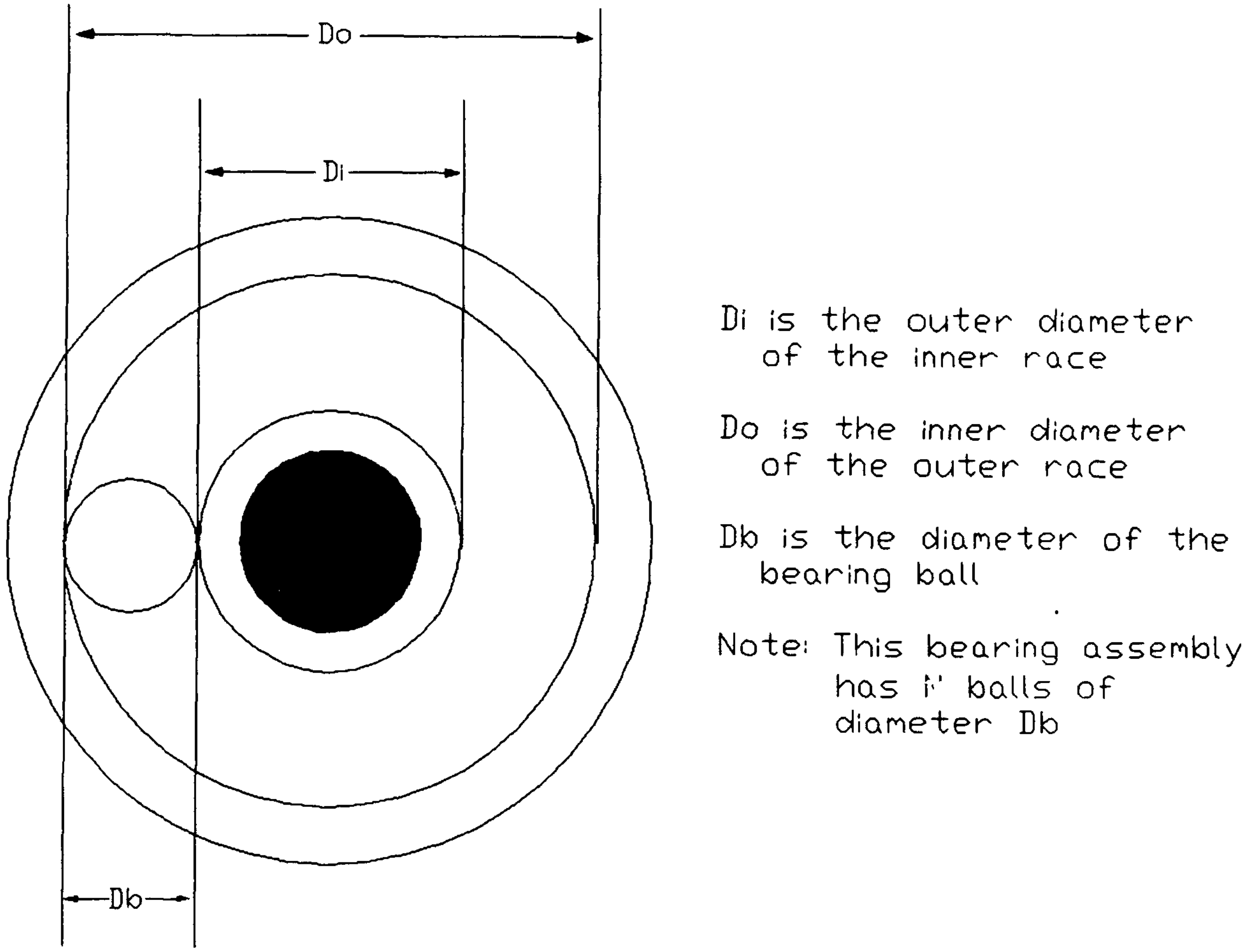

Figure G1. Dimension Definitions for a Roller Bearing.

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