RELIABILITY ISSUES OF MOTOR, BLOWER & PUMP INSTALLATIONS

CONDITION MONITORING
MAINTENANCE & REPAIR

Prepared by:

Chip Corbin, President

IMPACT Engineering, Inc.
23412 68th Avenue South
Kent, Washington 9032

Tel: (253) 826-9003
Fax: (253) 826-9004
www.impactengineering.com
ABSTRACT

This report reviews maintenance and repair of motor coupled blowers and pumps. This review will be approached from a perspective of maximizing machinery reliability utilizing Predictive Maintenance techniques. Although we will focus largely on the Motor – Blower installations, the issues discussed apply to most types of rotating machinery.

Over the past decade, instrument and personal computer technology advances have provided capability to assess and predict machinery problems by measuring operating condition using vibration, ultrasonic, and thermal sensors. Although traditional repair approaches remain a valid part of corrective maintenance, these new techniques combine to pro-actively manage maintenance; where the goal is increased reliability and reduced overall maintenance cost.

Structural considerations, bearings, lubrication, dynamic balancing and alignment all play a role in proper operation. Each of these topics will be discussed with common faults detailed. Diagnostic methods and repair considerations will be examined.

Finally, condition monitoring technologies will be examined for benefits of extending machine life and reducing repair costs by uncovering developing faults, allowing scheduling of pro-active maintenance before failure. The strategy of Total Reliability Management to reduce overall maintenance costs will also be introduced.

Corrective maintenance experiences and results by Impact Engineering are detailed in this document for use by our clients in implementing repair and maintenance procedures. The following sections will detail issues about these vital pieces of machinery from installation to repair, where increased reliability and reduced maintenance costs are the fundamental objectives of Total Reliability Management.
TABLE OF CONTENTS

ABSTRACT

1. FOUNDATION CONSIDERATIONS ........................................................................................ 1
   1.1. WORN/DAMAGED VIBRATION ISOLATORS ................................................................. 1
      1.1.1. Purpose of Isolators .............................................................................................. 1
      1.1.2. Problem with Isolators .......................................................................................... 1
      1.1.3. Replacement of Isolators ...................................................................................... 1
   1.2. STRUCTURAL RESONANCE CONDITION AT RUNNING SPEED ...................................... 2
      1.2.1. Determining a Resonant Frequency ....................................................................... 2
      1.2.2. De-tuning a Resonance Using Structural Modifications ......................................... 2
   1.3. CRACKS/DAMAGE TO FOUNDATIONS ......................................................................... 3
   1.4. SUMMARY .................................................................................................................... 3

2. BEARINGS & LUBRICANTS ............................................................................................... 4
   2.1. MOTOR BEARING SELECTION ......................................................................................... 4
      2.1.1. Greaseable vs. Non-Grease type Shielded (2ZZ) / Sealed (2RS) ............................. 4
      2.1.2. Switching From Open/Greaseable to Sealed/Shielded Bearings ............................... 4
      2.1.3. Bearing Life ........................................................................................................... 5
      2.1.4. Motor Bearing Fit/Tolerance (C3) Issues ................................................................. 5
   2.2. FAN PILLOW BLOCK BEARING SELECTION ............................................................... 7
      2.2.1. Types of Bearings ................................................................................................. 7
      2.2.2. Proper Bearing Selection ....................................................................................... 8
   2.3. FAN PILLOW BLOCK BEARING INSTALLATION ........................................................... 8
      2.3.1. Basic Rules to Follow ............................................................................................ 8
      2.3.2. Shaft Mounting of Tapered Bore Spherical Roller Bearings .................................... 8
      2.3.3. Shaft Mounting of FAG - Split Race Spherical Roller Bearings ............................ 9
      2.3.4. Looseness Concerns of Bearing Assemblies ......................................................... 9
   2.4. GREASING ISSUES & HIGH TEMPERATURE LUBRICANTS ............................................. 12
      2.4.1. Under Lubricated/Cool Bearing ........................................................................... 12
      2.4.2. Frequency & Amount of Lubricant ....................................................................... 13
      2.4.3. Automatic Lubricators ......................................................................................... 14
      2.4.4. Typical Operating Temperatures (High Temperature Concerns) ............................ 15

3. COUPLINGS & LUBRICANTS ........................................................................................... 16
   3.1. COUPLING ASSEMBLY & PROPER LUBRICANTS ......................................................... 16
      3.1.1. Setting Axial Coupling Clearances ........................................................................ 16
      3.1.2. Selection of Proper Lubricants .............................................................................. 17
      3.1.3. Offset of Coupling Keys ....................................................................................... 17
      3.1.4. Proper Alignment of Shafting/Coupling ................................................................. 17

4. MACHINERY ALIGNMENT PROCEDURES ..................................................................... 18
   4.1. MACHINERY SHAFT ALIGNMENT .............................................................................. 18
      4.1.1. Angular Misalignment ........................................................................................... 19
      4.1.2. Offset Misalignment .............................................................................................. 19
      4.1.3. Combination Angular and Offset Misalignment .................................................... 19
   4.2. INSTRUMENTATION .................................................................................................... 19
   4.3. THE “REVERSE INDICATOR” ALIGNMENT METHOD .................................................. 20
4.4. LASER-OPTICAL ALIGNMENT SYSTEMS ................................................................. 22
4.5. SHAFT ALIGNMENT VS. COUPLING ALIGNMENT ............................................ 23
4.6. PROPER MOVEMENT/SHIMMING METHODS FOR THE MTBM ......................... 24
4.7. SOFT -FOOT ........................................................................................................ 24
   4.7.1. Basis for Understanding and Correcting Soft-Foot ............................................ 25
4.8. PROPER HORIZONTAL MOVEMENT OF THE MTBM ....................................... 25
   4.8.1. Jacking Bolts .................................................................................................. 25
   4.8.2. Dowel Pins .................................................................................................... 26
4.9. ALIGNMENT TOLERANCES ............................................................................... 27
   4.9.1. Angularity and Offset .................................................................................... 27

5. FAN/BLOWER BALANCING .................................................................................... 28
5.1. BASIC UNDERSTANDING OF UNBALANCE FORCES IN ROTORS ..................... 29
5.2. APPROVED METHOD FOR DYNAMIC BALANCING ........................................... 30
5.3. CLEANING AND INSPECTION FOR DAMAGE/CRACKS PRIOR TO BALANCING .... 30
5.4. OTHER INSPECTIONS/CHECKS PRIOR TO BALANCING ....................................... 31
5.5. SHOP BALANCING VS. IN-PLACE / FIELD TRIM BALANCING ......................... 31
5.6. METHOD OF BALANCE WEIGHT ATTACHMENT ................................................. 32
   5.6.1. Important Rules to Remember ........................................................................ 32
5.7. IN-PLACE BALANCE SPECIFICATIONS ............................................................ 33
   5.7.1. Motor Balance ............................................................................................... 33
   5.7.2. Fan/Blower Rotor Balance ............................................................................ 33
   5.7.3. Entire Unit ..................................................................................................... 33
5.8. THE AFFECTS OF BALANCE TOWARD EXTENDING MACHINERY LIFE ............. 34

6. CONDITION MONITORING .................................................................................... 35
6.1. CONDITION ASSESSMENT OF THE MACHINE ................................................... 35
   6.1.1. Condition Analysis Methods ........................................................................... 35
   6.1.2. Vibration Analysis & Assessment .................................................................. 35
   6.1.3. Ultrasonic Bearing and Coupling Analysis ..................................................... 35
   6.1.4. Vibration and Ultrasonic Tolerances .............................................................. 36
   6.1.5. Thermographic (Infrared) Analysis ............................................................... 37
6.2. TOTAL RELIABILITY MANAGEMENT ............................................................... 38
1. FOUNDATION CONSIDERATIONS

As with any structure, the foundation must be sound in order to build upon it. Over the past twenty plus years that Vibration and Ultrasonic Condition surveys have been conducted, several types of foundation problems have been encountered. These will be addressed in order of importance as well as how often these problems are detected. It should be noted that the most common problems and causes of failures related to rotating machinery are related to the machine components themselves and not the machinery base. However, the foundation inspection and repair work should be the first step of the inspection process.

1.1. Worn/Damaged Vibration Isolators

The first and most common problem to be discussed is looseness in the form of worn/damaged foundation vibration isolators. Many rotating machinery installations have vibration isolation mounts installed between the bedplate of the assembly and the foundation structure (typically I-beams or concrete).

1.1.1. Purpose of Isolators

To prevent the residual vibration due to operation of the rotating piece of machinery from being transmitted through the foundation to the surrounding structure and to other areas of the vessel where it might cause additional damage or discomfort.

1.1.2. Problem with Isolators

The problem occurs when, after a period of time, the isolators wear out and do not properly support/isolate the structure which they are designed to support.

1.1.3. Replacement of Isolators

During the machinery overhaul as well as normal in service operation, it is necessary to inspect these isolators for signs of wear/damage and replace them as necessary. In some cases, the Forced Draft Fans have been adjusted to operate with minimum vibration and the isolators have been replaced with steel pads due to continued problems with the isolators. The decreased level of vibration has allowed the removal of the isolators with no detrimental effect on the adjoining equipment and living spaces.
1.2. Structural Resonance Condition at Running Speed

The second most common problem encountered is a structural resonant condition and is often related to cracks/damage to the foundation (See Section 1.3). A structural resonant condition occurs when the natural frequency of the machine structure (and/or part of that structure) is equal to or close to the operating speed of that machine. The resulting condition is that vibration generated during normal operation by the rotating machinery is increased substantially by the structure’s natural tendency to vibrate at that frequency. Quite often, a resonant condition will increase the vibration at the resonant frequency by a factor of five (5) to ten (10) times acceptable limits.

1.2.1. Determining a Resonant Frequency

The resonant frequencies of a structure can be determined in basically two types of tests, the most common of which is the Impact Test for common machinery. This procedure involves striking the structure with a rubber mallet while measuring the resulting vibration of the structure. The machine is secured during this test.

If an Impact Test cannot be performed due to high levels of background vibration or if the structure of the machine is too large to excite, a coast-down or speed-up type test is then performed. This procedure involves measuring the vibration and phase at all operating speeds (i.e., during a coast down) and looking for major increases in vibration and phase shifts of the vibration, indicating the speed where a resonant condition exists.

Once the resonant frequency of the structure is determined, it is compared to the operating speeds of the machine. If the resonant frequency is equal to or close to (within 10%) of one or more of the operating speeds of the machine, modifications to the structure may be required to change the resonant frequency, reducing the vibration at operating speed.

1.2.2. De-tuning a Resonance Using Structural Modifications

One method of changing the resonant frequency (de-tuning) of a structure is to install stiffeners or additional structural supports to move the resonant frequency away from the machine running speed. In the case of the Forced Draft Fan, this type of modification has been accomplished on several vessels by adding diagonal stiffeners to the underside of the Forced Draft Fan foundation. The result can be a reduction in vibration levels at running speed that can often approach a factor of ten (10). It must
be understood that each structure is unique; therefore, identical structural changes to two (2) similar machines may not have the same de-tuning effect. The addition of mass to the structure can also be used to change the resonant frequency. In the case of most machinery, stiffening and/or the repair of cracks has proven to be the best approach.

1.3. Cracks/Damage to Foundations

1.1.1. The third issue relating to the foundation of typical rotating machinery is cracking/damage to the foundation, resulting in structural looseness. Foundation cracking/damage is typically caused by either excessive, prolonged vibration from Blower damage or looseness of the foundation due to machine base soft-foot conditions. Both conditions will cause increased levels of vibration. Although foundation cracking/damage is uncommon, it has been encountered in past surveys; therefore, it should be included as part of the overall inspection process.

1.4. Summary

For rotating machinery to operate reliably, the foundation must be in good condition without cracks, worn-out vibration isolators, damage, or major resonant conditions present.
2. BEARINGS & LUBRICANTS

2.1. Motor Bearing Selection

2.1.1. Greaseable vs. Non-Grease type Shielded Bearings

For those applications designed to have non-greaseable (i.e., sealed/shielded) type bearings installed, note that there is a significant difference between “Shielded” (2ZZ) and “Sealed” (2RS) type bearings. Many people mistake these terms to be one in the same. “Shielded” bearings are installed in applications to provide protection against the entrance of coarse dirt or particles without concern for retaining lubricants or keeping other fluids out. “Sealed” bearings on the other hand perform the same function as “Shielded” bearings in that they provide an effective barrier against the entrance of contaminants, but also contain an additional seal to prevent the loss of internal lubricants or entrance of external fluids.

2.1.2. Switching From Open/Greasable to Sealed/Shielded Bearings

Investigations regarding typical bearing failures have found:

Switching from open, greaseable type bearings to non-greaseable (i.e., sealed/shielded) bearings is not recommended. Past results of this change to a non-greaseable style bearing has been a substantially reduced bearing life due to dry-running bearings. This has been documented with Ultrasonic bearing analysis over the past five (5) years (see section 6.1.3). Non-greaseable type bearings are supposed to retain grease within the bearing throughout the life of the bearing. Unfortunately, this often does not happen and the result is a significant increase in the rate of bearing wear leading to premature failure once the lubricant has either dried up or broken down (i.e., oxidized).

Both “Sealed” and “Shielded” bearings are currently selected for various bearing applications. After consulting with FAFNIR, TORRINGTON & SKF, the technical advice was to switch from “Shielded” to the “Sealed” bearing type. It should again be noted that “Shielded” bearings, having no inside diameter “seal”, allows the oil from the grease base to leak out of the bearing during normal operation resulting in the bearing being under lubricated. The bearing nomenclature suffix designators, PP-Fafnir or 2RS-SKF, indicate the “Sealed” bearing style.
The engineering technical issue regarding the switch to the “Sealed” style bearings is that “Sealed” bearings run slightly hotter than “Shielded” bearings because of the contact of the rubber seal to the inside race. Consequently, a “Sealed” bearing has a lower rated operating speed due to increased heat at higher RPM. However, most electric motor applications (i.e., 1200 - 3600 RPM) are still well within the specified ranges for “Sealed” bearings.

**Example:**

- 6308 ZZ - Shielded Bearing = 7500 RPM MAX
- 6308 2RS - Sealed Bearing = 5000 RPM MAX

### 2.1.3. Bearing Life

Bearing manufacturers calculate bearing life expectancy, L10 rating, based on expected dynamic loads and service conditions. This life can be shortened if the bearing is installed in the wrong application, installed improperly, run too hot and/or under lubricated.

**Conclusion:** Use open, greaseable type bearings where originally installed and do not switch to non-greaseable (i.e., “Sealed”/“Shielded”) bearings. When installing non-greaseable type bearings, switch from the standard installed “Shielded” bearings to a “Sealed” style bearing.

### 2.1.4. Motor Bearing Fit/Tolerance/Clearance (C3) Issues

Bearing manufacturers conform to ISO standards for two types of dimensional specifications: Tolerance and Internal Clearance.

Tolerance defines the +/- radial dimensional range between the inside (bore) and outside (housing) diameters. Class 1 (ABEC-1) is the most common type produced; applications include electrical motors, pumps, etc. Special applications may require up to a Class 7 (ABEC-7) fit.

Internal clearance defines the radial clearance between moving and fixed components inside the bearing. Nomenclature is different for each manufacturer but the clearance ranges from small to large as described, for example, by: SKF C0-C1-C2-Normal-C3; and Fafnir R-T-H-P. Bearing installation with interference, as opposed to clearance, fits on the shaft or in a housing and elastically deforms the race and reduces the internal clearance. Thermal growth of bearing components will effectively reduce this clearance as well.
There have been premature bearing failures due to the improper selection of motor bearings based on Tolerance and Internal Clearance. It is possible that the normal installation of a bearing into an end bell may have slightly less than perpendicular dimensions relative to the shafting (therefore, the bearing would be installed mis-aligned/cocked on the shaft). If the bearing’s internal clearance is too small (i.e., C1, C2) when this condition exists, then premature bearing failure may result.

**Conclusion:** Specify a normal Internal Clearance, Normal or C3 (SKF), and a normal Tolerance, Class 1, for bearings unless operating conditions require selecting a special dimensional fit.
2.2. Pillow Block Bearing Selection

There are currently several different types of spherical roller bearings used in the pillow block bearing assemblies. The following sections summarize these bearing types and installations:

2.2.1. Types of Bearings

The two principal types of Spherical Roller bearings are the standard Tapered Bore and the Split Race type-bearing configuration. These bearings are made by a variety of manufacturers including: SKF, Torrington, Cooper and FAG. Both bearing types support radial and axial (when optionally provided) loads on the shaft.

2.2.1.1. Standard Spherical Roller Bearings

The standard Tapered Bore Spherical Roller bearing requires sliding the complete bearing and taper insert along the shaft from the open end of the shaft (See Appendix # 1, “Mounting Procedures for Spherical Roller Bearings” from the Torrington Engineering Manual). This procedure requires the coupling to be removed and may also require the motor to be rotated out of the way. Additionally, the Blower casing may also have to be split in order to allow the shaft/blower to be elevated to remove the bearings from the Pillow Block bearing housings. This procedure is very time consuming and should be planned accordingly.

2.2.1.2. Split Race Standard Spherical Roller Bearings

The Split Race Spherical Roller bearing configuration on the other hand allows both the inner and outer races to be split in half and assembled without having to slide the bearing completely down the shaft from the open end (i.e., coupling end). FAG manufactures a standard bearing insert for a typical FAG Pillow Block bearing assembly. The FAG Split Roller bearing is a unique precision product that must be properly clamped, torqued onto a clean, uniform shaft that is within specified limits and is adequately lubricated (See Appendix # 2, “Mounting Instructions for FAG Grease Lubricated Split Roller Bearings and Pillow Blocks”).
Cooper also manufactures a Split Race Spherical Roller bearing. This bearing utilizes the same principles of the FAG design, but incorporates a custom/unique Pillow Block. The issue here is that the Pillow Block height is usually not compatible with the numerous FAG Pillow Block type assemblies currently installed. Therefore, in order to use this type of bearing, spacer plates must be custom manufactured for each installation. Also, the thrust bearing insert is different than the floating/expansion bearing and must be ordered accordingly (See Appendix # 3, Cooper Bearing Installation).

2.2.2. Proper Bearing Selection

Steam Vessels currently have a combination of all the bearing types noted above. At this time, there is no preference for any particular type of bearing. Instead, it is the intent of this diagnostic review to identify differences and describe proper installation of these bearings for the greatest possible reliability.

2.3. Pillow Block Bearing Installation

Depending on the size of the bearing and type of bearing application, there are different methods for mounting either Tapered Bore or Split Race Spherical Roller bearings onto the shaft. The following sections highlight these issues:

2.3.1. Basic Rules to Follow


2.3.2. Shaft Mounting of Tapered Bore Spherical Roller Bearings

2.3.2.1. Overview

The basic principle of this bearing design is to utilize a tapered bore on the inside surface of the inner race with a matching taper sleeve adapter piece that is split axially to allow it to expand/contract. When the tapered adapter piece is placed on the blower shaft with the bearing’s tapered inner race riding over it, the spool piece can be expanded by pulling it axially under the bearing’s inner race. This is accomplished by adjusting a threaded ring on the taper sleeve adapter piece.
Although the fit of a Tapered Bore Spherical Roller bearing could be determined by measuring the distance the bearing is forced on the tapered seat, it is more practical to measure the reduction of the radial internal clearance (RIC) caused by the expansion of the inner ring. This procedure requires determining the initial roller-element to outer-race clearance before mounting, then checking the RIC during mounting until the proper reduction of the RIC has been accomplished. NOTE: Tightening of the tapered sleeve adapter piece adjusting ring must be made using a spanner, not a drift pin (See Appendix # 1, page 2, “Shaft Mounting Tapered Bore Spherical Roller Bearings” from the Torrington Engineering Manual).

2.3.3. Shaft Mounting of FAG - Split Race Spherical Roller Bearings

2.3.3.1. Overview

Unlike a tapered bore bearing, the Split Race design relies solely on the compression of the inner race to the shaft to hold it in place both radially as well as for thrust loading. The outer race is also split and compressed around the split set of rollers to adjust the bearing for correct fit and tightness when assembled.

For detailed installation instructions, see Appendix # 2, “Mounting Instructions for FAG Grease Lubricated Split Spherical Roller Bearings and Pillow Blocks”.

2.3.4. Looseness Concerns of Bearing Assemblies

2.3.4.1. Overview

Bearing looseness is a major issue that either is present at the time of installation or may develop due to wear caused by misalignment, imbalance or premature component failure. This condition can be detected by either Vibration Spectral monitoring or Ultrasonic Condition monitoring. These monitoring methods can be used in combination to detect the early stages and progression of this condition (See Section 6.0 - Predictive Maintenance).
2.3.4.2. Pillow Block Bearing Looseness

Pillow Block bearing internal looseness is the most common type of looseness associated with type of bearing. Since the installation of these bearings is a detailed process, a common mistake is to improperly set the reduction of the radial clearance (RIC) or fail to torque the races of the split race bearings to the correct specifications.

*Tapered Bore Spherical Roller Bearings:*

Often the bearing radial clearance is either too loose or too tight. Too loose a clearance will result in looseness that can be extreme enough to allow the blower shaft to move axially inside the bearing’s inner race (i.e., if it is installed as the thrust bearing with the required thrust ring) and result in the blower backing plate rubbing against the fan casing. This could result in catastrophic damage.

Too tight a clearance will result in overheating of the bearings, resulting in lubricant breakdown and eventual bearing failure. Again, this could be avoided by the proper setting of the RIC.

It is imperative to check the bearing RIC after it has been installed if looseness is suspected. By removing the pillow block upper cap, a feeler gage can be used to measure the current gap of the installed bearing.

*Note:* The bearing must be loaded downward while taking this reading. The coupling side bearing of an overhung fan installation will have to be pulled down (i.e., by use of strap/chain fall) since this bearing is normally loaded in the upward direction by the cantilever affect of the overhung arrangement.
Once the installed bearing radial clearance is known, it can be compared against the clearance of a new/spare bearing and then the RIC can then be calculated. Adjustment is then possible by use of a spanner (not a drift pin and hammer) to correct the tightness/fit of the tapered adapter (See Appendix # 1, page 2 - “Shaft Mounting Tapered Bore Spherical Roller Bearings” from the Torrington Engineering Manual).

**Split Race Spherical Roller Bearings:**

As noted previously, unlike a tapered bore bearing, the Split Race design relies solely on the compression of the inner race to the shaft to hold it in place both radially as well as for thrust loading. The outer race is then compressed to set the proper radial clearance to tolerance (i.e., adjusted based on torque values). Looseness can and will result if these tolerances are not set properly and can result in a catastrophic failure or if caught early just damage to the bearing. Again the checking of these clearances / torque’s could save a costly failure. Past experience from a fan with slight looseness present, as predetermined in the condition data, found a damaged bearing (i.e., plastic retainer ring broken) and improper torque on the outer race machine screws.

**2.3.4.3. Motor Bearing Looseness**

The most common cause of motor bearing looseness is End Bell (i.e., bearing housing) wear/damage that allows the bearing to move, spin and vibrate within the bearing housing. Extreme cases of looseness can cause the outer race to fix its position in a severely misaligned condition, leading to premature bearing failure. The solution to this problem is that motors with End Bell damage must be repaired with machined inserts or replaced.
2.4. **Greasing Issues & High Temperature Lubricants**

The bearings installed in the motor and pillow blocks of Forced Draft Fans operate in a high temperature environment under high dynamic loads. These bearings require consistent and adequate lubrication in order to achieve the designed L₁₀ life of the bearing. It is often debated as to how much grease the bearing needs and how often the bearing should be lubricated. Condition analysis of these bearings provides an effective tool for monitoring lubrication effectiveness and quality.

2.4.1. Under Lubricated/Cool Bearing

An under lubricated bearing usually runs cool but sounds rough (i.e., clicking, popping) when analyzed with Ultrasonic Testing (see Section - 6.0). Greasing immediately reduces the elevated ultrasonic emissions and clicking or popping sounds; but also elevates the operating temperature.

Since the bearing was designed to operate at a specified temperature (accounting for internal, radial clearance reduction relative to thermal growth) a cool, under lubricated, bearing runs with elevated vibration which is the result of internal bearing looseness. Normally, greasing the bearing reduces overall vibration levels.

To further document this circumstance, a major Japanese motor manufacturer states the following on its motor nameplate:

“When re-filling during running, using the following guide, fill new grease by opening grease outlet cap/cover and then shutting it after discharging used grease”.

Re-greasing Interval = Approx. 2000 hours.
Re-greasing Quantity = Approx. 50 grams.

They go on to note the following:

“Operate after filling grease quantity above in case of test working or if stopped more than two months. Don’t mix grease with lithium base”.

In summary, the motor manufacturer is saying to re-grease the bearings while the unit is running with the proper quantity of grease at least four (4) times per year (i.e., quarterly). They also note that after extended periods of shutdown (i.e., 2 months or approx. 1000 hrs), the bearings should be re-lubricated at start-up.
SKF goes into further detail to discuss re-lubricating hours for the Spherical Roller Bearing installed in pillow block bearings (See Appendix #4, “Lubrication” from the SKF Bearing Installation and Maintenance Guide - Figure 15). This table addresses the increased re-lubrication rates for Spherical Roller bearings. As noted, an 1800 RPM, 60 mm diameter bearing requires greasing every 1500 hours when mounted horizontally or 750 hours if mounted vertically. This assumes that the bearing is operating below 160 °F and a “good” quality lithium based grease is used.

The review also discusses re-lubrication procedures for removing the old “used” grease. Here SKF advises the following:

“If the normal re-lubrication interval is shorter than six (6) months, then it is recommended that the grease fill in the bearing be replenished (topped up) at intervals corresponding to [0.5 x re-lubrication interval]; and the complete grease fill should be replaced after three (3) replenishments.”

“If the normal re-lubrication interval is longer than six (6) months, then it is recommended that the grease fill in the bearing be completely removed and replaced with fresh grease.”

2.4.2. Frequency & Amount of Lubricant

Based on typical manufacturer recommendations and assuming the unit is running 8000 hours/year, re-greasing of the spherical roller bearings should be performed at least every 1500 hours and replenished at 750 hours. SKF also addresses replenishment as follows: “by adding small quantities of fresh grease at regular intervals”

The problem with this lubrication cycle is that it produces a condition where the bearings are possibly over lubricated (i.e., at time of greasing - 50 grams) and then under lubricated between the replenishment intervals. This method or type of greasing has been called “spike” lubrication by one of the automatic lubricator manufacturers. One method to deal with this “spike” in lubrication times/levels in the bearing is to provide a steady flow of fresh grease to the bearings by use of an automatic lubricator. An automatic lubricator is simply a container of grease that is mounted either on or next to the bearing and provides a constant supply of lubricant via the grease fitting to the bearing.
2.4.3. Automatic Lubricators

The automatic lubricator uses either a spring, screw assembly or gas charge device that pushes a piston which in turn releases grease out of the reservoir, into the bearing, at an adjustable, metered rate. The gas charge type unit typically works better over longer lubrication periods since the unit does not depend on spring tension and can be set for a controlled flow rate for up to one (1) year. The Electro-Luber™ and PERMA™ is discussed here in detail since it has been field-tested and shown excellent performance on the pillow block bearings.

2.4.3.1. Electro-Luber™ Working Principles

The components of this dispenser include time selector switches, two (2) 1.5 volt batteries, electrochemical cell and electrolyte, gas chamber, piston, lube reservoir, all mounted inside the 4.5 ounce cylinder and mounting nipple.

When one of the selector switches is closed, an electrochemical reactor cell is activated, and the resulting reaction takes place by which electrical energy is converted into nitrogen gas. As the gas is produced, internal pressure builds up, which is applied against a piston to force the lubricant out of the reservoir into the lube point. The amount of electrical current determines the volume of gas produced, which in turn, controls the rate of lubricant flow (see Appendix # 5 - The Electro-Luber™ Dispenser).

2.4.3.2. Electro-Luber™ - Grease Rates for Proper Lubrication

The Electro-Luber™ Dispenser has recommended lubrication rates based on shaft diameters. Since most Forced Draft Fans have shaft diameters that are within the range of 49 to 62 mm., the setting for this size range would be 6 months to dispense 4.5 ounces of grease (see Appendix # 5, page 8 - Lube Dispensing, Roller & Ball Bearings). Since the temperature around the fan often approaches 120 °F, the rate may be slightly higher (see Appendix # 5, page 9 - Adjustment for Temperature Variations).
2.4.4. Typical Operating Temperatures (High Temperature Concerns)

Since the motor and pillow block spherical roller bearings operate in a high temperature environment (i.e., 115-120°F), bearing cap temperatures may often approach 140-150°F with internal temperatures even higher. The combination of a high temperature environment with long periods between lubrication has typically resulted in break-down/oxidation of the grease. This state of oxidation, and associated lubrication quality degradation, can be recognized by the very dark/black color. Note: Condition surveys have revealed that the dark/black color of the grease is primarily the result of oxidation and not dirt/soot entering the bearing.

To deal with the high temperature environment, increase the hand greasing interval as required for the size/diameter of shaft and clean the grease completely out of the bearing every third (3rd) “re-greasing interval” as recommended by SKF (See Appendix # 4, page 2 - Re-lubrication Procedures). Grease renewal should also be performed if the grease becomes dark/black when using an automatic lubricator.

If lubricant break-down/oxidation continues to be a problem, switch to a synthetic, high temperature lubricant that is rated at operating temperatures of at least 300 °C. Synthetic greases also have ratings for Shear Stability, Resistance to Water, and Extreme Pressure Performance and Wear Protection (See Appendix 6 - ThixoGrease®/Sample of Tests). ThixoGrease® has performed extremely well in both motor and pillow block bearings and has maintained proper lubrication in environments where other greases have broken down, causing catastrophic bearing failure.
3. COUPLINGS & LUBRICANTS

3.1. Coupling Assembly & Proper Lubricants

3.1.1. Setting Axial Coupling Clearances

The axial clearance of a coupling is the distance that separates the two (2) coupling halves when installed on their respective shafts. Couplings are designed to allow a small amount of axial movement of one (1) or both of the shafts due to the forces on the shafts during operation from the presence of misalignment and/or thermal expansion. The manufacturer’s installation instructions state the minimum and maximum allowable axial clearance dimensions.

3.1.1.1. Minimum Allowable Dimensions

If the axial clearance is smaller than the minimum allowable dimension for a specific coupling, the possibility exists that the coupling halves will be pushed together (“lock-up”) during operation. As a Forced Draft Fan operates, the various parts absorb heat and expand slightly. The blower dynamic forces of operation will also tend to shift the blower shaft axially toward the coupling as the machine comes up to speed. If the coupling axial clearance is insufficient, they will be forced together, causing the coupling halves to rub. This condition will also place additional loads on the motor bearings. Both these problems will lead to accelerated wear and premature failure of the bearings.

3.1.1.2. Maximum Allowable Dimensions

If the axial clearance is larger than the maximum allowable dimension for a specific coupling, assembly of the coupling will be extremely difficult and the normally installed dental (i.e., gear) type coupling will not have the proper gear tooth contact. This condition can result in excessive wear and/or a failure that will cause the coupling teeth to fail leaving the coupling halves to spin freely (see Section - 4.9.1).

**Note:** Installation of jacking bolts in the axial direction on the Motor feet will allow quick and accurate adjustment of the coupling axial clearance during the alignment process.
3.1.2. Selection of Proper Lubricants

Due to high loads, high temperatures and the relatively dirty environment in which these couplings are installed, it is highly recommended that an EP-synthetic grease be used to lubricate the couplings. Ordinary lithium based greases do not provide adequate lubrication to the coupling over the extended periods of time in which the Forced Draft Fans are operated. Only the synthetic greases have the necessary capability to prevent thermal breakdown (i.e., oxidation) during the harsh operating condition typical of Forced Draft Fans. Almost all manufacturers of lubricants make an EP-synthetic grease which can be used in the Forced Draft Fan couplings. These greases are slightly more expensive but are much cheaper than the cost of replacing either the coupling or the entire fan assembly and the associated downtime.

3.1.3. Offset of Coupling Keys

Condition monitoring data review has shown that a common cause of vibration in the Forced Draft Fans is the imbalance of the coupling assembly/keys. The keys used in the couplings for Forced Draft Fans are relatively large compared to the coupling and a key is installed in each coupling half. If the coupling is assembled with both keys in line (i.e., on the same side) with each other or even relatively close to each other, an imbalance condition will result which may elevate the running speed vibration above acceptable limits. The way to prevent this is to install the coupling with the keys offset at the 180-degree position (i.e., keys on opposite sides). This effectively balances out the weights of the imbalance forces due to the rotating mass of the keys about the shaft centerline. The result will be considerably smoother operation of the Forced Draft Fan and the best reference point to begin dynamic balancing if still required.

3.1.4. Proper Alignment of Shafting/Coupling

Procedures and tolerances for machinery alignment as noted in Section 4.0 should be followed. Often coupling manufacturers attempt to give tolerances that allow greater amounts of misalignment as a marketing tool to sell their coupling. The coupling may have been designed to take the additional wear and tear that will be produced. The problem with this is that the excessive forces and vibration caused by poor alignment tolerances damage other internal components of the machine (i.e., seals, bearings, etc.) thereby increasing the maintenance costs.
4. MACHINERY ALIGNMENT PROCEDURES

4.1. Machinery Shaft Alignment

The definition of “perfect” Machinery Shaft Alignment is as follows:

“Positioning of two (2) or more machines so that their rotating shaft centerlines are co-linear at the coupling center under operating conditions”.

Attaining this type of “perfect” alignment in the real world is unrealistic. Therefore, tolerances have been developed to standardize the amount of allowable misalignment. This will be discussed in detail in Section 4.9.

![Figure 1. Example of Misalignment](image)

In Figure 1 above, shaft misalignment is apparent in the vertical plane. Misalignment could also be present in the horizontal plane.

Misalignment causes excess bearing loads which destroy the critical parts of the machine (bearings, seals, couplings, etc.). Failures of this type are usually evident; however, they are often mistakenly identified as “normal wear and tear”. To decrease the chance of failures, it is important to realign your equipment to standard acceptable tolerances. High quality craftsmanship is necessary, but high quality does not mean harder work. It means smarter work.

In summary, why perform precision shaft alignments?

To:
- Reduce vibration and noise
- Reduce bearing, coupling and/or seal wear/damage
- Reduce maintenance costs and downtime
- Save money
There are three types of misalignment which affect coupled rotating machinery.

4.1.1. Angular Misalignment

Angular misalignment is a condition in which the centerline of one (1) machine forms an angle with the shaft centerline of another machine.

4.1.2. Offset Misalignment

Offset misalignment is a condition where the centerline of one (1) machine shaft is displaced some distance from the centerline of another machine at the point where both shaft centerlines meet at the coupling.

4.1.3. Combination Angular and Offset Misalignment

This is where a combination of both angular and offset misalignment are present. It is the most common form of machinery shaft misalignment and therefore a common starting point to all alignment work.

Even today, much machinery alignment work is done by trial and error with old and out dated methods. While some of these methods may eventually produce acceptable alignment results, they can be very inaccurate and extremely time consuming. As a result of the time consuming nature of this work, machinery alignment is occasionally considered to be “good enough” when it actually is not even close.

It is often said that a machinist or engineer who does a good alignment job in a reasonable amount of time has a “feel” for alignment. This comment suggests that alignment is an “art”, not a “science”. However, a good machinist or engineer must have a basic understanding of the mathematical principles involved in the process and the proper tools to accomplish the task.
4.2. Instrumentation

Two (2) types of measuring tools and instruments are in common use that we can define as “Approximate or Rough” and “Precision”:

“Approximate” measuring tools include the use of a straight edge, taper gauge, and feeler gauge. When used during the alignment process, their values are affected by: a) smoothness of coupling surfaces, b) trueness of coupling bore (concentricity and angularity) and c) differences in the outside diameter (OD) of the coupling halves.

Because many couplings installed onboard a vessel will not be perfectly machined, it is best to assume the alignments made with these tools will not achieve acceptable results. Instead, these tools can be used to “rough align” the machinery before the “Precision” measuring instrumentation is required for the final alignment.

“Precision” measuring instrumentation includes the various types of dial indicators (both mechanical/electronic) and laser-optical measurement devices. Accurate readings can be obtained regardless of the condition, trueness, matching, and/or spacing of the coupling halves. The only disadvantage versus using the “approximate” measuring instrumentation is the amount of time needed for initial set-up and the added cost of the instruments. However, the initial set-up time is actually less, in most applications, because the flexible coupling does not have to be disassembled and the accuracy of the readings makes the alignment job easier, much faster and more accurate.

4.3. The “Reverse Indicator” Alignment Method

Two (2) common methods for aligning machinery with dial indicators are as follows:

1) “Rim & Face” where dial indicator readings are taken at the coupling horizontal and vertical surfaces. Any axial movement in the shafts will cause error in alignment calculations. Some axial shaft movement is usually present, so this method is best avoided.

2) “Reverse Indicator” method which requires two (2) sets of rim readings; one (1) set each at the “Machine To Be Moved” (MTBM) and the “Stationary Machine” (SM) sides of the coupling. The shafts are then rotated together to acquire the radial dial readings. On most typical flexible coupling installations, the motor is the MTBM and the blower or pump shaft bearings make up the SM.
Using the “Reverse Indicator” method, it is usually possible to attach two (2) dial indicators to the machinery in such a manner that both sets of readings can be taken simultaneously. If only one (1) dial indicator can be attached at a time, it is permissible to take one (1) set of readings and then change the mounting arrangement and take the other required set of readings. The most common alignment mistake made when using dial indicators is attempting to align or check the alignment of two (2) pieces of machinery by taking a rim reading on only one (1) of the machines. This procedure provides only half the information needed for the “Reverse Indicator” method. It is possible to get a single set of rim readings that appear to be “perfect” on a machine that is still severely misaligned.

Typical “Reverse Indicator” systems include mounting hardware that consists of the brackets, posts, connectors, and other hardware used to properly attach the dial indicators to a shaft of the machine. The components are specifically designed for obtaining dial indicator rim readings for rotating machinery alignment.

An alignment computer, a calculator-like machine, is also provided as part of the “Reverse Indicator” alignment system to reduce the work required calculating the necessary alignment corrections. These results can be obtained using standard graph paper and a ruler but the process adds considerable time to the job and requires more operator training. This graphical method to obtain the alignment results will not be covered in this document.

Typical “Reverse Indicator” Alignment Computer System Overview:

A high quality “Reverse Indicator” alignment system with a computer and alignment data sheets (See Appendix # 7 - Horizontal and Vertical Shaft Alignment Worksheets) would allow the user to select either the Vertical or Horizontal alignment methods. (Note: Both vertical and horizontal capabilities are extremely important onboard a ship due to the rather large amount of vertical machinery installed in order to conserve space). The Alignment computer menu “prompting” procedures are built into the computer program to prevent entering data that is not applicable to the alignment method chosen. All required dial readings and distances are requested and the final calculations are automatically performed with the required alignment corrections (i.e., shimming or horizontal movement) shown on the screen.
Most important is that the alignment computer also calculates the alignment condition after each move by displaying the **offset** and **angularity** at the coupling center values so that alignment tolerances may be checked to ensure the job is done to the proper allowable limits. It is important to note that the final alignment will not be “perfect” but rather be within the acceptable range of the tolerances based on machine speed (see Section 4.9).

The alignment computer/program is designed for use with precision measuring instruments and available mounting hardware. There are more expensive alignment computers available with ports to attach electronic dial indicators or even laser-optical sensors which virtually eliminate indicator reading errors and provide even greater alignment accuracy and quicker results.
4.4. Laser-Optical Alignment Systems

The use of a laser-optical alignment system to align a machine accomplishes shaft alignment by the same numerical method used with dial indicators utilizing the “Reverse Indicator” setup. The major benefit to this method is that the system is usually easier to set up, faster, and more accurate since the readings are actively acquired and directly input to the computer via cables without mechanical errors. Laser-optical alignment systems are therefore much more expensive than a simple dial indicator based “Reverse Indicator” alignment system but can often be financially justified because of the decreased alignment time and increased accuracy.

Laser alignment systems consist of an alignment computer, a laser sender, laser receiver, bracket assemblies, and the associated wiring. Depending upon the manufacturer of the alignment system, the sender and receiver may be packaged in either two (2) separate housings, or mounted one (1) above the other in the same housing using a prism to reflect the laser beam from the sender across the coupling, into/out of the prism and back to the receiver, as shown above in Figure 2. Both configurations of laser sender and receiver are equally accurate. The best configuration depends upon the intended applications as well as the size constraints of the machinery to be aligned.

As with the “Reverse Indicator” bracketing, the laser-optical bracket assemblies supplied with the alignment system allow personnel to determine the alignment condition of the machine without the disassembly of the coupling. Bracketing assemblies rigidly attach the laser and receiver to the shafts. When both shafts are rotated together, the laser and receiver make concentric circles around the shaft centerlines.

Figure 2. Typical Laser-Optical System Mounting
With this method, any misalignment present between the two (2) shafts will be detected by the changing position of the laser beam on the laser receiver.

One benefit to this method is that there is no longer any need to watch the dial indicator as the shafts are rotated. The computer displays the values of misalignment based on the degree of rotation. Errors in reading dial indicators are no longer a problem. One drawback with this method is that the user will have a harder time visualizing the position of the motor relative to the blower. However, the computer will graphically display the alignment condition based on offset and angularity at the coupling center and the required movement at the motor feet.

4.5. **Shaft Alignment vs. Coupling Alignment**

Recall that the rotating centerlines of the two (2) shafts must form one (1) continuous straight line, at normally operating conditions, for proper “Shaft Alignment”. Although it is often referred to as “Coupling Alignment” (i.e., aligning the two coupling halves), the true intent is to make sure that the shafts are aligned, and not necessarily the coupling halves.

If all couplings were bored straight/true through their exact center or machined perfectly about their Rim and Face, it would be possible to “align the two (2) coupling halves” and obtain correct machinery shaft alignment. However, irregularities in the machining process and even the forces imposed on the coupling halves during installation (especially with “shrink fit” couplings that must be heated during installation) leave a significant concentricity error. Therefore, “coupling alignments” should be avoided. This is not to say that the dial indicators should not be placed on the coupling halves to obtain measurements; but rather, that the two (2) shafts must be rotated simultaneously to obtain the desired readings. When both shafts are rotated together, the couplings actually become an extension of the shaft centerlines (i.e., assuming they are not loose on the shaft) and irregularities do not affect the readings.

Another advantage of “shaft alignment” versus “coupling alignment” is that by aligning the two (2) shafts rather than the coupling halves, the need to eliminate all shaft run-out is not required. A shaft which is bent slightly outside the bearings (i.e., 0.005” run-out) can still be properly aligned using “shaft alignment” but cannot when using “coupling alignment” in an emergency situation.
4.6. **Proper Movement/Shimming Methods for the MTBM**

The standard practice for adjusting the position of a machine to be moved (MTBM), requires that the vertical position of this unit be adjusted first by adding or subtracting shims. Once the motor is at the correct vertical height, it can then be moved in the horizontal plane with minimal changes to the vertical plane alignment.

During this vertical adjustment process, shims are added or removed from under the mounting feet of the motor (MTBM). It is strongly recommended that pre-cut, stainless steel shims be used because they maintain their precision thickness due to minimal compressibility and will not corrode and deteriorate. Additionally and most importantly, these precut shims are cut to precise thickness and marked accordingly. This simplifies the alignment process and saves valuable time over hand cutting of rolled shim stock.

Since the precut, stainless shims are available in thickness up to 0.125", it is possible to limit the number of shims under each foot of the motor to **no more than three (3) shims**. This will further reduce the possibility of soft-foot and associated looseness (see Section 4.7).

**Note 1**: Shims hand-made from stainless steel sheets, laminate, brass, tin, etc., should only be used in emergency situations when pre-cut stainless steel shims are not available. These substitute shims should be replaced at the earliest convenience to avoid unnecessary soft-foot, looseness, and the potential resulting damage.

**Note 2**: Vertical lifting of the motor for shimming can be easily accomplished by using standard wedges under one side of the motor feet. In this case it is best to leave the opposite side foundation bolts snug so that the machine does not move in that direction. Horizontally oriented jacking bolts would also prevent this movement and are preferred because they make horizontal movement easier.

4.7. **Soft-Foot**

“Soft-foot” is the condition that exists when all four (4) feet of the machine are not supporting the weight of the machine. This condition will make precision alignment impossible since tightening the bolts at the motor feet will cause the machine to move. Uncorrected soft-foot is probably the largest cause of frustration and lost time in performing shaft alignment. More importantly, soft-foot conditions will elastically deform the motor frame and bearing housings; and the resulting stresses will cause premature bearing failure.
When soft-foot is present, the dial indicator or laser-optical readings can be different each time the hold-down bolts are tightened, loosened, and re-tightened. This is very frustrating because attempted alignment corrections may not produce the results desired until the soft-foot is corrected.

4.7.1. Basis for Understanding and Correcting Soft-Foot

The basis for understanding and correcting soft-foot is the knowledge that three (3) points determine a plane. In the case of a chair, the floor is the “plane” and the bottom tips of the legs are the four “points”. Three (3) tips will always rest on the floor, even if a person is sitting with his weight positioned above the short leg (the short leg will then be on the floor and the normal leg which is diagonally opposite the short leg will be off the floor).

By using this example, it can easily be seen that when a machine is initially placed on its base, it will often be resting on three (3) of its support feet unless the foundation and bottom of the motor feet are perfectly machined which is very unlikely. Also, because the feet of the machine are actually square pads (not true points) it is possible that the machine will be resting on only two (2) support feet that are diagonally opposite each other. In this case, the machine feet will have two (2) soft-feet.

To correct a soft-foot condition, all four (4) feet should be tightened then one foot loosened at a time. A dial indicator should be used to measure the rise of each foot and shimmed accordingly to correct the softness as required. In cases of extreme soft-foot (i.e., > 10 mils), the corrective shims should normally be split under the diagonally opposite foot (i.e., 5 mils and 5 mils respectively).

4.8. Proper Horizontal Movement of the MTBM

4.8.1. Jacking Bolts

A jacking bolt is a combination of a bolt and nut, where the nut is fixed to the machinery foundation, typically via a welded bracket arrangement. Ideally, a jacking bolt is positioned adjacent to each motor foot, facilitating the horizontal plane movement of the motor during the alignment process. It is estimated that at least one (1) man-hour of work can be saved on every alignment job if jacking bolts are properly installed.
The use of jacking bolts allow personnel to make minor changes to both the side-to-side position and coupling end gap adjustment in a controlled manner. Monitoring the horizontal movement of a machine is easily accomplished with the use of dial indicators attached to a magnetic base or by using a feeler gage between the jacking bolt and foot to measure the required movement.

The use of sledge hammers, portable jacks, and pry bars is not recommended for machinery horizontal position changes due to difficulty in making the required small movements for precision alignment. Moreover, excessive applied force could cause damage to the machine.

4.8.2. Dowel Pins

It is a common misconception that dowel pins prevent a machine from moving and can actually hold it in place if the foundation bolts become loose. Field testing has shown that with jacking bolts installed it is often possible to move the motor as much as .030-.040" with the dowel pins installed and the hold-down bolts slightly loose (this is not possible with the bolts tight). Since this amount of movement is unacceptable for precision alignment, we must conclude that the hold-down bolts secure the machine in place, and the dowel pins should not be relied on for this purpose.

The actual purpose of dowel pins is to keep track of the current and future alignment position of the motor relative to the blower shafts. Incorrectly, this often results in the motor being re-aligned (in the horizontal plane) using the dowel pin position holes rather than actually performing a new alignment.

Another problem that dowel pins create is that the motor feet and foundation mounting pads have so many holes that they both may need to be replaced. This condition will not occur if the dowel pins are re-installed properly into re-reamed holes.

In summary, when tapered dowel pins are re-installed after the alignment is complete, they should be put back into the same hole in the motor foot and reamed with a tapered reamer large enough to give full contact along the tapered pin. Placing the tapered reamer into a slow speed drill works very well for this task. Note that grease works best for lubrication while reaming. Apply anti-seize compound to the dowel pin prior to final installation.
Note: DO NOT HAMMER THE DOWEL PINS INTO THE REAMED HOLES. JUST TAP LIGHTLY TO ALLOW EASY, FUTURE REMOVAL.

If a dowel pin has been hammered into a machine foot it may take well over one (1) hour to get the pin out. In extreme cases, the pin will not be able to be removed and have to be hand cut between the bottom of the foot and the foundation pad using a hacksaw blade. This may also be required if a dowel pin has been driven into a blind hole. In this case, it is not possible to drive the pin out from the under side of the foot with a punch pin. If vise grips fail to twist/lift the pin up, then the only option is to cut the pin by hand.

The best way to secure the machine, in addition to the foundation bolts, is to tighten the horizontal jacking bolts at the conclusion of the alignment process.

4.9. Alignment Tolerances

Perfection may be possible under laboratory conditions but it is virtually impossible to obtain under conditions found in the field. Therefore, some misalignment must be permitted in normal field applications. This permissible misalignment is noted in alignment tolerance Table 1, p.33.

4.9.1. Angularity and Offset

Before the present method of establishing tolerances based on “Angularity” and “Offset” (at the coupling centerline), tolerances were usually stated in terms of “dial indicator readings”. The problem with this old method is that dial indicator readings are dependent upon the location at which they are taken. Because the exact location at which a person is going to take a reading is usually unknown, clarity and standardization with the old method is impossible. “Angularity” and “Offset”, as defined, standardize the specifications and tolerances because they accurately describe the position of the shaft centerlines relative to each other at a specific position. These are calculated values and cannot be directly measured.

Offset values are stated in “Thousandths of an Inch (Mils)”. Angularity values are stated in “Thousandths of an Inch (Mils) per 10” inches” of coupling diameter.
Table 1. Shaft Alignment Tolerance

<table>
<thead>
<tr>
<th>RPM</th>
<th>OFFSET (mils)</th>
<th>ANGULARITY (mils/10&quot; diameter)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Excellent</td>
<td>Good</td>
</tr>
<tr>
<td>600</td>
<td>5.0</td>
<td>9.0</td>
</tr>
<tr>
<td>900</td>
<td>3.0</td>
<td>6.0</td>
</tr>
<tr>
<td>1200</td>
<td>2.5</td>
<td>4.0</td>
</tr>
<tr>
<td>1800</td>
<td>2.0</td>
<td>3.0</td>
</tr>
<tr>
<td>3600</td>
<td>1.0</td>
<td>1.5</td>
</tr>
<tr>
<td>7200</td>
<td>0.5</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Soft Foot - Max. - 2.0 mils for all speeds @ each Foot.

Table 2. Coupling Axial Clearance

<table>
<thead>
<tr>
<th>SHAFT DIAMETER</th>
<th>MIN. (approx.) CLEARANCE</th>
<th>MAX. (approx.) CLEARANCE</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt; 2</td>
<td>1/8</td>
<td>1/4</td>
</tr>
<tr>
<td>2 - 3 1/4</td>
<td>1/8</td>
<td>3/8</td>
</tr>
<tr>
<td>3 3/8 - 9 1/2</td>
<td>1/4</td>
<td>1/2</td>
</tr>
<tr>
<td>9 5/8 - 12 1/2</td>
<td>3/8</td>
<td>5/8</td>
</tr>
<tr>
<td>12 5/8 - 16</td>
<td>1/2</td>
<td>3/4</td>
</tr>
</tbody>
</table>

Note: The actual coupling manufacturer's axial clearance specifications should be used.
5. Fan/Blower Balanced

5.1. Basic Understanding of Unbalance Forces in Rotors

In order to discuss balancing a fan rotor in general, one must first understand the basics of unbalance. Rotor unbalance is usually due to several sources, rather than just one, acting at several locations and/or planes on the rotor. For example, in one (1) plane, the unbalance may be due to a crack in the rotor or a piece of a blade missing. At another plane/location along the rotor axis, unbalance may be due to weights that have fallen off the rotor since the last balance job. And, another possibility of unbalance may be due to the coupling and its key position relative to each half of the coupling.

The end result is that the sum of all the unbalance forces can be resolved into an equivalent amount of unbalanced weight acting at a distance from the rotor’s centerline, as shown in Figure 4. Therefore, to be perfectly balanced, the center of mass of the rotor would be positioned at the center of rotation for a simple, single plane solution.

Unbalance units are expressed as the product of the unbalanced weight times the radius at which it is acting. Standard units would be ounce-inches (oz • in); Metric units are in gram-millimeters (g • mm). Often gram units will be used rather than ounce units since 28.35 grams = 1 ounce; thereby allowing much more sensitive weight units and easier numbers to work with. Consider that as long as a constant radius is used for trial/balance weight attachment, the radius can be dropped from the unit calculations since it just introduces a constant into the calculations.

Note: Pure unbalance is seen as an increase in vibration at exactly 1.0 x Shaft Speed (SS).
5.2. **Approved Method for Dynamic Balancing**

Current technologies allow accurate and much faster balancing utilizing a balance method known as the **Influence Coefficient Method (ICM)**. Older methods such as the “Four (4) Run Method” should no longer be used due to substantially less accurate results and increased job completion time.

During the start of the ICM balancing process, initial vibration (i.e., filtered at $1.0 \times$ Shaft Speed) and phase/direction measurements are acquired using a vibration analyzer, vibration pickup (accelerometer), phototachometer, and data-sheets (See Appendix # 8 - Rotor Balancing Report). Then, a trial weight is added to the rotor (to significantly influence the vibration), and a second set of balance data is acquired. The balance program then performs a vector summation of all the unbalance forces acting at the chosen measurement points/planes and calculates the correction weight for mounting at a specified rotational angle, at a chosen distance from the centerline. Note that the rotor is marked/indexed from 0 to 360 degrees and the phototachometer is referenced to the 0-degree position. The trial and also the final balance weights are attached to the rotor as determined by the balancing calculation (Example: 20 grams @ 180 degrees). While vibration values are normally discussed in velocity units, balancing vibration data is typically expressed in displacement, either Standard units (mils, \(0.001\)” =1 mil) or Metric units (mm).

5.3. **Cleaning and Inspection for Damage/Cracks Prior to Balancing**

The first task that must be performed prior to balancing a Forced Draft Fan is to water-wash the blower and then inspect the rotating element, blower housing, and foundation for signs of damage and/or cracks. An inspection of the housing and foundation can be done with the Fan in operation but the machine must be stopped, locked-out, and tagged in order to inspect the blower.

A visual inspection of the blower housing should include all the stiffeners that hold it in place, the blower housing foundation, and the areas around all bolt holes. Also, check each hold-down bolt for tightness.

Once the blower housing/foundation has been properly inspected, ensure that the blower is tagged out and not rotating. Then, remove the blower casing access panel.

Spin the blower slowly by hand while inspecting the blower blades for signs of cracks and/or damage. Inspect the volute plate and the backing plate for signs of cracks, paying particular attention to the areas on the
backing plate where balance weights may be installed. These areas are more susceptible to cracks/damage due to additional stress at these locations.

If there are balance weights mounted directly on the blower blades, make a note of their positions. The weights mounted on the blower blades should be removed (i.e., iterated off) during the balancing procedure (see Section 5.5).

5.4. Other Inspections/Checks Prior to Balancing

Since the ICM of balancing requires a filtered vibration reading at 1 x Shaft Speed (SS), other possible sources of vibration that may amplify the 1 x SS vibration must also be eliminated. The other major installation-related conditions that can increase 1 x SS vibration are misalignment, resonant conditions, and looseness. Therefore, a proper alignment and soft-foot check should also be performed with corrections as required.

The coupling should also be inspected to ensure that the coupling keys are indexed to be 180 degrees out of phase (i.e., opposite sides) to eliminate induced key unbalance.

A structural resonance test should be performed if a resonant condition is suspected. Structural modifications may be required prior to successful balancing.

5.5. Shop Balancing vs. In-Place / Field Trim Balancing

Prior to re-assembly of a blower rotor onto a fan shaft, the blower should have been shop balanced ashore prior to returning to the vessel. It must be noted that this procedure will balance the rotor at a much slower RPM, well under the normal operating speed of the blower. Typically this procedure is limited to less than 600 rpm and may even be slower depending on the size and weight of the rotor being balanced. The shop balance machine typically tries to incorporate balance specifications that are referenced to the RPM at which the balance takes place in order to estimate the resultant vibration that would be measured at full operating speed. Unfortunately, this slow-speed balancing is rarely good enough for the typical higher operating speed at 1800 rpm.

Consequently, final trim balancing of the blower, after it has been mounted on its respective shafting, will have to be performed in-place at full operating RPM. Never count on the shore-side shop balance to reduce the normal operating vibrations to within acceptable limits. In extreme/high vibration conditions, low speed balancing in-place may need to be performed first, prior to the final balancing at high speed (1800 rpm).
5.6. **Method of Balance Weight Attachment**

The backing plate of the blower is the best place to attach weights for balancing the blower. The backing plate of the blower is approximately twice as thick as the volute plate and the blower blades. The added thickness gives the backing plate the strength necessary to support the weights on the blower during operation and resist cracking and/or other damage. It may be necessary to attach weights to actual blower blades in special circumstances. In this case, it is best to use custom clamp-on (u-shaped) weights with set screws for securing to the blade, on the **leading edge** of the blade. It is not recommended to attach this type of weight to the trailing edge of the blade, even for temporary trial weight runs.

If weights have previously been attached to the blower fan blades, they should be removed during the balancing process by iterating them off one at a time. To accomplish this, remove the weight on a blade by telling the balancing software that a trial weight had been attached 180 degrees offset (opposite side) from that location. The program will then calculate the required weight and location to reduce the vibration amplitude. This process will limit vibration levels during balancing to avoid extremely high levels and potential rotor damage from removing all the old weights at one time.

5.6.1. **Important Rules to Remember**

1. Do not remove balance weights from the fan blades without adding/removing weight to compensate for the change (i.e., reduce the vibration again).
2. Only add or remove one (1) weight at a time unless the balance instructions specifically state otherwise.
3. Severe damage to machinery and danger to personnel may result from operating a severely unbalanced Forced Draft Fan. In this case, slow speed balancing may be required prior to balancing at high speed.
5.7. In-Place Balance Specifications

Balance standards are typically based on allowable unbalance and are given in oz\(\times\)in per pound of rotor weight or equivalent metric units. This type of standard does not take into account the rather weak type of foundation structure (i.e., steel rails rather than concrete) that a shipboard Forced Draft Fan is mounted on, the bearing type, or bearing configuration.

Since the overhung fan configuration has proven to be much less stiff and rugged (as measured by vibration condition readings) compared to a center supported blower installation, we must adopt balancing acceptance standards that correlate to this over-hung type of installation.

Every balance job ends up being a compromise between perfection and reality that is ultimately governed by economics and technical practicality. Therefore, we will use the vibration readings acquired on the bearing housings as the pass/fail criteria for the fan installation. The following simplified tolerance guideline is based on actual field balancing experience as well as years of periodic vibration data:

5.7.1. Motor Balance

The motor should be shop balanced to produce vibration levels measured on the motor bearings that are well below 0.10 in/sec (Pk) velocity when run uncoupled from the fan shafting. Other causes of vibration, such as foundation structural resonance and/or looseness, may be given special consideration.

5.7.2. Fan/Blower Rotor Balance

The fan should be field balanced “in-place” to produce vibration levels measured on the pillow block bearings that are no higher than 0.20 - 0.25 in/sec (Pk) either radially or axially. Other causes of vibration, such as foundation structural resonance and/or looseness, may be given special consideration.

5.7.3. Entire Unit

The Entire unit should be balanced to the best possible value as allowed by the entire structure with a maximum allowable value of 0.30 in/sec (Pk) at any one (1) measurement point either radially or axially to the shafting. Again, other causes of vibration, such as foundation structural resonance and/or looseness, may be given special consideration.
5.8. The Affects of Balance Toward Extending Machinery Life

Field balancing is not only a valuable procedure for machinery which has seen extended service in the field, but also is required for Forced Draft Fan Rotors following shore-side overhaul and shop balancing. Operating RPM, rotor stiffness, bearing support and stiffness, load, and drive conditions all contribute to the final running condition (i.e., vibration levels) of the complete assembly. Almost always specifications, which were met during shop balancing, do not satisfy the on-site condition requirements following re-installation. Additionally, once put into service, the rotor balance may change due to stress relieving, erosion, soot buildup, or balance weight loss, thus requiring re-balancing of the blower assembly.

Balancing rotating assemblies in the field has the following significant advantages:

- The rotor is balanced in its own bearings.
- The rotor is balanced at normal operating speed.
- The rotor is balanced at normal load.
- The rotor is driven by being coupled to its own motor.
- Tear down, re-assembly, and realignment are not normally necessary.
- Downtime and related costs are substantially reduced.

In conclusion, balancing machines that are severely out-of-balance may reduce catastrophic failures but will not address extending the life of the machine’s most vital component; the bearings. Unbalance is a major contributor to decreased bearing life and premature failures. Table 3, comparing vibration levels to bearing life, demonstrates this relationship for a rotor operating at 1800 rpm.

<table>
<thead>
<tr>
<th>Vibration [In/sec]</th>
<th>Bearing Life [Years]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6</td>
<td>1.14</td>
</tr>
<tr>
<td>0.5</td>
<td>1.47</td>
</tr>
<tr>
<td>0.4</td>
<td>1.94</td>
</tr>
<tr>
<td>0.3</td>
<td>2.63</td>
</tr>
<tr>
<td>0.2</td>
<td>3.70</td>
</tr>
<tr>
<td>0.1</td>
<td>5.44</td>
</tr>
<tr>
<td>&lt; 0.1</td>
<td>Up to 8.0</td>
</tr>
</tbody>
</table>

Table 3. Vibration Levels vs. Bearing Life

Consequently, it is extremely important to reduce normal operating vibration levels well below the 0.30 in/sec acceptance threshold.
6. CONDITION MONITORING

6.1. Condition Assessment of the Machine

6.1.1. Condition Analysis Methods

It is recommended to monitor the Forced Draft Fan installation utilizing the following techniques: Spectral Vibration Analysis (Semi-Annual), Ultrasonic Bearing & Coupling Analysis (Semi-Annual), and Thermographic/Infrared Electrical Analysis (Annual). The Vibration and Ultrasonic analyses are performed during surveys following ABS Guidelines and performed by an ABS recognized, Third Party Condition Monitoring Specialist Company on a Semi-Annual basis. The Electrical Analysis surveys are performed by recognized Thermographers and also follow ABS guidelines.

6.1.2. Vibration Analysis & Assessment

Currently there are two (2) major screening factors for SUSPECT MACHINERY used during the vibration condition survey. First, Vibration at machine running speed or multiples (harmonics) is examined. The classic rule is that levels approaching and/or exceeding 0.3 in/sec \([P_k]\) velocity are considered excessive. Second, an anti-friction bearing HFD (High Frequency Detection) indicator number is calculated from the vibration data. When this number exceeds 2 - 3 g’s (1 g = 32.2 ft/sec\(^2\)), bearing load and/or wear is excessive. High values, > 5 g’s, are usually an indication of bearing damage or failure.

6.1.3. Ultrasonic Bearing and Coupling Analysis

Ultrasonic Testing (UT) detection of bearings utilizes a contact probe to monitor, via headphones or calibrated LED display in dB, high frequency sound (40 kHz - above human audible range) that is generated by friction between moving parts. With the probe placed on the grease fitting or bearing housing, a good bearing produces a smooth whistling sound with an amplitude of 0 - 25 dB. A bearing that is about to fail will always sounds like crushing glass with an amplitude of 40 - 50 dB or higher. A typical trouble-shooting criteria chart is illustrated below in Table 4, Section - 6.1.4. The information recorded will vary according to the bearing type, load, RPM, and method of lubrication. The quality of the UT sound is extremely important in distinguishing a problem from a lack of lubrication or lubrication problem (grease oxidation) as opposed to a bearing defect.
In addition to bearing lubricant quality or bearing fault detection, coupling analysis is also performed using a non-contact UT microphone that records the coupling emission amplitude and sound quality during normal operation. Clicking sounds are often an indication of lubricant breakdown, misalignment, or coupling damage.

6.1.4. Vibration and Ultrasonic Tolerances

See Appendix #9 - General Machinery Vibration Severity Charts, for two (2) different types of vibration criteria charts. The first looks at the Overall Level of vibration from 600 cpm (cycles per minute) to 60,000 cpm. This criteria classifies machines by size and type and has ranges as follows: “Acceptable”, “Tolerable”, and “Not Permissible” based on the Overall Vibration levels. The second chart deals with vibration at just one (1) frequency (i.e., 1 x SS) and its amplitude relative to acceptable levels as noted. **Note that both these criteria classify vibration levels over 0.30 in/sec. within the “Not Acceptable” range.**

The following table summarizes typical guidelines for ultrasonic data screening for bearings:

<table>
<thead>
<tr>
<th>STATE</th>
<th>UT SOUND QUALITY</th>
<th>UT [dB] Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Good Bearing</td>
<td>Whistling - Static Sound</td>
<td>0 - 25 dB</td>
</tr>
<tr>
<td>Lubrication Problems</td>
<td>Noise Increasing / Decreasing - Clicking</td>
<td>25 - 40 dB</td>
</tr>
<tr>
<td>Careful Surveillance</td>
<td>Clicking / Popping</td>
<td>40 - 50 dB</td>
</tr>
<tr>
<td>Bad Bearing / Failed</td>
<td>Popping / Glass Crushing Like Sound</td>
<td>50 - 70 dB</td>
</tr>
</tbody>
</table>

Table 4. Ultrasonic Sound Quality Criteria

Coupling UT data is also highly dependent on the quality of the sound (i.e., clicking/popping) as well as the overall UT level. Typically, levels over 30 dB are considered suspect unless no characteristic clicking/popping sounds are present. External noise from surrounding UT sources (i.e., air or steam leaks) will also be present in the coupling airborne data and will often increase the UT levels above the 30 dB threshold. In this case, clicking/popping of the coupling would have to be present to be considered suspect.
6.1.5. Thermographic (Infrared) Analysis

Infrared thermal imaging systems are valuable tools that are used to pinpoint problem electrical components and avoid costly plant shutdowns or unscheduled outages. The use of thermal imaging does not eliminate the need for a thorough preventive maintenance program but serves to identify critical areas and to locate trouble spots that help determine maintenance priorities between scheduled shutdowns.

The theory of operation of a thermal imaging system is based on the principle that all objects above absolute zero (-273 °C) radiate infrared energy, the level and wavelength of that are proportional to the temperature of the object. Since infrared energy is invisible to the naked eye, some means must be provided to transform it into a visual image. Basically, an infrared imaging system consists of a camera that remotely senses the infrared radiation being emitted, detectors which transform the radiation to electrical signals, an amplifier to boost the signal to suitable levels, and a monitor to view the radiation as an image with some method of image storage to disk.

Increased temperatures accompany most equipment failures in the shipboard environment; in other words, they get hot. Infrared thermal imaging systems can detect this rise in temperature with no physical contact between the objects being scanned and the test equipment thereby allowing use during normal full-power operations. Consequently, the Thermographer can pinpoint trouble “hot” spots with no interruption in service.

During normal equipment operation, thermal imaging systems are used to check insulation efficiency, optimize control settings and locate any thermal losses or hot spots. Prior to scheduled maintenance shutdowns, it can be used to identify areas requiring immediate attention. After shutdowns, it may be used to verify and evaluate the repair work.

Thermal Imaging technology provides the following benefits:

- Pinpoint potential problems.
- Reduce downtime and emergency repairs.
- Reduce man-hours spent on preventive maintenance by pinpointing areas that need repairs.
- Extend equipment life and increase its overall reliability.
- Evaluate and verify repair work.
• Minimize scheduled maintenance times by allowing for spare parts to be on-hand prior to shutdowns.
• Promote a more efficient preventive maintenance program.
• Prevent accidents, personal injury, and equipment damage.
• Reduce risk of fire due to electrical failure.
• Reduce casualty loss of equipment.

Current policy is to schedule Thermographic surveys on at least an **Annual Basis**. Experience has revealed that electrical connection looseness is the primary anomaly that is either found in the motor controller or pothead connections and often re-occurs annually.

### 6.2. **Total Reliability Management**

Traditionally, two types of maintenance strategies have been utilized: Reactive Maintenance (RM), run until failure; and Preventive Maintenance (PM), overhaul at designated time interval before expected failure. Predictive Maintenance (PDM) describes a maintenance approach of determining a machine’s condition and its need for maintenance without disturbing normal operation.

PDM is accomplished by utilizing vibration analysis, ultrasonic analysis, lubricating oil analysis, and thermographic inspection for determining operating condition and detecting faults. From trending and diagnostic results of PDM, an active approach of systematically finding and eliminating the underlying causes of machine problems can then be initiated.

This corrective phrase is called Proactive Maintenance (PAM). IMPACT’s mission has been to efficiently balance the selected practices of RM, PM, PDM, and PAM into a defined strategy called **Total Reliability Management** (TRM). A comprehensive TRM maintenance approach ultimately eliminates unexpected failures of equipment and decreases overall maintenance costs resulting from unnecessary overhauls and emergency repairs.

This overall management process has been designed for the Marine Industry to minimize initial installation related problems, highlight developing machinery defects, reduce vessel down-time, and defer Classification Society open-and-inspect requirements. Well maintained, reliable operating machinery produces substantial, wide-ranging cost savings for the vessel.