Application Technique

Original Instructions

Applying Condition Monitoring to Various Machinery

Allen-Bradley • Rockwell Software

Rockwell Automation

Important User Information

Read this document and the documents listed in the additional resources section about installation, configuration, and operation of this equipment before you install, configure, operate, or maintain this product. Users are required to familiarize themselves with installation and wiring instructions in addition to requirements of all applicable codes, laws, and standards.

Activities including installation, adjustments, putting into service, use, assembly, disassembly, and maintenance are required to be carried out by suitably trained personnel in accordance with applicable code of practice.

If this equipment is used in a manner not specified by the manufacturer, the protection provided by the equipment may be impaired.

In no event is likely to Rockwell Automation, Inc. be responsible or liable for indirect or consequential damages resulting from the use or application of this equipment.

The examples and diagrams in this manual are included solely for illustrative purposes. Because of the many variables and requirements associated with any particular installation, Rockwell Automation, Inc. cannot assume responsibility or liability for actual use based on the examples and diagrams.

No patent liability is assumed by Rockwell Automation, Inc. with respect to use of information, circuits, equipment, or software described in this manual.

Throughout this manual, when necessary, we use notes to make you aware of safety considerations.

	WARNING: Identifies information about practices or circumstances that can cause an explosion in a hazardous environment, which may lead to personal injury or death, property damage, or economic loss.
	ATTENTION: Identifies information about practices or circumstances that can lead to personal injury or death, property damage, or economic loss. Attentions help you identify a hazard, avoid a hazard, and recognize the consequence.
IMPORTANT	Identifies information that is critical for successful application and understanding of the product.

Labels may also be on or inside the equipment to provide specific precautions.

	SHOCK HAZARD: Labels may be on or inside the equipment, for example, a drive or motor, to alert people that dangerous voltage may be present.
	BURN HAZARD: Labels may be on or inside the equipment, for example, a drive or motor, to alert people that surfaces may reach dangerous temperatures.
	ARC FLASH HAZARD: Labels may be on or inside the equipment, for example, a motor control center, to alert people to potential Arc Flash. Arc Flash is likely to cause severe injury or death. Wear proper Personal Protective Equipment (PPE). Follow ALL Regulatory requirements for safe work practices and for Personal Protective Equipment (PPE).

Chapter 4

Machines with Fluid Film Bearings

Chapter 11

Vibration Relationships

Chapter 12

Vibration Severity

Chapter 13

Vibration Signature Analysis

This application guide is intended to provide general guidance in how to apply a Dynamix™ 1444 Series condition-monitoring system to various types of machinery.

The basic objective of a condition-monitoring system is to detect faults that manifest as increased vibration. When applying an integrated condition-monitoring system with a Logix controller and Dynamix monitors, the objective can still be just that, or it can be much more—it can be to implement a Smart Machine.

Regardless the scope of the solution, this document provides the guidance necessary to implement it: From basic monitoring or protection, to a smart machine that can specifically identify select common or expected faults that can occur for a given class of machine.

Guidance is provided for monitoring the most common class of machines, which are fitted with rolling element bearings. These machines are monitored with accelerometers as well as guidance for monitoring typically larger machines that are fitted with fluid film bearings and monitored with eddy current probes. The guide also includes significant reference information such as applicable standards and vibration severity limits, and discusses bearing types, electric motors, hazardous area applications, vibration signal relationships and more.

Additional Resources

These documents contain additional information concerning related products from Rockwell Automation®.

Resource	Description
Dynamix 1444 User Manual, publication 1440-UM001.	Instructions on installation and configuration of the modules.
1443 Accelerometer Selection Guide, publication 1443-TD001.	Details about sensors.
Eddy Current Probe Selection Guide, publication 1442-TD001.	Details about sensors.
Industrial Automation Wiring and Grounding Guidelines, publication 1770-4.1.	Provides general guidelines for installing a Rockwell Automation® industrial system.
Product Certifications website, http://www.rockwellautomation.com/global/certification/overview.page.	Provides declarations of conformity, certificates, and other certification details.

You can view or download publications at http://www.rockwellautomation.com/global/literature-library/overview.page. To order paper copies of technical documentation, contact your local Allen-Bradley distributor or Rockwell Automation sales representative.

Analysis Considerations

The guidance that is included in this document assumes that only a limited knowledge of vibration analysis is available. This document therefore makes various generalizations and accommodations to simplify implementation of a solution. When a vibration analyst is available, either on staff or contracted, any recommendation with regard to implementing a Dynamix solution is to be considered more authoritative and accurate than the recommendations provided here.

There are two key assumptions/accommodations made in this guide that should be considered when implementing a solution. The first assumption is that no tachometer is available. The second is that all bands for rolling element bearing machines are assumed measured in velocity.

No Tachometer is Available

For most machines, a speed sensor is not available for input to the monitoring system. Therefore guidance assumes the four tracking filters per channel, which provide magnitude and phase values, are not available.

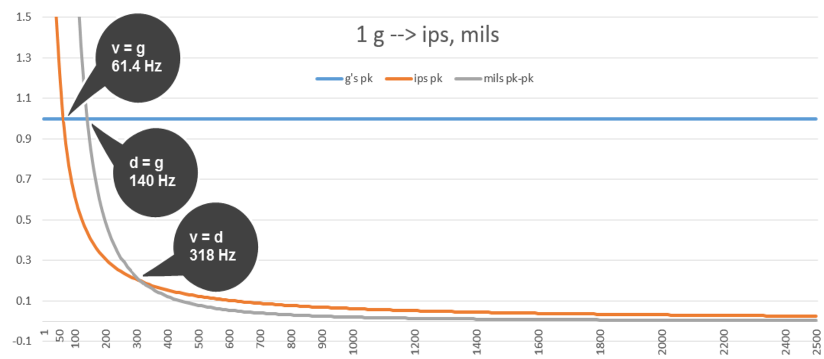
If a speed sensor, commonly a 1/rev signal, is available then the implementation can use the tracking filters for, at least, the first order measurements. In addition, at least the first order phase value can be monitored.

If monitoring a machine fitted with rolling element bearings, then the signal source for the band measurements can be changed to use an acceleration signal. The acceleration signal is likely to be an alternate path. Tracking filters should be configured to monitor the lower frequencies, typically the first few band definitions. See the discussions in Measured Velocity for further information.

Measured in Velocity

This guidance assumes that machines with rolling element bearings are monitored with accelerometers and that the measurement is integrated to velocity. This assumption is because all ISO Standards for overall vibration on these class machines are in velocity.

When signals are measured at frequencies near or over 1 kHz, the preferred units are g's (acceleration). This is because velocity measures are significantly tempered at higher frequencies. Velocity measures at higher frequencies are not always apparent or can fall below the noise floor for low amplitude signals.



The likely impact of this, relative to recommendations in this document, are concerning stage 1 bearing fault indication and probably gear mesh signal detection.

Stage One Bearing Fault

Stage one bearing fault is indicated by excited rolling element bearing natural frequencies, which are present between about 1 kHz and 2 kHz.

The guidance for machines with rolling element bearings recommends that the last band is configured to measure a frequency range from 50…100% of maximum frequency (FMAX) when FMAX is set to 2 kHz.

Attributes		Band 7
Band Name		Bearing Higher Harmonics and Natural Frequencies
Frequency Range		50% …100% FMAX
% of Overall Level	…	15%
Alarm 1 mm/s		0.67
Alarm 2 mm/s		1.02
Alarm 1 in/s		0.038
Alarm 2 in/s		0.056

The recommended alarm limits for the band are 15% of the overall level, which is specified in velocity. Because velocity measurements above about 1 kHz are significantly muted, it can be difficult to see a significant change in velocity as the bearing natural frequencies get excited. Nevertheless, there will be an increase which will be proportional to the change in acceleration. It is just a matter of how significant the change is.

When natural frequencies are excited, it typically results in 1+ g vibration in the 1…2 kHz band and sometimes well over that range. If 1 g is present in the band measurement, the velocity reading would be around 0.03…0.06 ips.

$$v = \frac{61.48 \times g}{Hz}$$

g's to ips	
Hz	0.1 g's
1 kHz	0.006 ips
2 kHz	0.003 ips

The suggested alarm limits are based on the velocity increase, which are based on ISO standards. Typically, these limits resolve to approximately 0.04 ips (alert) and 0.06 (alarm), which are consistent with the previous examples.

Other signals in the same frequency range can be present that could complicate bearing fault indication. Gear mesh related frequencies, harmonics of vane/blade pass, and so on, could be present in the frequency range being monitored. When a bearing fault is indicated, these other sources should be discounted before resolving that it is a bearing fault.

When the bearing natural frequencies are excited, it creates an easily discerned pattern in that multiple frequencies are excited. See Figure 32 on page 159.

If preferred, the signal source for the band measurements can be configured to read from an acceleration data signal source instead of velocity. In this case, the alarm limits for all bands would be adjusted accordingly. Note that in most cases, the limits specified by the ISO standards for overall vibration level are in velocity.

Gear Mesh

Gear mesh frequency depends on the number of teeth and the speed of the shaft. The frequency is usually above 1 kHz, sometimes significantly. On slow speed machines, the frequency can be well below 1 kHz.

Attributes	…	Band 4
Band Name		Gear Mesh Frequency
Frequency Range		0.8…1.2 GMF
% of Overall Level		25%
Alarm 1 mm/s		1.75
Alarm 2 mm/s		2.63
Alarm 1 in/s		0.097
Alarm 2 in/s		0.146

The same issues and answers that are discussed in stage 1 bearing fault detection apply to gear detection problems using the 1x gear mesh frequency. Since gear mesh frequency can be higher than 2 kHz or well less than 1 kHz, consideration of how to manage the issue must be done on a case by case basis.

Application Overview

This guide is organized first by bearing type, and then by machine.

Bearing Type refers to either rolling element or fluid film. The type of bearings a machine uses is fundamental to determine the sensing solution, and many of the possible faults that must be monitored.

For each bearing type guidance is provided for a range of common machinery. The machines that are listed are the basic components. These components are motors, fans, gearboxes, and so on. When using the guide each machine component must first be considered uniquely, and then considered as a whole.

For example, for a common motor driven pump; consider the sensor requirements for the motor and pump individually. Then consider the total sensor solution and adjust as appropriate:

- In many cases, it is not practical, or possible, to mount a sensor at some bearing locations. When this situation occurs, consider adding an axially oriented sensor for the (entire) machine, preferably at either of the inboard locations.
- In some cases, the total number of sensors require adding a monitor for just one or two measurement channels, which is not always practical economically. In those cases, a sensor can be excluded based on its proximity to other sensors, the importance of the asset, or other factors.

This guide discusses concepts and uses terms that are associated with machine condition monitoring, diagnostics, and signal processing that are not always familiar to the reader. When encountering an unfamiliar term, consult the Glossary for a definition or explanation.

General

Guidance is provided for a wide range of common machines. However, there are thousands of unique designs for each of the machines. If the design of any specific machine is substantially different from the examples that are provided, then consult a condition monitoring professional to verify an appropriate monitoring solution is determined.

Machines are qualified by bearing type. For example, rolling element, fluid film or plain (sleeve) bearings, by function such as motor, fan, or pump. They are qualified by size in horse power (HP) or kilo watts (kW). In some cases machines are also classified as horizontal or vertical, which refers to the orientation of the shaft.

Applicable Standards

This guide references various standards throughout. The following table lists these and other standards. In general, it is not necessary to review any of the listed standards. However, where there is any question about the suggested limits, particularly the overall vibration level limits, then the guidance of the applicable standard for the specific machine must take precedence over the values that are provided in this document.

IMPORTANT With Machinery Protection, the applicable standards and any OEM requirements must be referenced independently of this document and adhered to as appropriate.

See Machinery Protection for additional information.

Table 1 - Applicable Standards

Standard	Description
API Standard 670	Machinery Protection Systems, 5th Edition, November 2014 This standard covers the minimum requirements for a machinery protection system (MPS) measures radial shaft vibration, casing vibration, shaft axial position, shaft rotational speed, piston rod drop, phase reference, overspeed, surge detection, and critical machinery temperatures (such as bearing metal and motor windings). It covers requirements for hardware (transducer and monitor systems), installation, documentation, and tests.
ISO/TR 19201	Mechanical vibration—Methodology for selecting appropriate machinery vibration standards
ISO 7919-1	Mechanical vibration of non-reciprocating machines—Measurements on rotating shafts and evaluation criteria—Part 1: General guidelines
ISO 7919-2	Mechanical vibration --Evaluation of machine vibration by measurements on rotating shafts—Part 2: Land-based steam turbines and generators in excess of 50 MW with normal operating speeds of 1500 r/min, 1800 r/min, 3000 r/min, and 3600 r/min
ISO 7919-3	Mechanical vibration—Evaluation of machine vibration by measurements on rotating shafts—Part 3: Coupled industrial machines
ISO 7919-4	Mechanical vibration—Evaluation of machine vibration by measurements on rotating shafts—Part 4: Gas turbine sets with fluid-film bearings
ISO 7919-5	Mechanical vibration—Evaluation of machine vibration by measurements on rotating shafts—Part 5: Machine sets in hydraulic power generating and pumping plants
ISO 13373-2	Condition monitoring and diagnostics of machines—Vibration condition monitoring—Part 2: Processing, analysis, and presentation of vibration data

Table 1 - Applicable Standards

Standard	Description
ISO 14694	Industrial fans—Specifications for balance quality and vibration levels
ISO 10816-1	Mechanical vibration—Evaluation of machine vibration by measurements on non-rotating parts—Part 1: General guidelines
ISO 10816-2	Mechanical vibration—Evaluation of machine vibration by measurements on non-rotating parts—Part 2: Land-based steam turbines and generators in excess of 50 MW with normal operating speeds of 1 500 r/min, 1 800 r/min, 3 000 r/min, and 3 600 r/min
ISO 10816-3	Mechanical vibration—Evaluation of machine vibration by measurements on non-rotating parts—Part 3: Industrial machines with nominal power above 15 kW and nominal speeds between 120 r/min and 15 000 r/min when measured in place.
ISO 10816-4	Mechanical vibration—Evaluation of machine vibration by measurements on non-rotating parts—Part 4: Gas turbine sets with fluid-film bearings
ISO 10816-5	Mechanical vibration—Evaluation of machine vibration by measurements on non-rotating parts—Part 5: Machine sets in hydraulic power generating and pumping plants
ISO 10816-6	Mechanical vibration—Evaluation of machine vibration by measurements on non-rotating parts—Part 6: Reciprocating machines with power ratings above 100 kW
ISO 10816-7	Mechanical vibration—Evaluation of machine vibration by measurements on non-rotating parts—Part 7: Rotor dynamic pumps for industrial applications, including measurements on rotating shafts
ISO 10816-8	Mechanical vibration—Evaluation of machine vibration by measurements on non-rotating parts—Part 8: Reciprocating compressor systems
VDI 3834	Measurement and evaluation of the mechanical vibration of wind energy turbines and their components. Onshore wind energy turbines with gears

Machinery Protection

This guide describes how to instrument and monitor machines to serve a condition-based maintenance application. While the instrumentation applied per this guide can also serve a machinery protection application, it does not discuss the specific requirements associated with protection systems.

Protection applications sometimes require local relays, specific relay configuration and voting solutions, and measurements that are not addressed in this document, such as thrust position or bearing temperature.

Machinery protection systems are defined by API-670, by applicable ISO, or other standards, and by machine-specific OEM provided documentation.

IMPORTANT	If the application requirement includes protection then identify and address the requirements of any required standards before finalizing a system design. In all cases when any requirements for protection conflict with any described in this document, the requirements for protection must take precedence.

While most machinery protection standards do not address condition monitoring, the 5th Edition of API-670 does. Paragraph 4.8 of API-670 addresses system segregation, requiring that a protection system is separate and diverse. However, the following important statement is included following paragraph 4.8.1.

IMPORTANT	It is not intended to prohibit the inclusion of condition monitoring functionality within the MPS, provided failure of those functions does not impact the protective functions.

The Dynamix™ 1444 Series system documentation makes the following statement in regards to compliance with the API standard.

> API-670 Compliance
>
> The 1444 Series is designed in accordance with the relevant sections of the Fifth Edition of the American Petroleum Institute (API) standards 670[1] 'Machinery Protection Systems'.

Faults

For each machine, a Machine Faults section lists the various faults that occur commonly, uncommon but not unusual faults, and faults that are unusual but possible. Which of the faults are monitored depends on the history of the machine, the capabilities of the organization outside of the integrated monitor application, and the required scope of the solution. In general, unless other faults are expected, monitor the common faults.

For each machine a **Band Definitions and Limits** section is provided that provides the band and band alarm details necessary to implement monitoring of common faults. While in many cases a solution could be implemented directly from the table as shown, in some cases the band definitions will have to be altered in order to monitor faults that are listed as uncommon or unlikely, for most machines. In these cases the selected bands will have to be adjusted per the guidance provided in the Machine Faults table. Also refer to the further detail provided in the **Vibration Signature Analysis** section.

(1) Whether a system is compliant is dependent on the components that are provided, the various optional elements of the standard that you require, and the configuration of the installed system.

Bearing Types

Perhaps the most important attribute of a machine relative to the fault diagnostics, is the type of bearings that the machine is fitted with. While there are other types of bearings, such as magnetic bearings, this guide addresses only common rolling element, fluid film, and plain bearings. If another type of bearing is used, similar principles as discussed in the following section still apply, particularly for the non-bearing related faults. However, in that case, consult a condition monitoring professional about the best way to monitor the machine.

Rolling Element Bearings

A rolling element bearing carries a load by placing balls or rollers between two rings called races. The relative motion of the races causes the rolling elements to roll with little rolling resistance.

Rolling-element bearings have the advantage of a good trade-off between cost, size, weight, carrying capacity, durability, accuracy, friction, and so on. Consequently they are the most common bearing for small to medium size machines of all types.

Most rolling-element bearings feature cages. The cages reduce friction, wear, and bind by helping to help prevent the elements from rubbing against each other.

When rolling element bearings fail, they do so by progressing through fault stages that are highly predictable and can be detected.

Expected bearing life is defined by ISO standards as the L_{10} Bearing Life. L_{10} bearing life can be used to calculate the expected life of a bearing depending on the machine speed and load. See the glossary for further information on L_{10} bearing life.

Machines that are fitted with rolling element bearings use seismic type vibration sensors such as accelerometers mounted as close to the bearing as is practicable. These sensors are normally mounted using a stud that is fitted directly into the bearing housing. In cases where it is not possible to drill and tap into the housing or machine case it is possible to glue a mounting pad to the target surface. Whether you are mounting with a stud or glue pad, care must be taken in the preparation of the surfaces. This preparation is to achieve a smooth, flat, mating surface to provide accurate measurements and, if glued, to achieve a durable bond for the mounting pad.

Vibration sensors are normally mounted in the radial direction, perpendicular to the axis of the shaft, and in either a horizontal or vertical orientation. In some situations vibration sensors are also mounted in the axial direction, parallel to the axis of the shaft, as some faults force a machine to vibrate in, or more prominently in, the axial direction.

For machines up to 500 HP (375 kW), one radial sensor per bearing is normally sufficient, normally mounted in the horizontal direction.

For machines greater than 500 HP (375 kW), the preferred solution is to have two radial sensors per bearing, one horizontal and one vertical plus at least one axial sensor.

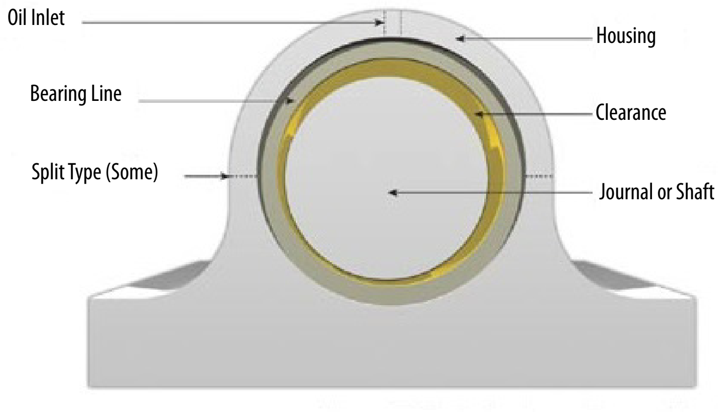
In some installations, it is not possible to install the vibration sensors in the preferred locations due to machine design, interference from pipework or other instrumentation, or other reasons. Consequently, selection of sensor mounting locations can often require compromise between mounting in the preferred vs. some other orientation, or even of monitoring a bearing at all. When it simply isn't practical to mount a radial sensor at a bearing, alternatively, consider mounting a sensor in the axial[(1)] direction.

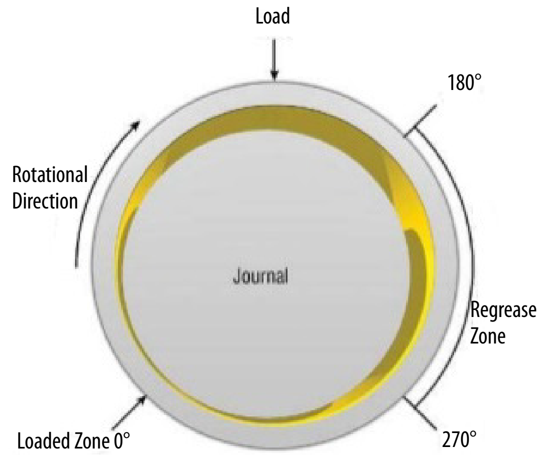
See Rolling Element Bearing Faults on page 158 for a discussion of faults associated with rolling element bearings and how they can be detected.

Fluid Film Bearings

Fluid film bearings use either the speed of the shaft (hydrodynamic) to create an oil wedge that the shaft then rides on, or pressurized oil (hydrostatic) to lift the shaft off the bearing surface. In these designs, the shaft does not contact the bearing surface, so little shaft vibration is transmitted to the bearing housing.

(1) In most cases, only one axial measurement is made per shaft.





Another type of fluid film bearing is a plain (sleeve) bearing. See Plain Bearings for further detail on that type bearing.

Consequently, machines that are fitted with fluid film bearings typically use non-contact pickups (eddy current probes) to measure the actual shaft vibration relative to the bearing housing.

The following section refers to common faults that are associated in terms of frequency orders, 1*x*, 2*x*, and so on, where 1*x* would be one times running speed, example 60 Hz, or 3600 rpm (50 Hz or 3000 rpm).

Fluid Film Bearing Faults

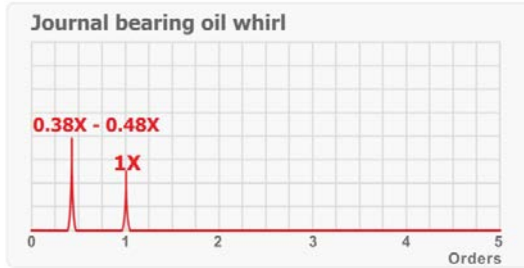
The following faults could apply to any machine fitted with fluid film bearings.

Table 2 - Fluid Film Bearing Faults

Faults	Possible Problems	Frequency (Order)	Primary Direction
Common Faults	Wear and clearance problems	1*x*,2*x*,3*x*,4*x*,5*x*,6*x* can be higher harmonics	Radial
	Loose bearing assembly	0.5*x* plus 1*x*, 2*x*, 3*x*, and possibly higher harmonics	Radial
Uncommon Faults	Rub	0.25*x*, 0.5*x*, 1*x*, 2*x*, 3*x*, and possibly higher harmonics	Radial
	Oil whirl instability	0.4*x*…0.48*x*	Radial
	Oil whip instability—for machines operating at or above twice the rotor critical frequency	Initially same as Oil Whirl but locks onto the rotor critical frequency, typically 0.5*x*	Radial

Other potential faults are associated with the specific type of machine, its foundation and structure, and its connected components.

Figure 1 - Journal Bearing Oil Whirl

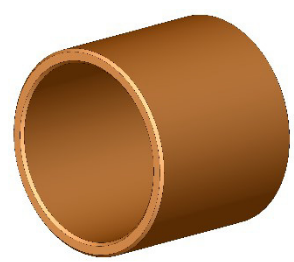


Plain Bearings

A plain, or sleeve bearing is the simplest form of bearing. It is essentially when a shaft rides on an oiled surface, or bushing. Bushings can be made of bronze or other metals, metal composites, even plastic. They can be lubricated using a pressurized oil feed, they can run dry (not oiled), or they can be self-lubricated.

Compared to rolling element bearings, which plain bearings can often be used in place of, sleeve bearings offer simpler designs and easier maintenance. These bearings are also capable of heavier loads and greater resistance to shock or misalignment. Consequently sleeve bearings are common in applications such as low speed, high horsepower machinery, and for bearings on crank shafts of many reciprocating engines.

Because little shaft vibration is transmitted to the housing of a sleeve bearing, similarly to any fluid film bearing, casing vibration measurement does not always provide a good indication of machine or bearing condition.



Consequently, machines that are fitted with plain bearings, where the bearings are accessible, are best monitored using eddy current probes. These probes measure the actual shaft vibration relative to the structure that the sensor is mounted on since bushings are not suitable to be drilled and tapped for a probe. However, in many applications, such as crank shaft bearings on a compressor, it isn't possible to measure the bearings directly, and accelerometers are used at the accessible bearing locations.

The following section refers to common faults in terms of frequency orders, 1*x*, 2*x*, and so on, where 1*x* would be one times running speed, example 60 Hz, or 3600 rpm (50 Hz or 3000 rpm).

Plain Bearing Faults

Because a plain bearing is similar to other fluid film bearings (hydrostatic and hydrodynamic), the potential faults that can occur are similar.

Table 3 - Plain Bearing Faults

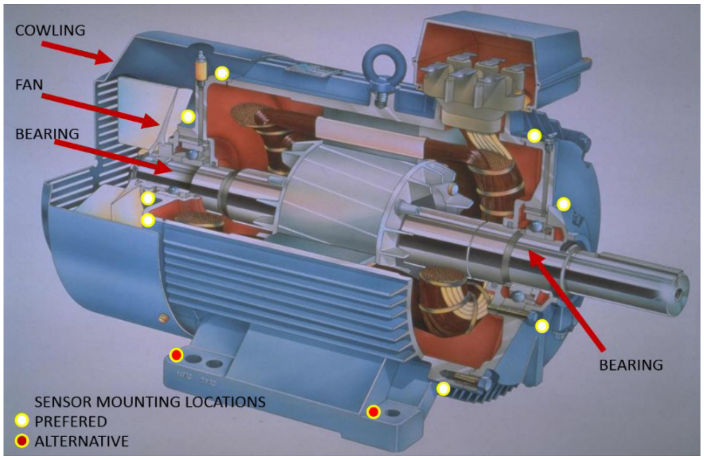
Faults	Possible Problems	Frequency (Order)	Primary Direction
Common Faults	Wear and clearance problems	1*x*,2*x*,3*x*,4*x*,5*x*,6*x* can be higher harmonics	Radial
	Loose bearing assembly	0.5*x* plus 1*x*, 2*x*, 3*x*, and possibly higher harmonics	Radial
Uncommon Faults	Rub	0.25*x*, 0.5*x*, 1*x*, 2*x*, 3*x*, and possibly higher harmonics	Radial
	Oil whirl instability	0.4*x*…0.48*x*	Radial
	Oil whip instability—for machines operating at or above twice the rotor critical frequency	Initially same as Oil Whirl but locks onto the rotor critical frequency, typically 0.5*x*	Radial

The most common problem with sleeve bearings is access. Eddy current probes are used in only a relatively few applications of these bearings, and often the bearing isn’t accessible at all. Casing vibration using accelerometers detect the vibration, but the magnitudes are reduced as shaft vibration amplitudes are attenuated up to 90%, so the vibration is not always apparent until a fault is pronounced. Consequently, in many cases the best method of sleeve bearing wear detection is oil analysis. But oil analysis does not identify the causes of bearing wear, such as severe misalignment or imbalance.

When monitoring machines that are fitted with sleeve bearings, use the guidance that is provided for fluid film bearings. But remember that if the measurements are made using accelerometers rather than eddy current probes, the measured vibration magnitudes are less than actual. In these cases, the focus of the monitoring has to be on the presence of fault frequencies, and their trended behavior, and not on the actual amplitude of the measurements.

Electric Motors

Because electric motors are the most common machine driver that is used in industry, they warrant special attention.



There are three types: AC Induction (asynchronous) motors, AC Synchronous motors, and DC motors. Of these types, the most common is the AC Induction motor. This document assumes common induction motors.

The images illustrate typical motors showing the various suitable locations where vibration sensors could be mounted. The requirement is to mount the sensor at a location that has a solid structural path to the bearing, and that the location is as close to the bearing as practical.

A common complicating issue with induction motors is the fan and its cowling. Sensors cannot be mounted on the cowling. When the cowling covers all suitable mounting locations, consider drilling a hole or cutting a notch in the cover to allow access to the mounting location. If that isn't possible, then the sensor can be mounted on the base, or the outboard bearing can be left unmonitored.

Another common problem, particularly on small motors, is space. An accelerometer requires a flat surface of approximately 25.5 mm(1 in.) diameter. The 1443 Series spot-face tools prepare a surface of 31.75 mm (1.25 in.) diameter.

IMPORTANT	Do not mount a sensor on a surface that is not flat as doing so results in measurement errors, including possible false signals.

How many sensors are required depends on the size of the machine. In general, the recommended number of sensors is one per bearing for motors less than 375 KW (500 HP), and two per bearing for larger motors.

When two sensors are applied, mount one as close to vertical as possible, and one as close to horizontal as possible.

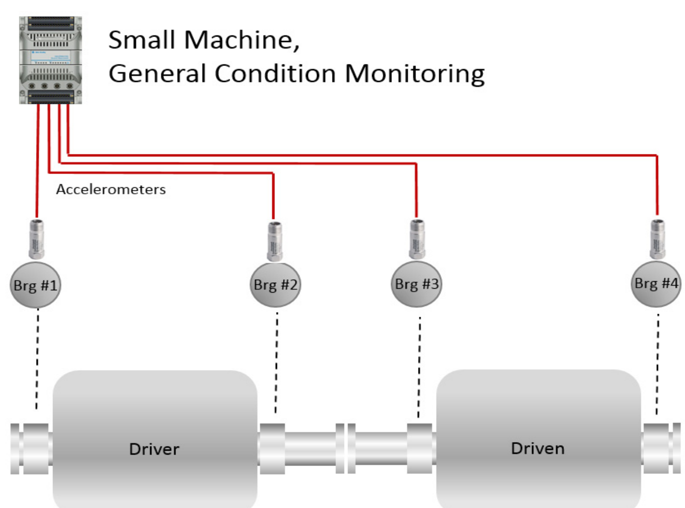
When only one sensor is used, horizontal is the default but consider vertical if the structure is obviously 'less stiff' in that direction. Whenever possible though, be consistent so that all sensors on the machine are mounted in the same orientation. For example, if at one location the sensor must be mounted vertically due to space limitations, then mount all sensors vertically.

System Arrangement

System Arrangement refers to where the sensors are mounted and how the sensors are associated with the instrumentation (monitors). In most cases, the arrangement is either '*Small Machine, General Condition Monitoring*', '*Large Machine, General Condition Monitoring*', or '*API-670 Compliant Machinery Protection System*' as described in the following. Note that in each of the following system arrangement drawings, the wiring illustrated exiting the modules is in order of channels from 0 (furthest left) to 3 (furthest right).

Small Machine, General Condition Monitoring

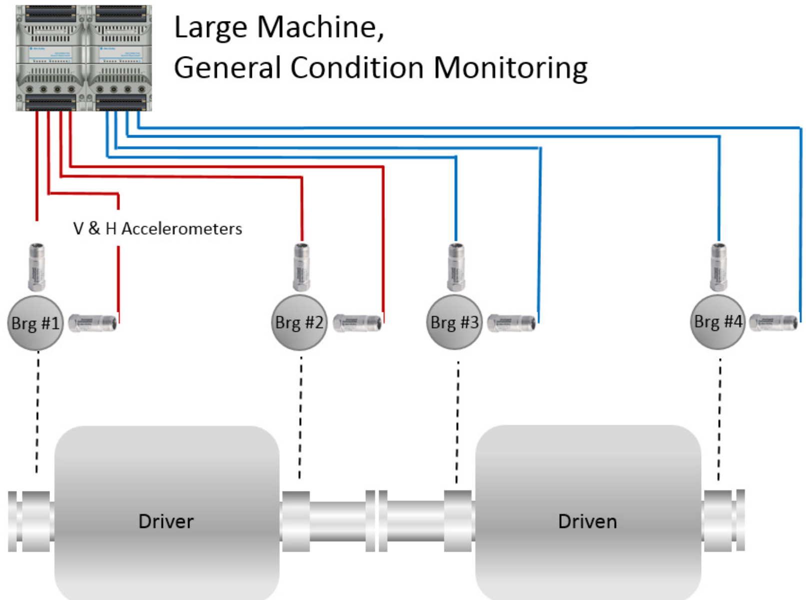
This document recommends that machines less than 500 HP and fitted with rolling element bearings be monitored using just one sensor per bearing. For general condition monitoring, to simplify the installation, the sensors can be wired in order from the driver machine's outboard bearing to the driven machine's outboard bearing. In these cases the arrangement is likely to be similar to the following:



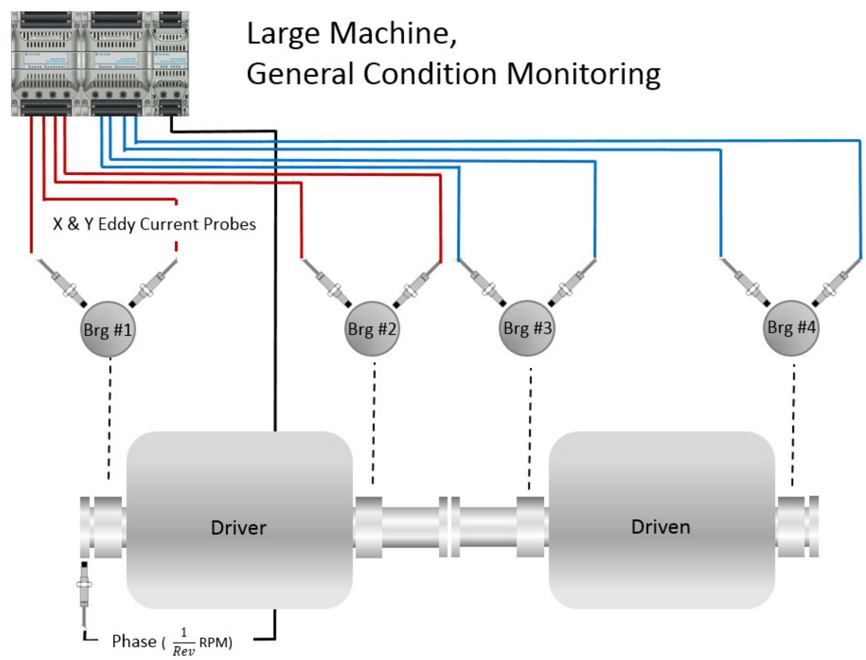
Note that in most cases, when a machine is fitted with a single sensor per bearing, the sensor will be mounted in the horizontal direction.

Large Machine, General Condition Monitoring

For large machines, greater than 500 HP, fitted with rolling element bearings it is recommended that two sensors be applied to each bearing. In these cases the arrangement bearings that it is likely to be similar to the following:



Similarly, if fitted with fluid film bearings and monitored with eddy current probes(1), the arrangement would look like the following:



(1) In most instances where eddy current probes are installed the application is for Machinery Protection. The arrangement illustrated for General Condition Monitoring is not recommended for machinery protection applications.

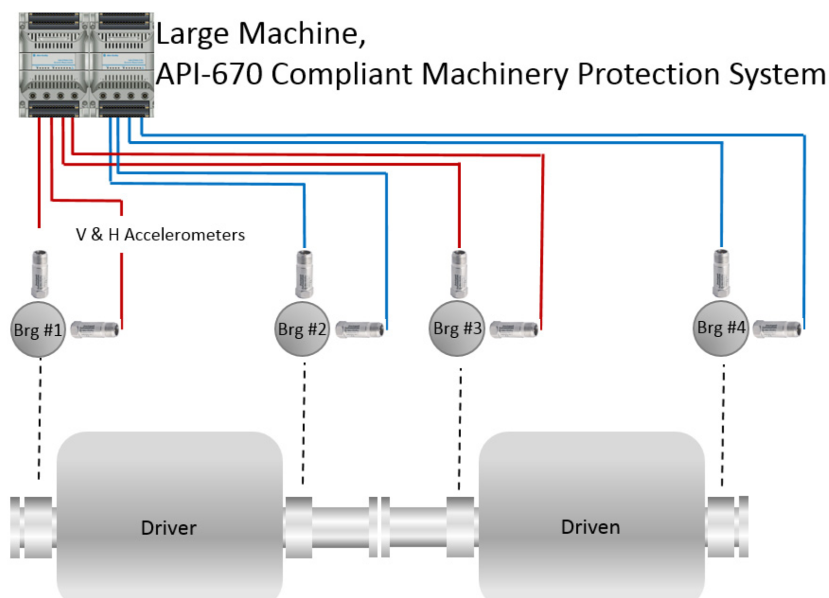
For applications that match either of the previous General Condition Monitoring arrangements, the recommended configurations can generally be used with little or no modification.

API-670 Compliant Machinery Protection System

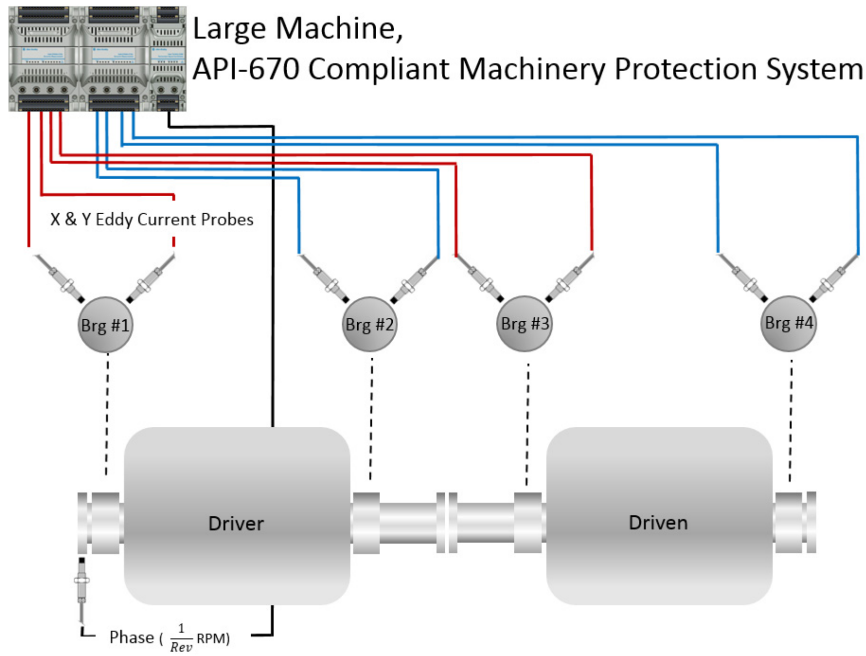
For installations where the application is protection and is required to be compliant to ***API-670 Machinery Protection Systems***, the recommended arrangement is to monitor just one bearing of a machine case with channels of any one monitor. This is to insure satisfying paragraph 7.1.3 of the standard.

7.1.3	At minimum, each monitor system is provided with the following features and functions. a)An installation design ensuring that a single circuit failure (power source and monitor system power supply excepted) shall not affect more than two channels (regardless of channels available on the monitor module) of radial shaft vibration, axial position, casing vibration, speed indicating tachometer, or six channels of temperature or rod drop on a single machine case. NOTE: The intent of this requirement is to ensure an installation design that is likely to not lose all monitoring on a machine case in the event of a single circuit fault
Terms, Definitions, Acronyms, and Abbreviations	3.1.100 **Machine case** A driver (for example, electric motor, turbine, or engine) or any one of its driven pieces of equipment (for example, pump, compressor, gearbox, generator, fan). An individual component of a machinery train

Following this guidance a typical system arrangement, for a two component, four bearing machine, can look similar to this, if accelerometers are used:



Or this if using eddy current probes:



The above arrangements wire just two of the four vibration sensors on each machine case to a single monitor. This can insure that in the event a monitor (module) fails that at most just two channels of radial or casing vibration would be lost on any one machine case (component), which satisfies section 7.1.3.a that is shown above.

Bill of Materials

This chapter references use of Dynamix™ 1444 Series monitors, 1442 Series Eddy Current Probes, and 1443 Series Sensors.

While the 1442 and 1443 series sensors provide a comprehensive offering, they do not always include every sensor required. The Dynamix and XM® series monitors work with any standards-based, third-party sensors.

WARNING: Hazardous Area Applications.

When an application requires installing monitors and sensors within a hazardous area issue such as area and classification, device and system approvals must be considered. Our monitors and sensors are not approved for use in all areas, and the approvals available vary by device. Be sure that when you are developing solutions for use in hazardous areas that only products that are approved for use in the area are selected.

See Hazardous Area Applications for further information.

XM Series

In some cases, the XM 1440 Series monitors are a more appropriate alternative, particularly if it is not possible or practical to include a Logix controller. Also, these monitors are more fitting in applications where only one or two channels of measurement are required. If an XM solution is considered, then the XM-124 Standard Dynamic Measurement Module is a two-channel instrument, so twice as many monitors are required compared to the following estimate.

Dynamic Measurement Module

The 1444 Series includes a four-channel Dynamic Measurement Module. So in most cases the number of modules that are required is simply the number of sensors divided by four. However, the following are some exceptions.

- If continuous spike energy or high frequency measurements, greater than 4.5 kHz, are required then only two channels can be used. The remaining two channels could be used for position measurements but not for vibration.
- In some cases, the practicalities of where to mount the modules makes it impractical to pull wire to the monitor location. So while Table 4 is a useful guide, the actual number of monitors required can be greater.

See publication 1444-TD001 for specifications and a list of accessories.

Table 4 - Dynamic Measurement Module

Dynamic Measurement Module			Number of Sensors		
			1…4	5…8	9…12
			Quantity		
	1444-DYN04-01RA	Dynamic Measurement Module	1	2	3
	1444-TB-A	Terminal Base for 1444-DYN04-01RA	1	2	3
Screw RPCs	1444-DYN-RPC-SCW-01	Screw Type RPCs for 1444-DYN04-01RA	1	2	3
	1444-TBA-RPC-SCW-01	Screw Type Connector set for 1444-TB-A	1	2	3
Spring RPCs	1444-DYN-RPC-SPR-01	Spring Type RPCs for 1444-DYN04-01RA	1	2	3
	1444-TBA-RPC-SPR-01	Spring Type Connector set for 1444-TB-A	1	2	3

Expansion Tachometer Signal Conditioner Module

If a tachometer is to be monitored, then the 1444 Series Tachometer Signal Conditioner Expansion Module is likely required as well.

One Tachometer Signal Conditioner includes two channels, so can measure two-speed signals, and can serve its TTL output signals to up to six Dynamic Measurement Modules (1444-DYN04-01RA). So in most cases one TSCX is all that is necessary per machine, or possibly per two machines.

Table 5 - Tachometer Signal Conditioner Expansion Module

Tachometer Signal Conditioner Expansion Module			Quantity
	1444-TSCX02-02RB	Tachometer Signal Conditioner Expansion Module	1
	1444-TB-B	Terminal Base for 1444 Series Expansion Modules	1
Screw RPCs	1444-TSC-RPC-SCW-01	Screw Type RPCs for 1444-TSCX02-02RB	1
	1444-TBB-RPC-SCW-01	Screw Type Connector set for 1444-TB-B	1
Spring RPCs	1444-REL-RPC-SPR-01	Spring Type RPCs for 1444-RELX00-04RB	1
	1444-TSC-RPC-SPR-01	Spring Type Connector set for 1444-TSCX02-02RB	1

Expansion Relay Module

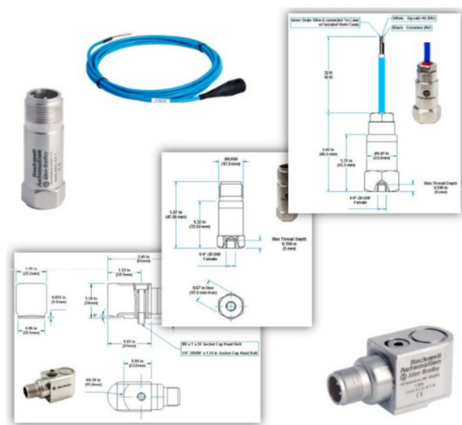
The Dynamic Measurement Module includes one Single Pole Double Throw (SPDT) relay onboard the module. However, in some cases additional relays are required. If additional relays are required, and how many, are dependent on the specific application and requirements. When required, each Dynamic Measurement Module, can host up to three Expansion Relay Modules (1444-RELX00-04RB) each of which includes four additional SPDT relays. The BOM for each is listed in Table 6.

Table 6 - Relay Expansion Module

Relay Expansion Module			Quantity
	1444-RELX00-04RB	Relay Expansion Module	1
	1444-TB-B	Terminal Base for 1444 Series Expansion Modules	1
Screw RPCs	1444-REL-RPC-SCW-01	Screw Type RPCs for 1444-RELX00-04RB	1
	1444-TBB-RPC-SCW-01	Screw Type Connector set for 1444-TB-B	1
Spring RPCs	1444-REL-RPC-SPR-01	Spring Type RPCs for 1444-RELX00-04RB	1
	1444-TBB-RPC-SPR-01	Spring Type Connector set for 1444-TB-B	1

Sensors

Vibration measurements on machines fitted with rolling-element bearings are made with accelerometers. The 1443 Series of sensors offers a family of accelerometers and accessories suitable for most applications.



See publication 1443-TD001 for a list of sensors, accessories, and options. For each sensor, select one sensor and one cable-based on Table 7.

Table 7 - 1443 Sensors and Cables

1443 Series Sensors and Cables	
Catalog Number	**Description**
1443-ACC-GP-T(1)	General-purpose Accelerometer, Top Exit, 100 mV/g
1443-ACC-GP-S(1)	General-purpose Accelerometer, Side Exit, 100 mV/g
1443-ACC-LF-T(1)	Low Frequency Accelerometer, Top Exit, 500 mV/g
1443-CBL-MS2IBC-16S(2)	16 feet of Silicone twisted shielded pair cable w/ molded 2-pin MIL connector and terminated to blunt cut, shield isolated from the connector
1443-CBL-MS2IBC-32S(2)	32 feet of Silicone twisted shielded pair cable w/ molded 2-pin MIL connector and terminated to blunt cut, shield isolated from the connector
1443-CBL-MS2IBC-64S(2)	64 feet of Silicone twisted shielded pair cable w/ molded 2-pin MIL connector and terminated to blunt cut, shield isolated from the connector

(1) Choose the top exit unless there is less than 6 in. (150 mm) space above the mounting location.

(2) Choose four of these cables depending on the distance from the sensor to the monitor.

Table 8 shows accessories to consider.

Table 8 - 1443 Series Mounting Pad Spot-face Tool

1443 Series Mounting Pad and Spot-face Tool		
Catalog Number	**Description**	**Quantity**
1443-PAD-075-0(1)	Mounting pad 1/4-28	4
1443-SFT-125-0(2)	Spot-face tool kit for sensor mounting, 1.25′ dia., 1/4-28 pilot, 2 Drill Bits, 3 Taps, Tap Wrench, Allen Wrench, Case	1 (total)
1443-SFT-125-M6(2)	Spot-face tool kit for sensor mounting, M6 pilot, 2 Drill Bits, 3 Taps, Tap Wrench, Allen Wrench, Case	
1443-SFT-125-M8(2)	Spot-face tool kit for sensor mounting, M8 pilot, 2 Drill Bits, 3 Taps, Tap Wrench, Allen Wrench, Case	

(1) Required if gluing the sensor on. Preferred solution is to drill and tap.

(2) If sensors are screw-mounted, use this tool to drill and tap a hole to mount the sensor. Required only per system or per facility. Drill and tap is the preferred mounting solution.

Sensor Mounting

Accelerometers must be mounted such that the junction between the sensor and machine is tight, and has a good fit between the two surfaces.

Method	Comments
Magnet	Magnet mounts are appropriate ONLY for temporary, attended installations. Never use magnet mounts for permanent installations. Magnet mounting is convenient but it provides the weakest interface between the sensor and structure, which means that higher frequency vibrations are attenuated, or removed, when used. Magnet-mounted sensors can provide accurate measurements up to 2 kHz to 5 kHz, depending on the type of magnet and surface preparation.
Epoxy	Suitable for temporary installations, and permanent installations where stud mounting isn't possible or practical. When using an adhesive mount, verify that the surface is flat, and that it is clean bare metal. Do not epoxy to a painted surface as the paint attenuates the signal. Epoxy mounted sensors can provide accurate measurements up to 10 kHz to 20 kHz, depending on the type of adhesive and surface preparation.
Stud	Suitable for permanent installations. When stud mounting (drill and tap), verify that the surface is properly prepared, preferably per API 670 requirements for surface finish and flatness, even for the non-API compliant installations. Stud-mounted sensors can provide accurate measurements up to 20 KHz to 50 kHz, or greater, depending on surface preparation. 

An improperly prepared surface or improperly mounted sensor could result in the following.

- Attenuated measurements that results in lower than actual-vibration amplitudes.
- Reduced frequency response that results in an inability to measure high frequency signals.
- Possible introduction of erroneous/false signals that could be misinterpreted as actual machine vibration or that could mask actual vibration.

Dual Output AT Sensors

Dual output AT type sensors output both acceleration and temperature measurements. The temperature measurement is from a thermistor mounted inside the accelerometer. The output is a proportional voltage signal with specifications such as 10 mV/°C from -55…+140 °C (-67…+284 °F).

While no applications require these sensors, they are popular due to their ease of use. However, when using AT sensors mounting is critical to assure the temperature measurement represents bearing temperature as closely as possible. When mounting an AT style sensor, verify that the sensor is placed directly on the bearing housing. The temperature is less accurate or representative the further the bearing is mounted from the sensor.

While vibration transmits with little attenuation through a metal structure or across metal support structures, the density of the material, the distance from the bearing, other heat sources, and surrounding airflow affect temperature.

In all cases, when using an AT sensor, consider temperature only as a trend-able indicator of bearing temperature and not as an absolute or accurate measurement. In this manner, it can be an additional and useful tool in monitoring bearing condition.

Eddy Current Probes

Vibration measurements on machines that are fitted with fluid film bearings are typically made with eddy current probes. The 1442 Series of sensors offers a family of eddy current probes and accessories suitable for almost any application.

When applying eddy current probes key attributes such as thread type, probe diameter (range), probe length, system length, and so on, must be matched to the specifics of the machine. Verify these and other key attributes before selecting specific probes for an application.

See the selection guide for the 1442 Series probes, publication 1442-TD001, for further details on available probes and probe accessories.

An eddy current probe system consists of the probe, extension cable, and driver. If it is necessary to replace components, replace with the same components from the same product family. It is not possible to replace a component of any suppliers eddy current probe system with components of another suppliers system. There is no assurance that measurement accuracy is retained.

Notes:

Machines with Rolling Element Bearings

This chapter explains various parts of machines with rolling-element bearings.

Small Electric Motors—20 HP (15 KW) and Below

This section describes small electric motors.

See Electric Motors for further detail on motors.

Sensor and Monitor Requirements

Depending on available space, mount one general-purpose sensor per bearing, one or two sensors total. One Dynamic Measurement Module is likely to be required. The monitor can be shared with the sensors that are used for the driven machine.

No Expansion Tachometer Signal Conditioner Module or additional Expansion Relay Module is required.

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

Possible machine faults are shown in Table 9.

Table 9 - Machine Faults—Electric Motor

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and axial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5*x* high levels of half order harmonics	Radial
	Eccentric air gap, shorted, or loose windings	2 *x* line frequency	Radial
	Loose rotor bar	1 *x* with 2 *x* slip frequency side bands	Radial
	Loose windings	2 *x* line and number of rotor bars x RPM	Radial

(1) 1*x*, 2*x*. . . N*x*—When not qualified, Nx refers to multiples of running speed.

Band Definitions and Limits

On small machines that are direct coupled to the driven component, significant vibration from the driven machine is present in the vibration measurements on the motor. Consequently consider the vane or blade pass frequency of the connected pump, fan, or blower in the configuration of the motor measurements.

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Set the Overall limits based on Table 10.

Table 10 - Alarm Limits—Small Electric Motor

Overall Alarm Limits	mm/s	In/s
Alarm 1	7.20	0.395
Alarm 2	10.65	0.593
Applicable ISO Standards (other standards can apply)	Not Applicable	

Band Alarms

Band alarms for a small motor will be the same as for a medium size motor. See. Table 12

Medium Electric Motors—15…375 KW (20…500 HP)

This section describes medium electric motors.

See Electric Motors for further detail on motors.

Sensor and Monitor Requirements

Mount one sensor per bearing, two general-purpose sensors total. One Dynamic Measurement Module is required. The monitor can be shared with the sensors that are used for the driven machine.

No Expansion Tachometer Signal Conditioner Module or additional Expansion Relay Module is required.

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

AC induction motor faults are the same regardless of size. See Table 9 for the faults applicable to motors.

Band Definitions and Limits

On moderate sized machines that are direct coupled to the driven component, significant vibration from the driven machine can be present in the vibration measurements on the motor. Consequently, and particularly if a smaller machine, consider the vane, or blade pass frequency of the connected pump, fan, or blower in the configuration of the motor measurements.

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2 x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Set the Overall limits based on Table 11.

Table 11 - Alarm Limits—Medium Electric Motor

Overall Alarm Limits	mm/s	In/s
Alarm 1	2.80	0.155
Alarm 2	4.20	0.234
Applicable ISO Standards (other standards can apply)	10816-3	

FFT Band Definitions and Alarm Limits

Apply the following FFT Band measurements to small and medium size electric motors.

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller.

Do not apply the eight bands that are defined for large motors in Table 12 to small motors. Small motors do not consider vibration from the driven component, which is prominent in small direct coupled machines.

Table 12 - Band Definitions—Medium Electric Motor

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	4*x*	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Misalignment, Looseness	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	3.2…4.2*x*	4.2…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	25%	25%	20%	15%
Alarm 1 mm/s		0.42	2.52	0.98	0.98	0.7	0.7	0.56	0.42
Alarm 2 mm/s		0.63	3.78	1.47	1.47	1.05	1.05	0.84	0.63
Alarm 1 in/s		0.023	0.140	0.054	0.054	0.039	0.039	0.031	0.023
Alarm 2 in/s		0.035	0.210	0.082	0.082	0.058	0.058	0.047	0.035

Large Electric Motors—375 KW (500 HP) and Above

This section describes medium electric motors.

See Electric Motors for further detail on motors.

Sensor and Monitor Requirements

Depending on available space, mount two general-purpose sensors per bearing, four sensors total. One Dynamic Measurement Module is required.

Machine Faults

AC induction motor faults are the same regardless of size. See Table 9 for the faults applicable to motors.

Band Definitions and Limits

On large machines, even when direct coupled to the driven component, a little vibration from the driven machine is present in the vibration measurements on the motor. Fault frequencies that are associated with the connected machine need not be considered in the configuration of the motor measurements, particularly on a large machine. If a problem develops in the connected machine that results in significant vibration, it is likely the vibration transmits to the motor and would see increased overall vibration levels.

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Set the Overall limits based on Table 13

Table 13 - Alarm Limits—Large Electric Motor

Overall Alarm Limits	mm/s	In/s
Alarm 1	4.5	0.25
Alarm 2	6.75	0.376
Applicable ISO Standards (other standards can apply)	10816-3	

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller.

See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

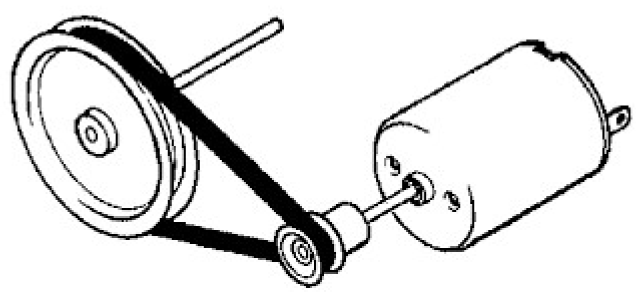
Table 14 - Band Definitions—Large Electric Motor

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	4*x*	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Misalignment, Looseness	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	3.2…4.2*x*	4.2…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	25%	25%	20%	15%
Alarm 1 mm/s		0.68	4.05	1.58	1.58	1.125	1.125	0.9	0.675
Alarm 2 mm/s		1.01	6.08	2.36	2.36	1.6875	1.6875	1.35	1.0125
Alarm 1 in/s		0.037	0.225	0.087	0.087	0.062	0.062	0.05	0.037
Alarm 2 in/s		0.056	0.338	0.132	0.132	0.094	0.094	0.075	0.056

Electric Motor with Pulley

In most cases when a motor is connected to a belt drive, the pulley wheel is affixed directly to the motor shaft. Consequently, in these cases, the vibration that is measured at the motor bearings includes vibrations that are associated with the pulley.

See Electric Motors for further detail on motors.



Sensor and Monitor Requirements

The number and location of general-purpose sensors are based on the motor size. It is either small, medium, or large, which can be determined with the preceding sections.

No additional sensors are required for the pulley.

Machine Faults

An electric motor with a pulley can exhibit vibration at frequencies that are associated with an AC induction motor, the same as any motor, and as described in Table 9. However, with a pulley that is directly attached to its shaft, the vibration that is measured at the motor bearing locations also include vibration that is associated with the pulley.

Table 15 - Machine Faults—Electric Motor with Pulley

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Worn or loose belts	Belt frequency, 1 *x* RPM[3]	Radial and axial
	Misaligned pulleys	1 *x* driver or 1 *x* driven machine	Axial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Vane pass or blade pass[2]	1 *x* V/BP, can be caused by Flow restrictions such as incorrect damper settings	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Radial
	Rotor rub	0.5/1/1.5/2/2.5/3/3.5*x* high levels of half order harmonics	Radial

(1) 1*x*, 2*x*. . . N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Vane pass or blade pass—Number of vanes or blades x running speed

(3) Belt frequency—See Belt Frequency on page 42 for a further discussion.

Belt Frequency

The most significant vibration that is associated with pulley systems occurs at a belt frequency, which is related to the pulley sizes and the belt length. The belt frequency is calculated as shown in Table 16.

Table 16 - Calculating Belt Frequency

Belt Frequency Calculation			
$BFx = \frac{3.142 \times PDi}{BL}$		$BL = 1.57 \times (PDi + PDo) + 2 \times Ds$	
BFx	Belt frequency (orders)	PDi	Drive (input) pulley diameter
BL	Belt length	PDo	Driven (output) pulley diameter
RPM	RPM of the drive pulley	Ds	Distance between shafts
Example			
BFx	Belt frequency (orders)	PDi	12
BL	Belt length	PDo	24
RPM	1500 rpm	Ds	48
$BFx = \frac{3.142 \times 12}{152.52}$		$BL = 1.57 \times (12 + 24) + 2 \times 48$	
$BFx = 0.2472$		$BL = 152.52$	

The Belt Frequency is in orders. Multiply by RPM to convert to CPM or RPM/60 for Hz.

- For the example, at 1500 rpm: BF = 1500 x 0.2472 = 370.81 cpm (6.18 Hz)

The Driven Shaft RPM is equal to the Input Shaft RPM x PDi/PDo,

- For the example: Driven Shaft RPM = 1500 x 12/24 = 750

The diameter and distance units cancel so can be any units but the same units must be used for each of the parameters.

Band Definitions and Limits

On belt driven machines, driven machine vibration is not normally transmitted to the motor through the belt drive. However, in many cases the motor is mounted to the same structure as the driven machine, so vibration that is associated with the driven machine is transmitted through the structure. Consequently, observe motor vibration to determine if driven component vibration is present, before finalizing band definitions.

The following assumes that the motor is relatively isolated from the driven machine.

If the Belt Frequency is known, then consider redefining the bands to monitor this frequency.

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2X the machine running speed, for example, 0.2 X 50 Hz = 10 Hz.

Set the Overall limits based on Table 17

Table 17 - Alarm Limits—Electric Motor with Pulley

Overall Alarm Limits	mm/s	In/s
Alarm 1	8.10	0.449
Alarm 2	11.20	0.623
Applicable ISO Standards (if >15 KW) other standards can apply.	10816-3	

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Table 18 - Band Definitions—Electric Motor with Pulley

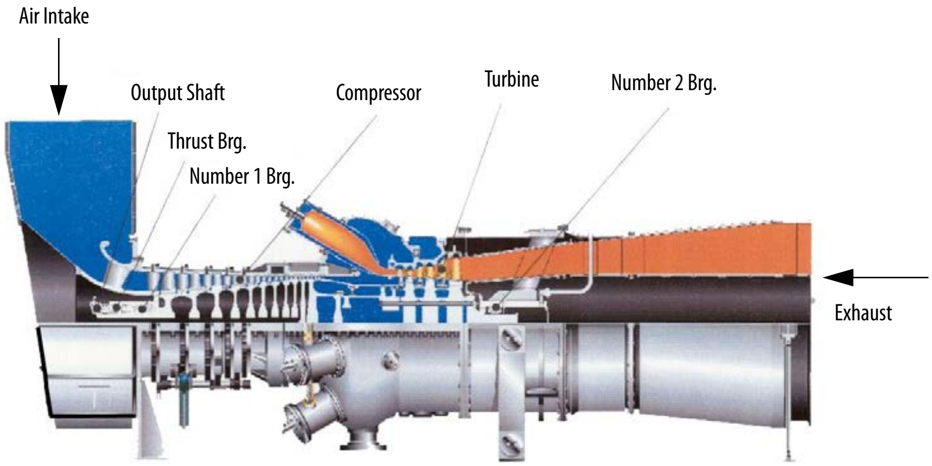
Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	4*x*	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Misalignment, Looseness	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	3.2…4.2*x*	4.2…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	25%	25%	25%	20%	15%
Alarm 1 mm/s		0.68	7.29	2.84	1.125	1.125	2.03	1.62	1.215
Alarm 2 mm/s		1.01	10.08	3.92	1.6875	1.6875	2.80	2.24	1.68
Alarm 1 in/s		0.037	0.404	0.157	0.062	0.062	0.112	0.090	0.067
Alarm 2 in/s		0.056	0.561	0.218	0.094	0.094	0.156	0.125	0.094

Gas Turbines

Most industrial gas turbines are fitted with fluid film (journal) bearings while most aeroderivative engines use rolling element bearings. Verify the bearing types used.

In most cases, industrial gas turbines are large engines that are often used in cogeneration applications where their exhaust gases are used to make steam for associated steam turbines or for other uses. Smaller aeroderivative engines are commonly used in mechanical drive applications such as for compressors or pumps.

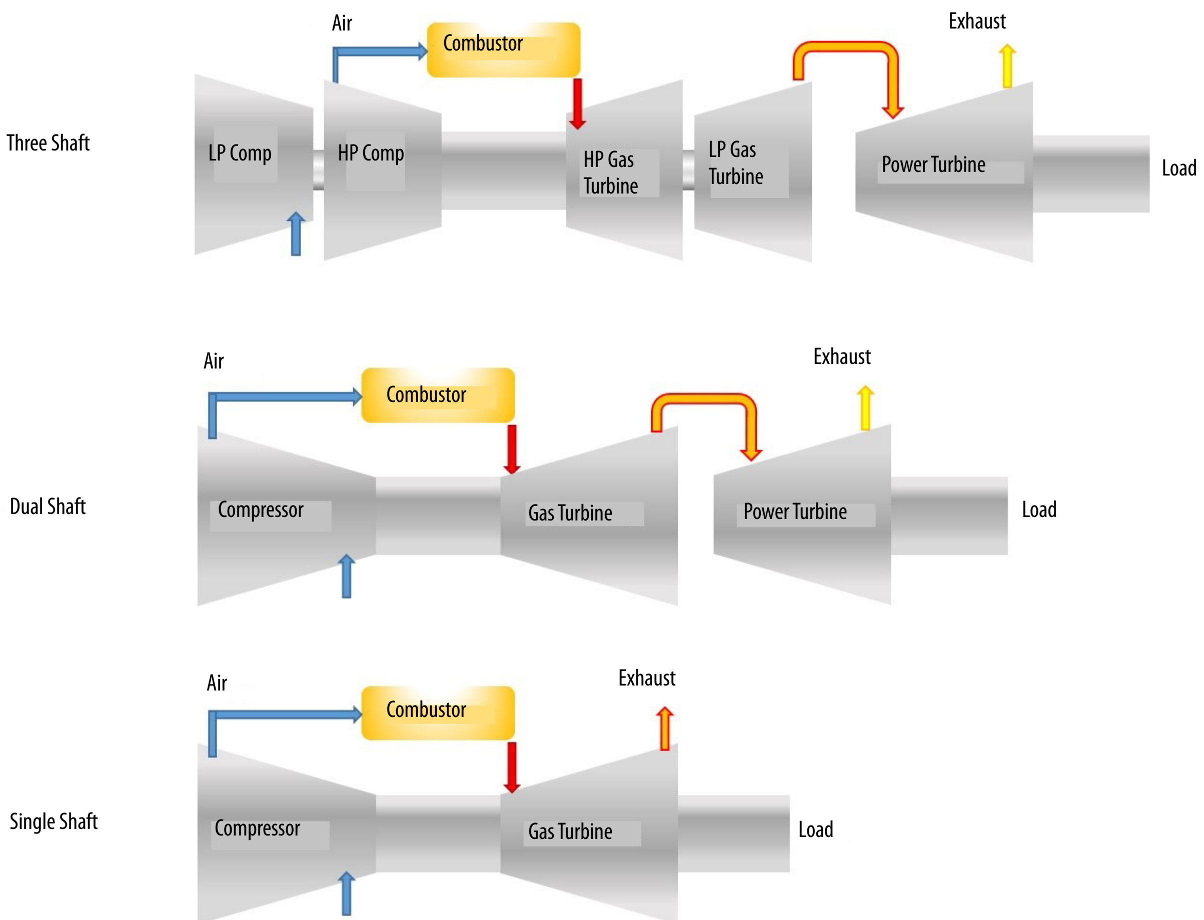
Figure 2 - Gas Turbine



If eddy current probes are used, then it is using fluid film bearings.

In addition to bearing type, an important consideration is the shaft arrangement. Is the engine one, two, or three shaft design? In particular, a three shaft design requires monitoring the high and low (or intermediate) pressure shafts using a common sensor as these designs do not generally allow mounting sensors on the LP shaft bearings.

Figure 3 - Turbine Shaft Arrangement



Sensor and Monitor Requirements

Sensor selection and placement for gas turbines are specified by the manufacturer.

In most cases, gas turbine applications are retrofits. In these cases, if the existing sensors must be replaced then equivalent 1443 Series sensors are available. However, some measurement locations on some gas turbines require sensors that are tolerant of high temperatures[1] that are not always available from Rockwell Automation®. When selecting sensors, verify that the existing sensors are identified and properly cross referenced to available 1443 Series sensors, or to third-party sensors if necessary.

Monitor selection is based on the number of sensors that are required, see Dynamic Measurement Module on page 28. In most cases, gas turbines also include one or two tachometers, which requires adding a Tachometer Signal Conditioner Expansion Module. In some cases, the entire machine has more than two-speed sensors, in which case an addition tachometer signal conditioner is required.

If the application requires local relays for trip or annunciation, and if multiple relays are required, then add additional Relay Expansion Modules.

Machine Faults

The order values that are indicated in the faults table refer to orders of the sensed shaft, which depends on the shaft arrangement.

Table 19 - Shaft Arrangement

Number of Shafts		
One	**Two**	**Three**
Gas turbine	Gas generator	Gas generator HP
		Gas generator LP (or IP)
	Power turbine	Power turbine

For three-shaft designs, the sensors that monitor the HP and LP bearings are common to both shafts. In these cases, both order frequencies are monitored from the same sensor.

(1) Extreme temperature sensors are piezoelectric accelerometers or velocity output sensors with charge mode amplifiers that are built into the sensor cable, rather than the sensor, to move the electronics as far from the heat as is practical.

Table 20 - Machine Faults— Gas Turbines Fitted with Rolling Element Bearings

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and axial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5/1/1.5/2/2.5/3/3.5*x* high levels of half order harmonics	Radial

(1) 1*x*, 2*x*. . . N*x*—When not qualified, Nx refers to multiples of running speed.

Band Definitions and Limits

The many varied types and designs of gas turbines make it impractical to define a specific set of FFT Band definitions to monitor them. Additionally, gas turbines likely require monitoring of the 1x vibration using tracking filters or consider band pass filters in any diagnostics applied.

Verify that any monitoring application that is implemented satisfies the protection monitoring of the engine manufacturer.

The following guidelines can be used to assess overall vibration level, the term Normal is used to indicate the expected maximum level of a new machine or a machine in good condition.

Guidelines for overall values of absolute vibration that is measured by accelerometers or velocity output devices such as integrating accelerometers mounted on the machine bearing housing or machine case, mm/s RMS (in/s Peak).

Table 21 - Alarm Limits—Gas Turbines

Shaft Speed RPM		3000 or 3600
Normal mm/s RMS (in/s peak)		4.5 (0.25)
Alarm 1	Presence of a possible anomaly requiring remedial action	9.3 (0.52)
Alarm 2	Machine is unsatisfactory for continued operation	14.7 (0.82)
Applicable ISO Standards (if >15 KW) other standards can apply.		7919-4, 10816-3

Wind Turbines

Industrial wind turbine designs vary from manufacturer to manufacturer and come in sizes from less than 1 to over 8 MW with new designs and larger size turbines being introduced continually. Consequently there is no standard definition of how to monitor a wind turbine, including the number and arrangement of sensors. A standard though is provided for the 'Measurement and evaluation of the mechanical vibration of wind energy turbines and their components, onshore wind energy turbines with gears', standard VDI 3834.

The two types of wind turbine generators are Double-Fed Induction Generators and Direct-driven Wind Turbine Generators.

Figure 4 - Double-fed Induction Generator

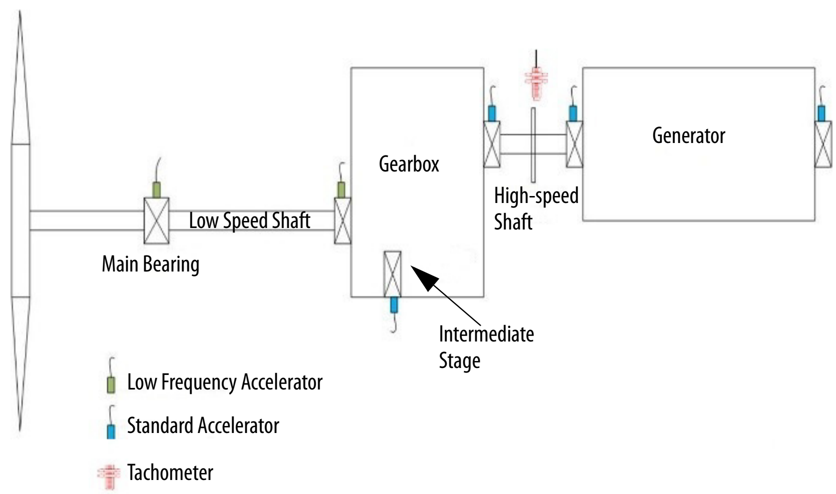
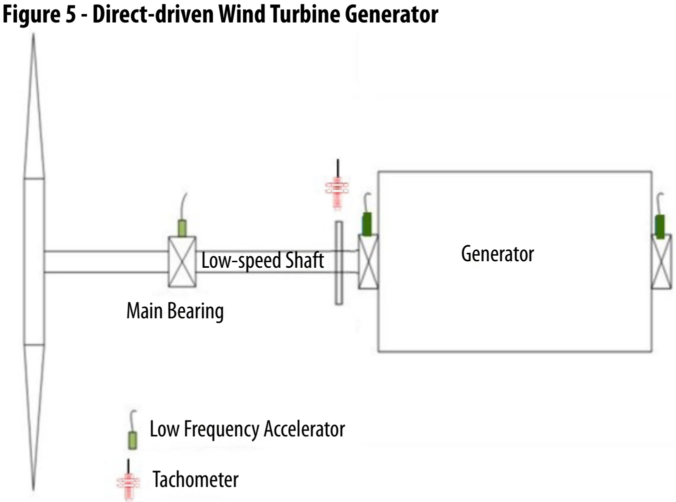


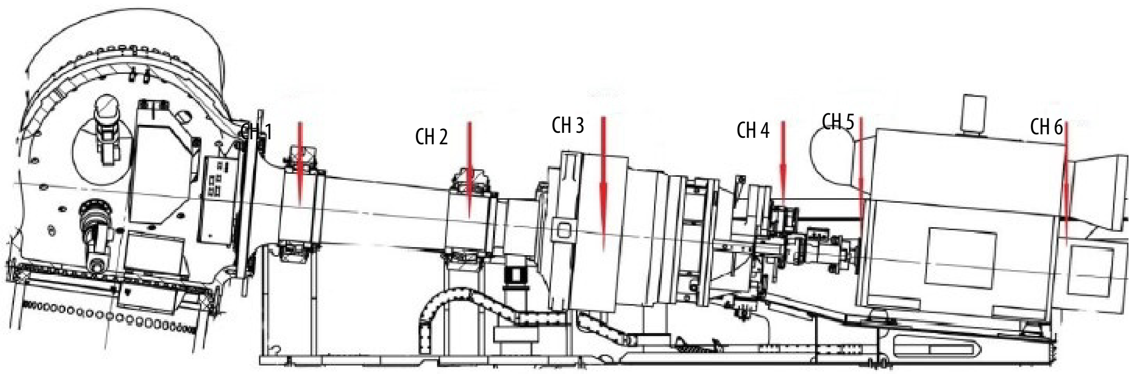
Figure 5 - Direct-driven Wind Turbine Generator



The shaft speed out of the gear set is fixed at either 1500 rpm or 1800 rpm. The low speed shaft varies in speed from as low as 5 to over 20 rpm, depending on the design.

Figure 6 shows a typical double-fed example. The turbine blades drive a low speed shaft through a speed increasing single-stage planetary gear, two stages spur gears arrangement(1) and, a generator. The shaft speed out of the gear set is fixed at either 1500 rpm or 1800 rpm.

Figure 6 - Double-fed Generator



The low speed shaft varies in speed from as low as 5 to over 20 rpm, depending on the design, see Table 22.

Table 22 - Common Turbines

Model	Capacity (MW)	Minimum RPM	Maximum RPM
Bonus (Siemens)	1.3	13	19
Bonus (Siemens)	2	11	17
Bonus (Siemens)	2.3	11	17
Clipper Liberty	2.5	9.7	15.5
Enercon E-126	7.6	5	11.7
Gamesa G87	2	9	19
GE 1.5s	1.5	11.1	22.2
GE 1.5sle	1.5	Unknown	
Goldwind	2.5	7	16
Mitsubishi MWT95	2.4	9	16.9
Repower MM92	2	7.8	15
Siemens	2.3	7	16
Suzlon 950	0.95	13.9	20.8
Suzlon S64	1.25	13.9	20.8
Suzlon S88	2.1	Unknown	
Vestas V100	2.75	7.2	15.3
Vestas V112	3	6.2	17.7
Vestas V82	1.65	Unknown	14.4
Vestas V90	1.8	8.8	14.9
Vestas V90	3	9	19

(1) There are other types of gear arrangements such as a two stage planetary gear with one stage spur gear.

Sensor and Monitor Requirements

At minimum, mount two low-frequency and four general-purpose sensors as illustrated in Figure 6. A total of six sensors and two Dynamic Measurement Modules are required.

Set the measurement parameters (example with single-stage planetary gear arrangement) shown in Table 23.

Table 23 - Sensor and Monitor Requirements

CH	Description	Sensitivity	Unit	HP Filter	FMAX
1	Main Bearing	500 mV/g	g	0.1	39
2	Inboard of Gearbox	500 mV/g	mm/s	1.0	107
3	Case of Ring Gear	100 mV/g	mm/s	1.0	1287
4	Outboard of Gearbox	100 mV/g	mm/s	5.0	1287
5	Inboard of Gearbox	100 mV/g	mm/s	5.0	1287
6	Outboard of Generator	100 mV/g	mm/s	5.0	1287

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

The FFT Bands for each channel of the Dynamix™ 1444 module must be configured to suit the measurement location. In most cases the general guidelines for the component type, described elsewhere in this document, apply.

Table 24 - Machine Faults—Wind Turbines

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	Main Shaft: 1*x* N_0 Gearbox: 1*x* N_1, 1x N_2, 1x N_4, Generator: 1*x* N_5	Radial
	Bearing Anomalies	Bearings of $B_0 \sim B_{11}$ Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Blade pass[2]	1 *x* BP, can be caused by flow restrictions such as incorrect damper settings	Radial
	Gear problems such as tooth wear, eccentricity, backlash, gear misalignment, which is cracked or broken tooth	Planetary drive (Z1, Z2, Z3): GM, SDF, RDF, PDF, 2 *x* GM, 3 *x* GM Frequency, and Gear Natural Frequencies. Spur Gear Drive (Z1p, Z4, Z4p, Z5): GM, 2 *x* GM, 3 *x* GM Frequency, and Gear Natural Frequencies	Radial and Axial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5*x* high levels of half order harmonics	Radial
	Eccentric air gap, shorted, or loose windings	2*x* line frequency	Radial
	Loose rotor bar	1*x* with 2*x* slip frequency side bands	Radial
	Loose windings	2*x* line and number of rotor bars *x* RPM	Radial

(1) 1*x*, 2*x*. . . N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Blade pass—Number of blades x running speed

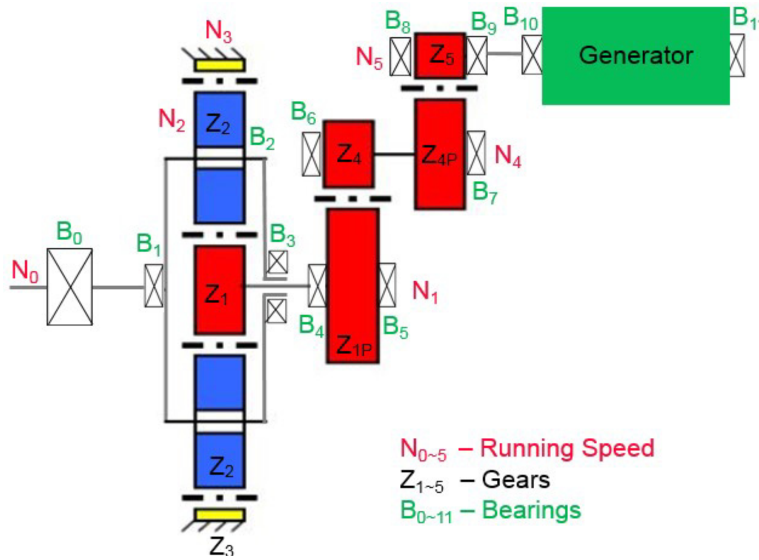


Table 25 - Parameters

Parameter	Description	Formula
N_0	Main shaft speed	Measured
N_1	Sun gear speed	$N_1 = \left(1 + \frac{Z_3}{Z_1}\right) \times N_0$
N_2	Planetary gear speed	$N_2 = \left(1 - \frac{Z_3}{Z_2}\right) \times N_0$
N_3	Ring gear speed	$= 0$
N_4	Output speed of first stage of spur gear drive	$N_4 = \frac{Z_{1P}}{Z_4} \times \left(1 + \frac{Z_3}{Z_1}\right) \times N_0$
N_5	Output speed of second stage of spur gear drive	$N_5 = \frac{Z_{4P}}{Z_5} \times \frac{Z_{1P}}{Z_4} \times \left(1 + \frac{Z_3}{Z_1}\right) \times N_0$

Wind Turbine Generator Limits

While the Dynamic module can perform the required measurements, over the required frequency ranges for each of the components, it cannot perform the required averaging of the measurements. Per VDI 3834, wind turbine monitoring requires that the (overall) vibration measurements be averaged using a method that is called energy-equivalent averaging. The key to this function is the period over which the measurement is averaged which, depending on the component that is measured, can be ten minutes or possibly longer.

The 1444 Series Dynamic Measurement Module cannot perform this averaging within the module, however, the required averaging, applied over the required period, can be implemented in the controller.

When monitoring a wind turbine, review OEM recommendations, standard VDI 3834, and other applicable standards, and then verify that an appropriate averaging and alarm solution is implemented in the controller.

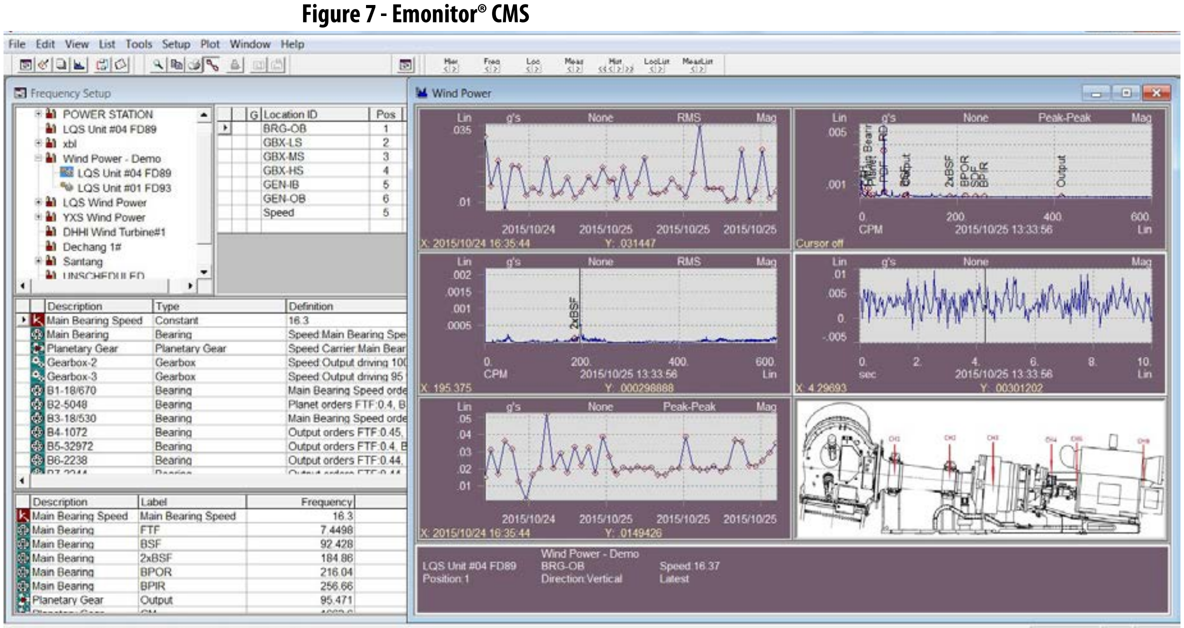
Per VDI 3834 monitoring wind turbines requirement, it can use as overall alarms typically configured in the monitor.

Table 26 - Alarm Limits—Wind Turbines

Overall Alarm Limits	Main Bearing	Gearbox	Generator
	g, RMS	mm/s, RMS	mm/s, RMS
Alarm 1	0.03	3.50	6.00
Alarms 2	0.05	5.60	10.00
Applicable ISO Standards (other standards can apply)	VDI 3834		

Typical measurement analysis is applied with Emonitor® CMS shown in Figure 7.

Figure 7 - Emonitor® CMS



Generators

While large generators are fitted with Fluid Film Bearings, smaller machines such as diesel, wind turbine driven, and gas turbine driven generators use common rolling element bearings. In these cases, the monitoring solution, expected behavior, and diagnostics are similar to a large electric motor.

Sensor and Monitor Requirements

Mount two general-purpose sensors per bearing, four sensors total. One Dynamic Measurement Module is likely to be required.

If a tachometer (speed) sensor is available, then include a Expansion Tachometer Signal Conditioner Module.

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

Electric generator faults are similar to AC induction motor faults.

Table 27 - Machine Faults—Generators

Types of Faults	Possible Problems(1)	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and Axial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5*x* high levels of half order harmonics	Radial
	Eccentric air gap, shorted, or loose windings	2*x* line frequency	Radial
	Loose rotor bar	1*x* with 2*x* slip frequency side bands	Radial
	Loose windings	2*x* line and number of rotor bars *x* RPM	Radial

(1) 1*x*, 2*x*. . . N*x*—When not qualified, Nx refers to multiples of running speed.

Band Definitions and Limits

The following guidelines can be used to assess Overall vibration level, the term normal is used to indicate the expected maximum level of a new machine or a machine in good condition.

Guidelines for Overall values of Absolute Vibration that is measured by Accelerometers or Velocity output devices such as integrating accelerometers mounted on the machine bearing housing or machine case, mm/s RMS (in/s Peak).

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Table 28 - Alarms Limits—Generators

Overall Alarm Limits	mm/s	In/s
Alarm 1	4.50	0.251
Alarms 2	6.75	0.376
Applicable ISO Standards (other standards can apply)	7919-2, 10816-2	

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

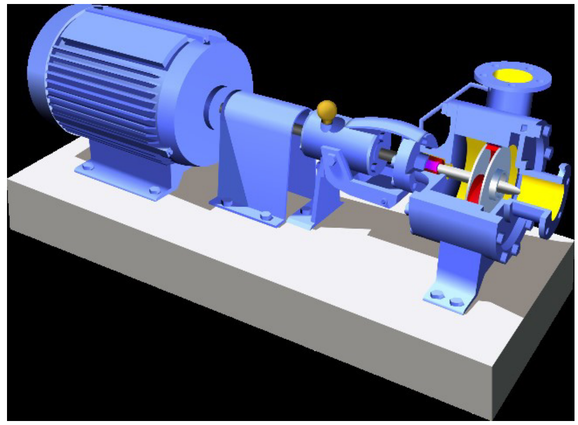
Table 29 - Band Definitions—Generators

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	4*x*	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Misalignment, Looseness	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	3.2…4.2*x*	4.2…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	25%	25%	20%	15%
Alarm 1 mm/s		0.68	4.05	1.58	1.58	1.125	1.125	0.9	0.675
Alarm 2 mm/s		1.01	6.08	2.36	2.36	1.6875	1.6875	1.35	1.0125
Alarm 1 in/s		0.037	0.225	0.087	0.087	0.062	0.062	0.05	0.037
Alarm 2 in/s		0.056	0.338	0.132	0.132	0.094	0.094	0.075	0.056

Pumps—Horizontal Mount

Pumps are the most common machine in industry. They are also the single largest consumer of power by machine type.

There are two basic types of pumps: positive displacement and centrifugal. This guide references only centrifugal pumps.



Pumps are classified many ways but the characteristics most important to monitoring are the number of impellers and type of volute.

Table 30 - Impellers and Volutes

Number of Impellers		
Single stage	Pump has one impeller only	For low head service
Two stage	Pump has two impellers in series	For medium head service
Multi-stage	Pump has three or more impellers in series	For high head service
Type of Volute		
Single	Volute has single lip that is easy to cast	Used in small low capacity pumps. Have higher radial loads.
Double	Volute has dual lips located 180° apart	Have balanced radial loads

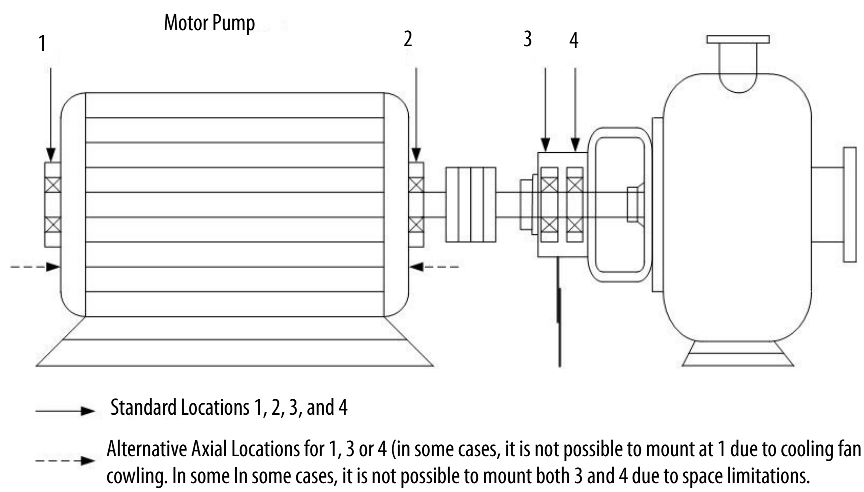
In addition to the impellers and volutes, pumps can fall into one of two basic machine configurations. These configurations are overhung (OH) and between bearings (BB). The overhung type can be close coupled where the impeller is attached directly to the motor shaft, above and below, or separately coupled where there is a coupling between the motor shaft and the pump impeller shaft, below. The between bearing type is likely to have a coupling between the motor shaft and pump shaft, and the impeller is likely to be supported by two bearings on either end of the shaft, below. Each design has specific uses based on flow rates, product to be pumped, and so on. For the purposes of this discussion, only the vibration monitoring considerations are included.

All machine faults and band definitions that are shown below are appropriate for each type. The main difference with overhung design is the fact that any tilt of impeller from perpendicular to shaft, 90° angle, can cause an increase in various measures in the axial direction. For that reason, it is recommended that one of the pump sensors be considered to be in the axial plane, similar to an overhung fan. Balance is more difficult and any shaft to impeller angle can cause accelerated bearing failure.

Figure 8 - Pump Bearing Configurations—Left: Between Bearings, Right: Close Coupled Overhung

Sensor and Monitor Requirements

Figure 9 - Motor Pump



For small pumps, less than 500 HP, place one sensor in the radial (horizontal) direction at each bearing.

For large pumps, greater than 500 HP, place two sensors at each bearing. If the pump is center mount, mount all sensors in the radial direction (vertical and horizontal). If the pump is overhung, such as illustrated in Figure 9, then mount one of the sensors at the pump inboard bearing in the axial direction (axial and horizontal).

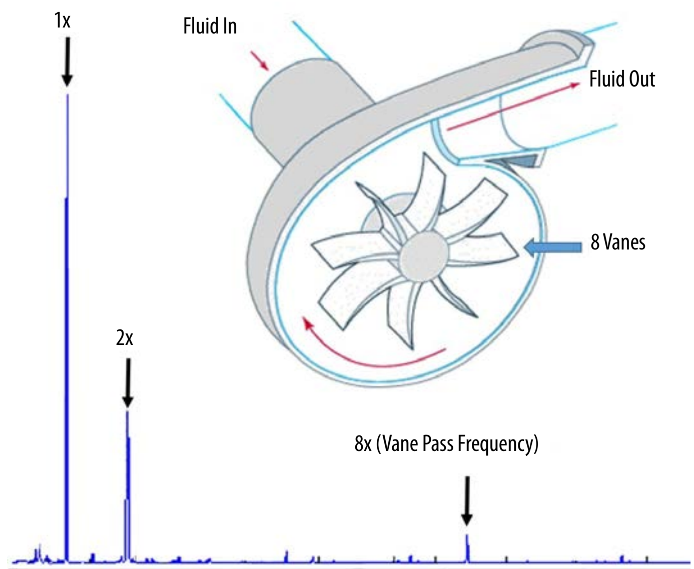
See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

The following table is for a single stage, single volute centrifugal pump.

For a multi-stage pump, the vane pass[1] of each stage is monitored. Each stage has another number of vanes.

Figure 10 - Pump Fan



The pressure at the volute (fluid out) varies as the impeller rotates, depending on how exactly the vanes align with the outlet at any given moment.

Consequently, for any centrifugal pump, the pressure at the volute pulsates at a frequency equal to the number of vanes that are multiplied by the speed of the pump. This value is called the Vane Pass Frequency.

For a double volute pump, the vane pass frequency is at two times the number of vanes.

(1) Figure 10 describes the fundamentals of vane pass in a pump and how it is depicted in the FFT spectrum.

Table 31 - Machine Faults—Horizontal Mount Pumps

Types of Faults	Possible Problems(1)	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment(2)	1*x*/2*x*/3*x*	Radial and Axial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/ roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Vane pass(3)	Number of vanes x running speed, can be caused by flow restrictions or uneven gap between rotating vanes and stationary diffuser.	Radial
	Cavitation(4)	Typically caused by low suction pressure.	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial

(1) 1*x*, 2*x*. . . N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Misalignment is not valid with a close coupled overhung impeller configuration.

(3) The numbers of vanes or blades on the impeller times running speed. If the number of vanes is unknown set this band to 4*x* to 7*x* running speed (Pumps often have from four to seven vanes).

(4) Indicated by random High frequency vibration above Vane Pass frequency. Vibration, related to excited structural resonance frequencies, can occur at frequencies from the Vane Pass frequency up to the Maximum Frequency. As cavitation-related vibration can be confused with bearing Anomalies, the presence of cavitation can be minimized before suspecting bearing problems.

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Table 32 - Alarms Limits—Centrifugal Pumps

Centrifugal Pumps	Overall Alarm Limits	mm/s	In/s
Centrifugal Pumps up to 500 HP	Alarms 1	2.80	0.156
	Alarms 2	4.20	0.234
Centrifugal Pumps greater than 500 HP	Alarm 1	5.40	0.300
	Alarm 2	8.10	0.451
Applicable ISO Standards (other standards can apply)	7919-2, 10816-2		

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Table 33 - Band Definitions—Centrifugal Pumps Greater Than 500 HP

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* BPF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Blade Pass Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 BPF	1.2 BPF…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	60%	25%	20%	15%
Alarm 1 mm/s		0.81	4.86	1.89	1.89	3.24	1.35	1.08	0.81
Alarm 2 mm/s		1.22	7.29	2.84	2.84	4.86	2.025	1.62	1.125
Alarm 1 in/s		0.045	0.270	0.105	0.105	0.180	0.075	0.060	0.045
Alarm 2 in/s		0.068	0.406	0.158	0.158	0.271	0.113	0.090	0.068

Pumps—Vertical Mount

Vertical pumps are in many respects identical to horizontal pumps other than the obvious orientation of the rotor shaft. Vibration is highly directional, so this orientation does change how the sensors are placed on the pump. Sensor mounting in the radial direction remains unchanged other than one sensor per bearing is all that is required. The axial, or vertical axis however becomes more important as the thrust of the entire motor rotor, shaft, and pump impeller is supported vertically. Note possible locations in the following drawing. Also, the highest radial vibration amplitude is typically in line with the direction of inflow into the pump from the pump-body horizontal piping connection. As close to that orientation as practical, is the best location around the outside of the pump and motor for the radial sensors.

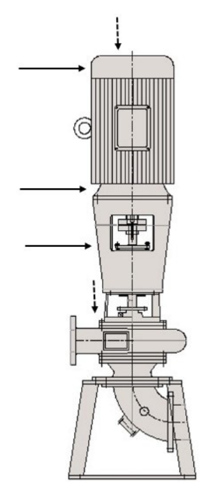
Table 34 - Vertical Pump Impellers and Volutes

Number of Impellers		
Single stage	Pump has only one impeller	For low head service
Two stage	Pump has two impellers in series	For medium head service
Multi-stage	Pump has three or more impellers in series	For high head service
Type of Volute		
Single	Volute has a single lip that is easy to cast	Used in small low capacity pumps Have higher radial loads
Double	Volute has dual lips located 180° apart	Most centrifugal pumps are of double volute design Have balanced radial loads

Sensor and Monitor Requirements

See the appropriate section for the size motor used. The pump sensors must be placed at each bearing. However, vertical pumps in particular often have bearings only above the pump. Before selecting sensors, review the pump design to determine appropriate mounting locations. Also consider adding a sensor in the axial direction if a suitable radial mounting location isn't available for one or more bearings.

- If the pump is submersed, then one or both pump bearings are not always accessible
- When possible sensors are mounted in-line with the discharge piping



See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

The following table is for a single stage, single volute centrifugal pump. For a multi-stage pump, the vane pass of each stage must be monitored. Each stage has a different number of vanes.

See Machine Faults for Pumps—Horizontal Mount for an explanation of Vane Pass Frequency.

If a double volute pump the vane pass frequency is at two times the number of vanes.

Table 35 - Machine Faults—Vertical Pumps

Types of Faults	Possible Problems(1)	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and Axial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Vane pass(2)	Number of vanes x running speed, can be caused by flow restrictions or uneven gap between rotating vanes and stationary diffuser.	Radial
	Cavitation(3)	Typically caused by low suction pressure.	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3x can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial

(1) 1*x*, 2*x*...N*x*—When not qualified, Nx refers to multiples of running speed.

(2) The numbers of vanes or blades on the impeller times running speed. If the number of vanes is unknown set this band to 4...7x running speed (Pumps often have 4...7 vanes).

(3) Indicated by random high frequency vibration above vane pass frequency. Vibration, related to excited structural resonance frequencies, can occur at frequencies from the vane pass frequency up to the maximum frequency. As cavitation-related vibration can be confused with bearing anomalies, the presence of cavitation can be minimized before suspecting bearing problems.

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Table 36 - Alarm Limits—Vertical Pumps

Overall Alarm Limits	mm/s	In/s
Alarms 1	7.10	0.395
Alarms 2	10.65	0.593
Applicable ISO Standards (other standards can apply)	10816-7	

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Table 37 - Band Definitions—Vertical Pumps

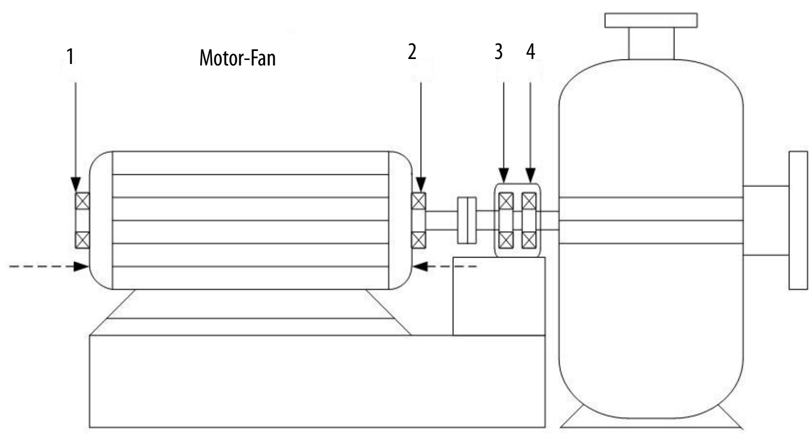
Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* BPF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Blade Pass Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 BPF	1.2 BPF…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	60%	25%	20%	15%
Alarm 1 mm/s		1.065	6.39	2.49	2.49	4.26	1.78	1.42	1.065
Alarm 2 mm/s		1.60	9.59	3.73	3.73	6.39	2.66	2.13	1.60
Alarm 1 in/s		0.059	0.356	0.138	0.138	0.237	0.099	0.079	0.059
Alarm 2 in/s		0.089	0.534	0.208	0.208	0.356	0.148	0.119	0.089

Fans and Blowers—General

Blowers and fans are among the most common machines in Industry. They are used for ventilation, aeration, exhaust, cooling, drying, and more in almost every industry from cement to petrochemical to metals processing. Available with flow rates up to 2,000,000 cubic feet (57,000 cubic meters) per minute, all sizes of electric motors can drive blowers with direct or belt drive systems. Gas or steam turbines can drive the blowers.

Sensor and Monitor Requirements

See the appropriate section for the motor size. The fan requires one or two general-purpose sensors that are mounted at the illustrated locations.



Standard Locations 1, 2, 3, and 4

Alternative Axial locations 1, 3, or 4 (in some cases, it is not possible to mount at 1 due to cooling fan cowling. In some cases it is not possible to mount both 3 and 4 due to space limitations).

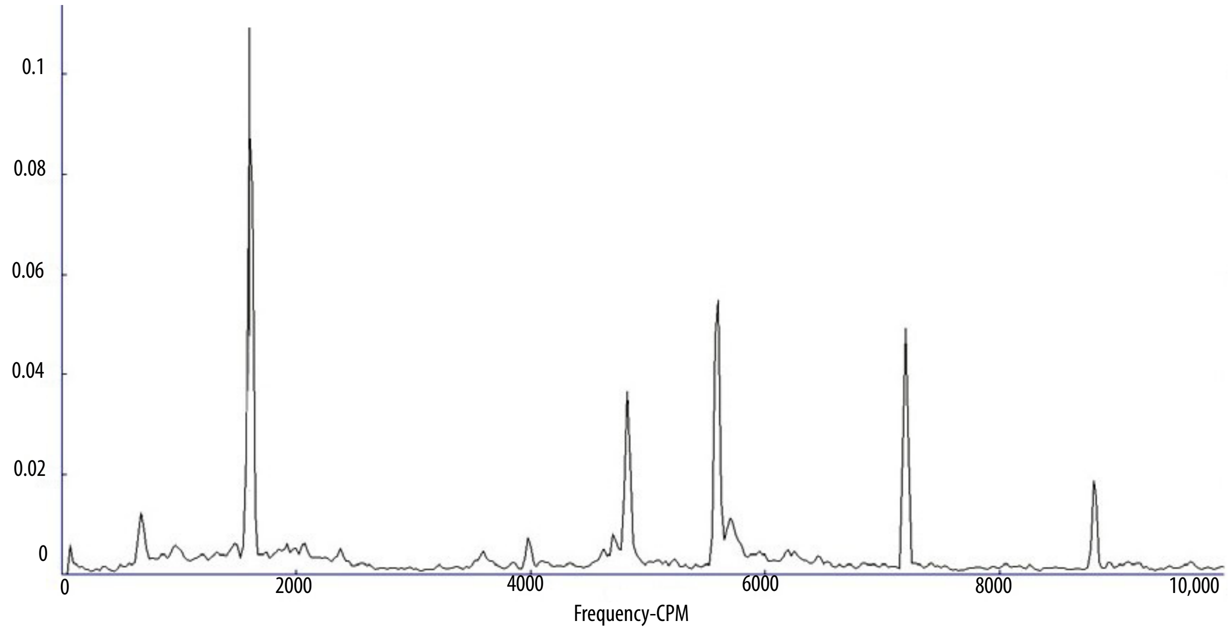
For small fans, less than 500 HP, place one sensor in the radial (horizontal) direction at each bearing.

For large fans, greater than 500 HP, place two sensors at each bearing. If the fan is center mount, mount all sensors in the radial direction (vertical and horizontal). If the fan is overhung, such as illustrated above, then mount one of the sensors at the fan inboard bearing in the axial direction (axial and horizontal).

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

The following table is for a single stage machine. For a multi-stage machines, the blade pass of each stage must be monitored. Each stage has a different number of blades.



The pressure at the outlet varies as the fan rotates, depending on how exactly the blades align with the outlet at any given moment.

Consequently, for any fan, the pressure at the outlet pulsates at a frequency equal to the number of blades that are multiplied by the speed of the fan. This value is called the blade pass frequency.

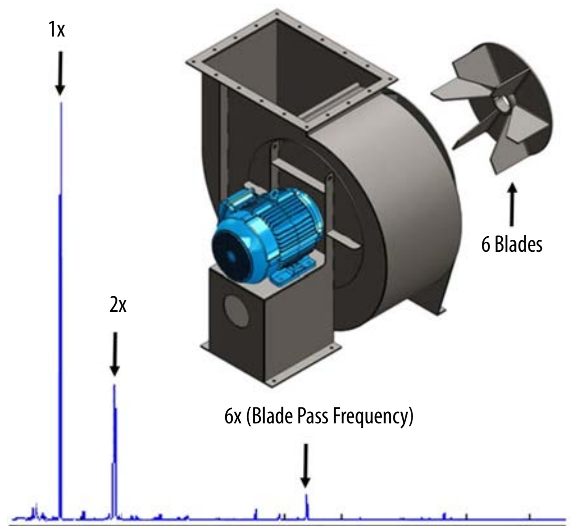


Table 38 - Machine Faults—Fans and Blowers—General

Types of Faults	Possible Problems(1)	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and Axial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Blade pass(2)	1 *x* BP, can be caused by flow restrictions such as incorrect damper settings	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial

(1) 1*x*, 2*x*…N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Number of blades *x* running speed

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Table 39 - Alarms Limits—Fans and Blowers—General

Overall Alarm Limits	mm/s	In/s
Alarms 1	4.5	0.251
Alarms 2	6.75	0.376
Applicable ISO Standards (other standards can apply)	14694, 10816-7	

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Table 40 - Band Definitions—Fans and Blowers—General

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* BPF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Blade Pass Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 BPF	1.2 BPF…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	60%	25%	20%	15%
Alarm 1 mm/s		0.68	4.05	1.58	1.58	2.7	1.13	0.90	0.68
Alarm 2 mm/s		1.01	6.08	2.36	2.36	4.05	1.69	1.35	1.01
Alarm 1 in/s		0.037	0.225	0.087	0.087	0.151	0.062	0.050	0.037
Alarm 2 in/s		0.056	0.338	0.132	0.132	0.226	0.094	0.075	0.056

Fans and Blowers—Large

Sensor and Monitor Requirements

See the appropriate section for the size motor. The fan requires one or two general-purpose sensors that are mounted at the illustrated locations.

For small fans, less than 500 HP, place one sensor in the radial (horizontal) direction at each bearing.

For large fans, greater than 500 HP, place two sensors at each bearing. If the fan is center mount, mount all sensors in the radial direction (vertical and horizontal). If the fan is overhung, such as illustrated previously, then mount one of the sensors at the fan inboard bearing in the axial direction (axial and horizontal).

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

The following table is for a single stage machine. For a multi-stage machines, the blade pass of each stage must be monitored. Each stage has a different number of blades.

See Machine Faults for Fans and Blowers—General for an explanation of Blade Pass Frequency.

Table 41 - Machine Faults—Large Fans and Blowers

Types of Faults	Possible Problems(1)	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Blade Pass(2)	1 *x* BP, can be caused by flow restrictions such as incorrect damper settings	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial

(1) 1*x*, 2*x*. . .N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Number of blades *x* running speed

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Table 42 - Alarm Limits—Large Fans and Blowers

Overall Alarm Limits	mm/s	In/s
Alarms 1	7.10	0.395
Alarms 2	10.65	0.593
Applicable ISO Standards (other standards can apply)	14694, 10816-7	

FFT Band Definitions and Alarm Limits

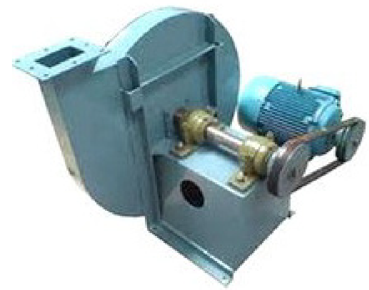
The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Table 43 - Band Definitions—Large Fans and Blowers

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* BPF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Blade Pass Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 BPF	1.2 BPF…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	60%	25%	20%	15%
Alarm 1 mm/s		1.065	6.39	2.49	2.49	4.26	1.78	1.42	1.065
Alarm 2 mm/s		1.60	9.59	3.73	3.73	6.39	2.66	2.13	1.60
Alarm 1 in/s		0.059	0.356	0.138	0.138	0.237	0.099	0.079	0.059
Alarm 2 in/s		0.089	0.534	0.208	0.208	0.396	0.148	0.119	0.089

Fans and Blowers —Belt Driven

Sensor and Monitor Requirements



See the appropriate section for the size motor.

The fan requires one or two general-purpose sensors that are mounted at each bearing.

For small fans, less than 500 HP, place one sensor in the radial (horizontal) direction at each bearing.

For large fans, greater than 500 HP, place two sensors at each bearing. If the fan is center mount, mount all sensors in the radial direction (vertical and horizontal). If the fan is overhung, such as illustrated above, then mount one of the sensors at the fan inboard bearing in the axial direction (axial and horizontal).

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

The following table is for a single stage machine. For a multi-stage machines, the blade pass of each stage must be monitored. Each stage has a different number of blades.

See Machine Faults for Fans and Blowers—General for an explanation of Blade Pass Frequency.

Table 44 - Machine Faults—Belt Driven Fans and Blowers

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Worn/Loose Belts	Belt frequency,[3] 1 *x* driver and 1 *x* driven machine	Radial and Axial
	Misaligned Pulleys	1 *x* driver or 1 *x* driven machine	Axial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Blade Pass[2]	1 *x* BP, can be caused by flow restrictions such as incorrect damper settings	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Radial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial

(1) 1*x*, 2*x*. . .N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Number of blades x running speed

(3) See Electric Motor with Pulley for information on how to calculate the belt frequency.

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Table 45 - Alarm Limits—Belt Driven Fans and Blowers

Overall Alarm Limits	mm/s	In/s
Alarms 1	8.10	0.449
Alarms 2	11.20	0.623
Applicable ISO Standards (other standards can apply)	14694, 10816-7	

FFT Band Definitions and Alarm Limits

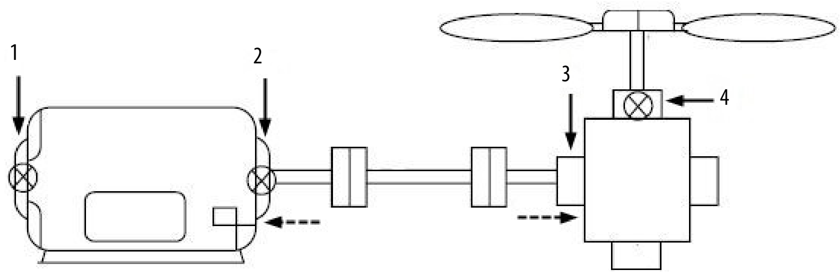
The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Table 46 - Band Definitions—Belt Driven Fans and Blowers

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* BPF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Blade Pass Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 BPF	1.2 BPF…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	60%	25%	20%	15%
Alarm 1 mm/s		1.215	7.29	2.84	2.84	4.86	2.03	1.62	1.215
Alarm 2 mm/s		1.68	10.08	3.92	3.92	6.72	2.80	2.24	1.68
Alarm 1 in/s		0.067	0.404	0.157	0.157	0.269	0.112	0.09	0.067
Alarm 2 in/s		0.094	0.561	0.218	0.218	0.374	0.156	0.125	0.094

Fans—Cooling Tower Fans

Figure 11 - Cooling Tower Fan



Sensor and Monitor Requirements

Mount two general-purpose sensors at the illustrated locations in Figure 11. Consider mounting a sensor in the axial direction at either of the inboard locations, which the dashed arrows show in Figure 11. Also consider the following options.

- If the fan speed(1) is less than approximately 700 RPM, then use a low frequency sensor at the fan bearing location, number four in Figure 11.
- Consider using an integral cable design sensor for sensor locations that are sometimes mounted in wet environments.

See Bill of Materials for the specific catalog numbers and quantities.

(1) Large cooling tower fans can operate at speeds as low as 125 RPM.

Machine Faults

See Machine Faults for Fans and Blowers—General for an explanation of blade pass frequency.

Table 47 - Machine Faults—Cooling Tower Fans

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Blade Pass[2]	1 x BP, can be caused by flow restrictions such as incorrect damper settings	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
	Gear problems such as tooth wear, eccentricity, backlash, gear misalignment, which is cracked or broken tooth.	1 *x* Gear Mesh Frequency.[3] Also 2 & 3 *x* Gear Mesh Frequency and Gear Natural Frequencies	Radial and Axial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial

(1) 1*x*, 2*x*. . . N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Number of blades x running speed

(3) Number of teeth x the speed of the gear

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 2000 Hz, the Minimum frequency, so the high pass filter setting is approximately 0.2x the machine running speed, for example, 0.2 x 50 Hz = 10 Hz.

Table 48 - Alarm Limits—Cooling Tower Fans

Overall Alarm Limits	mm/s	In/s
Alarms 1	7.10	0.395
Alarms 2	10.65	0.593
Applicable ISO Standards (other standards can apply)	14694, 10816-7	

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Table 49 - Band Definitions—Cooling Tower Fans

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* BPF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Blade Pass Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 BPF	1.2 BPF…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	60%	25%	20%	15%
Alarm 1 mm/s		1.065	6.39	2.49	2.49	4.26	1.78	1.42	1.065
Alarm 2 mm/s		1.60	9.59	3.73	3.73	6.39	2.66	2.13	1.60
Alarm 1 in/s		0.059	0.356	0.138	0.138	0.237	0.099	0.079	0.059
Alarm 2 in/s		0.089	0.534	0.208	0.208	0.356	0.148	0.119	0.089

Large FD and ID Fans, Centrifugal Compressors, Chillers

These classes of machines come in many different designs. Some are overhung, as illustrated in Figure 12 and Figure 13. Some have bearings on opposite sides, called center hung, and on some machines one or more of the bearings are not always accessible. Consider the specific machine design before resolving a monitoring solution.

Figure 12 - Motor—Fan

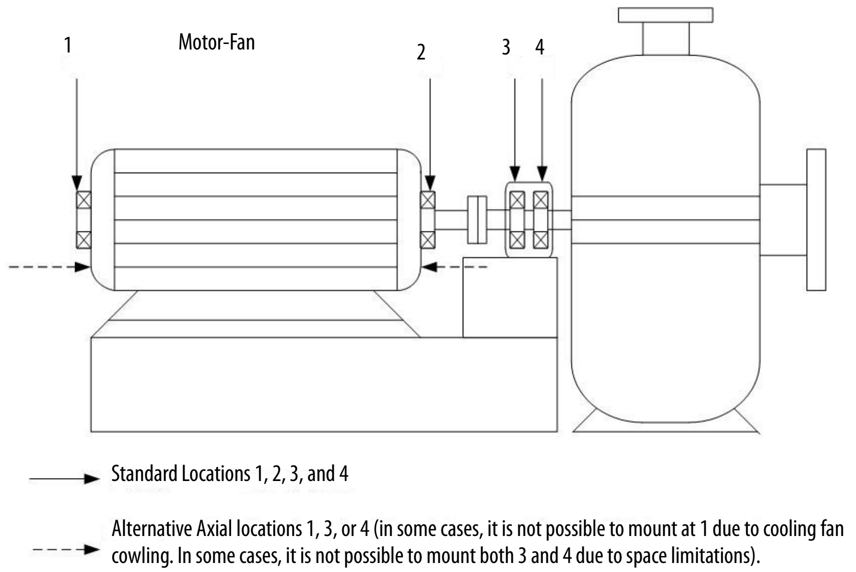
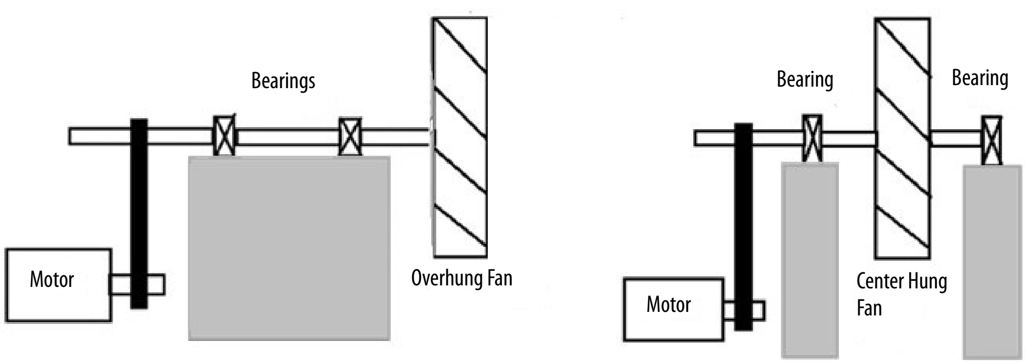


Figure 13 - Overhung and Center Hung Fans



Sensor and Monitor Requirements

See the appropriate section for the size motor.

The fan, compressor, or chiller requires one or two general-purpose sensors that are mounted at the illustrated locations, if an overhung design. For compressors, in particular, consider mounting a sensor in an axial orientation as vibration that is associated with compressor surge is more apparent in the axial direction.

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

See Machine Faults for Fans and Blowers—General for an explanation of Blade Pass Frequency.

Table 50 - Machine Faults—Fans, Compressors, and Chillers

Types of Faults	Possible Problems(1)	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Blade Pass(2)	1 *x* BP, can be caused by flow restrictions such as incorrect damper settings	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial
	Surge	0.1…0.5*x*	Axial

(1) 1*x*, 2*x*…N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Number of blades *x* running speed

Band Definitions and Limits

Table 51 - Machine Type and Alarm Limits—Fans, Compressors, and Chillers

Machine Type	Overall Alarm Limits	mm/s	In/s
FD fans, centrifugal compressors, centrifugal motor driven chillers	Alarms 1	5.40	0.301
	Alarms 2	8.10	0.451
ID fans and shaft-mounted integral fans	Alarms 1	4.80	0.267
	Alarms 2	7.20	0.401
Applicable ISO Standards (other standards can apply)	14694, 10816-3, 10816-8		

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Table 52 - Band Definitions—FD Fans, Centrifugal Compressors, and Centrifugal Motor Driven Chillers

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* BPF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Blade Pass Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 BPF	1.2 BPF…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	60%	25%	20%	15%
Alarm 1 mm/s		0.81	4.86	1.89	1.89	3.24	1.35	1.08	0.81
Alarm 2 mm/s		1.22	7.29	2.84	2.84	4.86	2.025	1.62	1.22
Alarm 1 in/s		0.045	0.270	0.105	0.105	0.180	0.075	0.060	0.045
Alarm 2 in/s		0.068	0.406	0.158	0.158	0.271	0.113	0.090	0.068

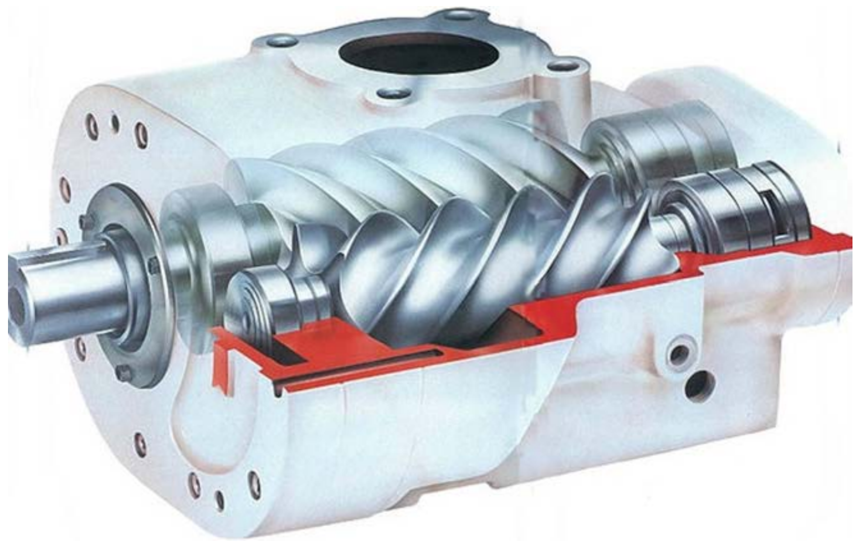
Table 53 - Band Definitions—ID Fans and Shaft Mounted Integral Fans

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* BPF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Blade Pass Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 BPF	1.2 BPF…12.2*x*	12.2*x*…50% FMAX	50% …100% FMAX
% of Overall Level		15%	90%	35%	35%	60%	25%	20%	15%
Alarm 1 mm/s		0.72	4.32	1.68	1.68	2.88	1.2	0.96	0.72
Alarm 2 mm/s		1.08	6.48	2.52	2.52	4.32	1.8	1.44	1.08
Alarm 1 in/s		0.04	0.240	0.094	0.094	0.160	0.067	0.053	0.04
Alarm 2 in/s		0.06	0.361	0.140	0.140	0.240	0.100	0.08	0.06

Screw Compressors

Among other uses, screw type compressors are used to supply large quantities of industrial air or natural gas. Vibration monitoring in these machines is typically accomplished using techniques that are typically associated with other rotating machinery. Differences result from compression of gas, unique internal make-up of the screws that are used to perform compression of the gaseous material, and the types of bearings

Figure 14 - Screw Compressor



Screw type compressors are typically motor driven, often within a common package. Depending on the size and required rotor speed, the motor can be directly coupled to the male rotor. Another configuration would have a bull gear and pinion set fitted between the motor and compressor.

Compressor Type

Screw compressors fall into two basic wet or dry categories. The wet compressor, using oil as a lubricant, typically runs from the motor shaft with much tighter clearances between rotors. The dry type can run much faster and use timing gears to keep the two unlubricated surfaces from touching. When considering the monitoring requirements, it does not matter if the wet or dry categories are selected. The key attributes that affect the monitoring solution are the type of bearings and whether the machine uses gears.

For the mentioned design characteristics, it is imperative that the compressor component specifications be considered before designing a vibration monitoring system.

Bearing Type

Depending on the size of the compressor and the manufacturer, the bearings can be either rolling element or fluid film sleeve type bearings. A combination of bearings can be used as well. Typically, the motor is fitted with common rolling element bearings and the compressor is fitted with sleeve type fluid film bearings.

In addition to the usual radial bearings, which support the shafts, screw compressors also typically employ thrust bearings on each of the rotor shafts as the compressive forces can create extreme lateral loads on the rotor.

Sensors

On extremely large compressors with sleeve type bearings, the manufacturer may have outfitted the machine with proximity probes. In other cases, the manufacturer may have prepared (drilled) the bearings for the installment of probes. If eddy current probes (ECPs) are present or can be installed, then ECPs should be used to monitor the fluid film bearings. If ECPs are not available, then the fluid film bearings can be monitored with accelerometers or velocity output transducers. All rolling element and thrust bearings should be monitored with accelerometers.

Sensor placement on large screw type compressors includes a radial, horizontal mounted, transducer over each bearing, including both motor and compressor bearings. Depending on the inclusion and design of a gear reductions system, there can be additional bearings that require monitoring. In some cases, the motor and compressor inboard transducers are sufficient to monitor the gear reduction system.

Thrust bearings can be monitored with a single accelerometer that is oriented in the axial direction. In this instance, the centerline of the transducer is parallel to the centerline of the motor shaft and compressor rotors.

In rare occasions, very large compressors can have Eddy current probes, mounted axially, to monitor thrust. These probes measure actual physical wear of the thrust bearing. More common is the use of an accelerometer to measure actual loss of the oil film in the thrust bearing. Use of an accelerometer to measure increased lobe mesh frequency (LMF) in the axial direction is another common use of these probes.

The vibration components of a screw compressor center around the following:

- Motor and gears
- Male and female rotor bearings
- Male and female rotor speeds
- Lobe mesh frequency
- Impact vibration due to failed thrust bearings
- Impacts of the male and female rotors due to radial bearing or other failure

The use of frequency bands to monitor various components and processes in screw compressors is dependent on the design of the compressor. There are fundamental frequencies that can be monitored and the bands can be calculated based on design specifics. The following two tables present examples of frequencies present in a screw type compressor and how to calculate the frequency ranges necessary to monitor them.

Detailed compressor design and component specifications from the manufacturer's owner manual should be obtained and substituted for the data contained here.

Table 54 - Screw Type Compressor Example Parameters

Parameter	Function	Example		
		RPM	CPM	Hz
Male Rotor Timing Gear	80 Teeth			
Female Rotor Timing Gear	120 Teeth			
Gear Teeth	52			
Pinion Teeth	38			
Gear Ratio	1.3684			
Male Rotor Lobes	4			
Female Rotor Lobes	6			
Motor Speed	= 1*x*	3575		

Table 55 - Screw Type Compressor Example Key Frequencies

Key Frequencies	Function	Example		
		RPM	CPM	Hz
Timing Gear Mesh			391,360	6522.67
Bull Gear Mesh Frequency	= Motor speed x number of gear teeth		185,900	3098.333
Pinion Gear Mesh	= Bull gear mesh		185,900	3098.333
Pinion Shaft Speed	= Pinion gear mesh/number of pinion teeth	4892		
Male Rotor Speed	= Pinion shaft speed	4892		
Male Rotor Lobe Mesh Frequency	= Male rotor speed x number of lobes		19,568	326.133
Female Rotor Lobe Mesh Frequency	= Lobe mesh frequency		19,568	326.133
Female Rotor Speed	= Female rotor lobe mesh frequency/ number of female rotor lobes	3262		
Gas Pulsation Frequency	= Female rotor speed		3262	326.133

Utilizing the sample data in Table 54, the motor speed multiplied by the speed increaser ratio yields the input rotor speed which is typically the male rotor. The calculations will be different if the female rotor is driven, hence the drive input must be known. The example also assumes a rotor design of four male lobes and six female lobes on two rotors. If the helical design of the rotors utilizes three male and five female lobes, the calculations will change and all calculated values depicted in Table 55 will also change. The following two tables are for setting band limits for the motor, the compressor and bull gear. Note that bull gear frequencies are likely to show up in the motor spectra and in the compressor spectra. Not all screw compressors have a bull and pinion gearing mechanism. Some compressors can have a direct motor drive where motor rpm is the rpm of the driven rotor.

Monitoring timing gear mesh frequencies may require Dynamix 1444 personality above 4 KHz. If gear mesh monitoring is required and the gear mesh is above 4 KHz, the channel count of some or all of the Dynamix 1444 modules may be reduced and should be considered in system design.

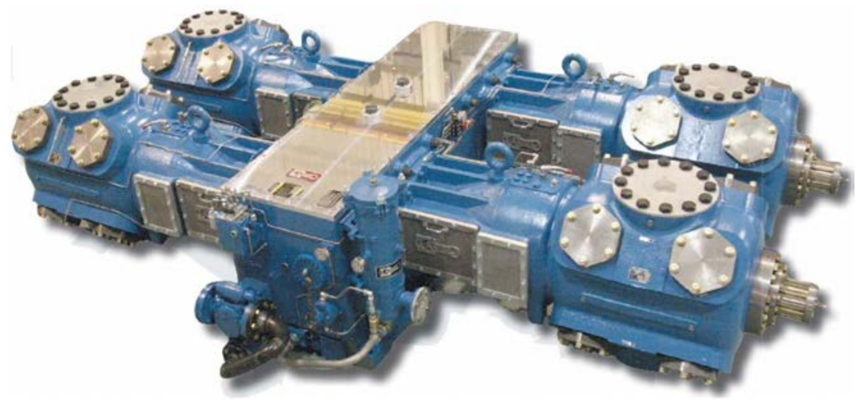
Table 56 - Band Definitions—Screw Compressor

<table>
<tr><th colspan="2">Attributes</th><th>Band 0</th><th>Band 1</th><th>Band 2</th><th>Band 3</th><th>Band 4</th><th>Band 5</th><th>Band 6</th><th>Band 7</th></tr>
<tr><td rowspan="2">Band Name</td><td>Fault Frequency</td><td>1x</td><td>1x</td><td>1x GMF</td><td>1x LMF</td><td>2x LMF</td><td>4.5…12x (typically)</td><td>12x…~1 kHz (typically)</td><td>~1…2 kHz (typically)</td></tr>
<tr><td>Indicated Fault or Fault Qualification</td><td>Male Rotor</td><td>Female Rotor</td><td>Bull/pinion</td><td>Lobes on Male Rotor Frequency</td><td>Lobes on Male Rotor Frequency</td><td>Male rotor bearing fundamental frequencies</td><td>Female rotor bearing fundamental frequencies</td><td>Bearing higher harmonic and natural frequencies</td></tr>
<tr><td colspan="2">Frequency Range</td><td>Pinion Speed</td><td>Pinion speed x male rotor lobes/female rotor lobes</td><td>Motor speed x #Teeth on bull gear</td><td>Pinion speed x #lobes on male rotor</td><td>(Pinion speed x #lobes on male rotor) x 2</td><td>4.2…12.2x Male rotor speed</td><td>4.2…12.2x Female rotor speed</td><td>50%…100% FMAX</td></tr>
</table>

Reciprocating Compressors

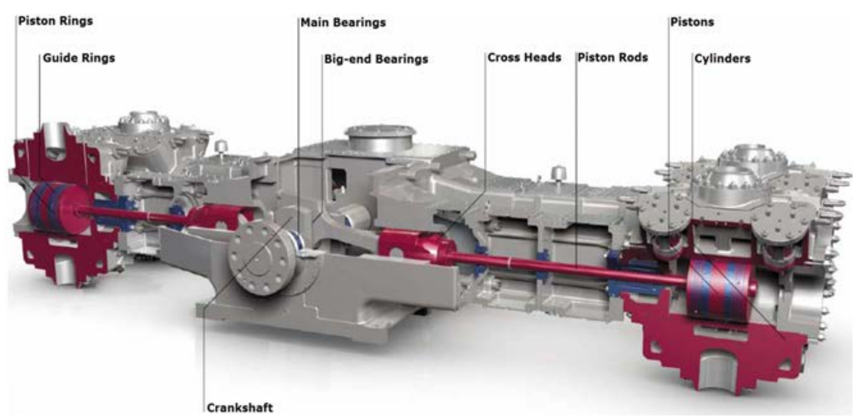
Reciprocating compressors are frequently used for compressing natural gas and air for industrial applications. The most common design of these large compressors is the horizontally opposed cylinder configuration.

Figure 15 - Reciprocating Compressor



Unlike typical machines monitored with vibration measurements, not all mechanical operation is rotational. In these machines, in addition to the usual rotational forces, there are significant mechanical forces from both the reciprocating components and the gas being compressed. For these reasons, while traditional vibration measurement techniques are still required, they are not in themselves a complete solution. The measurements necessary to provide comprehensive monitoring of a reciprocating compressor may include frame vibration, impact vibration, and rod drop. These measures are likely to provide the data necessary to detect and monitor the progression of mechanical and gas compression-related faults to avoid or mitigate associated damage.

Figure 16 - Reciprocating Components Example



Frame Vibration

Frame vibration results from the rotation of the crankshaft, rod, and piston assemblies direction change and the changes in compression forces. Based on typical running speeds for these compressors, frame vibration is best measured using velocity. Note that larger machines can have crank speeds below 600 RPM. In cases where crank speeds are below 600 RPM, consider using sensors suitable for low frequency measurements. These sensors are typically used with native velocity output units.

Frame vibration can be monitored with the transducer oriented so the mounting center line is in-line with the rod and piston travel. Machines with multiple banks of cylinders can have a transducer that is mounted on each bank and each crank of the horizontally opposed configuration. The transducer can be mounted to a frame component and centered on the crank throw. Do not mount the transducer on the flange or bracket.

Impact Vibration

Vibration from impacts is usually caused by loose components, issues that are associated with opening and closing valves, liquid slugs in gas being compressed, and so on. The vibration is in the form of a narrow peak or spike in the waveform. If the signal is repetitive, it will be represented in the spectrum at the frequency of the repetition. If the signal is not repeating, it may only be detectable from the time waveform, using true peak to peak signal detection.

Typically a fault such as loose components will cause cyclic, repetitive, forces which can be observed in the spectrum. Random occurring events, such as when a compressor ingests a liquid slug, will either not be indicated in the spectrum at all, or they may be indicated indirectly as they excite natural frequencies at the moment of the event. Typically impact type vibration is most readily detected by using accelerometers to measure the true peak to peak overall acceleration (g's) or spike energy (gSE).

The preferred location for the transducers is mounted perpendicularly to the centerline of the rod travel on each cylinder. It is important to follow sensor mounting best practices. Improper surface preparation, the use of magnets, adhesives, and so on, can cause significant loss of high frequency signals.

Assuming a compressor with four horizontally opposed cylinders, the suggested transducer count is eight. The transducer count is based on four transducers for the frame and four for the impacts at the cylinders. Impact vibration is likely to excite the g’s measurement much more than the velocity measurements. As such, impact vibration, in g’s, at the cylinders will be detectable sooner than other vibration measures, such as velocity measurements of frame vibration. The use of gSE as a measure of impacts and loose parts is discussed in more detail in the Spike Energy (gSE) Measurements chapter. See Random Impact Event Detection on page 130.

Vibration Alarms

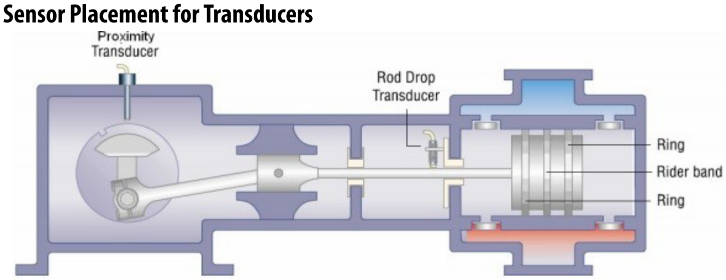
The use of alerts and alarms with these monitoring systems is not as straight forward as with typical rotational machinery. Always consult the machine manufacturer for guidance and suggested limits for both frame and impact vibration. If no information is available, it is a best practice to start with a baseline reading on a known good machine. From this starting point, trended data is then consulted periodically to look for any noticeable changes. See Statistical Alarms on page 156 for additional information on using trended data and statistical alarms.

The use of bands with these systems is also not straight forward. Case-mounted velocity transducers are likely to yield amplitudes in the 1x and 2x range. These amplitudes are frequently in the harmonics of 1x due to gas load vibration. There is little information as to the relative amplitudes that can be assumed to be significant. Trended data from each compressor is the best source of information as to what is considered excessive. Spectrum data is also the best indicator of frequency ranges, other than 1x and 2x, that can be monitored for each with a real-time analyzer such as Emonitor RTA with the Dynamix 1444 FFT module.

Rod Drop and Rider Band Wear

Rod drop and rider band wear are terms that describe the measurement of wear of a band that is used in horizontally opposed compressors to support the piston in the cylinder. The rider band wears as the piston travels and must be periodically replaced. If the band wears too far, damage to the piston and cylinder is likely. Impact vibration measurement is likely to pick up this damage but serious damage to the compressor can have already occurred. For this reason, a system is often in place, which measures actual physical wear of the rider band in units of displacement. This measurement is then used to alert operators before the band wears out.

Figure 17 - Sensor Placement for Transducers

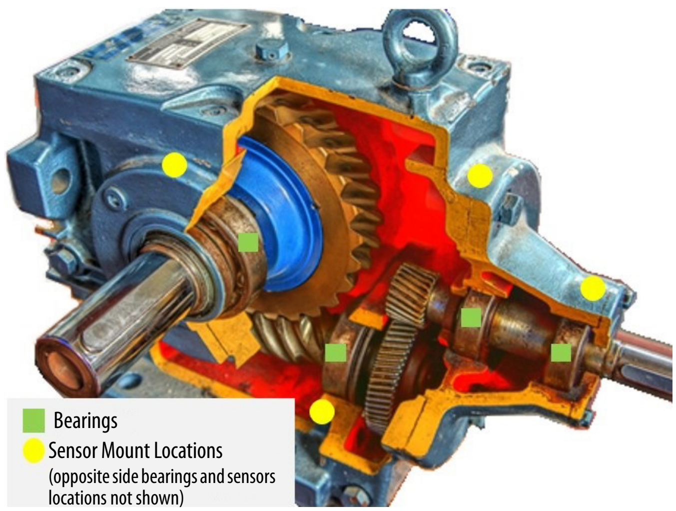


These monitoring systems require an Eddy current proximity probe be installed to measure the actual position of the rod relative to the wall of the cylinder. The systems also require a transducer to measure the travel location of the rod linearly during one stroke of reciprocating motion. The measurement module measures the position (bias voltage) of the rod at the exact same time during the reciprocating motion that is based on a pulse from the transducer that measures the rotation of the crank (phase angle). Rod drop systems are typically installed during the manufacture of the compressor. These systems may be difficult to add after the fact. Monitoring the transducers requires a measurement module with rod drop capability such as the Dynamix 1444 Series' Dynamic Measurement Module.

Gearboxes

Gearboxes are used in every industry. They can reduce or increase speed from a driver, or they can change the direction of rotation. A simple gear set has one input and one output, but gearboxes with multiple input shafts or multiple output shafts are also common.

Figure 18 - Complex Gearbox



Sensor and Monitor Requirements

Gear sets come in many different forms, some complex and some simple like the gearboxes shown in Figure 18 and Figure 19.

Figure 19 - Simple Gearbox

To monitor a gear set, a sensor is mounted at each bearing on each shaft. Each shaft in almost all gear sets has two bearings, regardless the number of gears that are mounted to the shaft.

To determine the number of sensors, consider the number of shafts. Find a suitable mounting location for a sensor at either of its two bearings. Some gearboxes do not have suitable mounting locations at each bearing. Gears are susceptible to faults that create axial forces. Consider adding one sensor that is mounted in the axial direction for each shaft (ideally) or for the input and output shafts.

Table 57 - Reduction Gearbox

Reduction Gearbox	Number of Shafts	Number of Bearings	Axial Sensors	Total Sensors
Common single reduction gearbox	2	4	2	6
Common double reduction gearbox	3	6	2	8

IMPORTANT	If the speed of either the input or output shaft is greater than 10,000 RPM, high frequency sensors must be used. (1443-ACC-HF-T, High Frequency Accelerometer, Top Exit, 50 mV/g, 20 kHz)

Once the number of sensors is known, see Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

The following table is for measurements on a single shaft. Gearboxes with more than two shafts can transmit via structural elements or the gears. Vibration from one shaft can transmit to other shafts or to the structure that other sensors are mounted on. This transmission means that it isn't uncommon to see vibration that is associated with other shafts in a gear set besides the shaft that the monitored bearing is supporting. For example, on a simple two shaft reduction gear, the sensor that is monitoring the output can measure vibration at the running speed frequency of the input shaft. This measurement depends on the severity of the vibration, the design of the gearbox, and the sensor location.

Table 58 - Machine Faults—Gearbox

Types of Faults	Possible Problems(1)	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Bearing Anomalies	Inner/Outer race anomaly frequencies, ball/roller anomaly frequencies, cage anomaly frequencies, bearing natural frequencies	Radial
	Gear problems such as tooth wear, eccentricity, backlash, gear misalignment, which is cracked or broken tooth.	1 *x* Gear Mesh Frequency.(2) Also 2 & 3 *x* Gear Mesh Frequency and Gear Natural Frequencies	Radial and Axial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial

(1) 1*x*, 2*x*. . . N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Number of teeth *x* the speed of the gear

Band Definitions

The overall limits vary based on the type and size of gearbox. The allowable or expected vibration on a relatively small gear set, such as an automotive transmission, is far different than on gear set used in a car crusher. The following limits can be used as a starting point if more specific data is not available from OEMs or other sources.

Set the maximum frequency (FMAX) to encompass the 1 x Gear Mesh frequency(1) (GMF) and, if 2 x GMF is less than 4.8 kHz then to encompass the 2 x GMF. When applied to high-speed machinery, where the input or output shaft is greater than 10,000 RPM, the gear mesh frequency can be up to 20 kHz.

Table 59 - Machine Type and Alarm Limits—Gearbox

Gearbox Type	Overall Alarm Limits	mm/s	In/s
Small Worm and Wheel Gears	Alarm 1	7.00	0.390
	Alarm 2	10.50	0.585
Large Worm and Wheel Gears	Alarm 1	9.00	0.501
	Alarm 2	12.50	0.696
General Reduction or Speed Increasing Gears	Alarm 1	4.50	0.251
	Alarm 2	6.75	0.376
Applicable ISO Standards (other standards can apply)		-	

(1) For measurements greater than 4.8 kHz FMAX, the 1444 Series Dynamic Measurement Module must be configured to use only two channels. This must be considered when specifying the number of required modules. See the User Manual, 1444-UM001, for further information.

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Table 60 - Band Definitions—Small Worm and Wheel Gears

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* GMF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Gear Mesh Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 GMF	1.2*x*…12.2*x*	12.2*x*…50% FMAX	1.0 …2.0 kHz
% of Overall Level		15%	90%	35%	35%	25%	25%	20%	15%
Alarm 1 mm/s		1.05	6.30	2.45	2.45	1.75	1.75	1.40	1.05
Alarm 2 mm/s		1.575	9.45	3.68	3.68	2.63	2.63	2.10	1.575
Alarm 1 in/s		0.058	0.351	0.136	0.136	0.097	0.097	0.078	0.058
Alarm 2 in/s		0.088	0.526	0.205	0.205	0.146	0.146	0.117	0.088

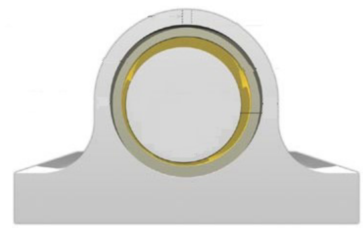
Table 61 - Band Definitions—Large Worm and Wheel Gears

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* GMF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Gear Mesh Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 GMF	1.2*x*…12.2*x*	12.2*x*…50% FMAX	1.0 …2.0 kHz
% of Overall Level		15%	90%	35%	35%	25%	25%	20%	15%
Alarm 1 mm/s		1.35	8.10	3.15	3.15	2.25	2.25	1.80	1.35
Alarm 2 mm/s		1.875	11.25	4.38	4.38	3.13	3.13	2.50	1.875
Alarm 1 in/s		0.075	0.451	0.175	0.175	0.125	0.125	0.100	0.075
Alarm 2 in/s		0.104	0.626	0.244	0.244	0.174	0.174	0.139	0.104

Table 62 - Band Definitions—General Reduction or Speed Increasing Gears

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	2*x*	3*x*	1*x* GMF	4.5…12*x* (typically)	12*x*…~1 kHz (typically)	~1…2 kHz (typically)
	Indicated Fault or Fault Qualification	Bearing Cage Anomalies	Unbalance	Misalignment, Looseness	Misalignment, Looseness	Gear Mesh Frequency	Bearing Fundamental Frequencies	Bearing Lower Harmonic Frequencies	Bearing Higher Harmonics and Natural Frequencies
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…2.2*x*	2.2…3.2*x*	0.8…1.2 GMF	1.2*x*…12.2*x*	12.2*x*…50% FMAX	1.0 …2.0 kHz
% of Overall Level		15%	90%	35%	35%	25%	25%	20%	15%
Alarm 1 mm/s		0.67	4.05	1.58	1.58	1.13	1.13	0.90	0.67
Alarm 2 mm/s		1.02	6.08	2.36	2.36	1.69	1.69	1.35	1.02
Alarm 1 in/s		0.038	0.225	0.088	0.088	0.063	0.063	0.050	0.038
Alarm 2 in/s		0.056	0.338	0.132	0.132	0.094	0.094	0.075	0.056

Machines with Fluid Film Bearings



In industrial machinery applications, fluid film bearings are bearings that support their loads solely on a thin layer of oil.

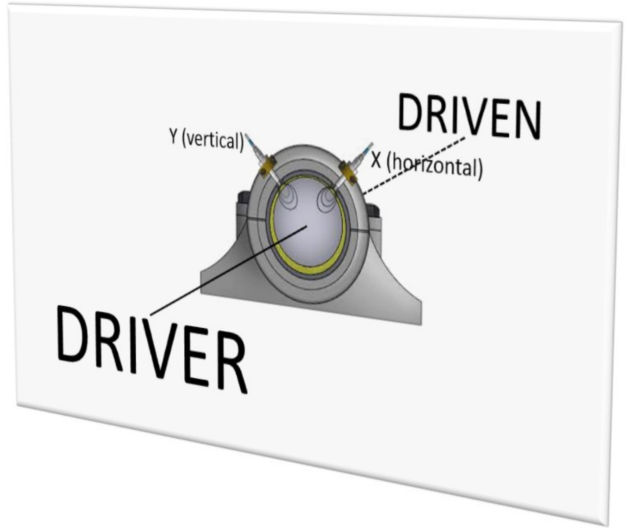
They can be broadly classified into two types, Hydrodynamic (fluid dynamic) bearings and hydrostatic bearings. Hydrodynamic bearings rely on the high speed of the journal, the part of the shaft that rests on the fluid, to pressurize the fluid in a wedge between the faces. Hydrostatic bearings are externally pressurized fluid bearings.

Fluid film bearings[1] are frequently used in high load, high speed, or high precision applications where ordinary ball bearings would have short life or cause high vibration.

(1) Another type of fluid film bearing is the plain bearing as describe on page 20.

Fluid Film Bearing Sensor Requirements

Because the shaft inside a fluid film bearing rides on an oil film or 'wedge', it does not contact the bearing housing. Shaft vibration is significantly dampened, if present at all, on the bearing housing. Consequently, to measure shaft vibration, sensors must directly measure the movement of the shaft rather than the vibration of the bearing housing. To obtain this measurement, use eddy current probes (proximity sensors).



Eddy current probes are typically mounted by drilling and tapping a hole through the bearing housing, which the sensing part of the probe is then threaded into. Normally two sensors are installed, offset one to the other by 90° (±5°), so that shaft movement (vibration) can be measured in all directions. The sensors are referred to as X and Y. When viewed from the driver end of the machine train, the Y (vertical) probe is on the left side of the vertical center. The X (horizontal) probe is on the right side of the vertical center regardless of the direction of the shaft rotation.

For applications that require eddy current probes see selection guide for the 1442 Series probes, publication 1442-TD001.

Using Velocity Output Sensors

There are times when it is not possible to install eddy current probes (ECP), such as when monitoring sleeve bearings on a crank shaft or floating bearings in turbines. And there are times when installing ECPs simply isn't practical due to costs[1]. In these instances, the use of velocity output transducers and accelerometers may be necessary.

An accelerometer that is mounted on the surface of a fluid film bearing will detect the same frequencies of vibration as an ECP. The magnitude of the vibration is likely to be muted by the dampening caused by the fluid film. This dampening can be as much as 90%. This dampening significantly complicates alarm limits determination or severity estimation. To determine alarm limits and velocity measurements, 10% of the differentiated displacement limit can be a reasonable starting point. A complex analysis is required to determine shaft movement accurately from velocity measurements and more precisely determine alarm limits[2]. The mass of the machine, foundation, and rotor are all factors that must be considered. Other considerations include bearing pedestal stiffness, rotor speed, and so on. Because of this complexity, any provided displacement alarm limits cannot be converted to velocity and used as is.

See Statistical Alarms on page 156 if alarm limits appropriate to the sensing solution are not available from the manufacturer. In this case, the assistance of a trained vibration analyst familiar with the machines, transducers, and alarms can be consulted.

Velocity measures can be used as a diagnostic tool. Velocity measures can provide similar, but not all, indications of machine condition to that of an eddy current probe. The overall vibration levels can be monitored. Baselines can be established when the machine and bearings are in known good working order. The measures can be trended to provide indication of change and deterioration. Vibration at specific frequencies indicative of fault can be monitored and trended. Phase measurements, when a tachometer is available, can be monitored to provide indication of change in mass and possibly other faults. Phase measurements are NOT to be used as input to a balance equation.

(1) An eddy current probe is a bit more expensive than a typical accelerometer. The real difference is the installation costs. The eddy current probe is mounted inside the bearing. This installation requires a hole to be drilled through the bearing cap. The installation expense for installing ECPs when the bearing caps were not pre-drilled by the OEM can be significant.

(2) Manufacturers' specifications for vibration limits that are associated with fluid film bearings are typically for shaft displacement measurements relative to the bearing, which is what an ECP measures. If monitoring with bearing case-mounted sensors, the machine OEM can be consulted regards appropriate alarm limits for the applied sensing solution.

Electric Motors

The following assumes a motor greater than 375 KW (500 HP) since in almost all cases only large electric motors are fitted with fluid film bearings, see page 91.

Sensor and Monitor Requirements

Sensor requirements vary depending on the requirements for the machine, which can include measurements other than vibration. This document addresses only the requirements that are associated with the vibration measurements.

The vibration measurements that are made on a large electric motor vary depending on the design, size, and sometimes the age of the machine. The following two measurement solutions can be applied.

Common Displacement Measurements

In most cases when fluid film bearings are used, the eddy current probes are installed in the X and Y direction at each bearing.

Determine the specific sensor requirements by obtaining a list of the existing sensors. Cross-references are available for common sensor families that provide an equivalent 1442 Series eddy current probe system. If in doubt, engage a condition-monitoring professional to help identify the appropriate 1442 Series sensor.

Acceleration or Velocity Measurements

The preferred monitoring solution for any fluid film bearing application is eddy current probes. Some large electric motors use accelerometers or velometers due to the cost of instrumenting eddy current probes. In these cases, mount sensors in the vertical and horizontal directions at each bearing.

Determine the specific sensor requirements by obtaining a list of the existing sensors. Cross-references are available for common sensor families that provide an equivalent 1443 Series sensors. If in doubt, engage a condition-monitoring professional to help identify the appropriate 1443 Series sensors.

Monitor selection is based on the number of sensors that are required, see Dynamic Measurement Module in Bill of Materials.

If the application requires local relays for trip or annunciation, and if multiple relays are required, then add additional Expansion Relay Modules.

The Dynamix™ 1444 Series excludes temperature or speed/overspeed measurement modules.[1] However, these variables are in plan, so review current availability.

Large Electric Motors

In many cases, large electric motors also include a tachometer, possibly requiring a tachometer signal conditioner module addition, page 28.

If the application requires local relays for trip or annunciation, and if multiple relays are required, add additional relays, page 29.

Machine Faults

The order values that are indicated in the faults table refer to orders of the sensed shaft, which depends on the shaft arrangement.

Possible problems that are associated with large electric motors that are fitted with fluid film bearings.

Table 63 - Machine Faults—Electric Motors

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and Axial
	Oil Whirl/Whip	0.3…0.49*x* then 0.5*x* typical. Can start as whirl and lock onto a rotor critical frequency typically at 0.5*x* running speed.	Radial
Uncommon faults	Rotor Rub (light)	0.5*x*, Can also be 1/3 or ¼ of 1*x* RPM	Radial
	Rotor Rub (heavy)	1.5*x*, 2.5*x*, 3*x*, 3.5*x*, 4*x*, 4.5*x*, 5*x*, 5.5*x*. Can be multiple harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial
	Eccentric air gap, shorted, or loose windings	2*x* line frequency	Radial
	Loose rotor bar	1*x* with 2*x* slip frequency side bands	Radial
	Loose windings	2*x* line and number of rotor bars *x* RPM	Radial
	Loose rotating parts	1*x* Phase change. Change in phase over time, can be 360°	Radial

(1) 1*x*, 2*x*…N*x*—When not qualified, N*x* refers to multiples of running speed.

(1) Speed measurements are included in the Dynamic Measurement Module, which can use the Tachometer Signal Conditioner to power and condition the speed sensor. However, this serves the 'Phase (1/Rev RPM' measurement function. It is not intended to provide the speed/overspeed functions. Monitor selection is based on the number of sensors required.

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 10x running speed or about 1500 Hz. If low frequency noise is expected due to flow induced vibration that is associated with the driven machine, apply a high-pass filter to a frequency sufficiently high to filter the noise from the measurement.

Set the overall limits based on Table 64.

Table 64 - Alarm Limits—Electric Motors

Shaft Speed RPM		3000	3600
Normal Peak to Peak Displacement µm (mils)		90 (3.5)	80 (3.2)
Alarm 1	Presence of a possible anomaly that requires remedial action	165 (6.5)	150 (6.0)
Alarm 2	Machine is unsatisfactory for continued operation	240 (9.5)	220 (8.7)
Applicable ISO Standards (other standards can apply)		—	

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Set limits based on maximum percentage of the overall level and deviations from observed norms.

Table 65 - Band Definitions—Electric Motors with Fluid Film Bearings

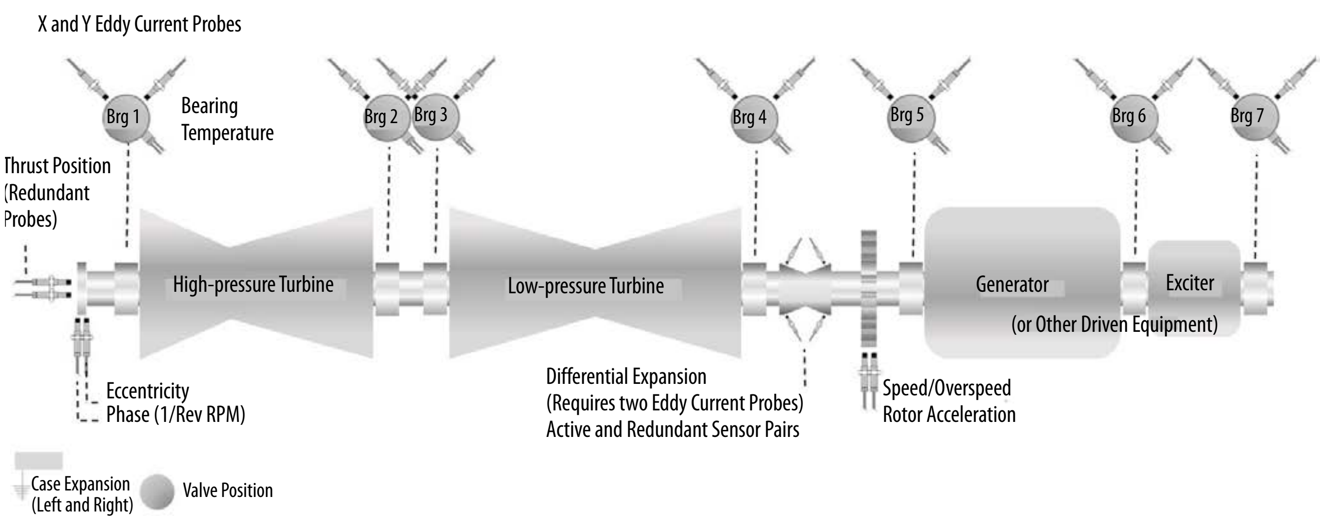
Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	1.5*x*	2*x*	2*x* LF	2.5*x*	3*x*	4*x*
	Indicated Fault or Fault Qualification	Oil Whirl/ Whip/Rub	Unbalance	Rub	Misalignment, Looseness	Electrical	Rub	Misalignment, Looseness	Misalignment, Looseness
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…1.8*x*	1.8…2.2*x*	1.95…2.05*x* LF	2.2…2.8*x*	2.8…3.2*x*	3.2…4.2*x*
% of Overall Level		10%	90%	10%	20%	5%	10%	10%	10%

Steam Turbines

Steam turbines are among the most complex machines to monitor. The instrumentation that is required is referred to as Turbine Supervisory Instrumentation[1] (TSI) and can include all instrumentation in the following illustrations.

Some small and probably old turbines sometimes exclude eddy current probes for vibration monitoring. However, a Steam Turbine Instrumentation System (TSI) will always require thrust position, expansion, and speed measurement systems as a steam turbine simply cannot be operated without them. Consequently, a steam turbine monitoring project can be a retrofit of existing instrumentation. So the sensor solution is known.

Steam turbine designs do not allow access to most of the sensors while the machine is running. Many of these machines are old and records that provide the specifics of the installed sensors are not always readily available. For speed, position, and expansion measurements that use eddy current probes, the sensors simply cannot be identified without knowing the specifics of the sensors already installed.



(1) Turbine Supervisory Instrumentation (TSI) can also refer to the instrumentation required for gas, hydro, or wind 'turbines'. But in most cases, it is regarding steam turbines. These turbines have unique instrumentation requirements that are associated with managing their long, heavy rotors, which can bend when not operating and grow due to thermal expansion as they heat up.

Sensor and Monitor Requirements

Sensor requirements vary depending on the requirements for the machine, which can include many measurements other than vibration. The measurements include thrust position, eccentricity, case and differential expansion, and possibly over speed. This document addresses only the requirements that are associated with the vibration measurements.

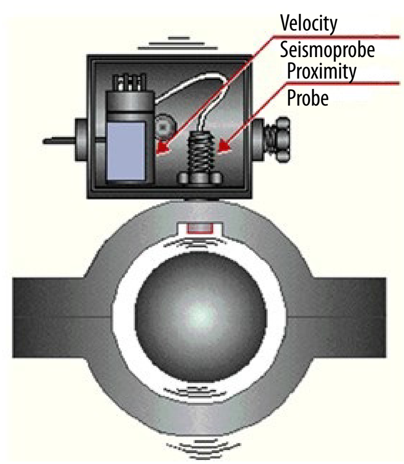
The vibration measurements that are made on a steam turbine vary depending on the design, size, and sometimes age of the machine. In general, there are three measurement solutions that can be applied. The measurements are common displacement, absolute shaft, and acceleration.

Common Displacement Measurements

The vibration sensors are eddy current probes. Determine the specific sensor requirements[1] by obtaining a list of the existing sensors. Cross-references are available for common sensor families that provide an equivalent 1442 Series eddy current probe system. However, if any doubt, engage a condition-monitoring professional to help identify appropriate 1442 Series sensor.

Absolute Shaft Measurements

On some turbines, usually turbines generators that are greater than 500 MW, absolute shaft vibration measurements are sometimes required.



Absolute shaft vibration is a measure of the motion of the shaft relative to free space. The measurement is typically applied when the rotating assembly is five or more times heavier than the case of the machine. Absolute shaft motion is proportional to the vector addition of the casing absolute motion and the shaft relative motion.

Measurement requires use of an eddy current probe and an accelerometer or velocity output accelerometer mounted so that they measure in the same plane, at the same location. Usually they are held in a common enclosure.

The vibration sensors include both accelerometers and eddy current probes. Determine the specific sensor requirements(1) by obtaining a list of the existing sensors. Cross-references are available for common sensor families that provide an equivalent 1442 Series eddy current probe system and 1443 Series accelerometer. However, if in doubt, engage a condition-monitoring professional to help identify appropriate 1442 and 1443 Series sensors.

IMPORTANT	Consider the temperature requirement before selecting an accelerometer. In some cases, the application requires a high temperature sensor, rated higher than standard 1443 sensors.

Acceleration Measurements

Most steam turbines are not instrumented with accelerometers. However, there are three cases where they can be:

- Absolute shaft measurement
- Some small, old, turbines can be instrumented only with accelerometers. While eddy current probes are a better solution, the expense necessary to add eddy current probes is sometimes prohibitive as that requires machining the bearing caps.
- Some turbines have accelerometers that are mounted in addition to the eddy current probes for diagnostic(1) purposes.

Determine the specific sensor requirements by obtaining a list of the existing sensors. Cross-references are available for common sensor families that provide an equivalent 1443 Series sensors. However, if in doubt, engage a condition-monitoring professional to help identify appropriate 1443 Series sensors.

IMPORTANT	If accelerometers are used, consider the temperature requirement before selecting a sensor. In some cases, the application requires a high temperature sensor that must be rated higher than standard 1443 sensors.

Monitor selection is based on the number of sensors that are required, see Dynamic Measurement Module in Bill of Materials.

In most cases, turbine supervisory instrumentation (TSI) application requires local relays. If the application requires local relays for trip or annunciation and if multiple relays are required, add additional Expansion Relay Modules.

The Dynamix 1444 Series excludes temperature or speed/overspeed measurement modules.(2) However, Dynamix Series products that serve these functions are in plan, so review current availability.

(1) If the turbine has a history of blade problems, or the customer is concerned with blade problems, then consider adding one or two accelerometers to each turbine bearing. These accelerometers allow for monitoring the blade pass frequencies, which indicate disturbed flow due to blade or control valve problems.

(2) Speed measurements are included in the Dynamic Measurement Module, which can use the Tachometer Signal Conditioner to power and condition the speed sensor. However, this serves the '**Phase (1/Rev RPM**' measurement function. It is not intended to provide the speed/overspeed functions.

Machine Faults

The order values that are indicated in the faults table refer to orders of the sensed shaft, which depends on the shaft arrangement.

Table 66 - Machine Faults—Steam Turbines

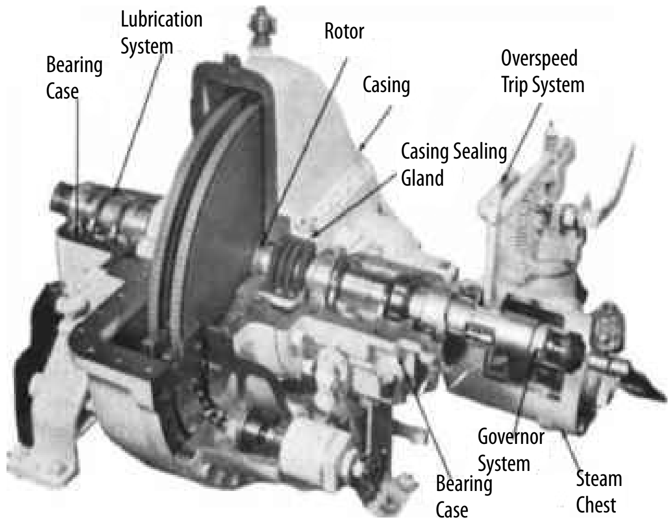
Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and Axial
	Oil Whirl/Whip	0.3…0.49*x* then 0.5*x* typical. Can start as whirl and lock onto a rotor critical frequency typically at 0.5x running speed.	Radial
Uncommon faults	Rotor Rub (light)	0.5*x*, Can also be 1/3 or 1/4 of 1*x* RPM	Radial
	Rotor Rub (heavy)	1.5*x*, 2.5*x*, 3*x*, 3.5*x*, 4*x*, 4.5*x*, 5*x*, 5.5*x*. Can be multiple harmonics	Radial
	Resonance Frequency (RF)	Structural resonance, possibly rotor critical speed frequency	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial
	Loss of mass	1*x* Phase change. Instantaneous change in 1*x* phase, can be 360°, with accompanying change in 1*x* magnitude (possibly lower).	Radial
	Loose rotating parts	1*x* Phase change. Change in Phase over time, can be 360°	Radial
	Bent or cracked rotor	2*x* Phase	Radial
	Disturbed flow	Blade pass frequencies.[2] Damaged blades, control valve problems	Radial

(1) 1*x*, 2*x*…N*x*—When not qualified, N*x* refers to multiples of running speed.

(2) A steam turbine can have from as few as about 20 blades to well over 100 blades on a wheel. And it can have just one wheel, a single stage turbine, to over a dozen.
Some small single stage turbines, one wheel, has only a few blades, so the blade pass frequency is within the normal FMAX measured. And on large turbines, the leading stages can have a relatively few number of small high-pressure blades, so also possibly within the normal FMAX.
If the number of blades is known, and the resulting blade pass frequency (number of blades x RPM) is within the FMAX, then a band is defined that encompasses this or these frequencies.
If an accelerometer is available, then configure that measurement to include the blade pass frequencies up to at least (approximately) 5 kHz and possibly up to 10 kHz or 20 kHz to monitor the blade pass frequencies.

Band Definitions and Limits

The band definitions assume relative displacement measurements using eddy current probes. They are also applicable to absolute shaft measurements. However, if accelerometers are used or are available in addition to probes, bands must be uniquely defined for those sensors. Consult a condition-monitoring professional to aid in defining those sensors.



Set the maximum frequency (FMAX) to approximately 40 orders of rated operating speed.

$$\mathrm{FMAX(Hz)} = 40 \times \frac{\mathrm{NormalOperatingSpeed}}{60}$$

In most cases, displacement measurements on steam turbines do not present significant low frequency noise. In some cases, flow noise can be substantial. In such a case, consider applying a high pass filter at a frequency that attenuates the noise and is not greater than approximately 0.4x.

Set the overall limits as shown in Table 67.

Table 67 - Alarm Limits—Steam Turbines

Shaft Speed RPM		3000	3600
Normal Peak to Peak Displacement µm (mils)		90 (3.5)	80 (3.2)
Alarm 1	Presence of a possible anomaly requiring remedial action	165 (6.5)	150 (6.0)
Alarm 2	Machine is unsatisfactory for continued operation	240 (9.5)	220 (8.7)
Applicable ISO Standards (other standards can apply)		7919-2, 10816-2	

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

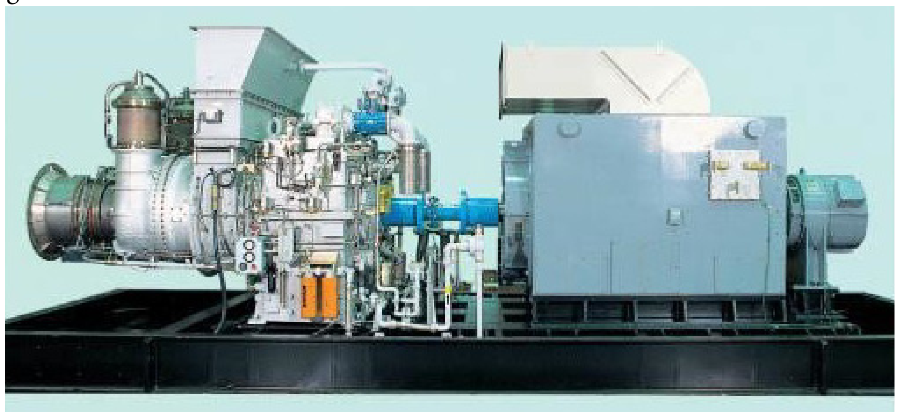
Set limits based on maximum percentage of overall level and deviations from observed norms.

Table 68 - Band Definitions—Steam Turbines

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	0.3*x*…0.5*x*	1*x*	1.5*x*	2*x*	RF	2.5*x*	3*x*	1*x BPF*
	Indicated Fault or Fault Qualification	Oil Whirl/ Whip/Rub	Unbalance	Rub	Misalignment, Looseness	Resonance	Rub	Misalignment, Looseness	Blade Pass
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…1.8*x*	1.8…2.2*x*	Machine Specific	2.2…2.8*x*	2.8…3.2*x*	1 *x* BP, or 10*x*—FMAX
% of Overall Level		10%	90%	10%	20%	5%	10%	10%	10%

Gas Turbines

See Gas Turbines fitted with rolling element bearings, for a general discussion of gas turbines.



Sensor and Monitor Requirements

Monitor selection (page 28) is based on the number of sensors required. In most cases, gas turbines include one or two tachometers which require adding a tachometer signal conditioner (page 28). In some cases, the entire machine can have more than two speed sensors, which requires an addition tachometer signal conditioner.

If the application requires local relays for trip or annunciation, and if multiple relays are required, add additional Expansion Relay Modules.

Machine Faults

The order values that are indicated in the following faults table refer to orders of the sensed shaft, which depends on the shaft arrangement.

Table 69 - Machine Fault—Gas Turbines

Number of Shafts		
One	**Two**	**Three**
Gas Turbine	Gas Generator	Gas Generator HP
		Gas Generator LP (or IP)
	Power Turbine	Power Turbine

For three shaft designs, the sensors that monitor the HP and LP bearings are common to both shafts. In these cases, it is best to monitor both order frequencies from the same sensor.

Possible problems that are associated with gas turbines that are fitted with fluid film bearings.

Table 70 - Common Problems—Gas Turbines

Problem	Frequency(1)	Comments
Oil Whirl/Whip	0.3…0.49*x* then 0.5*x* typical	Can start as whirl and lock onto a rotor critical frequency typically at 0.5X running speed.
Unbalance	1*x*	Possible loss of turbine blades or damage to rotating parts.
Shaft Crack	1*x* and 2*x*	Monitor the 1*x* and 2*x* magnitude and phase vectors A ±30% change in magnitude or ±20% change in phase from the normal values could indicate a problem and it would be best to investigate.
Misalignment	1*x* and 2*x*	For rigid couplings 1*x* flexible couplings typically 2*x* and 1*x*
Rotor Rub (light)	0.5*x*	Can also be 1/3 or 1/4 of 1*x* RPM
Rotor Rub (heavy)	1.5*x*, 2.5*x*, 3*x*, 3.5*x*, 4*x*, 4.5*x*, 5*x*, and 5.5*x*	Can be multiple harmonics
Loose Rotating Parts	1*x* Phase Change	Change in phase over time, can be 360°

(1) 1*x*, 2*x*…N*x*—When not qualified, Nx refers to multiples of running speed.

Band Definitions and Limits

The many varied types and designs of gas turbines make it impractical to define a specific set of FFT Band definitions to monitor them. Additionally, gas turbines likely require monitoring of the 1x vibration using tracking or band pass filters to consider in any diagnostics that are applied.

Set the maximum frequency (FMAX) to approximately 10x running speed or about 1500 Hz. If low frequency noise is expected, which is common in gas turbines, apply a high pass filter at a frequency sufficiently high to filter the noise from the measurement.

Confirm that any monitoring application that is implemented satisfies the protection monitoring of the engine manufacturer.

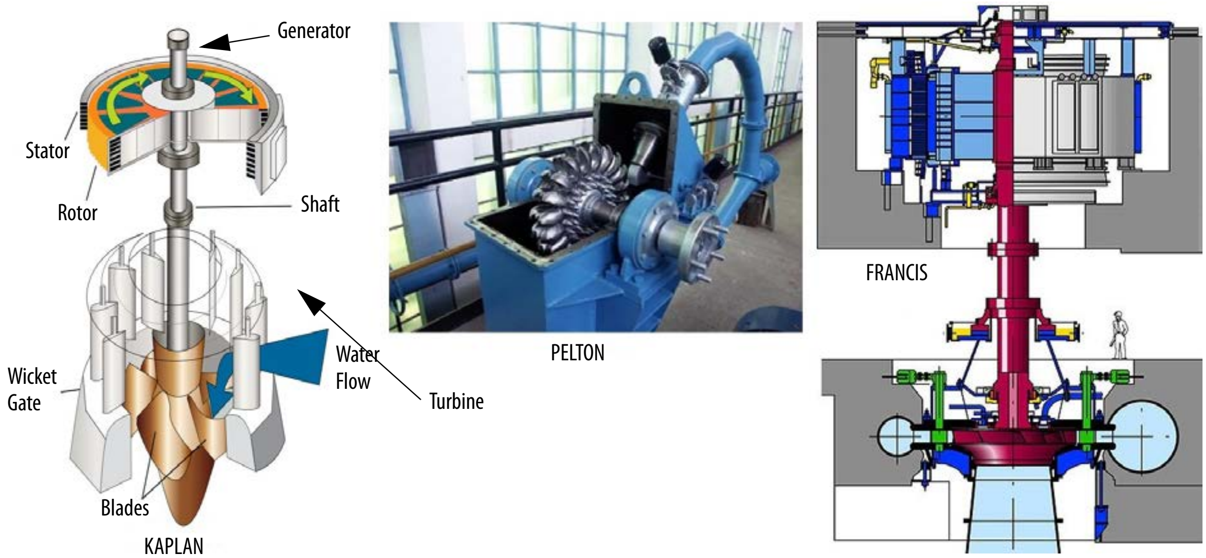
The following guidelines can be used to assess overall vibration level, the term 'normal' is used to indicate the expected maximum level of a new machine or a machine in good condition.

Table 71 - Alarm Limits—Gas Turbines

Shaft Speed RPM		3000	3600
Normal Peak to Peak Displacement µm (mils)		90 (3.5)	80 (3.2)
Alarm 1	Presence of a possible anomaly requiring remedial action	165 (6.5)	150 (6.0)
Alarm 2	Machine is unsatisfactory for continued operation	240 (9.5)	220 (8.7)
Applicable ISO Standards (other standards can apply)		7919-2	

Hydro Turbines

This section illustrates a typical hydro turbine but there are many different designs(1) and size turbines. Consult the manufacturer documentation for specifics on vibration alarm limits and how to monitor any specific turbine.



Hydroelectric turbines can have requirements unlike many other machines in that speeds are typically low, sometimes as low as 100 RPM. Low speeds, which are coupled with the high energy electrical fields that surround the generator, produce unique monitoring requirements. The sensors and monitoring strategy are unique.

Sensor and Monitor Requirements

Hydro turbines typically include bearings above and below the generator, a single turbine bearing, and a thrust bearing. The radial bearings are mostly fluid film type bearings and must be monitored using standard XY eddy current probes. In many cases, measurement of absolute vibration using low-frequency accelerometers are taken.

For a common three radial bearing plus thrust, arrangement of up to eight eddy current probes and/or six accelerometers are sometimes required. The design of the turbine and access to the bearings is of utmost importance. Typically in vertical shaft units, the upper guide bearing and lower guide bearing are accessible in the open pit under the generator and above the turbine impeller. Two proximity probes per bearing are used here in an XY configuration to monitor shaft vibration and shaft location in the sleeve bearing. The purpose is not only to monitor impeller vibration but also to verify that the shaft stays centered in the bearing. This monitoring avoids metal to metal contact between the bearing surfaces. Any contact between bearing surfaces forces all oil out and can quickly result in catastrophic failure.

(1) There are three main types of hydroelectric turbine. The Kaplan, which is designed for low pressure and high flow, the Pelton, which is designed for high pressure and low flow, and the Francis, which is a compromise and the most common turbine in use.

Eddy current probes must be selected based on the thread type and dimensional requirements of the specific machine. Determine the specific requirements based on existing sensor models or from manufacturers specifications before selecting the specific model 1442 Series sensors. Manufacturer specifications and assistance from plant personnel is required to mount probes.

Generator upper bearings are subject to extreme magnetic fields and are typically not monitored with Eddy Current probes. These probes would be in a continual state of electrical saturation. At these locations, capacitive probes are required which are not affected by the electromagnetic forces surrounding the rotor. Capacitive probes(1) can also be used to measure the position and run-out of the rotor inside the generator.

Use low-frequency accelerometers such as catalog number 1442-ACC-LF-T. See Sensors for further information. Accelerometers are mounted on the head cover and near the draft tube. Accelerometers are mounted this way to monitor such issues as cavitation in the impeller and absolute vibration measurements on the guide bearings and generator.

Thrust is best measured with proximity probes that target the thrust bearing mounted parallel with the shaft centerline. This method is not always possible in a retrofit situation. In such cases, general-purpose accelerometers can be used to provide a warning that the thrust surfaces have come together and require immediate attention.

For the typical eight sensor solution, two measurement modules are required.

Many hydro turbines include a tachometer. If one is present, then include an Expansion Tachometer Signal Conditioner Module.

See Bill of Materials for the specific catalog numbers and quantities.

(1) A capacitive probe is typically a flat device that allows mounting inside the generator between the rotor and stator. Vibration measurements can be taken with these probes as well to give an indication of end winding vibration and other electrical problems. Capacitive probes are available from MC-Monitoring.com.

Machine Faults

Table 72 - Machine Faults—Hydro Turbines

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and Axial
	Vane pass[2]	1*x*VP, can be caused by flow restrictions or uneven gap between rotating vanes and stationary diffuser	Radial
	Cavitation[3]	Typically caused by low suction pressure	Radial
Uncommon faults	Loose stator laminations	2*x* Line and Number of rotor bars x RPM	-
	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial

(1) 1*x*, 2*x*. . .N*x*—When not qualified, N*x* refers to multiples of running speed.

(2) Number of Vanes *x* Running Speed

(3) Indicated by random High frequency vibration higher than Vane Pass frequency. Vibration, which is related to excited structural resonance frequencies, can occur at frequencies from the Vane Pass frequency up to the Maximum Frequency.

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 10x running speed or about 1500 Hz. If low frequency noise is expected, due to flow induced vibration that is associated with the driven machine, apply a high pass filter at a frequency sufficiently high to filter the noise from the measurement.

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller. See Controller Based Alarm Detection for examples of how to program the controller to perform the alarm detection.

Set limits based on maximum percentage of overall level and deviations from observed norms.

Table 73 - Band Definitions—Hydro Turbines

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	1.5*x*	2*x*	2.5*x*	3*x*	1*x* VPF	Cavitation Frequency
	Indicated Fault or Fault Qualification	Rotor Vortices	Unbalance	Rub	Misalignment, Looseness	Rub	Misalignment, Looseness	Vane Pass	Cavitation
Frequency Range		0.2. . .0.8*x*	0.8. . .1.2*x*	1.2. . .1.8*x*	1.8. . .2.2*x*	2.2. . .2.8*x*	2.8. . .3.2*x*	0.8. . .1.2*x* VPF	RF, 1 *x* BP, or 1.05*x* VPF—FMAX
% of Overall Level		10%	90%	10%	20%	5%	10%	10%	10%

Generators

Large electric generators, such as those in most power plants, are fitted with Fluid Film Bearings. In most cases, the sizes of these generators are 100 MW and up but fluid film bearings can be applied to much smaller machines as well.

Steam, hydro, industrial gas, and wind turbines can power generators. New, larger wind turbines are being introduced continually. Few, if any, generators that are driven by small hydro turbines, diesel engines, small aeroderivative gas turbine, or most wind turbines are fitted with fluid film bearings.

Most large generators use field coils, rather than permanent magnets, to create the magnetic field necessary for power generation. When field coils are used, an exciter must be used to create the magnetic field during and after startup to maintain the required output voltage.

Sensor and Monitor Requirements

See Bill of Materials for the specific catalog numbers and quantities.

Monitor selection (page 28) is based on the number of sensors required. If the application requires local relays for trip or annunciation, and if multiple relays are required, then add additional Expansion Relay Modules.

Machine Faults

Table 74 - Machine Faults—Generators

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and Axial
	Oil Whirl/Whip	0.3…0.49*x* then 0.5*x* typical. Can start as whirl and lock onto a rotor critical frequency typically at 0.5*x* running speed.	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5*x* high levels of half order harmonics	Radial
	Eccentric air gap, shorted, or loose windings	2*x* line frequency	Radial
	Loose rotor bar	1*x* with 2*x* slip frequency side bands	Radial
	Loose windings	2*x* line and number of rotor bars *x* RPM	Radial

(1) 1*x*, 2*x*…N*x*—When not qualified, N*x* refers to multiples of running speed.

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 10x running speed or about 1500 Hz.

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller.

Table 75 - Band Definitions—Generators with Fluid Film Bearings

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	1.5*x*	2*x*	2.5*x*	3*x*	1*x* LF	2*x* LF
	Indicated Fault or Fault Qualification	Oil Whirl/ Whip/Rub	Unbalance	Rub	Misalignment, Looseness	Rub	Misalignment, Looseness	Electrical	Electrical
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…1.8*x*	1.8…2.2*x*	2.2…2.8*x* LF	2.8…3.2*x*	0.95…1.05*x* LF	1.8…2.2*x* LF
% of Overall Level		10%	90%	10%	20%	10%	10%	10%	10%

Pumps—Horizontal Mount

Sensor and Monitor Requirements

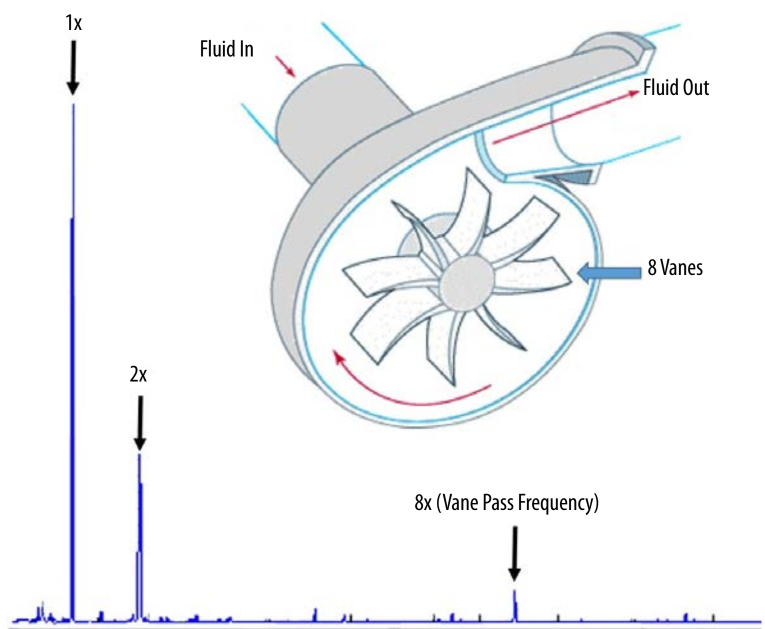
For small pumps, less than 500 HP, place one sensor in the radial (horizontal) direction at each bearing.

For large pumps, greater than 500 HP, place two sensors at each bearing. If the pump is center mount, mount all sensors in the radial direction (vertical and horizontal). If the pump is overhung, such as illustrated in Figure 9, then mount one of the sensors at the pump inboard bearing in the axial direction (axial and horizontal).

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

The following table is for a single stage, single volute, centrifugal pump. For a multi-stage pump, the vane pass of each stage must be monitored. Each stage has a different number of vanes.



The pressure at the volute (fluid out) varies as the impeller rotates depending on how exactly the vanes align with the outlet at any given moment.

For any centrifugal pump, the pressure at the volute pulsates at a frequency equal to the number of vanes multiplied by the speed of the pump. This value is called the vane pass frequency.

For a double volute pump, vane pass frequency is two times the number of vanes.

Band Definitions and Limits

Table 76 - Machine Faults—Horizontal Mount Pumps

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Misalignment	1*x*/2*x*/3*x*	Radial and Axial
	Oil Whirl/Whip	0.3…0.49*x* then 0.5*x* typical. Can start as whirl and lock onto a rotor critical frequency typically at 0.5*x* running speed.	Radial
	Vane pass[2]	1 *x* VP, can be caused by flow restrictions or uneven gap between rotating vanes and stationary diffuser	Radial
	Cavitation[3]	Typically caused by low suction pressure	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5x high levels of half order harmonics	Radial

(1) 1*x*, 2*x*…N*x*—When not qualified, N*x* refers to multiples of running speed.

(2) The numbers of vanes or blades on the impeller times running speed. If the number of vanes is unknown, set this band to 4…7 times the running speed (Pumps often have 4…7 vanes).

(3) Random High frequency vibration higher than Vane Pass frequency. Can occur at frequencies higher than Vane Pass frequency up to the Maximum Frequency and can be confused with bearing anomalies, the presence of cavitation must be ruled out before suspecting bearing problems.

Set the maximum frequency (FMAX) to approximately 10x running speed or about 1500 Hz. If low frequency noise is present, apply a high pass filter to a frequency sufficiently high to filter the noise from the measurement.

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller.

Table 77 - Band Definitions—Pumps with Fluid Film Bearings

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	1.5*x*	2*x*	2.5*x*	3*x*	1*x* VPF	Cavitation Frequency
	Indicated Fault or Fault Qualification	Oil Whirl/ Whip/Rub	Unbalance	Rub	Misalignment, Looseness	Rub	Misalignment, Looseness	Vane pass	Cavitation
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…1.8*x*	1.8…2.2*x*	2.2…2.8*x*	2.8…3.2*x*	0.95…1.05*x* VPF	RF, 1 *x* BP, or 1.05*x* VPF—FMAX
% of Overall Level		10%	90%	10%	20%	10%	10%	10%	10%

Fans, Compressors, and Blowers—Large

Sensor and Monitor Requirements

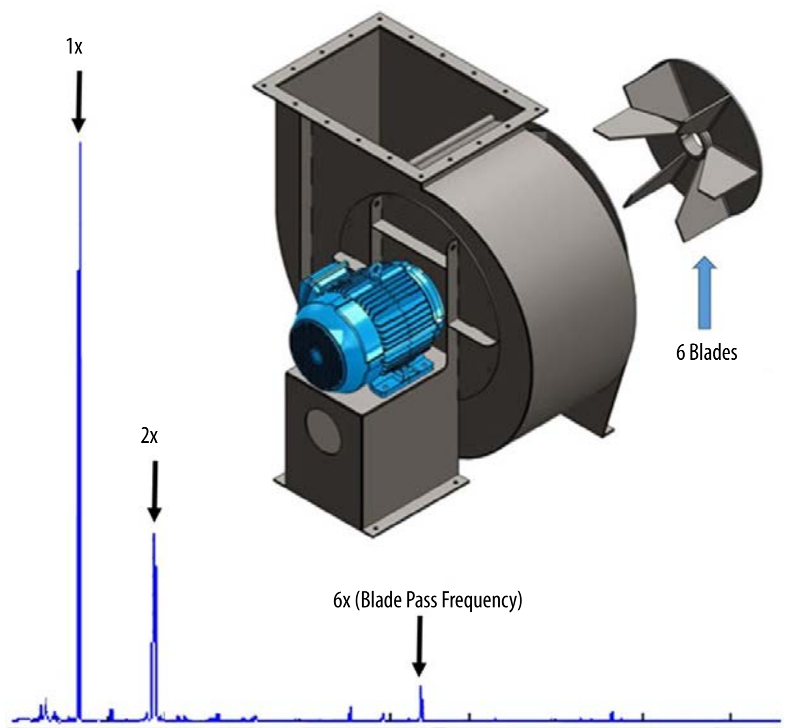
For small fans, less than 500 HP, place one sensor in the radial (horizontal) direction at each bearing.

For large fans, greater than 500 HP, place two sensors at each bearing. If the fan is center mount, mount all sensors in the radial direction (vertical and horizontal). If the fan is overhung, such as illustrated previously, then mount one of the sensors at the fan inboard bearing in the axial direction (axial and horizontal).

See Bill of Materials for the specific catalog numbers and quantities.

Machine Faults

The following table is for a single stage machine. For multi-stage machines, the blade pass of each stage must be monitored. Each stage has a different number of blades.



The pressure at the outlet varies as the fan rotates depending on how exactly the blades align with the outlet at any given moment.

For any fan, pressure at the outlet pulsates at a frequency equal to the number of blades multiplied by speed of the fan. This value is called blade pass frequency.

Table 78 - Machine Faults—Large Fans

Types of Faults	Possible Problems[1]	Frequency (Order)	Primary Direction
Common faults	Unbalance	1*x*	Radial
	Oil Whirl/Whip	0.3…0.49*x* then 0.5*x* typical. Can start as whirl and lock onto a rotor critical frequency typically at 0.5*x* running speed.	Radial
	Blade Pass[2]	1 *x* BP, can be caused by flow restrictions such as incorrect damper settings	Radial
Uncommon faults	Looseness—mounting looseness	1*x*/2*x*/3*x*/4*x* can be higher harmonics	Radial
	Looseness—component loose on shaft	0.5*x*/1*x*/1.5*x*/2*x*/3*x* can be higher harmonics	Radial
Unusual faults	Bent shaft	1*x*	Axial
	Rotor rub	0.5*x*/1*x*/1.5*x*/2*x*/2.5*x*/3*x*/3.5*x* high levels of half order harmonics	Radial

(1) 1*x*, 2*x*…N*x*—When not qualified, Nx refers to multiples of running speed.

(2) Number of Blades x Running Speed

Band Definitions and Limits

Set the maximum frequency (FMAX) to approximately 10x running speed or about 1500 Hz. If low frequency noise is present, apply a high pass filter to a frequency sufficiently high to filter the noise from the measurement.

FFT Band Definitions and Alarm Limits

The module does not have sufficient number of alarms to implement the alarming for all eight bands. In most cases then, only the critical measures/alarms should be implemented in the monitor, typically that means alarms that monitor the Overall and 1x magnitude measurements. Other alarms, typically only applied to provide diagnostics, can be implemented in the controller.

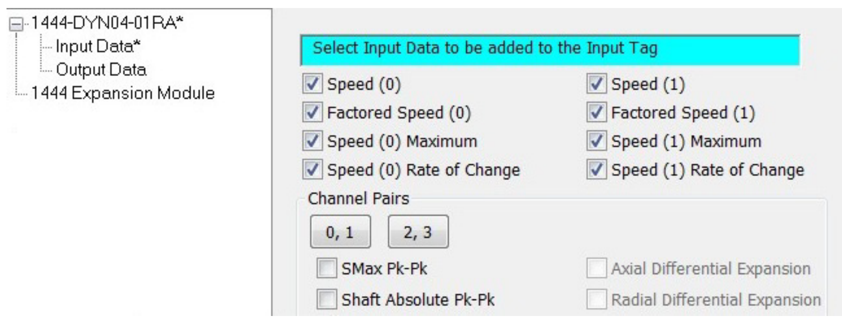
Table 79 - Band Definitions—Fans with Fluid Film Bearings

Attributes		Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	Fault Frequency	<1*x*	1*x*	1.5*x*	2*x*	2.5*x*	3*x*	1*x* BPF	RF
	Indicated Fault or Fault Qualification	Oil Whirl/ Whip/Rub	Unbalance	Rub	Misalignment, Looseness	Rub	Misalignment, Looseness	Blade Pass Frequency (Turbulence)	Resonance Frequency
Frequency Range		0.2…0.8*x*	0.8…1.2*x*	1.2…1.8*x*	1.8…2.2*x*	2.2…2.8*x*	2.8…3.2*x*	0.95…1.05*x* BPF	Machine Specific
% of Overall Level		10%	90%	10%	20%	10%	10%	10%	10%

Notes:

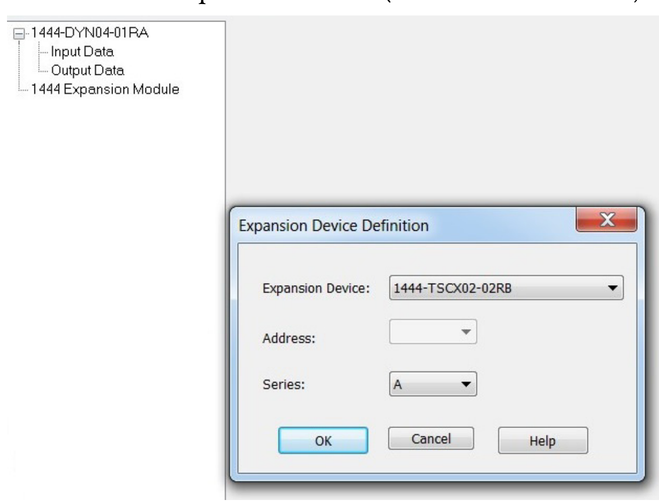
Speed Measurements

The Dynamix™ 1444 accommodates two-speed inputs. Each input can be measured from a speed signal or read from the module input assembly. For each speed, two-speed values can be applied. Values are the measured or input speed, which is multiplied by a user provided factor. The result is the factored speed. A factored speed is commonly applied when monitoring a machine with a gear set with a known gear ratio. Knowing the gear ratio makes it possible to calculate the output or intermediate shaft speed.



When a machine is not fitted with a tachometer or its signal cannot be provided to the module, the speed value (RPM) can simply be read from a controller output tag.

When a speed signal can be provided, the dynamic measurement module requires a TTL class signal input. Direct wiring can provide the TTL signal to the terminal base or the module interconnect transmits the TTL signals provided via an associated Tachometer expansion module (1444-TSCX02-02RB).



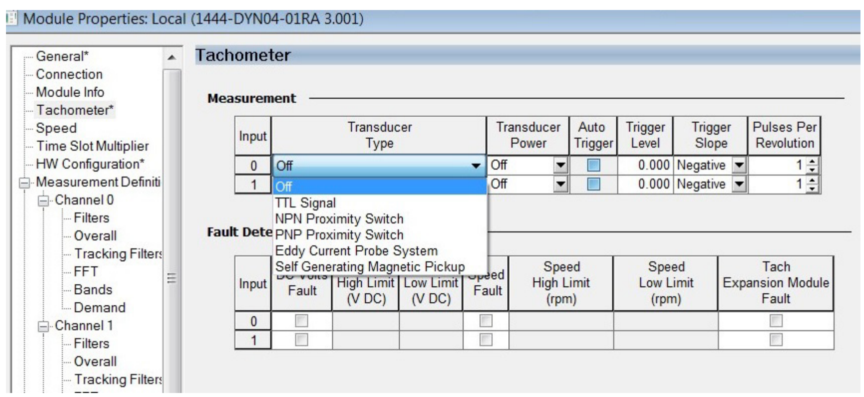
The tachometer expansion module can accept input signals from various standard speed sensors such as eddy current probes, self-generating magnetic speed sensors, NPN, and PNP proximity devices.

When required, the tachometer module provides power to the sensors either –24V DC or + 24V DC.

The tachometer expansion module is added to the system on the module definition page by right-clicking on 1444 expansion module.

Check the series revision of the hardware (label on rear of the module). Older modules are Series A while newer modules are Series B. The only difference between series A and B modules is that series B modules provide auto threshold level detection. Series B modules began shipping in August 2016.

The TTL signals from a tachometer expansion module can be supplied to up to six main modules. The module that is the host configures it. The tachometer configuration page is presented only when, and for, the host module that has the tachometer signal conditioner that is specified in its module definition. Dynamic measurement modules that are connected to a tachometer module but are not its host do not show the tachometer page since only the host tachometer module configures it.

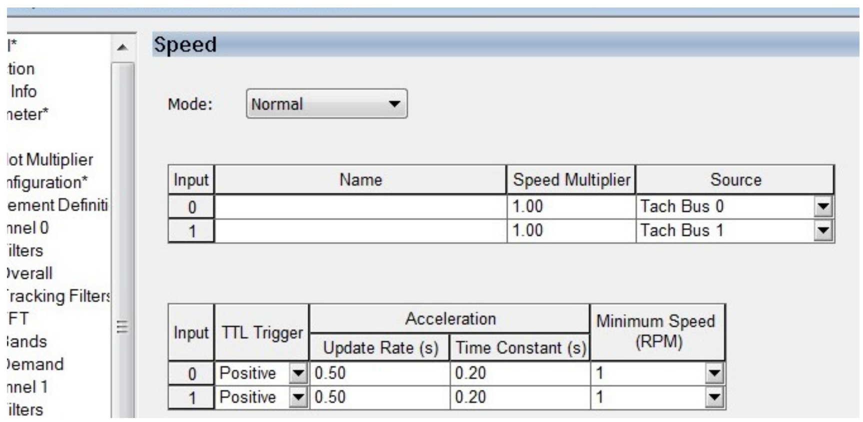


The auto trigger function is only available for series B hardware.

The tachometer is used for both speed measurements where the sensor can target a multi-toothed wheel and phase measurements. The sensor targets a single notch or shaft keyway which provides a single once-per-turn reference signal.

IMPORTANT	The Tachometer Signal Conditioner produces a once-per-rev TTL pulse from any multi-pulse input, such as a toothed wheel. This pulse could then be used to provide phase measurements, typically from a tracking filter. However, if phase measurements are made using a multi-pulse-per-rev source, then phase is not referenced to any specific machine location, such as a key way. In these cases phase is valid and is consistent while power is applied to the system. However, when power is cycled there is no assurance that phase does not change from any earlier operation.

If a multi-tooth wheel is being sensed, the numbers of teeth (pulses) are entered under pulses per revolution.



The speed multiplier column is used to enter the gear ratio if the factored speed value is being used.

Phase Measurements

Often the most important reason for speed measurement using a TTL rather than simply reading an RPM value from the input assembly is that measuring it allows the module to calculate phase measurements. When phase is important, confirm that the trigger slope specified on the speed configuration page is set the same as the trigger slope on the tachometer page as shown in Figure 20 and Figure 21.

Figure 20 - Trigger Slope—Speed Configuration Page

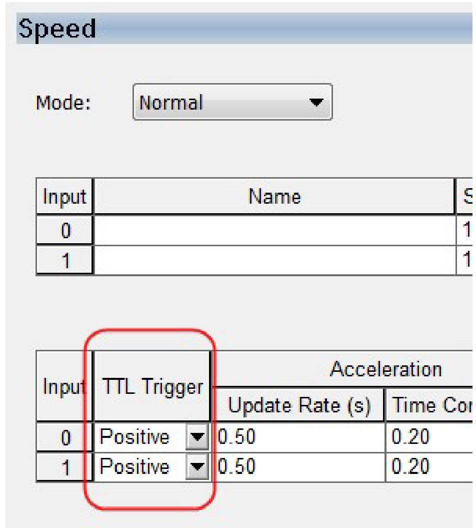
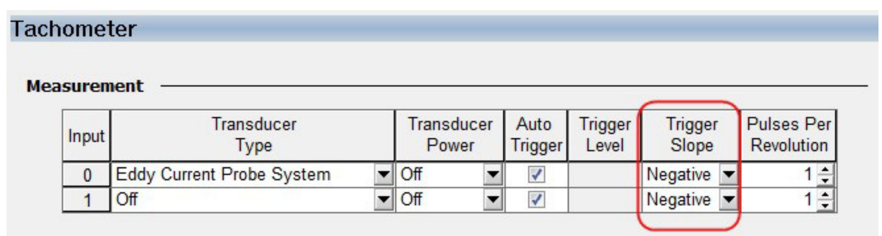


Figure 21 - Trigger Slope—Tachometer Page

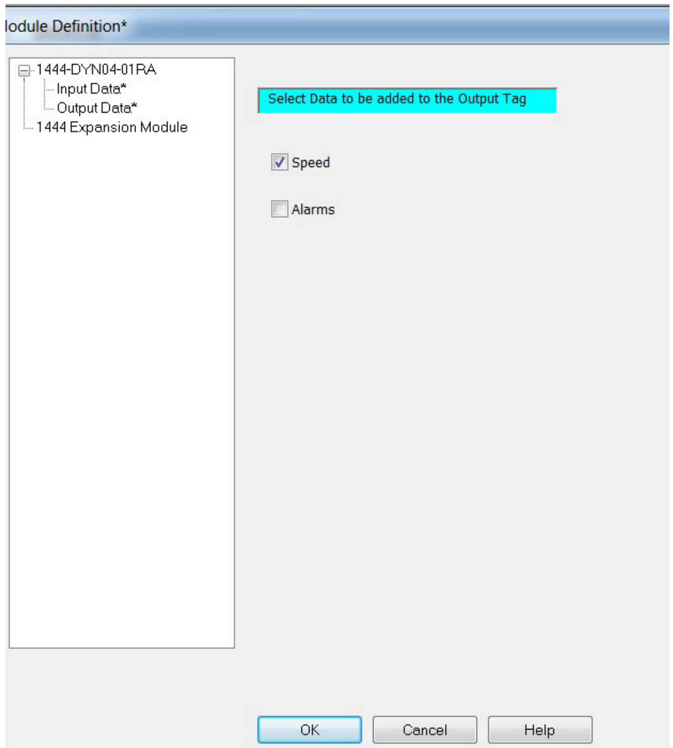


This tachometer page is only presented on the module that is hosting the tachometer signal conditioner.

Reading Speed Over I/O

The ability to obtain a speed value from the controller is of particular use when it is not possible or practical to install a separate speed sensor.

By default, the module input (controller output) assembly excludes the two optional speed values. To add them to the assembly, check the speed selection on the output data page in module definition.



When speed is checked, two floating point tags are added to the output assembly.

DYN_01:O.Speed	{...}	{...}	Float
DYN_01:O.Speed[0]	0.0		Float
DYN_01:O.Speed[1]	0.0		Float

The machine speed that is read from a drive or other I/O module must be copied to either of these output assembly tags. The tag value must be in RPM.

The output assembly tag can be used just like the other speed inputs, to control FFT bands for instance, but it cannot be used as an input to the Tracking filter or to control synchronous sampling. For these functions, an actual speed reference (pulse) signal is required.

Calculating Speed from an FFT Band

If there is no tachometer and the controller is not aware of machine speed, it is possible to measure speed by using an FFT band. To measure using an FFT band, define a band that spans a frequency range that includes the expected running speed and set the band type to frequency of maximum peak. The tag then includes the frequency of maximum peak within the range in Hz. In almost all cases, this measurement is running speed, simple logic can be applied to move this value to the output speed tag (after multiplying by 60).

In most cases, the magnitude of the maximum peak is checked first to verify that the machine is running. This confirmation makes an assumption that a running machine exhibits some minimum level of vibration at running speed frequency.

The following project is an example of this concept.

The project that is shown adds two tags to channel 3.

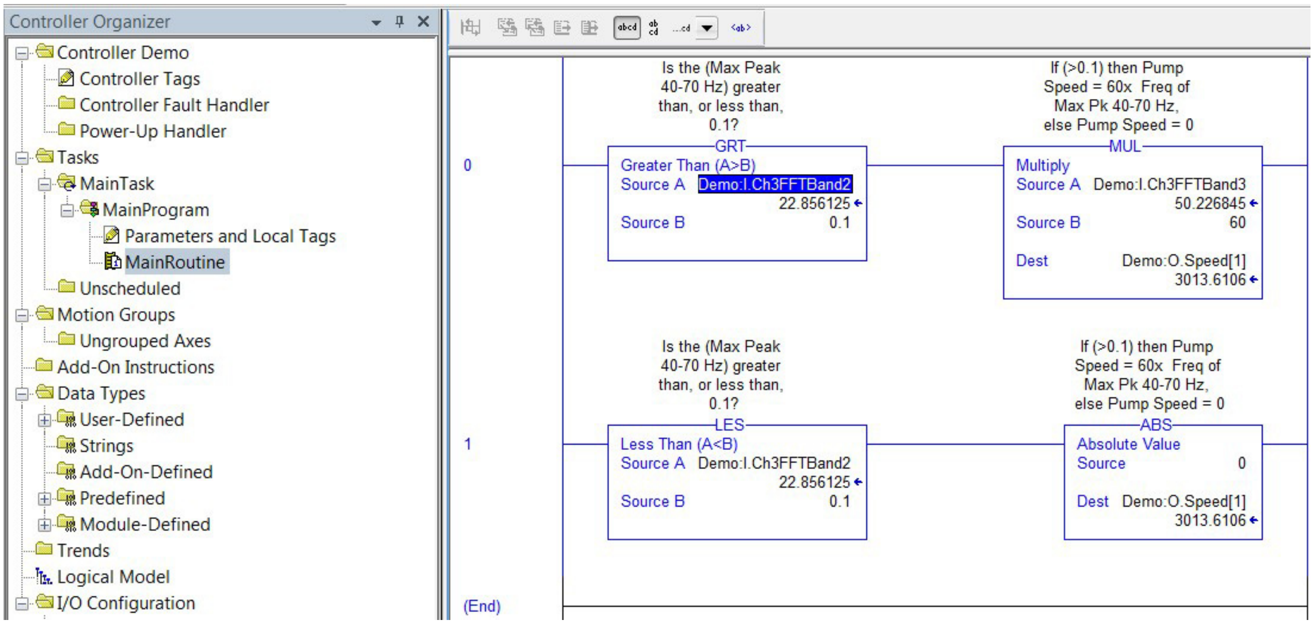
Bands

Band	Enable	Measurement Mode	Band Limit Begin	Band Limit End	Domain	Speed Reference
0	☑	Band Overall	0.10	100.00	Hz	
1	☑	Band Overall	4.90	5.10	Orders	Speed 1
2	☑	Band Maximum Pk	40.00	70.00	Hz	
3	☑	Freq of Band Maximum Pk	40.00	70.00	Hz	
4	☐	Band Overall	0.10	100.00	Hz	
5	☐	Band Overall	0.10	100.00	Hz	
6	☐	Band Overall	0.10	100.00	Hz	
7	☐	Band Overall	0.10	100.00	Hz	

Band2 is the magnitude of the maximum peak. Band2 is monitoring the frequency span of 40…70 Hz to confirm that it includes running speed.

Band3 is the frequency of the maximum peak that is measured in Band2.

Use the Band2 and Band3 tags, add a ladder logic that first checks the magnitude of the maximum peak (Band2). If Band2 is greater than 0.1 (in this example), multiply the value of the frequency of the maximum peak (Band3) by 60 and write the result to the output speed1 tag.



The following table is the result when the machine is running.

Demo:I.Ch1FFTBand0	10.130318		Float	REAL	OA 0 - 100 Hz
Demo:I.Ch2FFTBand0	24.856121		Float	REAL	OA 0 - 100 Hz
Demo:I.Ch3FFTBand0	24.863258		Float	REAL	OA 0 - 100 Hz
Demo:I.Ch0FFTBand1	1.33122085e-004		Float	REAL	Flow Turbulance: Vane Pass (5x)
Demo:I.Ch1FFTBand1	1.29499531e-004		Float	REAL	Flow Turbulance: Vane Pass (5x)
Demo:I.Ch2FFTBand1	1.80618600e-004		Float	REAL	Flow Turbulance: Vane Pass (5x)
Demo:I.Ch3FFTBand1	1.85695477e-004		Float	REAL	Flow Turbulance: Vane Pass (5x)
Demo:I.Ch3FFTBand2	22.856396		Float	REAL	Max Peak 40-70 Hz
Demo:I.Ch3FFTBand3	50.226845		Float	REAL	Freq of Max Peak 40-70 Hz
Demo:I.Speed0	3001.125		Float	REAL	Tacho Speed (0)
Demo:I.Speed1	3013.6104		Float	REAL	I/O Speed (1)
Demo:O	{...}	{...}		AB:1444_DY...	
+ Demo:O.Control	0		Decimal	INT	
Demo:O.TripInhibit	0		Decimal	BOOL	
Demo:O.TransientBuffer3Reset	0		Decimal	BOOL	
Demo:O.MaxSpeedReset	0		Decimal	BOOL	
− Demo:O.Speed	{...}	{...}	Float	REAL[2]	
Demo:O.Speed[0]	0.0		Float	REAL	
Demo:O.Speed[1]	3013.6106		Float	REAL	Pump Speed Calculated from Freq ...

For more information on configuration of the tachometer and speed pages, refer to the user manual 1444-UM001.

Notes:

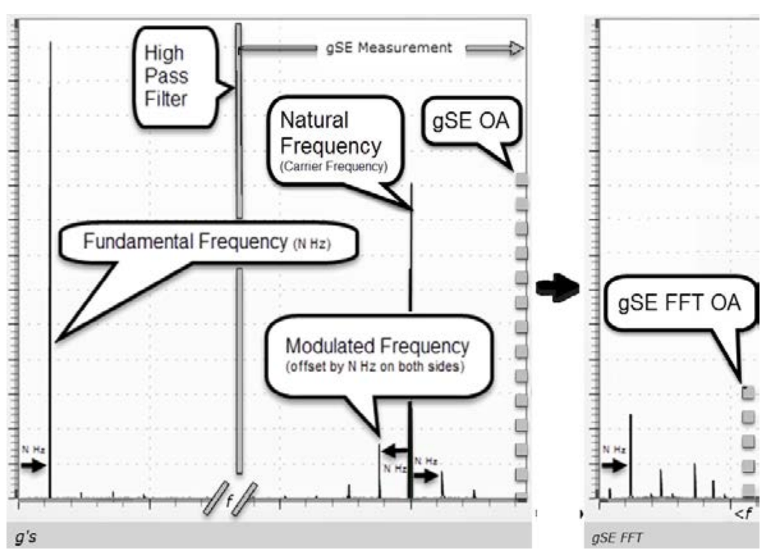
Spike Energy (gSE) Measurements

Theory

When a metal structure is impacted it rings (vibrates) at its natural frequency, just like a tuning fork. If the impact happens just once, then the ringing will slowly decay away. But if the impact is repetitive, such as from a scratch on a spinning ball that impacts a surface, or a damaged tooth on a rotating gear when it mates with the next gear, then the vibration at the natural frequency will never decay away, and further vibration will appear at frequencies on either side of the natural frequency as it is modulated by the rate of impacting—the frequency of the spinning ball or gear rotation.

Common integrated circuit piezoelectric (ICP) industrial accelerators are manufactured in such a way as to insure that the natural frequency of the sensor structure is well above the rated maximum frequency of the sensor, so that it does not interfere with measurements. However, if a measurement system samples fast enough, so at a high enough maximum frequency (FMAX), to span the natural frequency, it will see vibration at the sensors natural frequency if it is being impacted (excited).

This phenomenon is the basis for how and why enveloping/demodulating signal processing techniques such as Spike Energy™ work. These techniques require measuring at a sample rate that goes well above the rated FMAX and above the natural frequency of the sensor while filtering out the lower frequency signals, which are generally far larger in magnitude—so would dwarf the meaningful high frequency signals at and around the sensors natural frequency.



Spike Energy

The Spike Energy algorithm filters out low frequency content, and then measures the high frequency signals up to 40 kHz FMAX or higher depending on the measurement system. The measurement produces a gSE Overall (OA) value that is typically dominated by the magnitude of the sensors natural frequency, particularly when the natural frequency is in resonance (is excited). Additionally, the Spike Energy function produces a gSE Time Waveform and Spectrum that includes the low frequencies that were modulated around the natural frequency.

If a tuning fork is struck harder, it rings louder. Similarly, as the impacting becomes more severe, the magnitude of vibration at a sensor's natural frequency increases. Consequently, the most telling indication of a worsening condition with regard to the source of impacting, is an increase in the magnitude of the excited natural frequency. While the frequencies of modulated vibrations around the natural frequency can indicate the source of the impacting, the magnitude of the modulated signals may or may not contribute significantly to the overall vibration of the measurement.

So in most cases the value of the gSE measurement is simply the gSE Overall (OA) measurement. The gSE OA will provide clear indication of impacting, whether repetitive or random / discrete, without the need to observe the details of the spectrum. However, when it is necessary to diagnose the specific source of impacting, when it is repetitive, then the gSE FFT is available for the analyst to do that.

The Spike Energy measurement then is suitable for impact detection that is associated with:

Repetitive Faults
Rolling element bearing defects
Gear teeth defects
Flow turbulence
Rotor rub
Looseness (rotating element)

Random/Non-repetitive Events
Insufficient lubrication
Cavitation
Water ingestion in compressor
Non-rotating element impact events
Looseness (non-rotating element)

Repetitive events, such as those listed previously, occur continuously and can be detected by infrequent measures, including by use of portable data collectors. However, events that are process related, such as cavitation, or that are random, such as impacting from water ingestion into a compressor, require continuous monitoring.

Transducer Function

The gSE algorithm works best when the sensors natural frequency is near to the center of the measurement range. So, if a Dynamix™ 1444 monitor is used, which measures to 40 kHz and if set to a 5 kHz high pass filter, then the mid-range of the gSE measurement will be about 22.5 kHz.

Figure 22 and Figure 23 show the specification sheets of two accelerometers and their respective typical resonant frequencies. Note that the resonant frequency of the accelerometer in Figure 22 is 28,000 Hz while the resonant frequency of the accelerometer in Figure 23 is 23,000 Hz. In this example, while either sensor will work, the preferred sensor would be the one in Figure 23, if used with a Dynamix 1444-DYN04-01RA monitor.

The amplitude of the gSE overall is a function of impact signal amplitude and the sensitivity and frequency of the carrier (natural) frequency of the accelerometer. If two accelerometers with different resonant frequencies were mounted side by side, they would measure two different gSE overall amplitude values. Even two identical sensors mounted side by side would produce different values if the mounting of the sensors were at all different because mounting effects stiffness, which effects the natural frequency. For these reasons, it is important when using Spike Energy to consider the trended behavior of the values, whether gSE OA or peaks or band values from the gSE FFT, rather than the absolute magnitude of the value.

Figure 22 - Accelerometer Sheet

1443-ACC-GP-T Accelerometer

Table 6 - 1443-ACC-GPT-T Accelerometer Specifications

Attribute	Value	
Performance		
Sensitivity (±10%)(1)	10.2 mV/(m/s²)	100 mV/g
Measurement range	±785 m/s²	±80 g
Frequency range (±5%)(2)	2.0…14 kHz	120…840000 cpm
Frequency range (±3 dB)	0.8…19 kHz	48…1140000 cpm
Resonant frequency, typical	28 kHz	1680 kcpm
Broadband resolution (1…10000 Hz), typical	2943 µm/s²	300 µg
Non-linearity(3)	±1%	±1%
Transverse sensitivity	< 5%	< 5%

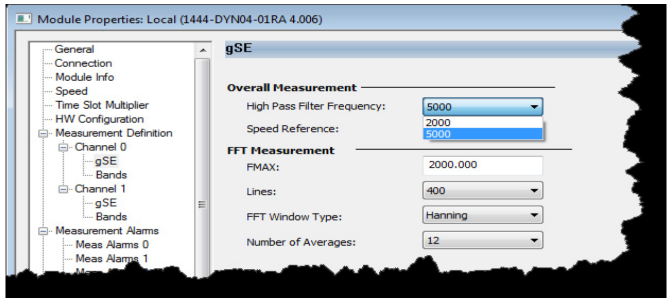
Figure 23 - Typical Resonant Frequency

Case Material	316L Stainless Steel	
Mounting	1/4-28	
[illegible]	[illegible]	
Resonant Frequency	1,380,000 CPM	23000 Hz
Mounting Torque	2 to 5 ft. lbs.	2,7 to 6,8 Nm
Mounting Hardware	1/4-28 Stud	M6x1 Adapter Stud

High Pass Digital Filters and Spike Energy

The Dynamix 1444 has two options for gSE filtering, a 2000 Hz filter and a 5000 Hz filter. Select the filter that is above the expected maximum frequency of vibrations occurring in the measurement such as gear mesh or blade pass frequencies. This selection is to insure that the gSE measurement is not influenced by vibrations other than those associated with the sensors natural frequency and modulated frequencies around it.

Figure 24 - Module Properties



The Dynamix 1444 monitor also allows setting a gSE FFT FMAX. This value must be set less than the selected High Pass Filter because the content of the gSE FFT is the high frequency de-modulated signals, above high pass filter.

<table>
<tr><th>High Pass Filter</th><th>Minimum FMAX</th><th>Maximum FMAX</th></tr>
<tr><td>5000 Hz</td><td rowspan="2">100 Hz</td><td>4999 Hz</td></tr>
<tr><td>2000 Hz</td><td>1999 Hz</td></tr>
</table>

Overall Vibration Upper Limits for Spike Energy

Displacement, velocity, and acceleration vibration measurements have established standardized tables, which suggest maximum not to be exceeded vibration amplitudes for various machine types. There are no such standard tables for Spike Energy measurements. The overall Spike Energy amplitude in units of gSE measured by the FFT analyzer is dependent on machine dynamic and structural characteristics, the type of accelerometer and mounting methods, the accelerometer mounting locations, the signal carrier frequency, and the fundamental frequency of the measured defect. All these variables preclude the establishment of 'expected' or 'typical' Spike Energy measurement amplitudes. In practice, normal Spike Energy gSE overall values may be 0.05 gSE or 8 gSE, depending on the relevant variables. Either may be perfectly acceptable. Consequently, the most meaningful use of Spike Energy is to trend measured amplitudes over time and look for shifts in the values from initial baseline amplitudes of known good components and machines.

Trending the Spike Energy gSE overall value will usually provide the earliest possible indication of a bearing fault. Spike Energy trending typically occurs over weeks and months duration. Rolling element bearing failure is typically highly predictable. This predictability does not rule out the rare catastrophic failure, but it is a common practice to calculate the actual life of a bearing based on load, speed of rotation, bearing geometry, and so on. An accepted failure calculation is the L10 Bearing Life formula. See L10 Bearing Life on page 212 in the glossary for more information.

Change in amplitude due to bearing early stage failure will typically be very gradual, and with a steadily increasing slope. As indicated previously, periodic monitoring is required. When the rate of change in the trend 'increases', failure is becoming more imminent and indicates that more frequent monitoring is required. At this stage, in addition to periodic viewing of the amplitude data, system alarming may become useful. Personnel familiar with each machine may be of assistance in to determine setpoints for the alerts and alarms. If assistance is not an option, statistical alarms based on the variance of amplitudes may prove more useful. Figures 3 and 4 below show examples of trended gSE data. Any historian that can accept ControlLogix® tag data is useful for this purpose when the Dynamix 1444 vibration module is utilized. Note that the slope starts to increase rapidly around the 75th sample in Figure 25.

Figure 25 - Turntable

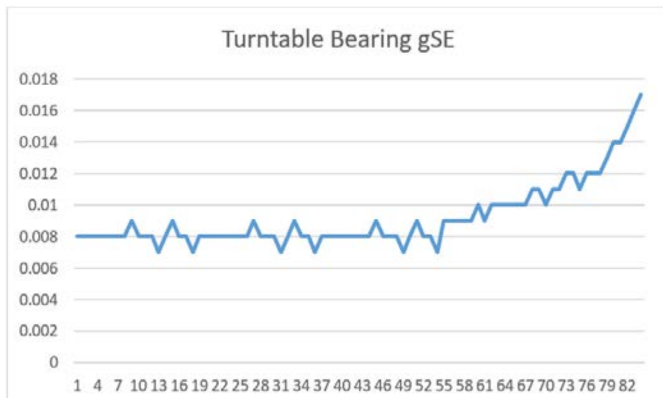
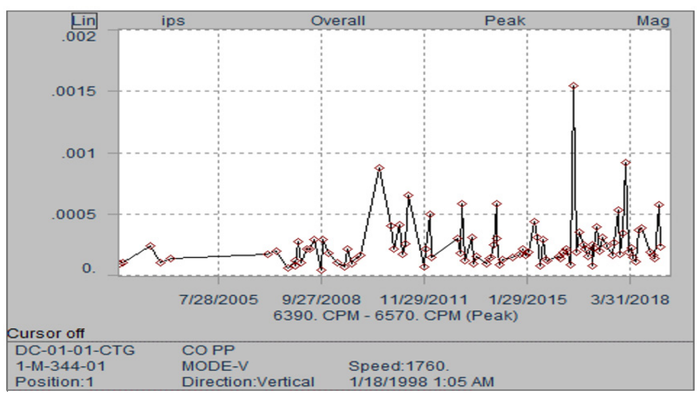


Figure 26 - CPM



Spike Energy Fault Frequencies in Bearings

While it is possible to monitor band frequencies of the Spike Energy spectrum, and to record the spectrum into advanced analytical Emonitor® software, in most cases only the Spike Energy overall measure need be monitored for online systems. In more critical and advanced situations, it is possible to break the Spike Energy spectrum into bands that correspond to known fault frequencies in a specific component such as a bearing or a known gear mesh frequency. Table 80 and Table 81 represent examples of eight possible bands which could be utilized with Spike Energy in the Dynamix 1444 module profile. Use of these bands assumes an advanced knowledge of vibration analysis and bearing failure identification. In Table 80, four measures that are associated with bearing defect monitoring are utilized. These measures are bearing cage frequency, ball spin frequency, outer race frequency, and inner race frequency. These values must be obtained from the bearing manufacturer and requires the specific bearing part number. This is an example only as there may be a need to monitor one specific bearing defect frequency such as outer race issues and, in this scenario, it may be better to monitor the outer race frequency and multiple harmonics of that frequency. In the following two examples, note that the entire usable frequency range to the module FMAX setting is included.

Table 80 - Suggested Spike Energy FFT Band Configurations

Attributes	Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Band Name	FTF Bearing Cage Defects	BSF Ball Spin Frequency	BSOR Outer Race Defect	BSIR Inner Race Defect	2x *BSF*	2x *BSOR*	2x *BSIR*	Bearing Higher Harmonics and Natural Frequencies
Frequency Range	0.2…1.2x *FTF*	0.8…1.2x *BSF*	0.8…1.2x *BSOR*	0.8…1.2x *BSIR*	1.8…2.2x *BSF*	1.8…2.2x *BSOR*	1.8…2.2x *BSIR*	2.2x FMAX

As an alternative solution, the example in Table 81 divides the spectrum into equal divisions and is not designed to identify specific fault frequencies that are associated to bearings or gear teeth. In cases where the fault frequencies may not be known, this spectrum may provide a solution where the Spike Energy spectrum is utilized without knowing the specific frequency ahead of time. As machine defects start to form, peaks of vibration will begin to form in the Spike Energy spectrum. These peaks, while not necessarily associated to a specific defect frequency, do indicate the beginning of failure. Increases in amplitude in any band suggest the use of either an acceleration or velocity spectrum to identify an increase in any suspect fault frequency such as bearing outer race, gear mesh, ball spin, and so on. At this point, the amplitude in the acceleration and velocity spectrum will be very low but should be visible in the spectrum floor. The increased amplitude may not guide the user to a specific defect, but does indicate that maintenance is beginning to be a requirement.

Table 81 - Frequency Range Spectrum

Attributes	Band 0	Band 1	Band 2	Band 3	Band 4	Band 5	Band 6	Band 7
Frequency Range	5…250 Hz	251…500 Hz	501…750 Hz	751…1000 Hz	1001…1250 Hz	1251…1500 Hz	1501…1750 Hz	1751…1999 Hz

As stated earlier, amplitude increases in the Spike Energy bands do not necessarily indicate imminent bearing failure. Increased amplitude should be interpreted as an early warning. The ability to switch profiles in the Dynamix 1444 module should be utilized to switch to either an acceleration measurement profile or a velocity measurement profile. The bearing condition will eventually appear in the velocity or acceleration spectrum when the bearing defect has become more pronounced. Again, use of these tools assumes an understanding of vibration analysis and bearing failure dynamics. Two examples of the use of trended overall Spike Energy amplitude in conjunction with the Spike Energy spectrum are shown in Figure 27 and Figure 28. Note the peak amplitude in Figure 27 that went into alarm and the corresponding higher vibration peaks in the spectrum waterfall. In this example, the problem was a bad motor bearing that was replaced. Note the dramatic shift in overall amplitude. Also note in Figure 30 that the amplitude in the Spike Energy trend was near the alarm level for some time, and then suddenly trended up. At this point, significant spikes in the spectrum waterfall also appeared.

Figure 27 - Alarm 1

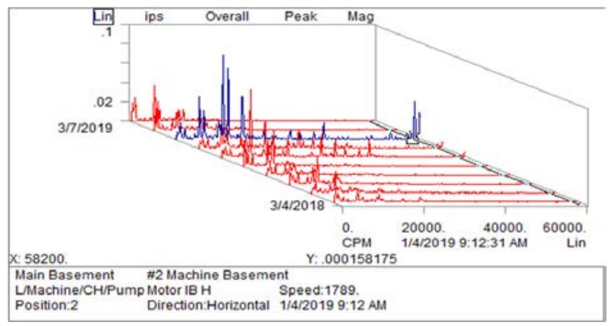


Figure 28 - Alarm 2

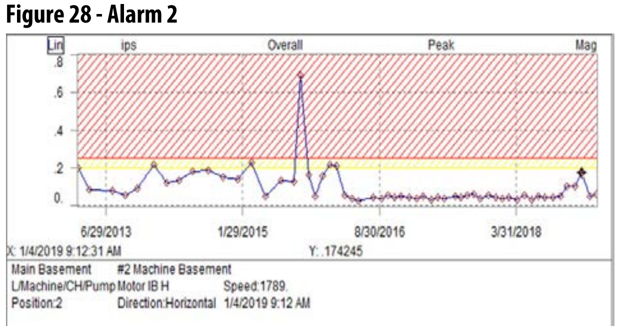


Figure 29 - Alarm 3

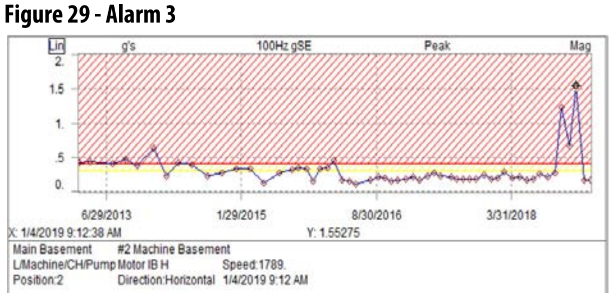
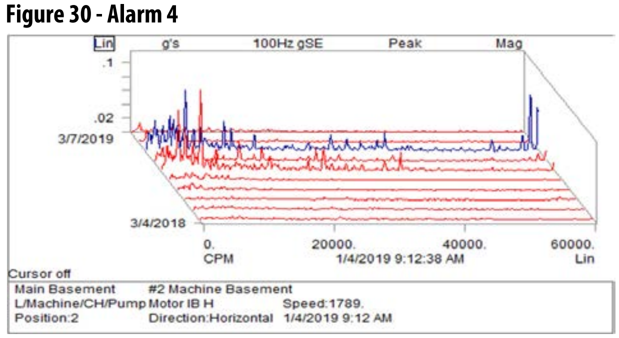


Figure 30 - Alarm 4



Spike Energy Fault Frequency Alarm Limits for Bearings

As with overall amplitude values in Spike Energy, there is no suggested upper acceptable limit for Spike Energy amplitudes that is related to fault frequencies in bearings. The same factors affect the overall Spike Energy amplitude level such as mounting method, proximity of the accelerometer to the bearing and the transmission path between the bearing and the accelerometer are equally relevant to individual fault frequencies.

It is the change in amplitude level over time and the presence of specific fault frequencies that are the best indicators of bearing problems, rather than the actual numeric value. Therefore, a baseline Spike Energy level should be established during machine operation in known good condition and warning levels set accordingly. As these baseline overall amplitudes begin to increase, amplitudes in one or more bands will also likely start increasing. Standard vibration measures including velocity and acceleration will now also show increased levels of amplitude in the previous frequency bands that indicate the bearing need to be changed.

Random Impact Event Detection

While Spike Energy is most commonly used for bearing fault detection, it is also useful for detecting random impact events in process applications. In these cases, the measurement must be performed continuously, and limits set appropriately based on observation (of the target process). For these applications, only the Spike Energy Overall measurement in gSE units is typically required.

Examples of applications that generate random impact events include:

Pump Cavitation

In most cases pump cavitation will be readily observable using standard FFT Band measurements, over the observed frequency range that is excited when cavitating. However, in some cases the spike energy measurement may prove a better indicator of cavitation.

Loose Parts Detection

Any application where it is possible for a part to randomly come loose and an impact will occur. Examples of impacts that occur from loose parts are in assembly processes where a misapplied part may impact a structure. In many such cases, the impact is detectable using the Spike Energy measurement.

Extremely Slow Rotating Bearings

Most industrial machines rotate at speeds somewhere between 600…3600 RPM. While infrequent, machines do operate below 600 RPM, and as low as 100 RPM. The optimal measurement for general monitoring of all these frequencies is velocity, which may require the use of a low frequency accelerometer at speeds below 600 RPM. There are those cases, while even more infrequent, where the speeds of machine shafts and bearings operate well below the practical limits of the FFT spectrum. Mathematically, the FFT spectrum is only usable at speeds above 34 RPM. Shaft speeds of 1 RPM and 3 RPM may be encountered in certain applications, well below the usable FFT range. While these speeds are too low to create imbalance issues, and so on, they often lead to premature bearing failure due to excessive force in the load zone.

The use of Spike Energy has proven useful in monitoring these extremely slow speed applications. The Spike Energy measurement is not intended to measure rotational vibration in these instances, but the 'ring' of the rolling element that contacts the defect in the mating bearing component. The Spike Energy overall amplitude will increase as the defects grow in the bearing. The amplitudes encountered that will have no established setpoint for alert and alarm purposes, but like all Spike Energy measurements, provide useful information as trended values. In these cases, the initial Spike Energy amplitude value may be quite small, such as 0.005, or 0.05 gSE. An increase to 0.008 or 0.08 may be significant. Monitoring the values in a trend will be required to establish a point at which bearing defects have developed to a level that require maintenance intervention.

Place the sensor as near the bearing as possible while still protecting the sensor from damage. It is also imperative that proper mounting techniques be followed as the fault vibration signals in these situations will virtually always be of very low amplitude. As stated earlier, the mechanical impacts tend to excite the mounted natural frequencies of the accelerometers, so the natural frequencies of the sensors must be considered. Typically, the highest frequency response and mounted natural frequency is the best choice for Spike Energy.

Notes:

Hazardous Area Applications

Hazardous locations are typically designated as such because of the presence, or possible presence, of a flammable substance. Such substances include flammable liquid, gas, or particulate (dust). The specific designations and associated criteria vary depending on the certifying authority or standard.

Area Classifications

Two common standards for designating hazardous areas are National Fire Protection Association (NFPA) and ATEX.

NFPA

In North America, the most widely used classification system is NFPA publication 70. Per NFPA, hazardous areas fall into the categories in Table 82.

Table 82 - NFPA

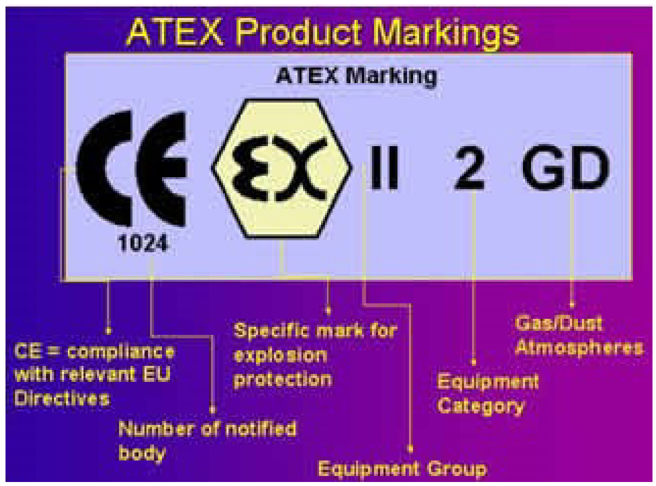
Class	Divisions	Groups	
Class I (flammable gases or vapors)	DIV 1 Gases and vapors under normal conditions.	Group A • Acetylene	
		Group B • Hydrogen • Butadiene • Ethylene Oxide • Propylene Oxide	
	DIV 2 Gases and vapors are normally confined but could be present under abnormal circumstances.	Group C • Ethylene • Ethyl Ether • Cyclopropane	
		Group D • Acetone • Benzene • Butane • Ethanol • Gasoline • Hexane • Methanol • Natural Gas	
Class II (combustible dust)	DIV 1 Dust exists under normal conditions.	Group E • Aluminum • Magnesium • Commercial Alloys	Group F • Coal • Carbon Black • Charcoal • Coke Dust
	DIV 2 Dust is normally confined but could be present under abnormal circumstances.	Group G • Flour • Grain • Wood • Plastic • Chemicals	
Class III (ignitable fibers and flyings)	DIV 1 (fibers are handled, manufactured used)	No specific groups in this category • Rayon • Cotton	
	DIV 2 (fibers are stored, handled)		

ATEX

Outside of North America, and particularly in Europe, the most commonly followed standard is ATEX. ATEX is a European directive much like OSHA in the United States. The directive, as it applies to hazardous locations and electrical installations, is ATEX 95, Equipment Directive 94/9/EC.

Flammable Vapors	Combustible Dust	Required Equipment Category
Zone 0	Zone 20	1
Zone 1	Zone 21	2
Zone 2	Zone 22	3

Equipment that is certified for each category has a label that identifies the zone, atmospheres, and so on. The equipment is certified to help protect.



Some of the certification, or notifying bodies are Sira, Intertek, Baseefa, Lloyds, and TÜV.

Using Safety Barriers

Intrinsic safety barriers are commonly used with monitor systems where various kinds of sensors must be placed in hazardous locations that are connected to monitor systems located outside the hazardous area. A barrier is an energy-limiting device that is used to limit the voltage and the current levels. This limit precludes the possibility of a spark at the sensor, which would risk igniting any surrounding flammable substance.

It is not the intent of this section to provide detail necessary to design a system architecture. Several factors are brought up here that provide a starting framework for what to consider in these situations.

Barriers are commonly used in petrochemical facilities, natural gas compression, and distribution facilities, agricultural extraction facilities using hexanes, and sewage treatment facilities with methane present. Barriers are also used around coal dust, in milling and grain handling facilities, near coating lines in metals and plastics where volatile coating materials are used, and any other facilities where flammable gasses, liquids, or dusts can be present.

The basic system configuration for the use of safety barriers requires the barrier be located electrically in the sensor signal cable. In addition, any sensor that is used must be certified for the area classification that it is used in.

Examples of intrinsically safe sensors are listed in Table 83.

Table 83 - Intrinsically Safe Sensors

Series	Cat. No.	Description
1443 Series Accelerometers	1443-ACC-IS-T	Hazardous Area Approved Accelerometer, Top Exit, 100 mV/g
	1443-ACC-IS-S	Hazardous Area Approved Accelerometer, Side Exit, 100 mV/g
1442 Series Eddy Current Probes(1)	1442-PS-0503M0010N	1442 Series NCPU Probe, 5 mm Tip, 150 mm Length Body, M8x1 Threads, 1.0 m Integral Cable
	1442-PS-0809M0005N	1442 Series NCPU Probe, 8 mm Tip, 90 mm Length Body, M10x1 Threads, 0.5 m Integral Cable

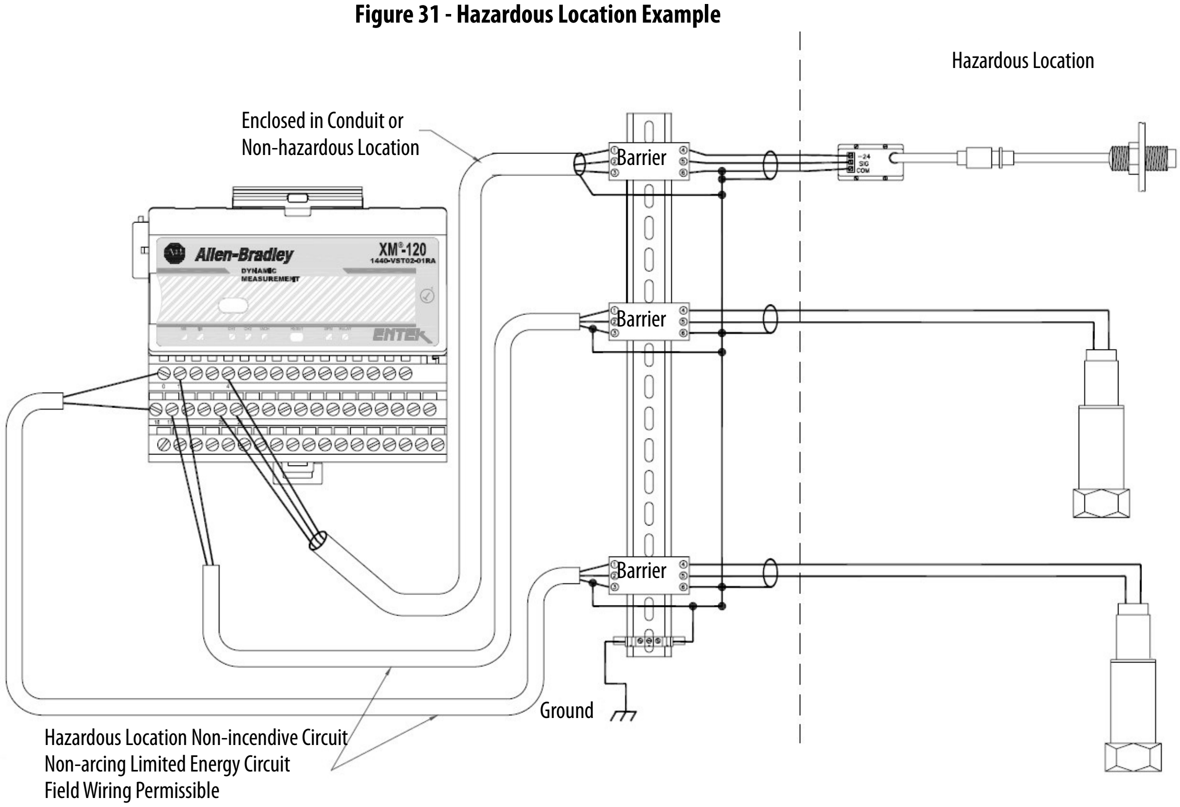
(1) All 1442 Series eddy current probes are also available from various other vendors.

Hazardous location certified accelerometers and eddy current probes are also available from various other vendors.

Accelerometers that are used for these areas have a safety certification rating. For instance, an ATEX rated sensor is approved for European use which is based on the ATEX directive, much like OSHA in the US. An FM rated sensor meets the requirements of FM approval, which is an independent testing arm of FM Global Insurance. A CSA approval meets the requirements of the Canadian Standards Association. Any of these approvals are adequate for specific installation, but it is also best to verify whether a specific standard is required and provide a sensor that is certified to that standard.

The second basic component of a safety barrier system is an approved barrier. The barrier is wired electrically into the sensor cabling that supplies power to the sensor and the return signal to the monitor. See Figure 31.

Figure 31 - Hazardous Location Example



Barriers that are typically used in these installations are supplied by MTL or Pepperl and Fuchs. The MTL part number for proximity probes is 7796-, and for accelerometers is 7728P+. The Pepperl and Fuchs part number is FFD2-VR4-Ex1.26 and this barrier works for both types of sensors. The function of the barrier is to limit both current and voltage. This limit exists so that sensor overheating and electrical sparks are not a risk if an electrical component fails in the sensor or monitor system. This precaution is typically taken with circuitry consisting of Zener diodes and resistors.

Approved Systems

Any approved installation must be implemented per applicable standards and local and regional codes.

In some cases, a system can be approved as a 'system' with approved drawings provided. Our XM® Series monitors, when used with certain 9000 Series accelerometers or 2100 Series eddy current probes, were approved this way. In these cases, detailed drawings must be provided and followed.

In most cases, monitors and sensors are approved as entity devices. This classification means that they can be used as part of an approved 'system' where the specific devices approvals and ratings satisfy the location requirements. However, in these cases the final assembled system is often approved by a local inspector.

Caution Over Cost

When evaluating an installation of a monitoring system in a hazardous area, always favor caution over cost. There are situations where a safety approved sensor with an approved monitor is an acceptable solution. If there is any doubt, assume that a barrier is required as the initial cost differential is negligible compared to the future risk. Safety systems are just that, a system. A safety system that includes not only the sensor and barrier but the wiring in some cases. The system also includes the enclosure if the monitoring system is in the hazardous area, the conduit, epoxy installation if sensor leads must go through a wall, and so on. The system must be deemed safe. The system must be designed by someone knowledgeable about hazardous locations and electrical installations.

Controller Based Alarm Detection

Dynamic Measurement Module

The dynamic measurement module allows you to define up to 23 different measurements from each dynamic channel (see table). In most applications, only a few of these, typically FFT Bands, need be defined to monitor and generally characterize the behavior of a machine. The module performs 92 unique measurements (23x4) but includes only 24 measurement alarms. If only FFT bands are applied as described in this guide the module allows definition of 32 of them (8x4), more than the module can alarm on. In many applications, alarm detection must be completed, in whole or part, in the controller.

Table 84 - Dynamic Measurement Module Measurements

Measurement	Description	Quantity Per Channel
FFT Band	Measures the overall energy, the maximum amplitude peak, or the frequency of the maximum amplitude peak, within a specified frequency or order range.	8
Order Magnitude	Measures the overall energy within a narrow pass band filter that follows the specified order, from 0.25*x* to 32*x* running speed.	4
Order Phase	Measures the phase of the signal within a narrow pass band filter that follows the specified integer order, from 1*x* to 32*x* running speed.	4
SMAX Magnitude	The greatest peak magnitude around the orbit.	1
SMAX Phase	The phase at which the greatest peak magnitude occurs around the orbit.	1
Overall	The total energy of the measurement. Two overalls are provided, one before application of any integration and high pass filter and one after.	2
Not 1*x*	Equals the overall measurement minus the order magnitude of a tracking filter that is configured to track the 1*x* running speed.	1
Shaft Absolute	An overall calculation that adds the absolute vibration, from a seismic sensor that is integrated to displacement, and a relative displacement measurement from an eddy current probe. Shaft Absolute is the amount of vibration relative to ‘free space’.	1
Bias/Gap	The sensor bias value or the gap measurement for the probe.	1

Ladder Logic

This example illustrates how to implement alarm detection in the controller with standard ladder logic that monitors the tag values that the monitor presents.

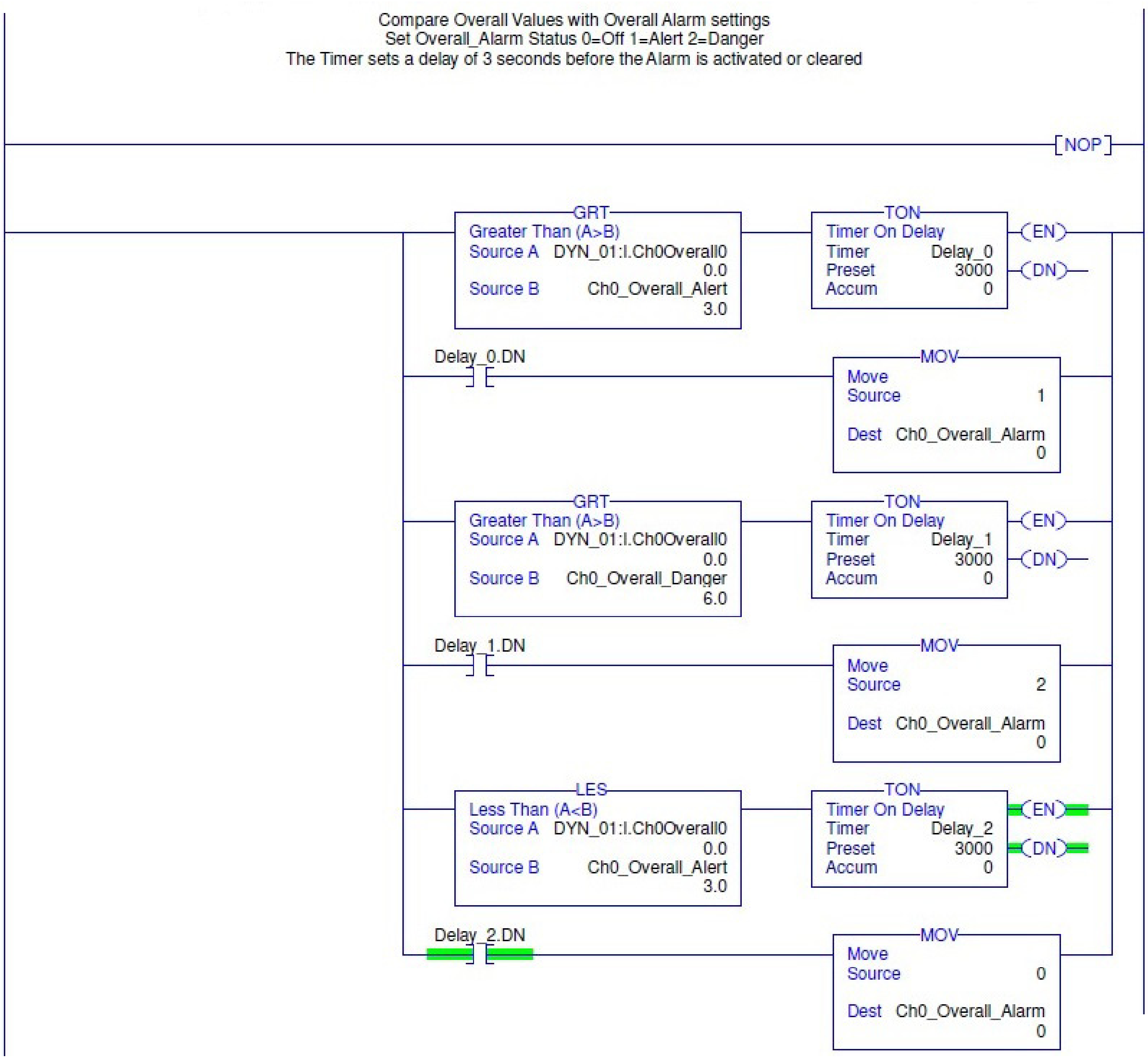
The provided example is for a common motor and pump with rolling element bearings.

The alarm alert and danger setpoints are stored as Tags in Logix.

Ch0_Overall_Alert	Local	3.0		Float	REAL		☐
Ch0_Overall_Danger	Local	6.0		Float	REAL		☐

The input tag values are compared against the alert and danger setpoint with a timer delay of three seconds to verify that the alarm state is not due to a momentary spike in the value.

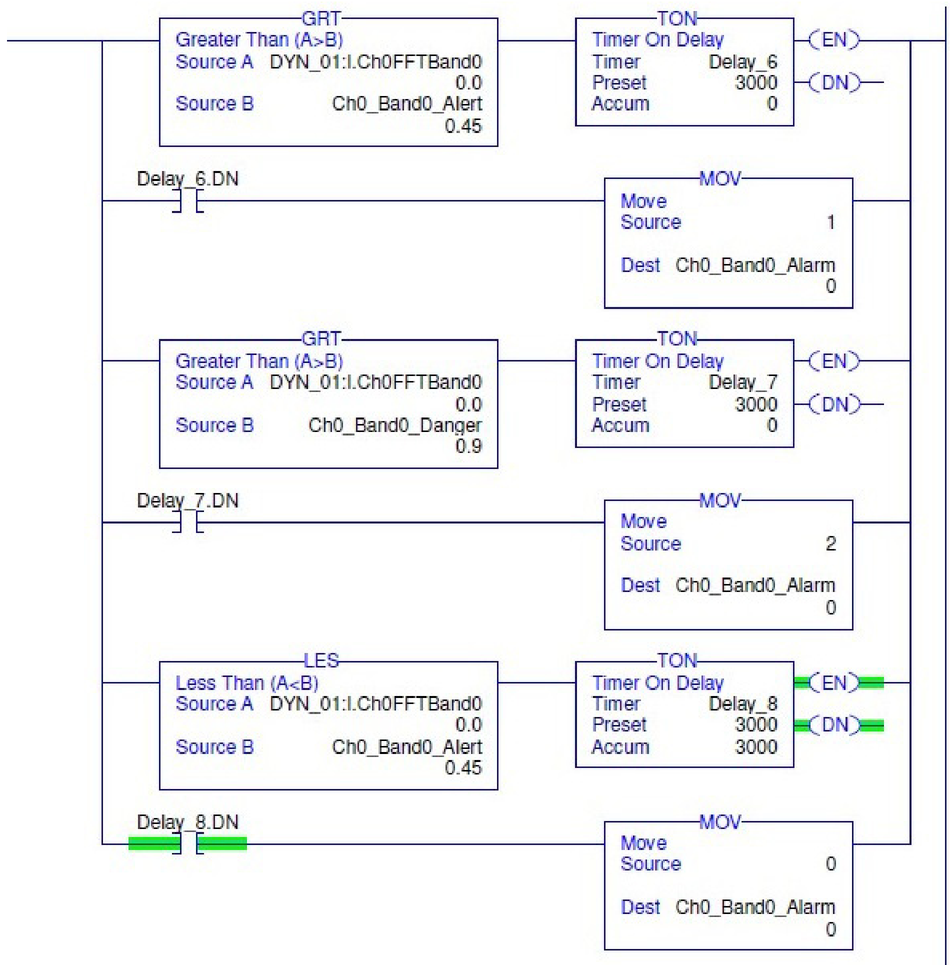
An alarm status tag, for example Ch0_Overall_Alarm, is set according to the alarm state, 0 = Normal, 1 = Alert and 2 = Danger.



The logic is repeated for overall (0) for each channel.

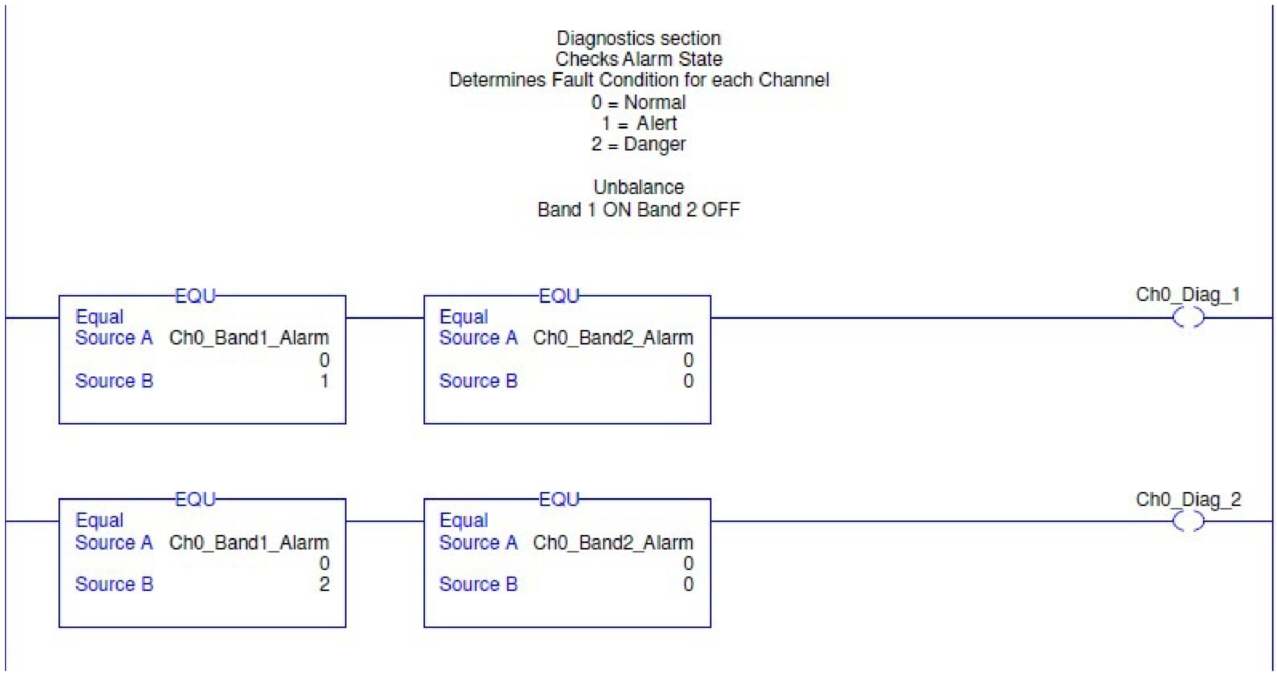
The individual band values are then compared against alarm setpoints created for each of the FFT Bands.

Ch0_Band0_Alert	Local	0.45		Float	REAL	
Ch0_Band0_Danger	Local	0.9		Float	REAL	



The logic is repeated for all FFT bands for each channel.

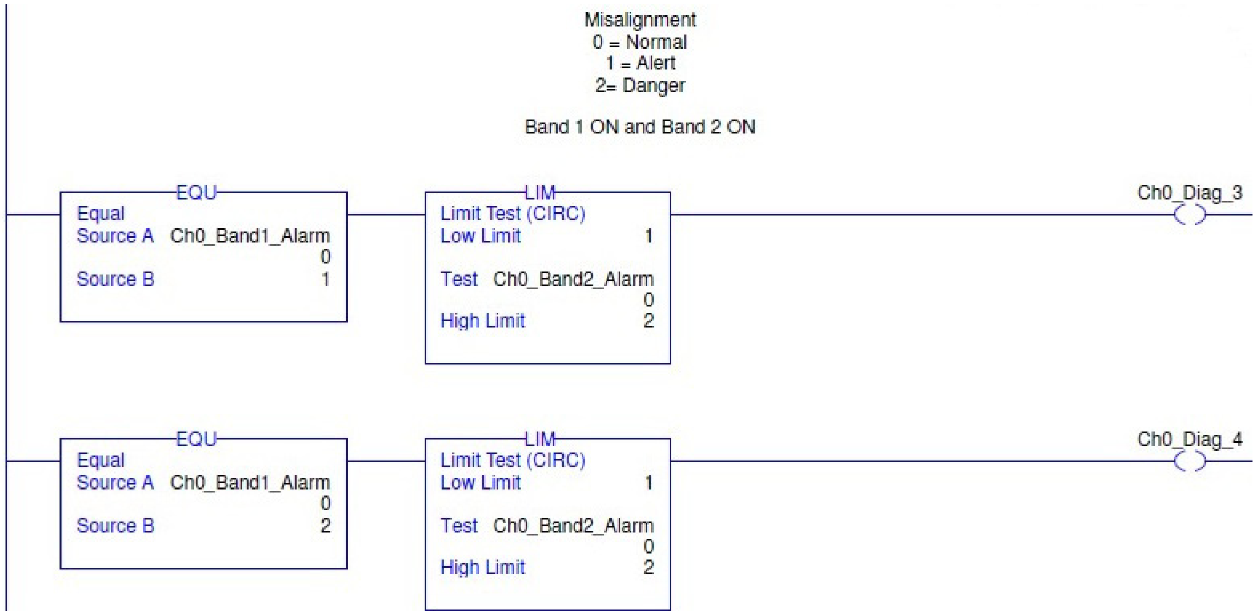
The status of the FFT band alarms is now used in diagnostic logic to determine if a particular fault is present.



In this example, Band 1 is set around the running speed frequency (indicates unbalance). Band 2 is set around twice running speed frequency (which would typically indicate misalignment).

The logic checks to see that only Band 1 is in an alarm state and sets either Ch0_Diag_1 for an alert state or Ch0_Diag_2 for a danger state unbalance condition.

The next section checks to see if both Band 1 and Band 2 are in alarm condition that signifies misalignment.



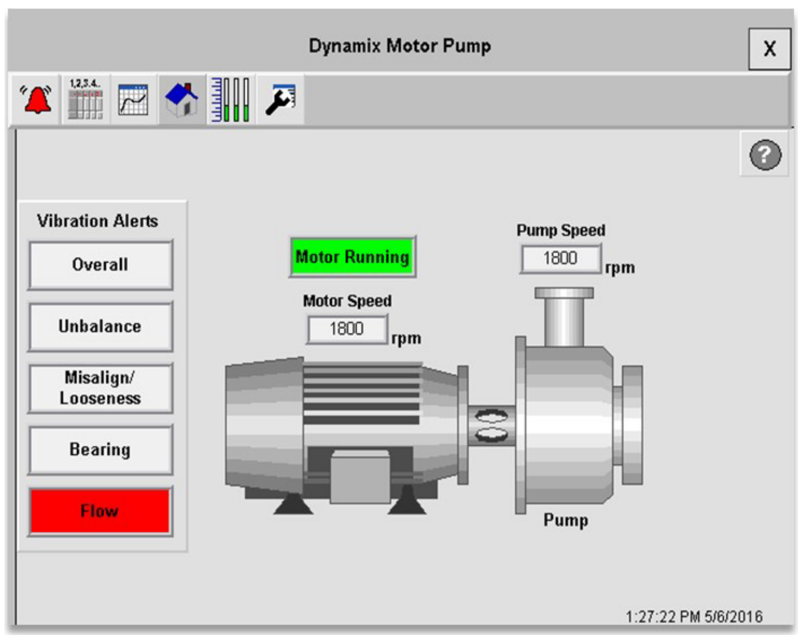
This Logic is repeated for various fault conditions, such as looseness, where Band 1, Band 2, and Band 3 must all be in an alarm condition.

Notes:

Dynamix Accelerator Toolkit

Accelerator Toolkit

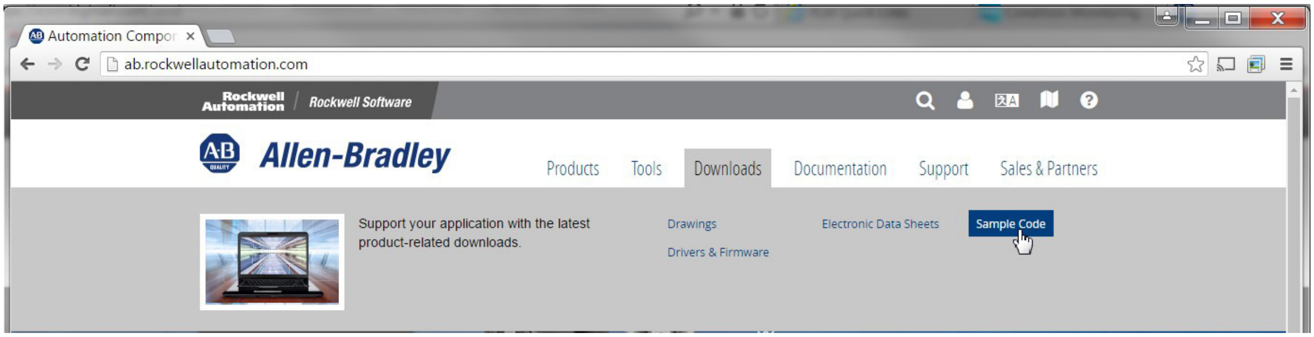
This chapter provides guidance on instrument, monitor, and information usage with the Logix controller. You are instructed on how to setup instruments and monitors. You are also shown how to produce tags that relay the machine condition to the Logix controller. Once the tags are sent, it is up to the user to provide appropriate visualization solutions. In some applications, management and presentation of tags is customized to blend into a larger supervisory system. In other cases, management and presentation of tags is applied uniquely within a dedicated visualization solution. Regardless of the level of integration necessary, Rockwell Automation® provides a set of examples. These examples are called the Dynamix™ Accelerator Toolkit. The toolkit can aid in the development of the required solution.



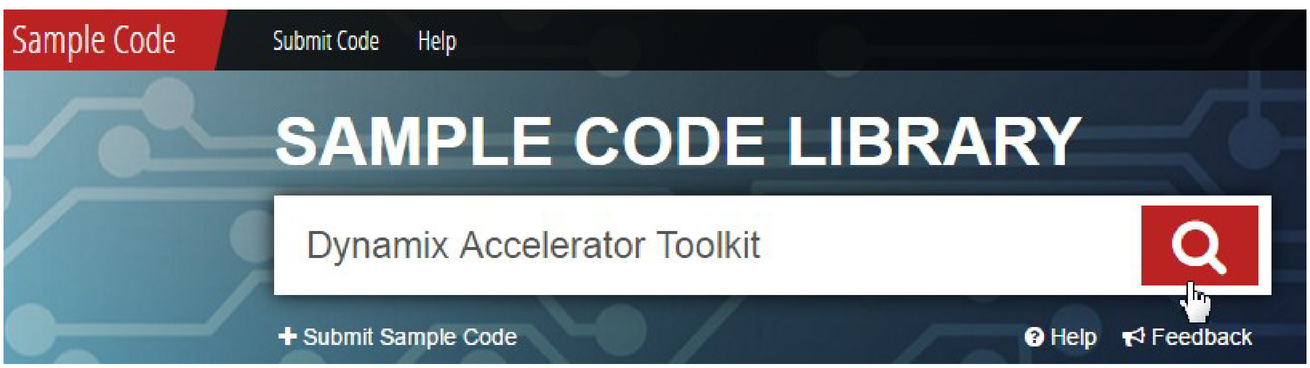
The Dynamix Accelerator Toolkit includes sample logic and HMI faceplates to integrate a Logix controller with a Dynamix 1444 Series Dynamic Measurement Module that uses a PanelView™ Plus for visualization. A quick start guide is included with instructions for using the toolkit. Also included is a sample drawing set with panel, power, and wiring diagram drawings for a typical Dynamix installation. This toolkit includes two example applications—a Motor Blower application and a Motor Pump application.

Downloads

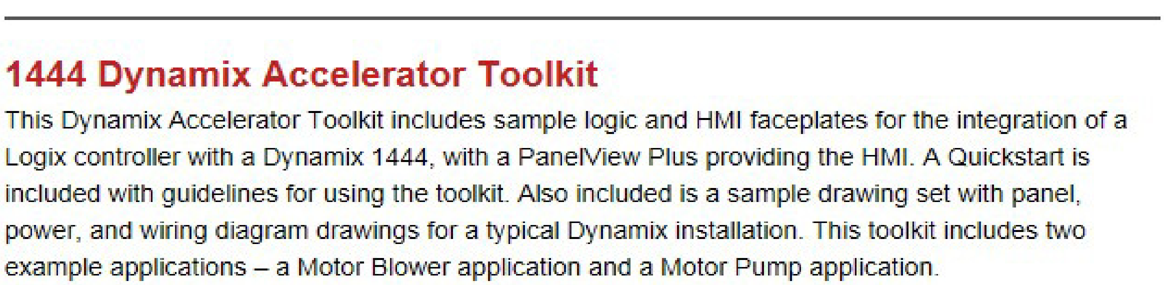
The toolkit is available on the Rockwell Automation Allen-Bradley® website. To download, go to www.ab.com, select Downloads, and then Sample Code.



From the Sample Code site, enter Dynamix or Dynamix Accelerator Toolkit.



In the results, find the entry for Dynamix Accelerator Toolkit.

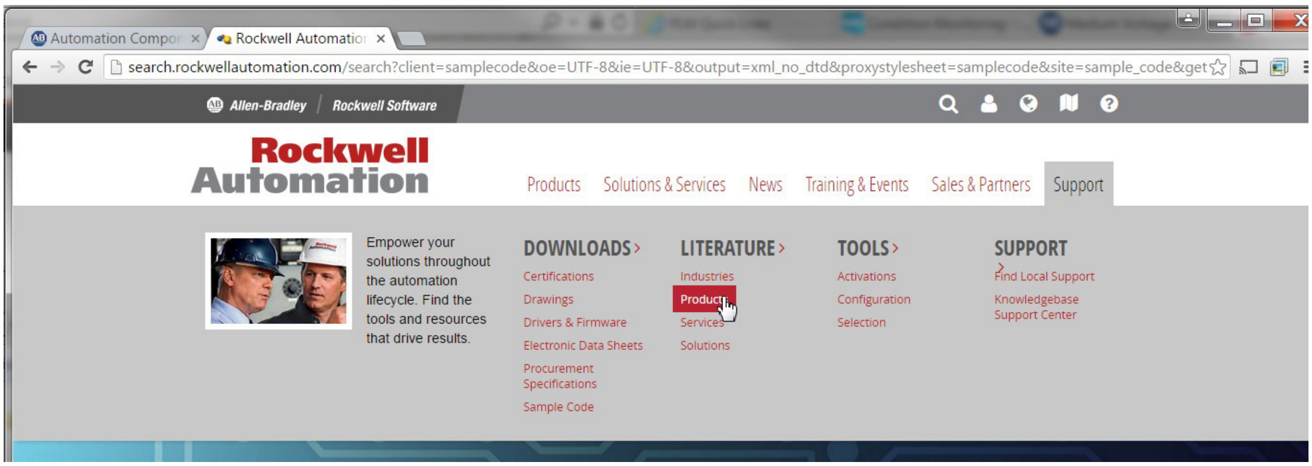


1444 Dynamix Accelerator Toolkit

This Dynamix Accelerator Toolkit includes sample logic and HMI faceplates for the integration of a Logix controller with a Dynamix 1444, with a PanelView Plus providing the HMI. A Quickstart is included with guidelines for using the toolkit. Also included is a sample drawing set with panel, power, and wiring diagram drawings for a typical Dynamix installation. This toolkit includes two example applications – a Motor Blower application and a Motor Pump application.

This toolkit link downloads a file that contains the kit and a Terms and Conditions document. After reviewing the terms and conditions, extract the container to access the kit materials, including the quick start guide.

Two documents, a product profile (1444-PP001) and data sheet (1444-TD001), are included in a Dynamix Publications folder. These documents represent the specific product capabilities that were available when the kit was published. For current product version and capabilities information, access the latest versions of these documents, and the Dynamix 1444 Series User Manual 1444-UM001, from the Rockwell Automation Literature Library.

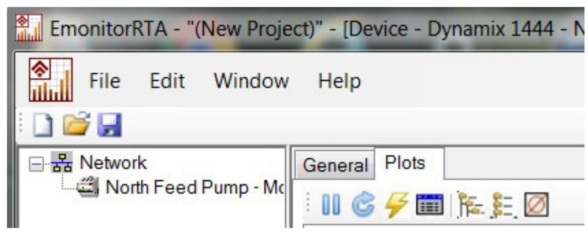


Notes:

CMS RTA Utility

CMS Real Time Analyzer (RTA)

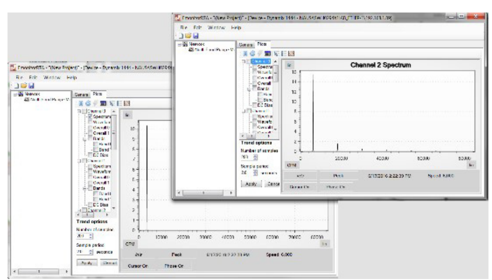
The CMS Real Time Analyzer (RTA) is a simple to use utility that allows spectrum viewing, time waveform, and trended parameters that are read directly from a Dynamix™ 1444 module. The utility is included in the Emonitor® CMS software product. The utility is not linked to the licensing requirements of Emonitor, which allows the tool to be deployed to any personal computer necessary. A separate RTA installation file is provided for this purpose.



Commissioning a 1444 Module Installation

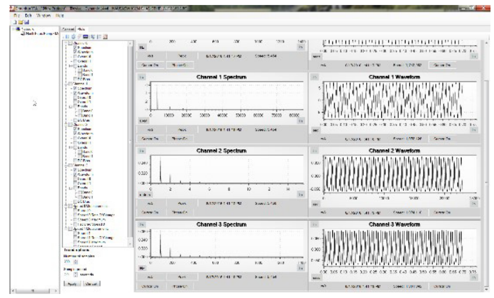
The tool is most useful as an aid in commissioning a 1444 installation.

During commissioning, the RTA can help perform an initial diagnosis of machine condition. RTA is used to identify vibration at frequencies that had not been anticipated or considered in the planned module configuration. The RTA is likely to provide help in adjustment or tuning the FFT bands to better suit the specific application.



In some cases, necessary details about machine components are not available at the time the module is configured. Details such as the number of vanes on a pump impeller or teeth on a gear are sometimes not available. In many cases, the RTA utility can be used to establish this kind of information.

The RTA can be used to verify that transducers are working as expected, that the signal wiring is correct, and that the module configuration is as expected.



After commissioning, the RTA allows you to observe machine condition in real time from anywhere if it can access a network.

When operator interfaces indicate problems with a machine, the first inclination is to examine the raw data, the spectrum, and time waveforms from the machine in real time. When serious problems arise and decisions have to be made immediately, this capability can be invaluable to analysts and reliability engineers by helping them make informed decisions that are based on what the machine is doing now.

Chapter 11

Vibration Relationships

You must know these three important characteristics when performing vibration measurement:

- Measurement type: Was it an acceleration, velocity, or displacement measurement?
- Signal detection: Was the measurement of the signals RMS, peak, or peak-to-peak value?
- Peak detection method: If signal detection is other than RMS, was it a true peak (or peak-to-peak) measurement or was the value calculated from the RMS value?

This section describes these characteristics and the mathematical relationships between them.

There are many online tools available that convert values between acceleration, velocity, or displacement. When using any of these tools, confirm that the source, target value units, and signal detection are considered.

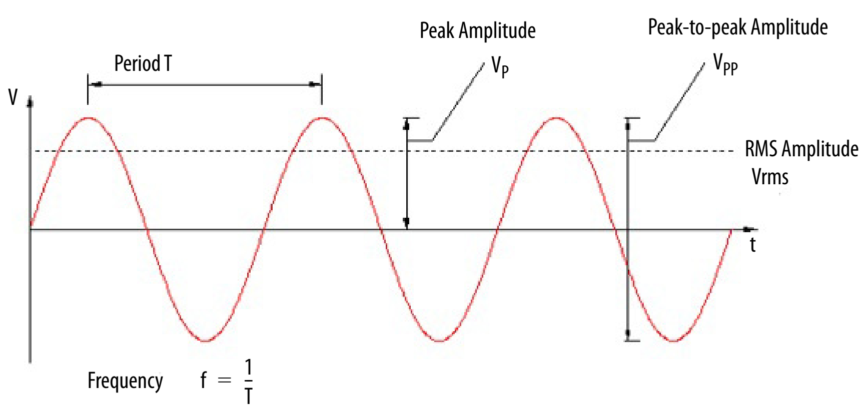
Signal Detection

$$\text{RMS} = 0.707 \times \text{peak}$$

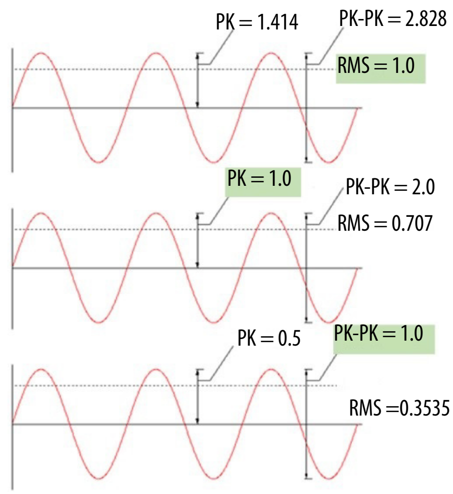
$$\text{Peak} = 1.414 \times \text{rms}$$

$$\text{Peak-to-peak} = 2 \times \text{peak}$$

$$\text{Crest Factor} = \frac{\text{peak}}{\text{rms}}$$



When measuring a sinusoidal signal, the measurement can be referenced to the RMS, peak, or peak-to-peak of the signal. If the signal is a pure sine-wave, then the mathematical relationships between them, as illustrated in the preceding signal detection graph, can be used to convert one to another. To understand what a measurement means, relative to signal magnitude, it is important to know the signal detection used.



For example, for a signal with a measured magnitude of '1.0', the RMS value could be 0.3535, 0.707 or 1.0, depending on the signal detection that was referenced for the measured value.

True vs. Scaled Peak

When measuring a signal, its magnitude can either be measured directly or inferred from the waveforms RMS value. Inferring a peak or peak-to-peak value from the RMS of a time waveform means that the magnitude is scaled from the RMS.(1) It is multiplied by 1.414 for peak or 2.828 for peak-to-peak.

If a signal is a pure sinusoid, then the scaled peak or peak-to-peak is identical to the value if it were measured directly. However, the greater the waveform varies from a pure sinusoid, the greater the error in any scaled peak or peak-to-peak value. How important this possible error is depends on the intent of the measurement. If the intent is to measure the actual deflection or distance that the signal indicates, the importance of the error can be significant. If the intent is to measure the energy within the signal, the RMS is the best indicator of this error, and a scaled (calculated) peak or peak-to-peak value is fine.

In condition-monitoring applications, it is usually the energy that the signal represents that is important. An example is the area under the curve. The energy or the RMS value, is representative of the sum of all vibration frequencies that comprise the waveform. Whereas the true peak or peak-to-peak value is a measure of the maximum deflection of the waveform, irrespective of the vibration frequencies that comprise it.

If a displacement measurement is being made, which is indicative of the physical movement of a shaft, then the measurement of interest is the TRUE peak or peak-to-peak value. When measuring displacement, contact between spinning and stationary parts can incur substantial damage if the shaft displacement was allowed to exceed a known limit.

(1) Scaled is also referred to as calculated. So calculated peak, or calculated peak-to-peak means the same thing as scaled peak or scaled peak to peak.

Engineering Units

Listed in Table 85 are the typical units for each domain and the usual (default) signal detection that is applied to each. There are other units that are used occasionally, such as meters or inches for length measures, but the table includes the units that are commonly used in condition monitoring.

Table 85 - Typical Engineering Units

Domain	Units	Comments	Metric/Imperial	Signal Detection
Acceleration	mm/s^2	Occasionally used	Metric	RMS
	$inch/s^2$	Rarely used	Imperial	Scaled peak
	g	Most common form	Metric	RMS
			Imperial	Scaled peak
Length	mm	Occasionally used	Metric	True peak-to-peak
	micron	Most common form	Metric	True peak-to-peak
	mil	Commonly used 1 mil = $1,000^{th}$ of an inch (0.001')	Imperial	True peak-to-peak
Velocity	mm/s	Most common form	Metric	RMS
	inch/s	Most common form	Imperial	Scaled peak
Bearing Anomaly Units	gSE	Units of a unique enveloped signal processing algorithm	Metric	RMS
			Imperial	Scaled peak

Integrated Measurements

Vibration measures are routinely integrated (converted) from acceleration to velocity or from velocity to displacement. It is also possible to differentiate measurements; from displacement to velocity or from velocity to acceleration. When digitally integrating or differentiating measures, a key assumption is that the signal is a pure sinusoid. The more complex or less sinusoidal a signal is, the greater the error in the conversion.

In most cases relative to machine diagnostics where the fundamental (1x) vibration is dominant, any integration error is small and can be discounted. However, when measuring impact events, signal noise and other non-cyclic signals the amount of error from integrating measurements can be significant. This error is not always a problem as the intent of the measure is to provide indication of change. For example, a trended parameter where change or rate of change provides information that is required rather than the absolute value itself.

Relationships (for Sinusoidal Signals)

Imperial

$V = \pi f D$	• D = displacement in inches peak to peak • V = velocity in inches per second • a = acceleration in (inches per second)^2 • g = acceleration in gravity (g's) • f = frequency in Hz
$V = \frac{61.48g}{f}$	
$a = 0.102f^2D$	
$g = 0.0511f^2D$	
$g = 0.016266Vf$	
$g = \left(386.09 \times \frac{in}{secs}\right)^2$	
$D = \frac{0.3183V}{f}$	
$D = \frac{19.57g}{f^2}$	

Metric

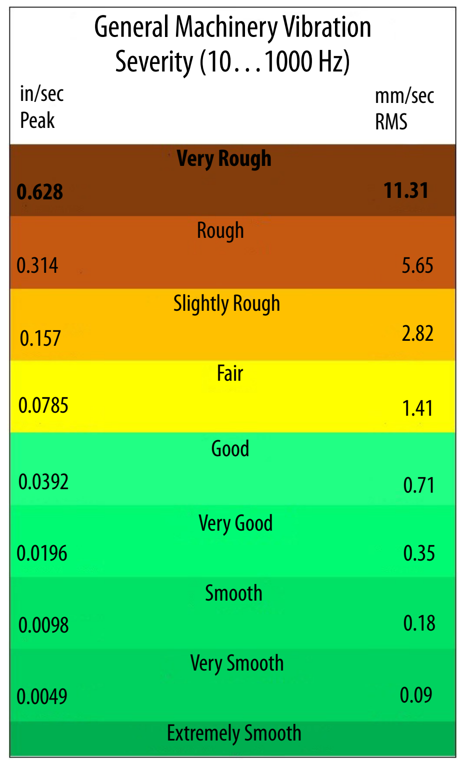
$V = \pi f D$	• D = displacement in meters peak to peak • V = velocity in meters per second • a = acceleration in (meters per second)^2 • g = acceleration in gravity (g's) • f = frequency in Hz
$V = \frac{1.56g}{f}$	
$a = 4.026f^2D$	
$g = 2.013f^2D$	
$g = 0.641Vf$	
$g = \left(9.80665 \times \frac{m}{secs}\right)^2$	
$D = \frac{0.3183V}{f}$	
$D = \frac{0.4968g}{f^2}$	

Vibration Severity

General Machinery Vibration Severity

The vibration alarm settings throughout this document are intended as a guideline only. These settings are not absolute vibration limits for a given machine type and are not intended to be used as acceptance criteria for new machine condition.

Settings are based on experience with machines of the given type[1] that have been in normal operation for some time.



The alarm points need adjustment after a period of normal operation. For instance, normal operation can be determined by using principles of statistical analysis. Determine the standard deviation by storing trend information over a period of months and using average values. See Statistical Alarms.

Further guidance regarding vibration severity can be found in the chart to the right.

Although these guidelines are useful as an aid to determine the machine condition initially, it is the change in vibration over time that is of most importance. The ability to trend vibration values over time is likely to give the most successful results.

A doubling of the vibration results in movement from one severity region to the next. An increase in severity from 2.82 mm/sec RMS (a Slightly Rough condition) to 5.65 mm/sec (a Rough condition) elevates you to the next region of the chart. If the vibration doubles from one measurement to the next, a significant change in machine condition has occurred and requires further investigation.

(1) While identical machines present the same spectral components, the magnitudes of these machines can vary significantly between machines. These differences are due to differences in structure or foundation, differences in process, differences in wear or other reasons. Consequently, in many cases only experience and observation can discern what is normal, or alert/alarm, for a machine.

When trying to establish acceptable levels of vibration for new machinery, consider Vibration Standards such as ISO 10816 for Bearing Casing measurements or ISO 7919 for Shaft vibration measurements.

Statistical Alarms

Statistical alarms can be used to provide levels of severity in machinery with trended data available. This data, with a number of samples that are collected over several months, is best if also collected during any variations of operation of the machine such as speed, load, product variances, and so on. We recommend that you use statistical alarms both for overall alarms and for band alarm values. The purpose behind statistical alarms is to account for individual vibration amplitudes and signatures that use trended actual data. Just as two identical machines can have vibration characteristics that vary under identical conditions, machines vary as to the amount of change in vibration amplitude during operating conditions that vary. These variations, while normal, can cause false alarms if two machines have the same alarm setpoints. A machine can experience little change in vibration amplitude that is based on loading and an identical machine can vary considerably more. If the setpoint is established for the smoother machine, the rougher machine, although still normal, can be in alarm.

Software is readily available to calculate statistical alarms. Most vibration analysis packages can arrive at statistical alarms. The statistical test that is performed is called an analysis of variance.

The basic procedure is as follows.

Find the arithmetic average of each of the data samples.

$$X_{AVE} = \frac{X_1 + X_2 + X_3 + X_4 \ldots X_N}{N}$$

Next, subtract the mean value, X_{AVE}, from each data point value and square each result. Add these results, divide by the total number of samples minus 1, and then find the square root of the result to determine the Standard Deviation of the data.

$$\sqrt{\frac{(X_1 - X_{AVG})^2 + (X_2 - X_{AVG})^2 \ldots (X_N - X_{AVG})^2}{N - 1}}$$

These calculations consider the variance of the data over time. The basic alarming function then is to set the alert value at two standard deviations above the mean, and the danger value at three standard deviations above the mean. This setting can be weighted if there is reason to believe that the result is a nuisance alarm by adding a percentage to alarm values, such as 10% higher, or even 25…50% higher with worn machines. Take care not to set the alarms too high as undesired results can occur.

Over time, experience is likely to allow fine-tuning of these alarm values.
Until then, application of statistical alarms helps verify meaningful alarm limits are applied.

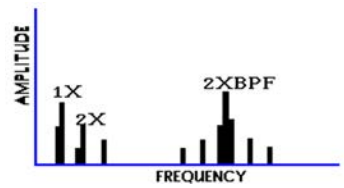
Chapter 13

Vibration Signature Analysis

This section attempts to describe the rules and techniques in applying vibration analysis to resolve common faults in industrial machinery.

Application of these rules requires a few key, common, steps:

- Identify the frequencies of peaks in the vibration spectrum
- Determine relationships between peaks in this spectrum, such as order multiples or harmonic relationships
- Consider what faults could cause vibration at these frequencies



- Confirm the indication of considered faults by determining if other vibration indicators, such as peaks at other frequencies or in other measurement locations, are present

IMPORTANT	Read and understand the following information regarding rules in this chapter.

The rules simply consider if vibration is present at the prescribed frequency or order, when measured in the prescribed orientation (radial vertical, radial horizontal, axial). They do not consider the availability of the measurement, or any particulars of how the measurement is made.

Because of costs, online systems are not likely to include sensors in every possible orientation, at every bearing. An online system might also be limited in the variability of its measurements, particularly when the monitoring system must also serve protection or quality applications.

Consequently, an online monitoring solution may not provide or process out all of the data values required for the applicable rules.

In addition to the required measures, rules require applying logic such as ‘if A and B then C’. While a Dynamix™ module can provide the measures, in most cases it cannot apply the logic(1).

(1) The Dynamix monitors alarming system allows logical comparisons of up to four inputs. So it may be possible, in some cases, to use the alarm system to perform the required logic

Rolling Element Bearing Faults

The following faults could apply to any machine fitted with rolling element bearings. Note that other potential faults will be associated with the specific type of machine, its foundation and structure, and its connected components.

Bearing anomalies create forces that manifest at four specific frequencies as shown in the following table.

Table 86 - Rolling Element Bearing Faults

Possible Problems	Frequency (Order)	Primary Direction
Ball Damage	1 *x* BSF—Harmonics of BSF	Radial
Inner Race Anomaly	1 *x* BPFI—Harmonics of PBFI	Radial
Outer Race Anomaly	1 *x* BPFO—Harmonics of PBFO	Radial
Cage Anomaly	1 *x* FTF—Harmonics of FTF	Radial

When present, the above faults will create forces at the frequencies resolved from the calculations in the following table. The table presents two formula:

- The Specific Frequency Formula provides the math necessary to determine the exact fault frequency
- The Estimated Frequency Formula provides a method of calculating the fault frequencies when the specific bearing dimensions are not known. These formula assume that the fundamental train frequency (FTF) is between 0.33x and 0.50x times running speed, which is true for the majority of rolling element bearings

Table 87 - Fault Frequency Formulas

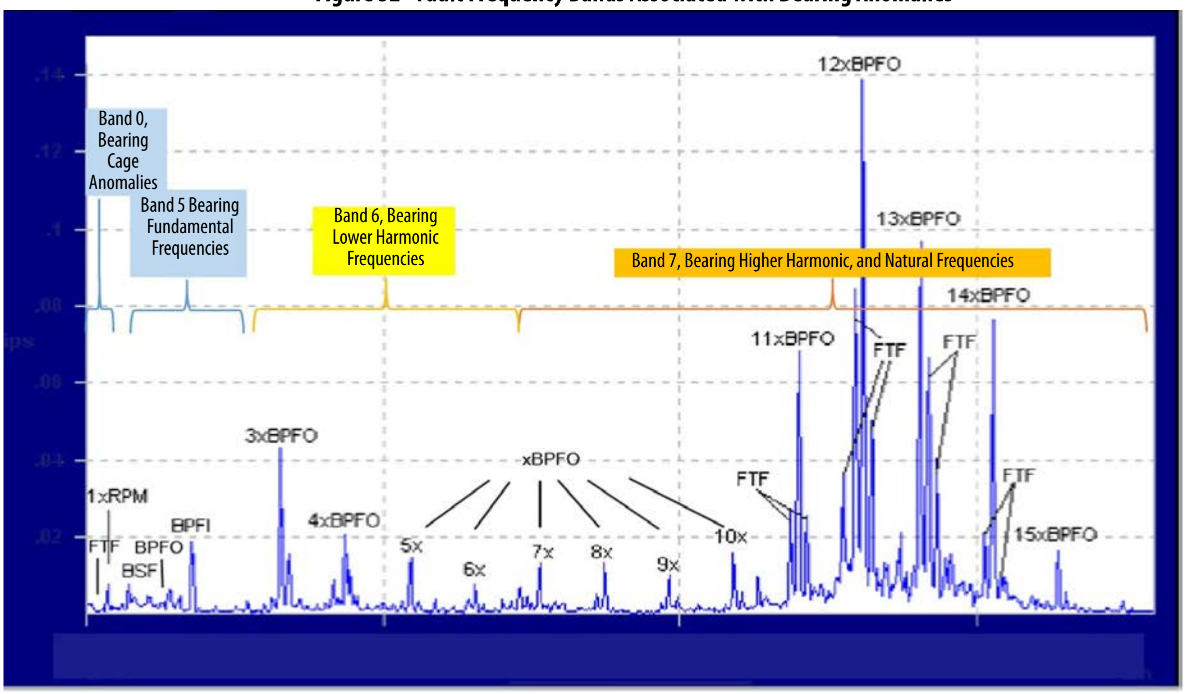
Fault Frequency	Specific Frequency Formula	Estimated Frequency Formula	
Ball Pass Frequency Outer Race	$BPFO = \frac{FN}{2}\left\{1 + \frac{N}{D}\cos\phi\right\}$	$BPFO = F \times 0.45 \times N$	> 2 x F, < 15 x F
Ball Pass Frequency Inner Race	$BPFI = \frac{FN}{2}\left\{1 + \frac{N}{D}\cos\phi\right\}$	$BPFI = F \times R \times N$	> 4 x F, < 15 x F
Fundamental Train Frequency	$FTF = \frac{F}{2}\left\{1 - \left(\frac{d}{D}\cos\phi\right)^2\right\}$	$FTF = F \times 0.4$	< 0.5 x F
Ball Spin Frequency	$BSF = \frac{D}{2d}\left\{1 - \left(\frac{d}{D}\cos\phi\right)^2\right.$	$BSF = F \times 3.5$	> 5 x F, < 15 x F
Definitions	Φ = Contact angle: The angle of load from the radial plane R = A factor that can range from 0.55…0.6 F = Running speed frequency, in Hz, is equal $\frac{RPM}{60}$		D = Pitch diameter d = ball diameter N = Number of balls

Rolling Element Bearing Fault Progression

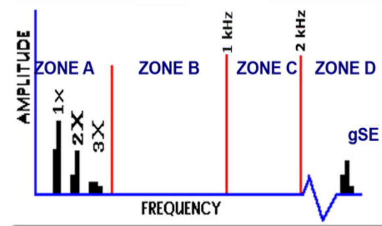
When faults occur in rolling element bearings the progression of that fault, from its initial instantiation to failure, usually follows five stages of detectable propagation. In the recommended configurations these stages, except for the 1st stage, are monitored using four FFT Bands.

Attribute	Band 0	…	Band 5	Band 6	Band 7
Band Name	<1*x* Bearing Cage Anomalies	…	Bearing fundamental frequencies	Bearing lower harmonic frequencies	Bearing higher harmonic and natural frequencies
Frequency Range	0.2…0.8*x*	…	1.2 VPF…12.2*x*	12.2*x*…50% FMAX	50%…100% FMAX

Figure 32 - Fault Frequency Bands Associated with Bearing Anomalies



Stage 1: Elevated Spike Energy



The earliest indication of bearing fault will be an elevated spike energy measurement, which can be far in advance of other indications.

- An elevated Spike Energy measurement may also indicate a lubrication problem. See FTF Frequency on page 163
- In most cases a Bearing Fault will be indicated by a Spike Energy measure as low as 0.25 gSE Overall

The Dynamix 1444-DYN04-01RA can measure spike energy, but doing so requires certain channel application restrictions. It cannot measure gSE while also performing normal acceleration or velocity measures[1]. The gSE measurements are not included in the recommended configurations due to these complexities.

While when using Spike Energy a fault may be detected 2 to 6 months[2] ahead of failure, a bearing fault can be detected when it reaches stage 2, described below, at typically 1 to 2 months[2] ahead of failure.

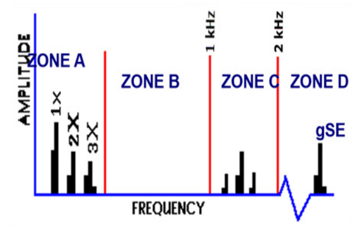
(1) This will change beginning with version 5.0 Firmware which is expected to be available in early 2021. Contact Rockwell Automation® for availability.

(2) How soon before failure a fault is detected depends on more than just the presence of the fault indicators in the data. It also depends on; how fast the fault propagates—a function of bearing load, temperature, lubrication and use of the machine; the complexity of the data other than from bearing fault indication—other signals within the monitored frequency ranges, such as blade pass or gear mesh frequencies; and the tolerance for risk that the operator is willing to accept—fault propagation is not liner, it will typically propagate exponentially near end of life, so deferring maintenance will require some risk.

Stage 2: Excited Bearing Natural Frequencies

Each of the components in a rolling element bearing will have one or more natural frequencies. These are the frequencies that a structure will 'ring' at when a force, such as an impact, is applied. As it rotates, a spall on a bearing, even before it is visible, will create an impact force when it contacts the inner or outer race, which will excite these natural frequencies.

Bearing natural frequencies are typically between 1 kHz(1) and 2 kHz. Consequently, when vibration elevates in this range, it can be an early indication of a bearing fault. However, depending on the machine, there may be other possible causes of vibration in this range, such as gear mesh, that should be discounted before determining that the vibration is being caused by a bearing fault.



- Slight anomalies begin to ring bearing component natural frequencies
- These frequencies occur in the range of 1…2 kHz
- At the end of Stage 2, sideband frequencies appear above and below natural frequency
- Spike Energy grows, for example, 0.25…0.50 gSE

What to Look For

Sample profiles for machines with rolling element bearings specify the last FFT band to monitor the ZONE C frequency range, '*Bearing Higher Harmonics and Natural Frequencies*'.

Attribute	Band 0	…	Band 5	Band 6	Band 7
Band Name	<1*x* Bearing Cage Anomalies	…	Bearing fundamental frequencies	Bearing lower harmonic frequencies	Bearing higher harmonic and natural frequencies
Frequency Range	0.2…0.8*x*	…	1.2 VPF…12.2*x*	12.2*x*…50% FMAX	50%…100% FMAX

In most cases, the recommended FMAX is about 2000 Hz, so 50% of FMAX will be about 1 kHz. If on a given machine there are no discrete vibration signals at frequencies between 800 and 1000 Hz, then consider lowering the Band 7 beginning frequency to 800 Hz.

Where to Look

Indicators of anomalies in rolling element bearings are likely to be present in measurements made in the radial direction on the bearing.

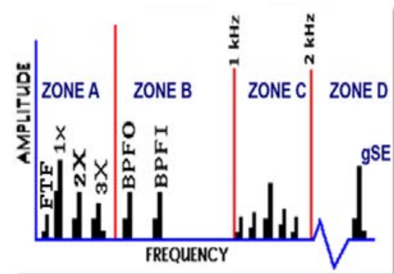
When two bearings are mounted closely together, or when the sensor is not mounted directly on a bearing, consider that the vibration observed may not be coming from the expected bearing.

(1) The natural frequencies of some bearings may be as low as 500 Hz.

Stage 3: Discrete Fault Frequencies

The next indication will typically be vibration at the associated bearing fault frequency; one or more of the BPFO, BPFI, BSF or FTF frequencies. When vibration is present at these specific frequencies there is almost no doubt that a bearing fault is present. The key here is to observe the specific frequency that the vibration is occurring at, and to identify the exact fault frequency as described above.

- Bearing anomalies frequencies and harmonics appear
- Many anomalies frequency harmonics appear with wear the number of sidebands grow
- Wear is now visible and can extend around the periphery of the bearing
- Spike Energy increases to between 0.5…1.0 gSE



What to Look For

Most of the fault indicators are at frequencies above 1x RPM but one fault indicator, the Fundamental Train Frequency (FTF), is almost always below the 1x RPM frequency. Consequently the recommended configurations specify two bands to monitor the frequency ranges applicable to a bearings Fundamental Fault Frequencies.

Attribute	Band 0	…	Band 5	Band 6	Band 7
Band Name	<1*x* Bearing Cage Anomalies	…	Bearing fundamental frequencies	Bearing lower harmonic frequencies	Bearing higher harmonic and natural frequencies
Frequency Range	0.2…0.8*x*	…	1.2 VPF…12.2*x*	12.2*x*…50% FMAX	50%…100% FMAX

Where to Look

The first band measures vibration less than 1x RPM, where the Fundamental Train Frequency is, and a second band monitors a frequency range above 1x RPM to detect the other fault frequency indicators.

Vibration in either of these bands when present along with the natural frequency indication (band 7), generally indicates the next stage of bearing fault propagation. However, if only the FTF frequency is elevated, there may be another explanation.

FTF Frequency

Vibration at the Fundamental Train Frequency, the 'cage' frequency, is unique in that this often indicates poor lubrication rather than a fault. If the lubrication is known to be adequate and the vibration persists, a fault is likely.

The sample profiles apply a band specifically to monitor for indication of the FTF indication.

Attribute	Band 0	…	Band 5	Band 6	Band 7
Band Name	<1*x* Bearing Cage Anomalies	…	Bearing fundamental frequencies	Bearing lower harmonic frequencies	Bearing higher harmonic and natural frequencies
Frequency Range	0.2…0.8*x*	…	1.2 VPF…12.2*x*	12.2*x*…50% FMAX	50%…100% FMAX

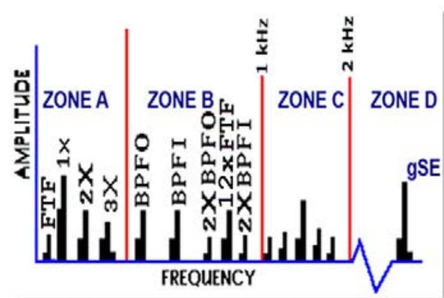
Where to Look

Indicators of anomalies in rolling element bearings are likely to be present in measurements made in the radial direction on the bearing.

Stage 4: Discrete Fault Frequencies and Harmonics

The next indication will typically be vibration at harmonics of one or more of the BPFO, BPFI, BSF or FTF frequencies. When vibration is present at harmonics of these frequencies the bearing fault is pronounced and the bearing is likely close to failure.

While there is a later stage of fault propagation, as described in Stage 5 below, by that point the bearing is essentially failed and continued operation would likely result in collateral damage and more expensive repairs. Therefore a bearing should be replaced as soon as possible after having propagated to the '*Fault Frequencies & Harmonics*' stage.



- Harmonics of bearing anomalies frequencies appear
- Sidebands of anomalies frequencies will appear around the natural frequencies and possibly around harmonics of other fault frequencies (example 12xBFFO ± 1xFTF)
- Wear is now visible and can extend around the periphery of the bearing
- Spike Energy increases to between 0.5...1.0 gSE

What to Look For

Vibration in this range is measured by the band labeled, 'Bearing Lower Harmonic Frequencies'.

Attribute	Band 0	…	Band 5	Band 6	Band 7
Band Name	<1*x* Bearing Cage Anomalies	…	Bearing fundamental frequencies	Bearing lower harmonic frequencies	Bearing higher harmonic and natural frequencies
Frequency Range	0.2…0.8*x*	…	1.2 VPF…12.2*x*	12.2*x*…50% FMAX	50%…100% FMAX

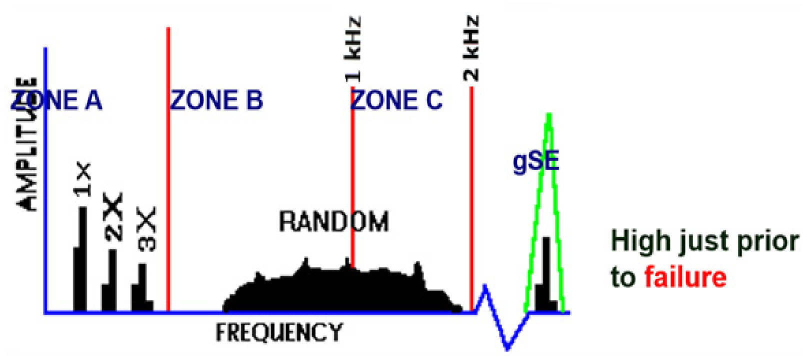
Where to Look

Indicators of anomalies in rolling element bearings will be present in measurements made in the radial direction on the bearing.

Stage 5: Failure Imminent

When the fault is close to failure the discrete frequencies will disappear, typically replaced by broad band noise as the impacting will excite natural frequencies throughout the structure. Eventually the bearing will fail to hold the shaft in its centered position, which may result in unrestrained imbalance forces. At any point in this stage the bearing could seize or simply come apart.

- Discreet bearing anomaly frequencies disappear and are replaced by random broad band vibration in the form of a noise floor
- Towards the end, even the amplitude at 1x RPM is effected as it will likely increase, possibly significantly
- High frequency noise floor amplitudes and Spike Energy may decrease
- Just prior to failure gSE may rise to high levels



What to Look For

The recommended configurations do not provide any further specific measurements to detect this stage of propagation. However, if a bearing were to reach this stage the measures defined in the recommended configurations would change. In most cases all of the magnitudes measured by the bands defined to detect bearing faults would likely drop although it's possible that one or more of them could increase. When this occurs the changes will likely occur very quickly.

Attribute	Band 0	…	Band 5	Band 6	Band 7
Band Name	<1*x* Bearing Cage Anomalies	…	Bearing fundamental frequencies	Bearing lower harmonic frequencies	Bearing higher harmonic and natural frequencies
Frequency Range	0.2…0.8*x*	…	1.2 VPF…12.2*x*	12.2*x*…50% FMAX	50%…100% FMAX

Where to Look

Indicators of anomalies in rolling element bearings will be present in measurements made in the radial direction on the bearing.

Fluid Film and Sleeve Bearing Faults

The faults listed below are associated with the condition of the bearing. These faults are common to any sleeve or fluid film bearing and have little to do with the type of machine or service to which it is applied.

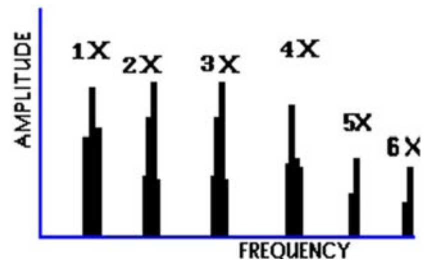
Sleeve Bearing Wear/Clearance Problems

A plain, or sleeve, bearing is the simplest form of bearing. It is essentially when a shaft rides on an oiled surface or bushing. Bushings can be made of bronze or other metals, metal composites, even plastic. They can be lubricated using a pressurized oil feed, they can run dry (not oiled), or they can be self-lubricated.

The most common fault is simply wear or clearance problems.

Symptoms

- Later stages of sleeve bearing wear are likely to give a large family of harmonics of running speed
- A minor unbalance or misalignment is likely to cause high amplitudes when excessive bearing clearances are present



Another problem, loose assembly, can provide similar indication except that it is likely to produce harmonics that begin with 0.5x.

What to Look For

The sample profiles, for use with fluid film bearings, define FFT Bands for monitoring 1, 2 and 3 *x* vibration.

Attribute	…	Band 1	…	Band 3	…	Band 5
Band Name	…	1*x* Unbalance	…	2*x* Misalignment/Looseness	…	3*x* Misalignment/Looseness
Frequency Range	…	0.8…1.2*x*	…	1.8…2.2*x*	…	2.8…3.2*x*

If this fault is present then each of these bands will elevate. Other bands may elevate as well, particularly any that include multiples of running speed, such as bands defined to monitor blade or vane pass on fans and pumps.

Where to Look

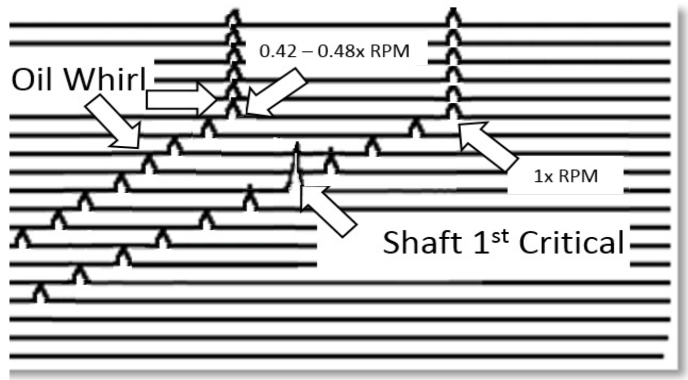
Indicators of this anomaly will be present in measurements made in the radial direction on the bearing.

When monitoring with eddy current probes the indicators of this condition should be present in both the X and Y directions. However, it is possible that it will be much more pronounced in one direction versus the other.

Fluid Film Bearing Oil Whirl

In hydrodynamic journal bearings, the oil within the bearing flows around the bearing in the direction of shaft spin, at a speed slightly less the 0.5x the shaft RPM. Normally the shaft rides on an oil wedge that forms at a slightly off-vertical, stable angle and location that is eccentric relative to the center of the bearing.

When an upset occurs the shaft can lift from the wedge, which allows pressurized oil to fill the gap. When this occurs the pressure of the wedge increases and forces the shaft to 'whirl' around the bearing with the oil. How serious this is will depend on parameters such as the load on the bearing and the dampening in the system. When serious the resulting vibration can be very violent.



Symptoms

- Usually occurs at 40...48% of running speed
- Vibration amplitudes are sometimes severe

What to Look For

The recommended configurations, for use with fluid film bearings, define one FFT Bands to monitor the frequencies that are indicative of oil whirl.

Attribute	**…**	**Band 1**
Band Name	…	1*x* Oil Whirl/Whip/Rub
Frequency Range	…	0.2…0.8x

In many machines, besides Oil Whirl, a rub condition can also create vibration within the frequencies that are monitored by this band, at 0.5 *x* running speed. Consequently, when oil whirl is suspected further investigation can be performed to verify that the frequency of the vibration is within the frequency range indicative of whirl (0.40...0.48*x*).

Where to Look

Indicators of this anomaly are likely to be present in measurements made in the radial direction on the bearing.

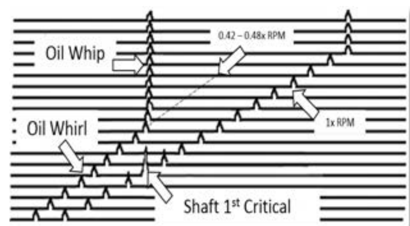
When monitoring with eddy current probes the indicators of this condition can be present in both the X and Y directions. However it is possible that it is likely to be much more pronounced in one direction versus the other.

Fluid Film Bearing Oil Whip

Oil Whip occurs when oil whirl is present and its frequency becomes coincident with the machines critical speed (natural frequency). After that the Oil Whip is likely to become 'locked' at the natural frequency, irrespective of machine speed.

Symptoms

- Oil whirl is present when machine speed is less than 2x the rotor's 1st critical (resonance) frequency
- Significant vibration is present at the rotor's 1st critical frequency when the machine speed is at or above 2x the rotor's 1st critical frequency



What to Look For

The recommended configurations, for use with fluid film bearings, define one FFT Bands to monitor the frequencies that are indicative of oil whirl and oil whip.

Attribute	…	Band 1
Band Name	…	<1*x* Oil Whirl/Oil Whip
Frequency Range	…	0.2…0.8*x*

When the shaft speed is <2x its critical (resonance) frequency an elevated measure in this band may indicate whirl. If the elevated vibration in the band continues, or more likely increases, once the speed exceeds 2x the shaft critical frequency then oil whip may be indicated.

In many machines, besides Oil Whip, a rub condition can also create vibration within the frequencies monitored by this band, at 0.5x running speed. Consequently, when oil whip is suspected further investigation should be performed to verify that the frequency of the vibration is coincident with the shaft resonance frequency (critical).

Where to Look

Indicators of this anomaly are likely to be present in measurements made in the radial direction on the bearing.

When monitoring with eddy current probes the indicators of this condition can be present in both the X and Y directions. However it is possible that it is likely to be much more pronounced in one direction vs. the other.

Rotor Shaft Faults

Unbalance, misalignment, rubs and the other faults listed below are associated with the condition of the rotor shaft. These faults are common to any rotor and have little to do with the type of machine or service to which it is applied.

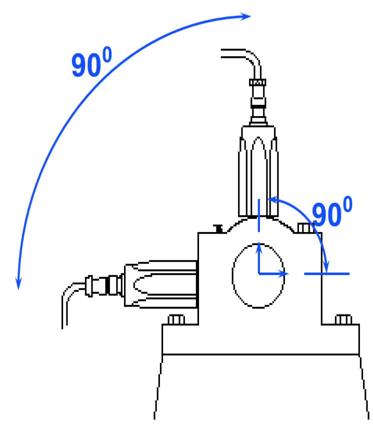
Phase Measurements

In many cases diagnosing faults associated with a shaft requires considering the effects of the fault on phase measurements. However, none of the recommended configurations are defined to measure phase. This is because measuring phase requires a tachometer signal input to the module, which requires the machine to have a speed sensor, and requires (usually) the monitor system include a Tachometer Signal Conditioner Expansion Module (1444-TSCX02-02RB). Because the monitoring systems for most common machines do not include a tachometer input, the recommended configurations do not assume one.

However, if the monitoring system does include a tachometer input, consider modifying the configuration to include phase measurements(1).

Unbalance

Unbalance can present in either of two ways, overhung or coupled. The difference is easier visualized. Consider two scenarios:



- A shaft supported by bearings on either end and fitted with bladed wheels or other masses, has an imbalance at some location other than dead-center.

  In this case, called 'coupled' unbalance, an imbalance nearer to one bearing than the other would obviously cause the shaft to, in a manner, wobble where one end would move differently from the other.
- An overhung shaft, supported only on one end, and that has an imbalance some distance from the supporting bearing.

  Since the mass relative to the bearing is consistent, shaft motion is likely to be consistent at every point. While it would move more at further distances from the bearing, it would always move in the same direction.

Notes to Consider

- It is possible for a change in balance to reduce the magnitude of vibration at 1 *x* RPM.

(1) Measuring phase requires definition of a tracking filter at the order of interest, typically 1x RPM

Consider an event where mass is lost at a location on a wheel that is near to the current heavy spot—where the current imbalance mass is. Or where mass is gained at a location opposite the existing heavy spot.

In these cases, the unbalance force would be lessened, but an event did occur. Regardless of any change in magnitude though, a change in mass distribution will cause a change in the phase angle. Monitor the 1x Phase value to detect these events.

- While unbalance can be the result of a change in mass, such as fouled or damaged blades on a fan or expander turbine, it can also be the final result of some other fault.
  - A failed rolling element bearing is likely to result in a, possibly dramatic, increase in unbalance.
  - A broken rotor bar on a motor is likely to create significant imbalance.

  Almost any fault, if left to go to failure, is likely to result in severe unbalance. The key is to identify and resolve faults before that happens.

Diagnosing unbalance generally follows these rules:
- Vibration frequency equals rotor speed
- Vibration predominantly radial in direction
- Stable vibration phase measurement
- Vibration increases as square of speed
- Vibration phase shifts in direct proportion to measurement direction

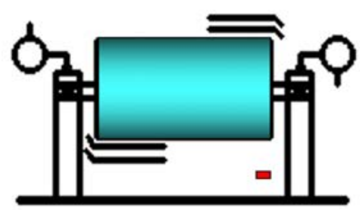
Symptoms

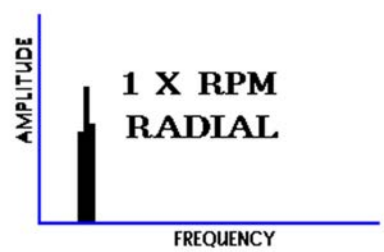
If phase measurements are available, then consider the further details regards coupled and overhung unbalance.

1. Coupled unbalance

   Couple unbalance is a condition that exists when the mass centerline axis intersects the shaft centerline axis at the center of gravity of the rotor. Equal heavy spots at each end of a rotor create a couple 180° opposite each other. Significant couple unbalance can introduce severe instability to the rotor that causes it to wobble back and forth like a 'seesaw' with the fulcrum at the rotor center of gravity.

   Couple unbalance exhibits each of the following characteristics:
   - 1*x* phase at each end of the shaft is 180° out one to the other
   - 1*x* RPM magnitude is always present and normally dominates
   - Amplitude varies with square of increased speed
   - Can cause high axial and radial amplitudes
   - Balance requires correction in two planes at 180°





2. Overhung Unbalance

   An overhung rotor is one that has the driven rotor that is placed outboard of the bearings. Overhung rotors can cause interesting vibration symptoms and can be difficult to balance.

   This configuration is often found in machines such as fans, blowers, and pumps. Because the planes where the balance correction weights are to be attached are outside the supporting bearings, these rotors are likely not to respond to normal single plane and two plane balancing techniques.

   Overhung rotors can cause high axial vibration at 1 x RPM.

   A large couple unbalance can be generated and the force unbalance.

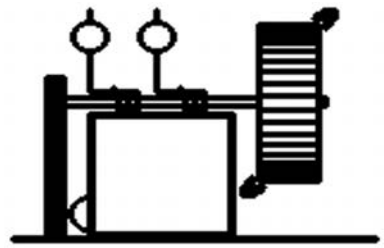
   For pure unbalance of an overhung rotor, the axial phase at bearing 1 is likely to approximately equal that at bearing 2 (±30°)

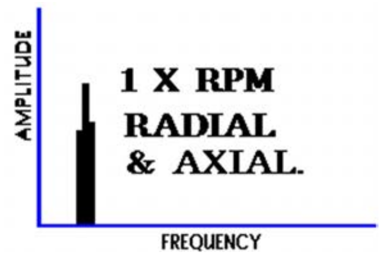
   Normally overhung rotor unbalance can be corrected by taking care of the force unbalance component first, which is likely to leave the remainder as couple unbalances with phase differences that approach 180°.

Because the unbalance planes are outside the support bearings, even a static unbalance is likely to cause a couple unbalances which is likely to be proportional to the distance of the unbalance plane from the rotor center of gravity.

Rules:

- 1 x RPM present in radial and axial directions
- Axial readings tend to be in-phase but radial readings can be unsteady
- Overhung rotors often have both force and couple unbalances each of which can require correction





What to Look For

The recommended configurations, regardless of bearing type, define one FFT Band to monitor the frequency associated with the vibration at the running speed of the machine, 1x RPM.

However, there are other faults that would cause high 1x vibration. So to resolve unbalance users should consider:

Attribute	…	Band 1
Band Name	…	1*x* Unbalance
Frequency Range	…	0.8x…1.2x

- On a two bearing supported shaft, so '**Coupled Unbalance**' mode, the unbalance forces can elevate 1 x RPM vibration at all radial measurements on both bearings

  While the magnitude of vibration, or the amount of change, at each measurement location can vary, even significantly, due to structure, unbalance is likely to be felt and is likely to be detectable at both bearings

- On an overhung shaft, so '**Overhung Unbalance**' mode, unbalance can elevate 1 x RPM vibration at all radial measurements on the supporting bearing

  For an overhung rotor, it can be important to measure the vibration in the **axial** direction and radial

  If an axial measurement is available, then unbalance can be resolved by high 1 x RPM vibration in both the axial and radial directions

- Because no shaft is perfectly balance, some amount of vibration at 1 x RPM is always present. So unbalance requires a bit more logic:
  - The vibration at 1 x RPM can comprise 90% or more of the total, overall, vibration
  - The 1 x RPM vibration magnitude, or the overall vibration magnitude, must exceed alarm thresholds applicable to the machine

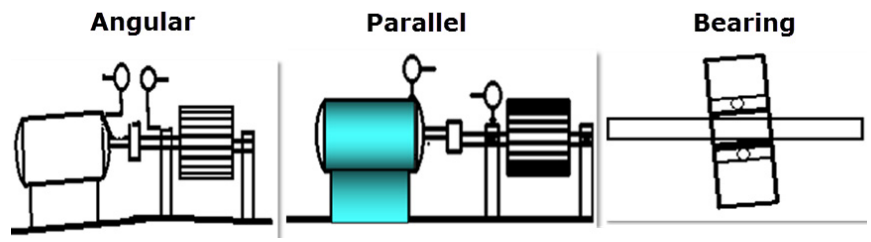
Where to Look

Indicators of this anomaly are likely to be present in measurements that are made in the radial direction on the bearing and, if Overhung Imbalance, in the axial direction.

When monitoring with accelerometers, differences in the stiffness of the structure can cause a significantly greater unbalance force and indication in the least stiff direction, usually horizontal.

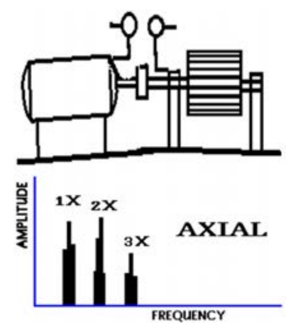
Misalignment

There are several ways misalignment can occur between two shafts including angular, parallel and, if fitted with rolling element bearing, the bearing could be misaligned with the shaft.

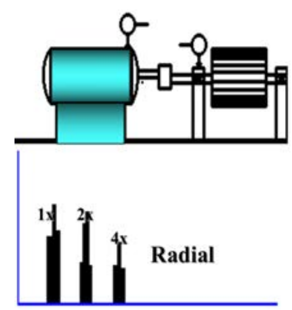


Symptoms

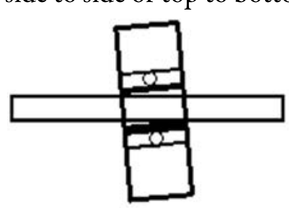
If phase measurements are available, consider further the details in regards to misalignment.

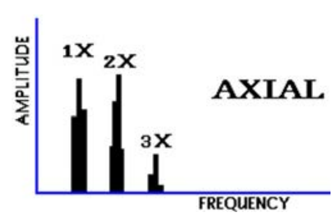


1. Angular Misalignment
   - Characterized by high axial vibration
   - 180° phase change across the coupling
   - Typically high 1 and 2 *x* axial vibration
   - Not unusual for 1, 2, or 3 *x* RPM to dominate
   - Symptoms could indicate coupling problems
2. Parallel Misalignment
   - High radial vibration 180° out of phase
   - Severe conditions give higher harmonics
   - 2*x* RPM often larger than 1 *x* RPM
   - Similar symptoms to angular misalignment
   - Coupling design can influence spectrum shape and amplitude



3. Bearing Misalignment
   - Vibration symptoms similar to angular misalignment
   - Attempts to realign coupling or balance rotor is likely not to alleviate the problem
   - Is likely to cause a twisting motion with approximately 180° phase shift side to side or top to bottom





What to Look For

The sample profiles, regardless of bearing type, define two FFT Bands to monitor the frequencies indicative of misalignment.

Attribute	…	Band 3	…	Band 5
Band Name	…	2X Misalignment/Looseness	…	3X Misalignment/Looseness
Frequency Range	…	1.8…2.2x	…	2.8…3.2x

There are other faults that would cause high 2 and 3 *x* vibration such as Looseness. To resolve misalignment, consider:

- In all modes of misalignment the key indicator is vibration at 2 *x* RPM, with magnitudes near or possibly even in excess of the vibration magnitude at 1 *x* RPM
- A high 2 *x* RPM in the axial direction, if measured, is a strong indicator of angular misalignment
- Some 3 *x* vibration can be present if the misalignment is angular, or if a misaligned bearing, but 3 *x* RPM vibration is not common if parallel misalignment

Where to Look

Indicators of this anomaly are likely to be present in measurements that are made in the radial direction on the bearing and, if Angular Misalignment, in the axial direction.

A key point of misalignment, other than Bearing Misalignment, is that indicators of it is likely to be present in measurements at bearings on both sides of the coupling.

If monitoring a fluid film bearing, the indicators of this condition can be present in both the X and Y directions. However, it is possible that the vibration in one direction is likely to be much more pronounced than the other.

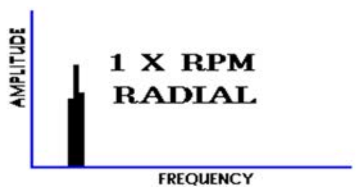
Looseness

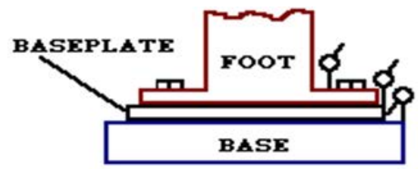
Like misalignment, eccentricity, unbalance, and others, there are several conditions that can be defined as 'Looseness' and not all of them cause similar forces (vibration).

Symptoms

If phase the measurements are available then consider these further details regarding looseness.

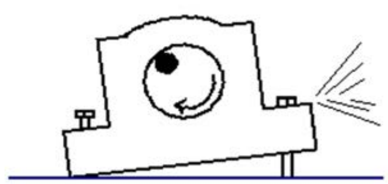
1. Loose Structural Mounting (Soft Foot)
    - Caused by structural looseness of machine feet
    - Distortion of the base is likely to cause 'soft foot' problems
    - Phase analysis is likely to reveal approximately 180° phase shift in the vertical direction between the baseplate components of the machine

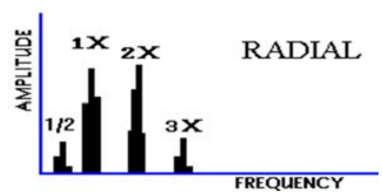




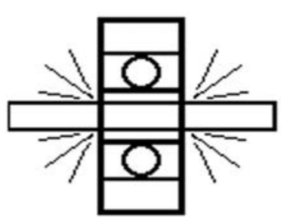
Note that this mode of looseness cannot be diagnosed by the recommended configurations.

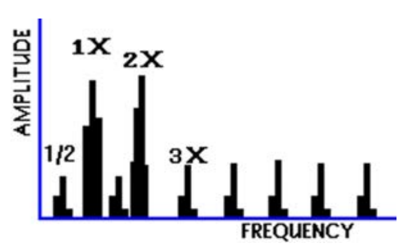
2. Loose Pillow Block
    - Caused by loose pillow-block bolts
    - Can cause 0.5x, 1x, 2x, and 3x RPM
    - Sometimes caused by cracked frame structure or bearing block





3. Mechanical Looseness
    - Phase is often unstable
    - Will have many harmonics
    - Can be caused by a loose bearing liner, excessive bearing clearance, or a loose impeller on a shaft





What to Look For

The sample profiles, regardless of bearing type, define three FFT Bands to monitor the frequencies indicative of looseness.

Table 88 - Machine Fitted with Rolling Element Bearings

Attribute	Band 0	…	Band 3	…	Band 5
Band Name	<1*x* Bearing Cage Anomalies	…	2*x* Misalignment/ Looseness	…	3*x* Misalignment/ Looseness
Frequency Range	0.2…0.8x	…	1.8…2.2*x*	…	2.8…3.2*x*

Table 89 - Machine Fitted with Fluid Film Bearings

Attribute	Band 0	…	Band 3	…	Band 5
Band Name	<1*x* Oil Whirl/Oil Whip	…	2*x* Misalignment/Looseness	…	3*x* Misalignment/Looseness
Frequency Range	0.2…0.8*x*	…	1.8…2.2*x*	…	2.8…3.2*x*

However, there are other faults that would cause high 2x and 3x vibration, such as misalignment. So to resolve looseness consider:

- Looseness is likely to force vibration at harmonics that start at ½x RPM, which is monitored by band 0 in the sample profiles.
- Depending on the type of looseness there could be many more harmonics, including ½ order harmonics
- In general, if there is abnormal (alert/alarm level) vibration at ½x, 2x and 3x, then the problem is looseness.

Where to Look

Indicators of this anomaly are likely to be present in measurements made in the radial direction on the bearing.

Looseness would likely be observed in measurements at any location on the same shaft, but not necessarily across a coupling.

When monitoring a fluid film bearing the indicators of this condition can be present in both the X and Y directions. However, it is possible that the vibration in one direction is likely to be much more pronounced than the other.

Bent Shaft

A bowed or bent shaft can generate excessive vibration in a machine, depending on the amount and location of the bend. Like eccentric shafts, the effects can decrease by balancing. However, more often than not, it is not possible to achieve a satisfactory balance in a shaft, which has any noticeable bend.

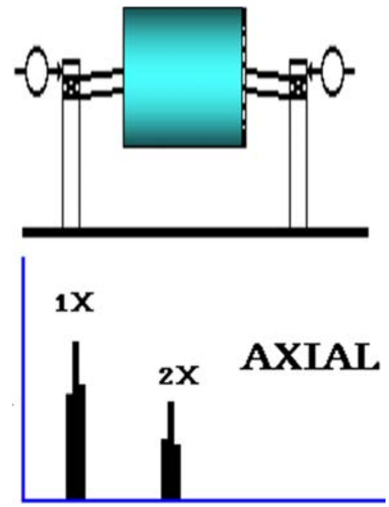
Symptoms

Bent shafts exhibit the following characteristics:

1. High axial vibration is generated by the rocking motion that is induced by the bent shaft. Dominant vibration normally is at 1 *x* RPM if bent near the shaft center, but a higher than normal 2 *x* RPM component can also be produced, particularly if bent near the coupling.
2. Axial phase change between two bearings on the same component (motor, fan, pump, and so on) is likely to approach 180º, dependent on the amount of the bend. In addition, if several measurements are made on the same bearing at various points in the axial direction, it is likely that phase differences, which approach 180º, occur between that measured on the left and right-hand side of the bearing, and also between the upper and lower sides of the same bearing.
3. Amplitudes of 1 *x* RPM and 2 *x* RPM is likely to be steady, assuming that 2 *x* RPM is not located close to twice line frequency (either 100 Hz or 120 Hz) which can induce a beat of the 2 *x* RPM component with 2 *x* line frequency if there is high electromagnetic vibration present.
4. If the shaft is bowed through or very near a bearing, you get a twisting motion by the bearing housing itself, which is likely to, result in significantly different phase readings on this bearing housing in the axial direction.
5. When much run-out is present at the rotating mass, it appears as unbalance. When run-out at coupling occurs, it appears as misalignment.
6. In bent shafts, amplitudes can vary with the square of speed and preload. If unbalance is more of a problem than bow, vibration is likely to decrease abruptly if operating below the first critical speed. However, if the rotor is brought above its first critical speed, unbalance amplitude is likely to change only a small amount, whereas if the dominant problem is a bent shaft, the amplitude is likely to drop significantly again as the speed is dropped towards the first critical speed.
7. If a rotor is located between bearings and can operate at or close to its fundamental natural frequency, it is likely to appear to be a 'bent' shaft and is likely to display these symptoms (see 'Resonant Vibration'). However, this is only temporary. When the machine is stopped or at another non-resonant speed, it will then 'straighten out'.
8. When electric motors have problems such as shorted lamination, they are likely to induce a bend thermally as the machine heats up, with the resultant vibration that is higher and higher as the rotor heats. This again is likely to introduce bent shaft symptoms (see 'Electrical Vibration'). In this case, the shaft again is likely to straighten when allowed to come back to the room temperature if the plastic limit of the shaft material has not been exceeded.

Rules:

- Bent shaft problems cause high axial vibration
- 1 *x* RPM dominant if bend is near shaft center
- 2 *x* RPM dominant if bend is near shaft ends
- Phase difference in the axial direction is likely to tend towards 180° difference



What to Look For

The recommended configurations, regardless of bearing type, define two FFT Bands to monitor the frequencies indicative of bowed or bent shaft.

Attribute	…	Band 1	…	Band 3
Band Name	…	1*x* Unbalance	…	2*x* Misalignment/Looseness
Frequency Range	…	0.8…1.2*x*	…	1.8…2.2*x*

There are other faults that would cause high 1 and 2 *x* vibration such as looseness. So to resolve a bowed or bent shaft consider:

- A bent shaft is likely to force significant 1 *x* RPM vibration in the radial direction, similar to unbalance. However…
- Unlike unbalance, a bent shaft is likely to produce significant 1 *x* RPM vibration in the axial direction
- Depending on the nature of the bow or bend, the problem can also product forces at 2x RPM

Where to Look

Indicators of this anomaly are likely to be present in measurements made in the radial direction on the bearings at either end.

A bent shaft is likely to present significant vibration in the axial direction.

A bent shaft would be observed in measurements at any location on the same shaft, but not necessarily across a coupling.

When monitoring a fluid film bearing the indicators of this condition can be present in both the X and Y directions. However, it is possible that the vibration in one direction is likely to be much more pronounced than the other.

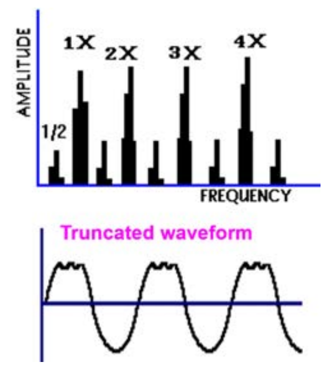
Rotor Rub

A rotor rub can occur in a radial direction at a seal, for example, or in the axial direction, due to uneven thermal growth between a turbine rotor and its casing. In any case, it is a rub, either through a complete shaft revolution, or just during a part of a revolution, between rotating and stationary components. Except in event of improper construction or assembly though, a rub is going to be caused by another problem; high vibration, unmanaged thermal growth, rotor bow, etc. So if a rub is indicated, then the corrective action is likely to be to determine what the cause is, and correct that.

A rotor rub is typically associated only with machines that are fitted with fluid film bearings.

Symptoms

- Similar spectrum to mechanical looseness
- Usually generates a series of frequencies, which can excite natural frequencies
- Sub harmonic frequencies can be present
- Rub can be partial or through a complete revolution



What to Look For

The recommended configurations, for machines fitted with fluid film bearings, define two FFT Bands to monitor the frequencies indicative of rotor rub.

Attribute	…	Band 2	…	Band 4
Band Name	…	1.5x Rub	…	2.5X Rub
Frequency Range	…	1.2…1.8x	…	2.2…2.8x

- A rub is likely to present vibration similarly to Looseness, but including ½x multiples
- Because of the many harmonics typically generated, there is a fair chance that one is likely to be coincident with a natural frequency, in which case there is likely to be a region of elevated vibration

Where to Look

Indicators of this anomaly are likely to be present in measurements that are made in the radial direction on one or more bearings, depending on where the rub is occurring.

If monitoring a fluid film bearing, the indicators of this condition can be present in both the X and Y directions. However, it is possible that the vibration in one direction is likely to be much more pronounced than the other.

Resonance, Natural Frequency, and Critical Speed

Any structure, or assembly of structures, is likely to have one or more natural frequencies. A resonance is an excited natural frequency where the system has locked on to the natural frequency. If a forcing function, a vibration, occurs at or near that same frequency it is likely to excite natural frequency into resonance.

Typical machine designs confirm that normal operating speeds, harmonics of machine speeds, vane pass frequencies, and other expected/normal forcing frequencies are well away from known natural frequencies. However:

- There can be natural frequencies other than those frequencies known to or specific to the machine. Typically these are the result of the specifics of the machine installation such as foundation and piping that are unique and external to the machine itself
- Natural frequencies are determined, in part, by the mass and stiffness of the system[(1)]. While the mass can be constant, the stiffness of a system can change as bearings wear, foundations loosen or degrade or due to changes in piping or other connected/supporting structures. In some cases even temperature changes can affect a natural frequency. As these characteristics change the natural frequency is likely to change, and possibly become coincident with a forcing frequency that had previously not been a concern

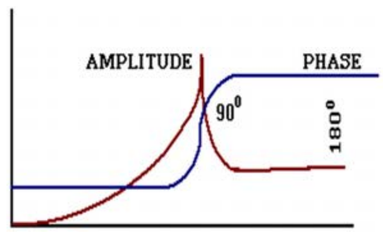
A critical is a natural frequency specific to a forcing vibration associated to the rotation of a rotor. A rotor critical differs from a common structural natural frequency because the stiffness of a rotor changes as it spins—the faster it spins the more stiff it becomes. A rotor critical then is the specific speed at which the natural frequency of the rotor becomes excited (goes into resonance). Rotor designs must consider its critical speed, and sometimes its 2nd critical speed, when the operating speed of the machine exceeds the critical speed as they have to account for the behavior (vibration) of the rotor as it accelerates through the critical speed.

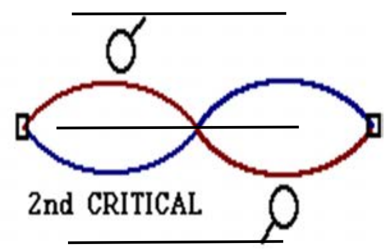
A resonance can be very destructive. Typical maintenance issues such as premature bearing failure, natural frequencies in resonance can cause fatigue, and cracking in machine structures that are not normally damaged by typical operation. How much damage depends on the force that is applied, the mass, stiffness, and dampening in the system and how sensitive the natural frequency is —how much it is likely to amplify the vibrating force.

(1) As a system becomes less stiff, the frequency of a natural is likely to move lower.

Symptoms

- Resonance occurs when the Forcing Frequency approximately coincides with a Natural Frequency
- 180° phase change occurs when shaft speed passes through its critical speed
- High amplitude of vibration is likely to be present when a system is in resonance





What to Look For

Because there is no association of the frequency that a resonance can occur at, and the mechanical attributes of the machine, or its speed, it isn't possible to define common rules for identifying a resonance.

If the monitored machine has a known natural frequency, an FFT Band can be defined to monitor it. This is particularly important if there is any possibility that a vibration could occur at that frequency.

If there is no reason to believe that a known natural frequency could be excited, no vibration could/can occur at that frequency, then the overall vibration can be carefully monitored. If the overall vibration increases without an associated increase in any of the monitored bands then the status of the natural frequency can be investigated—typically that means to review the FFT.

Where to Look

Indicators of this anomaly will be present in measurements made in the radial direction on each bearing. It likely will not be present across a coupling.

Electric Motor Faults

Vibration in electric motors occurs at the following frequencies:

Line Frequency (LF)	50 Hz (3000 cpm) or 60 Hz (3600 cpm) or variable if Variable Frequency Drive is used
Synchronous Speed (Ns)	$\frac{120 \times FL}{\text{Number of Poles}}$
Slip Frequency (SF)	$\frac{FL - \text{Rotor RPM}}{60}$
Running Speed(SF)	$Ns - 60 \times SF$
Rotor Bar Pass Frequency ((RBPF)	$\frac{\text{Number of Rotor Bars x } \textit{Rotor RPM}}{60}$
Running Speed(SF)	$\text{Number of Poles} \times SF$
Coil Pass Frequency (CF)	$\frac{\text{Number of Coils} \times \textit{Rotor RPM}}{60}$

Synchronous Speed

How fast a motor spins, relative to Line frequency, depends on the number of poles, calculated as shown above (Synchronous Speed (Ns)). If the motor is a synchronous motor then the motor speed will be as calculated by Ns and shown in the following table.

Synchronous speed (RPM) at 50/60 Hz per Number of Poles*

Poles	50 Hz	60 Hz		Poles	50 Hz	60 Hz
2	3000	3600		**12**	500	600
4	1500	1800		**14**	428.6	514.3
6	1000	1200		**16**	375	450
8	750	900		**18**	333.3	400
10	600	720		**20**	300	360

* Motors with more than 20 poles are available

Induction Motor Speed

Induction motors do not operate at Synchronous Speed (Ns). They spin at some speed less than Ns with the difference increasing with load. In most cases this difference, called Slip Frequency (SF), is somewhere between about 0.3 Hz at no load and 2 Hz at the motors rated load.

Induction Motor Fault Frequencies

Most electric motors faults create uneven interactions between the rotor and stator which results in forces that oscillate at Line Frequency, or more likely 2x LF, and often modulated by the Slip Frequency. This means that to diagnose most motor faults it is necessary to resolve vibration frequencies to resolutions sufficient to differentiate 1x LF and 1x RPM, i.e., the Slip Frequency. Since SF may be just 0.3 Hz (or less), to properly differentiate this the frequency of vibration must be resolvable to 0.1 Hz, or less.

This is problematic in that none of the recommended configurations are configured to provide this degree of resolution as they are intended for more general purpose applications.

If the FMAX is 2000 Hz, default for recommended configurations for machines with rolling element bearings, and the number of FFT Lines is 1800, then the resolution will be 1.11 Hz	$\frac{2000\text{ Hz}}{1800\text{ Lines}} = \frac{1.11\text{ Hz}}{\text{Lines}}$
If the FMAX is set at 1144 Hz, the lowest selectable FMAX from the ADC, the resolution would be 0.64 Hz	$\frac{1144\text{ Hz}}{1800\text{ Lines}} = \frac{0.64\text{ Hz}}{\text{Lines}}$

So when it is necessary to diagnose most electric motor faults, a more detailed investigation of the vibration spectra must be made(1). This means that it will be necessary to observe the FFT data directly, using a high resolution FFT, to identify and determine if vibration is occurring at the electrically related frequencies (1x LF, 2x LF, 2xLF ±1x SF).

Tools are available to accommodate this and the recommended configurations do configure the monitor to provide FFTs with up to 14,400 lines when requested. With an FMAX of 2000 Hz, a 14,400 line FFT would resolve frequencies to 0.14 Hz, or 0.08 Hz if the FMAX is 1144 Hz.

The tools necessary are either Emonitor® CMS, which can be configured to periodically sample and archive high resolution measurements, or the Emonitor RTA utility, a simple to use and freely available real time analyzer utility.

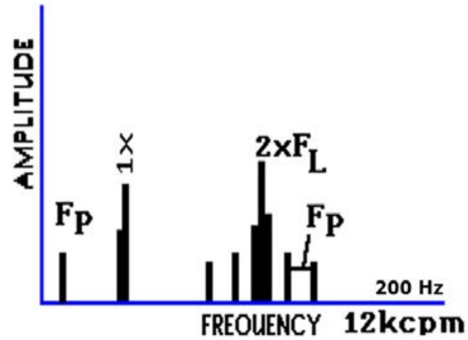
Induction Motor: Eccentric Air Gap

Eccentric air gap is usually the result of the centerline of the shaft being at an offset from the centerline of the stator. It can also result from a wiped bearing but that's another problem. Operation of a motor with an eccentric air gap is likely to cause more than just vibration. The operation is likely to accelerate deterioration of insulation, increase coil movement, and could result in rub between the rotor and stator.

(1) An alternative would be to define a module configuration that is specifically intended to monitor for electrical faults in motors. Such a configuration though would not be suitable for detecting the more common faults such as anomalys in rolling element bearings.

Symptoms

- Eccentric rotors produce a rotating variable air gap, this induces pulsating vibration
- Indicated by elevated 2x line frequency with slip frequency side bands
- Often requires zoom spectrum to separate 2FL and running speed harmonic
- Common values of SF range from 20…120 CPM (0.3…2 Hz)



What to Look For

An eccentric air gap will be indicated by vibration at 2x Line Frequency, typically with side bands at ±1x SF and ±2x SF (Slip Frequency). However, in most cases the recommended configurations will not provide the resolution necessary to differentiate 2xLF and 2xLF ±1xSF.

If high 2x RPM is indicated, and there is no 3x RPM or other fault indication, and the high 2x RPM is only on the motor, then further investigation(1) should be made to determine if any Slip Frequency side bands are present.

Attribute	…	Band 3
Band Name	…	2*x* Misalignment/Looseness
Frequency Range	…	1.8…2.2*x*

Where to Look

Indicators of this anomaly are likely to be present in measurements made in the radial direction on each bearing. It is likely not to be present across a coupling.

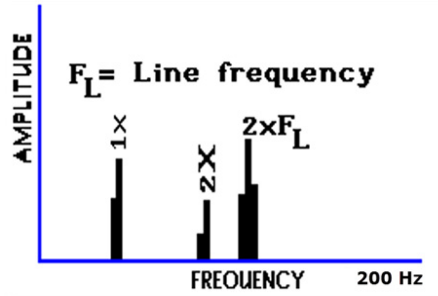
Induction Motor: Stator Eccentricity, Shorted Laminations, Loose Iron

Any of these stator problems; eccentric stator, which is shorted laminations or loose iron, can cause localized heating, which can cause warping of the structure, and is likely to generate similar vibration forces.

(1) To identify the presence of slip frequency side bands, observe an FFT with sufficient resolution to differentiate 2x RPM and 2x LF. Typically this will require an FFT resolution of 0.1 Hz to 0.2 Hz / FFT Line.

Symptoms

- Stator problems generate high amplitudes at 2x line frequency (2x LF)
- Stator eccentricity produces uneven stationary air gap, vibration is very directional Soft Foot can produce an eccentric stator



What to Look For

An eccentric stator, which is shorted laminations or loose iron in a stator is indicated by vibration at 2x line frequency. However, in most cases the sample profiles is likely not to provide the resolution necessary to differentiate between 2x RPM and 2x LF.

If high 2x RPM is indicated, and there is no 3x RPM or other fault indication, and the high 2x RPM is only on the motor, then further investigation[1] should be made to determine if any Slip Frequency side bands are present.

Attribute	…	Band 3
Band Name	…	2X Misalignment/Looseness
Frequency Range	…	1.8…2.2x

If the vibration is at 2x Line and no Slip Frequency side bands are present, then the problem is likely an eccentric stator, shorted laminations, or loose iron in the stator.

Where to Look

Indicators of this anomaly are likely to be present in measurements made in the radial direction on each bearing. It is likely not to be present across a coupling.

(1) To identify the presence of slip frequency side bands, observe an FFT with sufficient resolution to differentiate 2x RPM and 2x LF. Typically this will require an FFT resolution of 0.1 to 0.2 Hz / FFT Line.

Induction Motor: Rotor Bar Problems

A cracked or loose rotor bar is likely to create vibratory forces in either of two ways:

1. Vibrations at line frequency that are modulated at the pole pass frequency (PPF)[1]. This means that you would see high vibration at 1 *x* LF with side bands at ±PPF Frequency.

   It is also possible that elevated harmonics of 1 *x* RPM could also occur, also with PPF side bands.

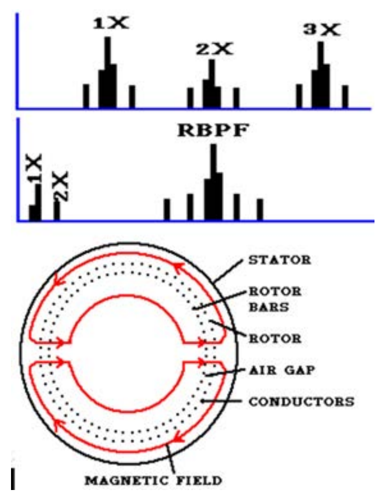
2. Vibrations at the rotor bar pass frequency (RBPD, are modulated by 2 *x* line frequency. This means that you would see high vibration at 1 *x* RBPF with side bands at ±2 *x* LF.

   The recommended configurations suggest an FMAX of about 2 kHz. While this should be adequate to observe the RBPF of most motors, if the motor has a large number of rotor bars the RBPF may be higher than 2 kHz.

A **broken rotor bar** is likely to increase the unbalance force significantly, indicated by high vibration at the 1 *x* RPM frequency.

Symptoms

- 1*x*, 2*x*, and 3*x* RPM with pole pass frequency sidebands indicates rotor bar problems
- 2 *x* line frequency sidebands on rotor bar pass frequency (RBPF) indicate loose rotor bars
- Often high levels at 2*x* and 3*x* rotor bar pass frequency and only low level at 1*x* rotor bar pass frequency



What to Look For

The sample profiles do not define FFT bands for rotor bar fault indication. Bands are defined that will detect 1*x*, 2*x* and 3*x* RPM vibration.

Attribute	…	Band 1	…	Band 3	…	Band 5
Band Name	…	1*x* Unbalance	…	2*x* Misalignment/Looseness	…	3*x* Misalignment/Looseness
Frequency Range	…	0.8…1.2*x*	…	1.8…2.2*x*	…	2.8…3.2*x*

If the 1 *x* RPM band is elevated, with or without the additional harmonics, and no other fault is indicated or apparent, then further analysis can be made to determine if there is a rotor bar problem.

(1) Pole Pass Frequency (PFF) = Number of Poles x Slip Frequency

The recommended configurations do not define an FFT band at the 1x RBP Frequency. However, if the motor is fitted with rolling element bearings then the RBPF may be within the span of the band defined to monitor the Bearing Natural Frequencies.

Attribute	Band 0	…	Band 5	Band 6	Band 7
Band Name	<1*x* Bearing Cage Anomalies	…	Bearing fundamental frequencies	Bearing lower harmonic frequencies	Bearing higher harmonic and natural frequencies
Frequency Range	0.2…0.8*x*	…	1.2 VPF…12.2*x*	12.2*x*…50% FMAX	50%…100% FMAX

The RBPF could be lower as well, within Band 6, or higher.

If the applied configuration does not define an FFT Band that encompasses the 1x RBPF, users should observe the Overall vibration and if elevated, and the defined bands do not explain the change, determine if the 1x BBPF is present, and if it has 2x LF sidebands.

Where to Look

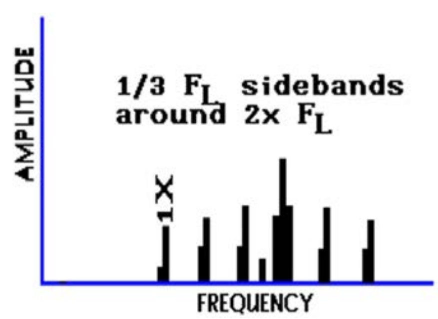
Indicators of these anomalies are likely to be present in measurements made in the radial direction on each bearing. It is likely not to be present across a coupling.

Induction Motor: Power Supply Phase Problems, Loose Connector

It is not possible to state conclusively what the consequence of one 3-phase connector becomes loose could be. This is because of the intermittent status of 'loose', or 'how loose', or what the impact would be if more than just one of the three connectors are loose. However, we do know what to expect if just one connector is 'loose'.

Symptoms

- Phasing problems can cause excessive vibration at 2 *x* LF with 1/3 LF sidebands
- Levels at 2 *x* LF can exceed 25 mm/sec RMS (0.7 ips pk) if left uncorrected
- Particular problem if the defective connector is only occasionally making contact



What to Look For

A loose connector is likely to cause vibration at 2 *x* LF, with side bands at one or more of ±1/3 *x* LF and ±2 *x* LF.

LF (Hz)	-2*x* 1/3 LF	-1*x* 1/3 LF	2*x* 1/3 LF	+1*x* 1/3 LF	+2*x* 1/3 LF
50	66.7	83.3	100	116.7	133.3
60	80	100	120	140	160

Since the sample profiles do not define FFT Bands at these specific frequencies, observe the 2x RPM band and if elevated, and no other fault is indicated, determine if the 1/3 LF sidebands are present.

Attribute	…	Band 3
Band Name	…	2X Misalignment/Looseness
Frequency Range	…	1.8…2.2x

Where to Look

Indicators of this anomaly are likely to be present in measurements made in the radial direction on each bearing. It is likely not to be present across a coupling.

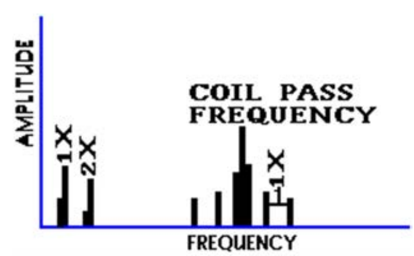
Synchronous Motor: Loose Stator Coils

Unlike common induction motors, synchronous motors do not slip. This means that the forces and diagnostic techniques that are associated with LF and SF used for induction motors, do not apply. So when monitoring a synchronous motor the key frequency to monitors is the Coil Pass Frequency (CF):

CF = Number of Coils x *Rotor RPM*/60

Symptoms

- Loose stator coils in synchronous motors generate high amplitude at CF
- The CF is likely to be surrounded by 1 *x* RPM sidebands



What to Look For

Loose coils in a synchronous motor are likely to create vibratory forces at the coil frequency that are modulated at the 1 *x* RPM frequency. This means that the indication is likely to be:

- High vibration at 1 *x* CF, with vibration at any of 1 *x* CF ±1 *x* RPM or 1 *x* CF ±2 *x* RPM

The recommended configurations do not define an FFT band at the 1x CF frequency. But if monitoring a synchronous motor the Coil Frequency should at least be determined and recorded.

If loose coils are a concern, particularly if there is history of the problem, then consider modifying one of the FFT Bands in the module configuration to monitor this frequency. However, if vibration is detected at CF, and there is no other possible source of vibration at or near that same frequency, even if from the driven machine, then a more thorough analysis of the FFT can be performed to resolve the presence of the 1 *x* RPM side bands to confirm the diagnosis.

Where to Look

Indicators of this anomaly are likely to be present in measurements made in the radial direction on each bearing. It is likely not to be present across a coupling.

DC Motor Problems

While historically DC motors were uncommon, which is changing rapidly—look up at your new sleek, quiet, and efficient DC motor driven ceiling fan that replaced its old clunky, noisy, and inefficient AC motor driven fan that had been the standard for ceiling fans for decades!

But in terms of faults and vibration and diagnostics, DC motors are very different from the still common AC induction motors. For DC motors, the key is the firing frequency of the Silicon Controlled Rectifier (SCR). The SCR frequency is related to the line frequency, and if it is a full or half wave rectifier.

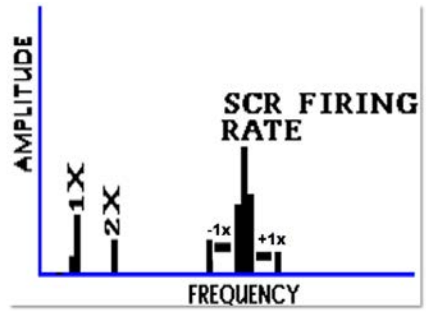
UF	SCR Frequency	
	Full Wave Rectifier (6x LF)	Half Wave Rectifier (3x LF)
50	300 Hz	150 Hz
60	360 Hz	180 Hz
VFD	6x LF	3x LF

The most common problems with DC motors are issues with the rectifier. Consequently the frequency that is indicative of these issues is SCR. So faults are likely to be indicative by vibration at 1 *x* SCR and possibly 2 *x* SCR, and often with 1 *x* RPM sidebands, so 1 *x* SCR ± 1 *x* RPM and possibly 2 *x* SCR ±1 *x* RPM.

Other DC motor problems, such as fuse or control card problems, or broken field windings, can cause vibration at 1 *x* RPM and its harmonics, up to 5 *x* RPM.

Symptoms

- DC motor problems can be detected by the higher than normal amplitudes at the Silicon Controlled Rectifier (SCR) firing rate
- These problems include broken field windings
- Fuse and control card problems can cause high amplitude peaks at frequencies of 1…5 *x* line frequency



What to Look For

The recommended configurations do not define an FFT band at the 1x SCR (or 2x SCR) frequency. But if monitoring a DC motor the SCR Frequency should at least be determined and recorded.

If a SCR tuning or other problem is a concern, particularly if there is history of the problem, then consider modifying one of the FFT Bands in the module configuration to monitor this frequency. However, if vibration is detected at 1 *x* SCR, and there is no other possible source of vibration at or near that same frequency, even if from the driven machine, then a more thorough analysis of the FFT can be performed.

Some faults are indicated at the 1 *x* RPM frequency and possibly at 2…5*x* RPM. If the bands that indicate vibration at these frequencies elevate, and no mechanical problem is indicated or apparent, then further investigations can be made into the possibility of an electrical problem.

Attribute	…	Band 3	…	Band 5
Band Name	…	2*x* Misalignment/Looseness	…	3*x* Misalignment/Looseness
Frequency Range	…	1.8…2.2*x*	…	2.8…3.2*x*

Where to Look

Indicators of these anomalies are likely to be present in measurements made in the radial direction on each bearing. It is likely not to be present across a coupling.

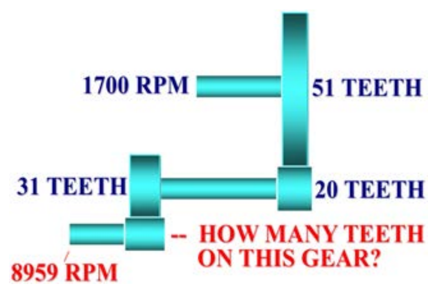
Gear Faults

A gearbox can be relatively simple; just one gear and pinion, two shafts and four bearings. Or, if a double reduction gear; two gear and pinion pairs, three shafts and six bearings. And some gearboxes are far more complex with multiple mating gears, several stages of reduction and various types of gears.

So while gear diagnostics are well-defined, the problem is more about the system complexity. Because of the often close proximity of components and often common structural elements, it is not unusual to observe vibration associated with multiple gears, shafts, and bearings at one sensor. Consequently, when gear vibration is analyzed, it is very important to use accurate speed and design information so the source of vibration can be correctly determined.

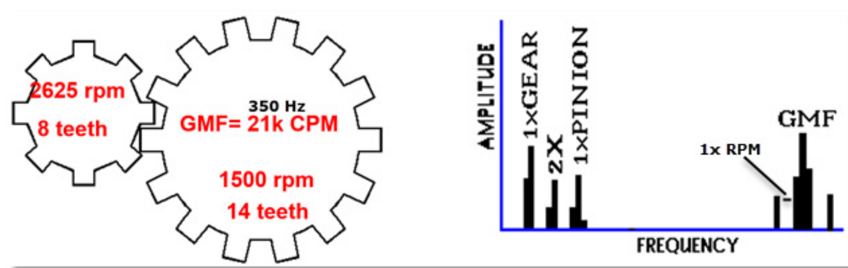
Vibration in gears occurs at the following frequencies:

Gear Mesh Frequency (GMF)	Number of Gear Teeth × Shaft RPM / 60
Natural Frequency (fn)	The natural frequency of the gear. It is likely to vary depending on the gear and system
Hunting Tooth Frequency (HT_f)	Common Frequency = Highest Common Denominator between the number of teeth on the pinion and bull gears GMF × Common Frequency / Number of Pinion Gear Teeth × Number of Bull Gear Teeth
1*x* GEAR	The 1 *x* RPM of the gear shaft
1*x* PINION	The 1 *x* RPM of the pinion shaft



Answer: 15

A **normal gear** is likely to present vibration at its GMF, but at relatively low amplitudes. It can also exhibit side bands of running speed (gear or pinion) around GMF.

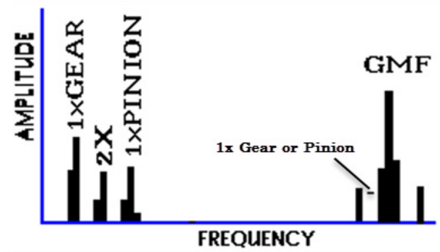


Tooth Load

It is not uncommon for the level of vibration associated with gear mesh to vary with load. However, when the level of vibration at a given load is not consistent with past performance, a Tooth Load problem may be indicated.

Symptoms

- Gear Mesh Frequencies (GMF) are often sensitive to load
- High GMF amplitudes do not necessarily indicate a problem
- If periodic (portable) measurements are taken then each analysis should be performed with the system at a consistent load



- If monitoring online then it may be important to correlate GMF amplitude and load, and alarm when the GMF increases above the norm for the current load

What to Look For

The recommended configurations defined for monitoring gears define just one FFT Band specific to monitoring gear issues.

Attribute	…	Band 4
Band Name	…	Gear Mesh Frequency
Frequency Range	…	1.8…1.2 GMF

While vibration at the 1 *x* GMF is the most common indicator of faults in gears, an elevated 1 *x* GMF band does not necessarily indicate a problem. This is because it is common for gear vibration to be sensitive to load. As tooth load increases, so does vibration at 1 *x* GMF.

Consequently, when monitoring gears, it is important to understand the normal behavior of the gear concerning load. Perform a more thorough review of the vibration data if the 1 *x* GMF vibration elevates above what is expected for the load conditions.

Where to Look

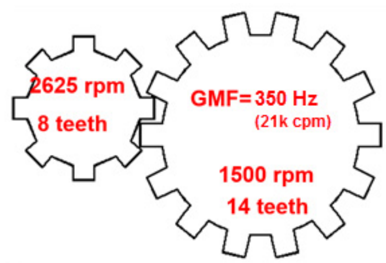
Indicators of anomalies in gears are mostly prevalent in the radial direction. Some gears and gear faults are likely to present significant vibration in axial direction.

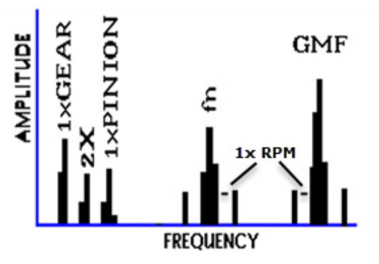
Tooth Wear

When gear teeth wear, they do not wear equally. This unequal wear means that the balance of the gear and/or pinion is likely to degrade. This degradation is always indicated by elevated vibration at the 1 *x* RPM of the gear and/or pinion.

Symptoms

- Wear is indicated by excitation of natural frequencies (fn) along with sidebands of 1 *x* RPM (gear or pinion) of the bad gear
- Sidebands of ±1*x* GEAR or ±1*x* PINION RPM around the natural frequency (fn)
- Sidebands of ±1*x* GEAR or ±1*x* PINION RPM around the 1 *x* GMF or 2 *x* GMF
- Sidebands are a better wear indicator than the GMF
- GMF is sometimes unchanging in amplitude when wear occurs





What to Look For

The sample profiles that are defined for monitoring gears define two bands that can indicate tooth wear.

Attribute	…	Band 1	…	Band 4
Band Name	…	1*x* Unbalance	…	Gear Mesh Frequency
Frequency Range	…	0.8…1.2*x*	…	1.8…1.2 GMF

While vibration at the 1 *x* GMF is sometimes not significantly affected, at least in the earlier stages, the band definition is wide enough to encompass the 1 *x* RPM side bands that are likely to be present and that is likely to become more prominent. Since these add to the 1 *x* GMF band magnitude a change in the 1 *x* GMF band magnitude, when accompanied by an increase in the 1 *x* RPM of the Gear or Pinion shaft band values, can indicate this condition.

Consequently, when monitoring gears, if an increase the Unbalance indicator is observed, note any change in the 1 *x* GMF. If both are elevated, then the FFT can be reviewed to determine if the 1 *x* GMF band is increasing due to vibration at the 1 *x* GMF, or at the 1 *x* RPM sidebands. If it is the sidebands, then damage is indicated. If it is at the 1 x GMF, then shaft unbalance, due to some other cause, is likely the problem.

Another indicator is that if damage is present then one or more natural frequencies can be excited. In this case, if the natural frequencies are known, observe their condition in the FFT. If the natural frequencies are not known, observe the FFT and identify any unexpected vibration that is modulated by (has side bands) the 1 *x* RPM of the gear or pinion shaft. If present then damage is likely.

Note that an eccentric gear, or excessive backlash, can exhibit similar symptoms but with vibration also present at harmonics of the 1 *x* GMF. See "Gear Eccentricity and Backlash" for further information.

Where to Look

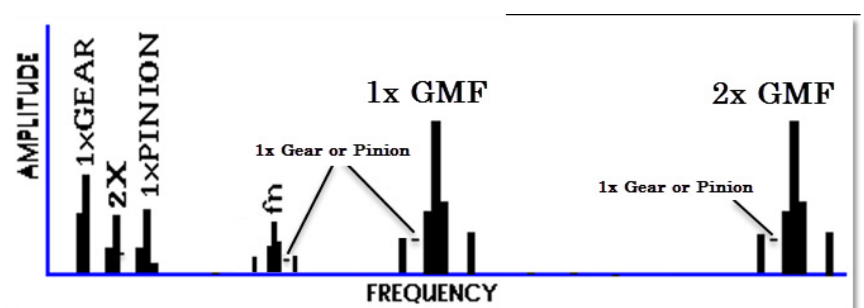
Indicators of anomalies in gears are mostly prevalent in the radial direction.

Gear Eccentricity and Backlash

An eccentric gear, or bent shaft, is likely to cause vibration at the 1x and harmonic frequencies (2 *x* GMF and/or 3 *x* GMF) of the affected gears Gear Mesh Frequency that are modulated by the 1 *x* RPM of the associated shaft. Gear backlash can cause similar indication but can also excite the gear natural frequency.

Symptoms

- Fairly high amplitude sidebands around GMF and GMF harmonics (2 *x* and/or 3 *x* GMF) suggest eccentricity, backlash, or non-parallel shafts
- The problem gear is likely to modulate the GMF and create sidebands at 1 *x* RPM (gear or pinion)
- Severe eccentricity can elevate vibration at the 1 *x* RPM and possibly harmonics of the 1 *x* RPM of the offending shaft
- Incorrect backlash can excite gear natural frequency (fn)



Note that the image does not illustrate 3 *x* GMF.

What to Look For

The sample profiles that are defined for monitoring gears define two bands that can indicate gear eccentricity and backlash.

Attribute	**…**	**Band 1**	**…**	**Band 4**
Band Name	…	1*x* Unbalance	…	Gear Mesh Frequency
Frequency Range	…	0.8…1.2*x*	…	0.8…1.2 GMF

While vibration at the 1 *x* GMF is sometimes not significantly affected, at least in the earlier stages, the band definition is wide enough to encompass the 1 *x* RPM side bands that are likely to be present and that is likely to become more prominent. Since these add to the 1 *x* GMF band magnitude a change in the 1 *x* GMF band magnitude, when accompanied by an increase in the 1 *x* RPM of the Gear or Pinion shaft band values, can indicate this condition.

Consequently, when monitoring gears, if an increase the Unbalance indicator is observed, note any change in the 1 *x* GMF. If both are elevated, then the FFT can be reviewed to determine if the 1 *x* GMF band is increasing due to vibration at the 1 *x* GMF, or at the 1 *x* RPM sidebands. Also observe if there is vibration at the 2 *x* GMF or 3 *x* GMF frequencies, also modulated (side bands) at the 1 *x* RPM of the offending shaft.

If (2x and/or 3x) GMF harmonics are present, then the problem is likely an eccentric gear or bent shaft. The problem could also be excessive backlash.

Excessive backlash is likely to excite the gears natural frequency. While the frequency of the natural is sometimes not known it is likely well below 1 *x* GMF frequency, and not coincidental with other normal vibration frequencies.

Where to Look

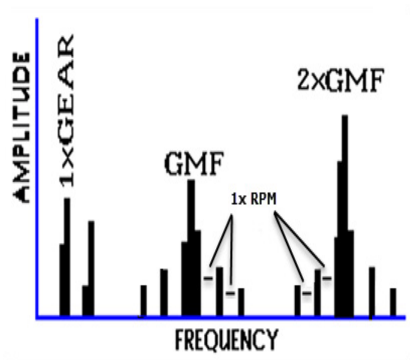
Indicators of anomalies in gears are mostly prevalent in the radial direction.

Gear Misalignment

Misaligned gears are likely to wear faster and the wear is likely to be uneven across the gear. Ultimately because a misaligned gear is likely to stress bearings, it is likely to result in premature bearing failure.

Symptoms

- Gear misalignment almost always forces vibration at harmonics (2*x* and/or 3*x*) of the GMF that are modulated at (so is likely to have sidebands) the shaft running speed (gear or pinion)
- Small amplitude at 1 *x* GMF but higher levels at 2 and 3 *x* GMF. The harmonic GMF frequencies are likely to be larger than the 1 *x* GMF
- Important to set FMAX high enough to capture at least 2 *x* GMF



Note that the image does not illustrate 3 *x* GMF.

What to Look For

The sample profiles that are defined for monitoring gears define just one FFT Band specific to monitoring gear issues.

Attribute	…	Band 4
Band Name	…	Gear Mesh Frequency
Frequency Range	…	1.8…1.2 GMF

Because a misaligned gear is likely not to increase vibration at 1 *x* GMF, the condition is signaled by an increase in the overall vibration without seeing an increase in any of the monitored bands. In this case, the FFT can be reviewed to investigate the problem.

When reviewing the FFT, if the vibration at 2 *x* GMF and/or 3 *x* GMF is present, with sidebands at 1 and/or 2 *x* RPM, then a misaligned gear is indicated.

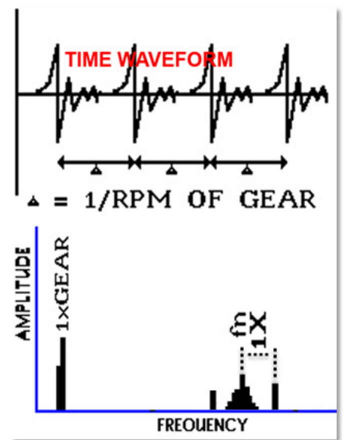
Where to Look

Indicators of anomalies in gears are mostly prevalent in the radial direction. However, depending on the type of gear, and the severity of the misalignment, the condition can be present, even more apparent, in the axial direction.

Cracked or Broken Gear Tooth

A gear tooth is similar to cantilever beam, which is supported on a long side, and that experiences forces at a specific contact point.

Symptoms



- A cracked or broken tooth is likely to generate a high amplitude at 1 *x* RPM of the gear
- It is likely to excite the gear natural frequency, which is likely to be modulated (side-banded) by the running speed (gear or pinion) fundamental
- Best detected using the time waveform
- Time interval between impacts is likely to be the reciprocal of the 1 *x* RPM

What to Look For

A broken gear tooth, and possibly a cracked tooth, is likely to unbalance the gear, which is likely to force increased vibration at the 1 *x* RPM of the gear. This is indicated by the FFT Band that is monitoring the '1 *x* Unbalance' vibration.

Attribute	…	Band 1
Band Name	…	1*x* Unbalance
Frequency Range	…	0.8…1.2*x*

The condition is likely to excite the gear natural frequency. However, since no FFT Band is configured to monitor the frequency, even if known, the only indication of this condition is likely to be an increase in the Overall vibration that is not accounted for by increases in other bands. When this occurs, the FFT can be reviewed. If the FFT shows a consequential vibration at an unexpected frequency, with sidebands at 1 *x* RPM, then a cracked or broken tooth can be indicated.

Where to Look

Indicators of anomalies in gears are mostly prevalent in the radial direction.

Hunting Tooth

Hunting Tooth vibration occurs when a gear and pinion each have one damaged tooth, and those two teeth mesh. So occurs at the frequency that the same two teeth mesh, which can be calculated as shown in the formula at the beginning of this section (Gear Faults).

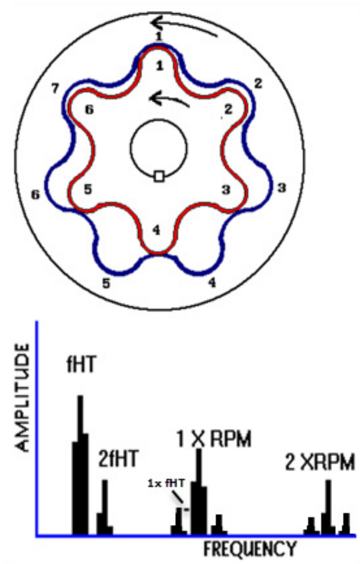
The HT_f is typically a low number. Depending on the result of the highest common denominator, whether something like '6' or '27', the resulting HT_f can be 5…10 Hz, or less than 1 Hz. It makes a significant difference in analysis because the resolution of the FFT (frequency per line) is sometimes not sufficient to differentiate the Hunting Tooth Frequency.

Another problem is that a Hunting Tooth fault is likely to propagate as the damaged teeth begin to damage other teeth, which means that the symptoms are likely to migrate toward 'Tooth Wear' eventually.

Symptoms

Hunting Tooth Frequency (HT_f) is how often (the frequency) that two specific teeth, one on each gear, which is damaged, come into contact with one another.

- Vibration at the $1\ x\ HT_f$ and possibly $2\ x\ HT_f$
- Sidebands at $1\ x\ HT_f$ around $1\ x$ RPM (for each shaft)
- Sidebands at $1\ x\ HT_f$ around the $1\ x$ GMF and $2\ x$ GMF
- The HTf can be at a very low frequency, so can often be missed
- Typically creates a growling sound



What to Look For

The recommended configurations do not define an FFT band at the $1\ x\ HT_f$ (or $2\ x\ HT_f$) frequency. In some cases the HT_f, or $2\ x\ HT_f$ may be within the Band 0 frequency span (0.2x…0.8x RPM) but in most cases HT_f will be less than $0.2\ x$ RPM.

If monitoring gears and either:

- HT_f is greater than $0.2\ x$ RPM and Band 0 is greater than normal.

Attribute	Band 0	Band 1
Band Name	1*x* Bearing Cage Anomalies	1*x* Oil Whirl/Oil Whip
Frequency Range	0.2…0.8*x*	0.2…0.8*x*

- HT_f is less than 0.2 *x* RPM and the Overall vibration is above normal and is greater than the sum of the measured bands.

Then a more thorough analysis of the FFT can be performed. Specifically the analysis can identify the presence of vibration (peaks) at the 1 *x* HT_f, 2 *x* HT_f frequencies, and at sidebands around 1 *x* RPM and the GMF frequencies. If vibration is observed at these frequencies, but not necessarily all of these frequencies, then a Hunting Tooth problem is likely.

Where to Look

Indicators of this anomaly are likely to be present in measurements made in the radial direction on the gear bearings.

Hydraulic and Aerodynamic Faults

Vibration in machines with blades or vanes (fans, pumps, and so on.) occur at the following frequencies.

Blade Pass Frequency (BPF)*	Number of Blades *x* Shaft RPM
Vane Pass Frequency (VPF)*	Number of Vanes *x* Shaft RPM
* If the machine has a diffuser then also consider the frequency that is calculated by VPF or BPF *x* Number of Diffuser Vanes. On machines with a diffuser, turbulence can manifest at this frequency, and multiples of it, rather than the base BFP/VPF. While BPF generally refers to fans and turbines, things with blades, and VPF for pumps, it is not uncommon that BPF and VPF are used in either case. This is particularly common where BPF is used to refer to vane pass on a pump	
Low Frequency Flow Noise	Typically somewhere between 0.8…33 Hz
Excited Natural Frequency	Typically somewhere between 1*x* VPF/BPF and 1000 Hz

Flow Turbulence

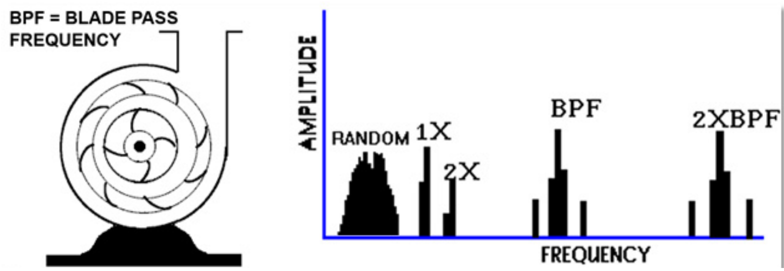
Machines that propel fluids, gaseous or liquid, can experience flow turbulence that manifests as vibration. Fluctuations in the variables associated with flow cause turbulent flow; typically pressure or possibly density (if gaseous).

Minor turbulence is common, can easily go unnoticed, and only slightly accelerates wear of components such as blades/vanes, seals, wear rings, impellers, and bearings. Severe turbulence can rattle deck plates and be very destructive.

In common centrifugal machines, pressure fluctuations occur when the clearances around the blades/vanes are not uniform due to a cocked or misaligned impeller or an eccentric rotor, or when the blades/vanes are damaged. Low suction pressure (starvation) can cause turbulence, high discharge pressure, or combinations of low/high suction/head pressures that are associated with process or system components external to the machine.

Symptoms

- Random low frequency vibration that is associated with flow noise can occur, typically in the 0.8…33 Hz (50…2000 CPM) range
- High vibration at 1*x*(1) and possibly 2*x* the Vane/Blade pass frequency or, if a diffuser is present, at the diffuser pass frequency and its multiples
- Flow turbulence has also been known to present as an apparently random 'haystack' at frequencies up to 10 kHz.



What to Look For

The primary indication of flow turbulence is vibration at the blade/vane pass frequency. The sample profiles for pumps and fans, regardless of bearing type, define one FFT Band to monitor the blade/vane pass frequency.

Attribute	…	Band 6
Band Name	…	Vane or Blade Pass Frequency
Frequency Range	…	0.8…1.2 V/BPF

If the BPF/VPF is elevated consider reviewing the FFT data to identify the presence of vibration at the 2 *x* BPF/VPF, and possibly the presence of low frequency flow noise. The presence of 2 *x* BPF/VPF is clear indication of a more pronounced turbulence problem.

Where to Look

Indicators of this condition are likely to be present in measurements that are made in the radial directions on the pump or fan bearings. On large machines, the condition can be much more pronounced at one end of the machine, typically the discharge end.

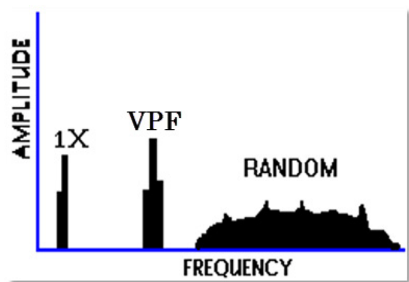
If monitoring a fluid film bearing, the indicators of this condition can be present in both the X and Y directions. However, it is possible that the vibration in one direction is likely to be much more pronounced than the other.

(1) Flow through a centrifugal fan or pump is not likely to ever be perfectly uniform, so there is always likely to be some 'normal' level of vibration at the 1*x* VPF/BPF. Higher harmonics of VPF/BPF are possible depending on the machine design, such as use of a split/double volute or a diffuser ring.

Cavitation

Cavitation is the formation and subsequent collapse of vapor bubbles in a pump. As the fluid passes from the suction side, bubbles form in the low-pressure areas, often along the back side of a vane. At some point, as each bubble moves toward the discharge side, and as the pressure increases, the bubble collapses. When the bubble implodes it triggers intense shockwaves that can damage the vanes, particularly if the bubble forms on and moves along the surface of the vane

Symptoms



- Cavitation is likely to generate random, high frequency broadband energy that is superimposed with VPF harmonics

  In some cases this will excite a natural frequency, which are commonly between 1 *x* VPF and about 1 kHz. If this occurs the resonance can cause far more significant, and possibly destructive, vibration
- Erosion of impeller vanes and pump casings can occur if left unchecked
- Sounds like gravel as it passes through the pump

What to Look For

The sample profiles do not define a band to monitor for cavitation. If a natural frequency is known, and it's between 1 *x* VPF and about 1 kHz, then it is likely that it is likely to be excited when cavitation is present. If it is or is not, can only be determined by observing the vibration while cavitation is known to be occurring.

Once an observed increase in vibration over a specific frequency range is identified caused by cavitation, then a band can be redefined to monitor it.

Attribute	…	Band 4	Band 6
Band Name	…	Vane Pass Frequency	Cavitation
Frequency Range	…	0.8…1.2 VPF	~~1.2 VPF…12.2x~~ **n1…n2 Hz**
% of Overall Level	…	60%	1/2…3/4 of observed value while cavitating

'n1…n2 Hz' are the beginning and ending frequencies of observed elevated region when cavitating. The band can be defined in the frequency domain.

Where to Look

Cavitation is detected by sensors that are mounted at pump output, rather than on the bearings. For sensors that are mounted on the bearings, cavitation is likely to be observed in the radial direction.

Another good method of cavitation detection is to use the Spike Energy (gSE) measurement. Spike Energy can be sensitive to vibration associated with cavitation so is often a more concise method than the earlier detailed method, particularly when cavitation does not excite a readily observed natural frequency.

Belt Problems

Pulley and belt drives are used on various types of machines such as fans, compressors, and machine tool spindles. While some of these machines can exhibit inherently high vibration, others can have quite low vibration. The basic considerations for vibration in belt and sheave systems are sheave alignment, sheave eccentricity, belt condition, and belt tension. There are differing belt types as well. Some are smooth while others are toothed, which add to the complexities of belt vibration monitoring.

When monitoring belts, sensor location, which is shown in the following figure, is based on the centerline between the two pulleys. Horizontal is radial in the direction of the centerline between the pulleys, and vertical is perpendicular to the centerline. Vertical can also be measured on both pulleys, in line with the shaft. Differentiating among various problems inherent in belt driven systems is significantly aided by comparing the horizontal and vertical amplitudes at various frequencies.

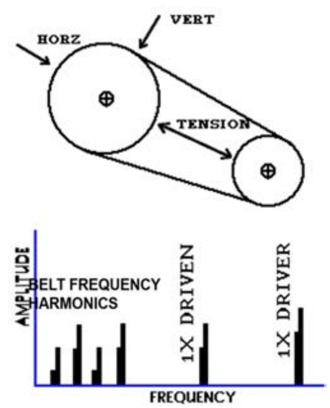
Problems such as high imbalance, misalignment, looseness, and so on often affect belt drives. Any of these can cause vibration leading to an incorrect belt system issue diagnosis. Care must be taken to differentiate among all of these symptoms and causes.

Vibration in belt driven systems occurs at the following frequencies.

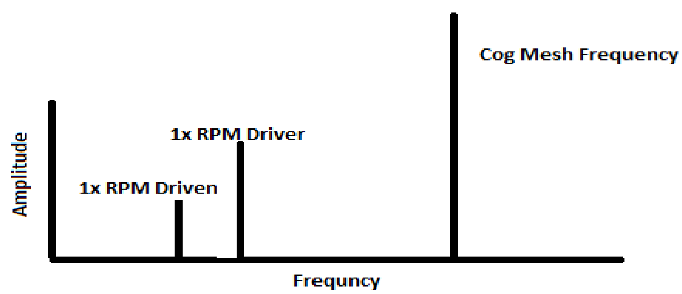
Belt Frequency	(3.142) *x* (Pulley RPM) *x* (Pulley Pitch Dia.)/Belt Length
U	Distance between belt tensile cord and bottom of belt tooth
Pulley Pitch Diameter	Pulley Outside Diameter + 2 *x* U
Cog Belt Mesh Freq	Number of Cogs *x* RPM / 60
	*Use RPM and Pitch Diameter of the same pulley

Worn, Loose, or Mismatched Belts

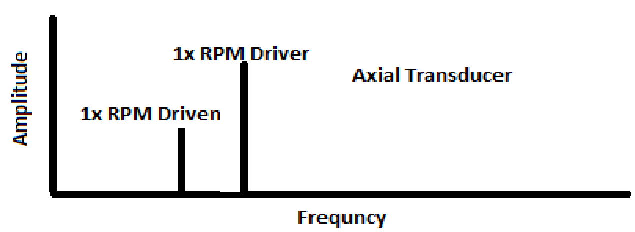
Worn belts are represented by amplitudes in the belt fundamental frequency (1*x*) range. The belt frequency is likely to be lower than both the driven and driving pulley 1 *x* RPM. Multiples of the belt frequency are likely to appear, often in multiples of 2*x*, 3*x* and 4*x*, and in some instances can elevate the whole sub-synchronous floor below the pulley 1 *x* frequency. The amplitudes of these sub-synchronous frequencies can also fluctuate during operation. The amplitude in the horizontal direction (parallel to belt tension) is likely to be the highest.



Loose cog belts are represented in the spectrum at the cog mesh frequency. The amplitude is likely to be highest in the horizontal (parallel to belt tension) direction.



Mismatched belts in multiple belt drive systems increase vibration much like worn belts. This is caused by using belts of unequal length. However, unique to multiple belt configuration is likely to be high axial vibration at the drive and the driven pulley shaft.



Symptoms

- Often 2 *x* RPM is dominant
- Amplitudes are normally unsteady, sometimes pulsing with either driver or driven RPM
- Wear or misalignment in timing belt drives is likely to give high amplitudes at the timing belt frequency
- Belt frequencies are below the RPM of either the driver or the driven

What to Look For

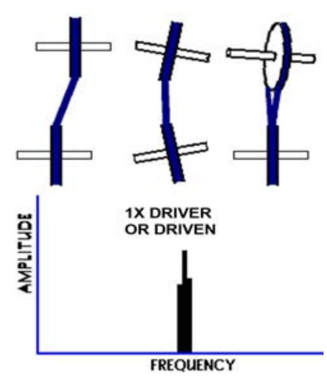
The recommended configurations look at the sub-synchronous frequency range band, for loose and mismatched belts, based on the driving pulley 1*x* RPM. Fluctuating amplitude in the sub-synchronous frequency band is also an indicator. Loose cog belts show up in a band of 0.9*x*…1.1*x* cog mesh frequency. Increasing amplitude is the indicator.

Where to Look

Indicators of these anomalies are also likely to be present primarily in the horizontal (parallel to belt tension) direction accelerometer. Mismatched belts are likely to also be confirmed using an axial accelerometer at either pulley shaft.

Belt/Pulley Misalignment

Vibration in machines with belt and pulley misalignment is likely to exhibit in high axial vibration. The dominant amplitude is likely to be at 1 *x* RPM of the driving pulley, although in some instances the driven pulley axial 1 *x* amplitude can be dominant. This is dependent on the mass and stiffness of the machine components driven by the belt/pulley system.



Symptoms

- Pulley misalignment is likely to produce high axial vibration at 1 *x* RPM
- Often highest amplitude on the motor is likely to be at the driven pulley RPM

What to Look For

The sample profiles review bands representing 1 *x* driver and 1 *x* driven vibration in the axially oriented sensor on the driven shaft.

Where to Look

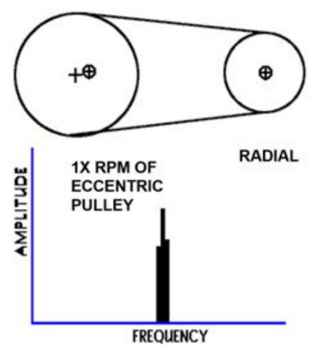
Indicators of these anomalies are likely to be present in the axial direction, parallel to the shaft centerline.

Eccentric Pulleys

Vibration in machines with eccentric pulleys is high at 1 *x* RPM of the eccentric pulley. Eccentric pulleys can be confused with imbalance, but cannot be corrected with balancing. The vibration is highly directional with eccentricity, in the orientation of a sensor parallel to the center line of the belt tension. Vibration amplitudes are highest when measured in this orientation, and significantly less when measured 90° from the centerline. Unbalance typically produces fairly consistent amplitudes around the entire rotation. Correction is only by replacing the pulley with a pulley with a concentric hub.

Symptoms

- Eccentric or unbalanced pulleys are likely to give a high 1 *x* RPM of the eccentric pulley
- The amplitude is likely to be highest in line with the belts
- Beware of trying to balance eccentric pulleys



What to Look For

The sample profiles review a band that is associated to 1*x* of eccentric pulley RPM, and associated to an accelerometer oriented in line with the belt tension. In the unlikely event, there is a second accelerometer that is oriented at 90° to the first accelerometer, a 1 *x* RPM band would also be monitored and compared to the first. If the amplitude is similar, then balance is suspected. If the second is less than half the first, then eccentricity is suspected.

Where to Look

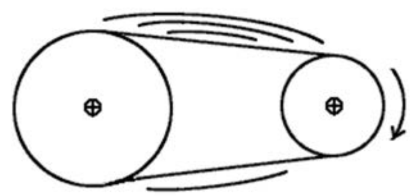
Indicators of these anomalies are likely to be present primarily with accelerometers that are oriented in the direction of the belt tension as eccentricity produces highly directional vibration.

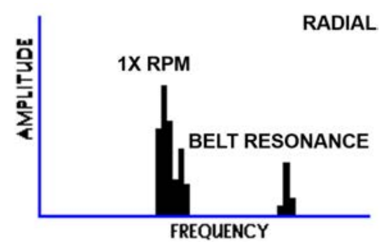
Belt Resonance

Vibration in machines with belt resonance exhibit high unsteady vibration at 1x RPM of either driven or driving pulley. In addition, the belt RPM harmonic could produce resonance in the belt. The belts typically are likely to start 'flopping' during resonance.

Symptoms

- High amplitudes can be present if the belt natural frequency coincides with driver or driven RPM, or belt speed harmonics
- The belts flop
- Vibration amplitude is typically inconsistent
- Belt natural frequency can be changed by altering the belt tension, length, or adding an idler pulley





What to Look For

The sample profiles review 1 *x* RPM of the driver and driven pulley. Resonance is differentiated from other symptoms at 1 *x* RPM in that the amplitude is unstable, and the belts move back and forth, 'flop' significantly.

Where to Look

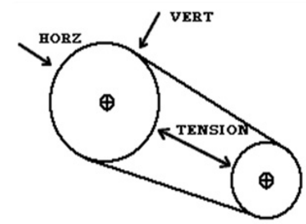
Indicators of these anomalies are likely to be present in an accelerometer that is oriented radially to either the driven or driving pulley.

Belt Anomalies and Loose Pulley

Vibrations in machines with belt anomalies exhibit higher vibration in the direction of belt tension. One missing piece or bump is likely to show up as a 1 *x* RPM of the belt. Crooked belts are likely to have higher vibration at belt RPM in the axial direction. A loose pulley or hub is likely to exhibit vibration at 1 *x* RPM of the loose pulley, and multiple harmonics of the pulley running speed.

Symptoms

- High amplitudes, possibly at belt RPM depending on the anomaly and number, is likely to be present in the direction of the belt tension
- Other anomalies can show up at belt RPM in the axial direction



What to Look For

The sample profiles review 1 *x* RPM of the belt, and bands associated with 1 *x* pulley harmonics for loose pulleys.

Where to Look

Indicators of these anomalies are likely to be present in an accelerometer oriented in line with the belt tension.

Glossary

The following terms and abbreviations are used throughout this manual. For definitions of terms that are not listed here, refer to the Allen-Bradley® Industrial Automation Glossary, publication AG-7.1.

1x Vibration The vibration frequency at the speed of the rotating shaft. Typically associated with imbalance.

2x Vibration The frequency at twice the running speed of the shaft. Typically associated to issues with machine couplings between components.

Accelerometer A piezo electric transducer with the piezo crystal that is compressed between two weights. Vibration energy compresses the crystal and the change in output voltage produces an AC current. This current uses a charge amplifier and the resulting AC signal is converted from the raw data form to usable data by an FFT analyzer. The native measurement of the accelerometer is g's of acceleration.

Accelerometer Mounted Resonant Frequency Each accelerometer has a frequency range considered the linear usable range of the transducer. This range is designated as the repeatable range of the accelerometer. In addition, each manufacturer specifies a natural frequency at which the accelerometer excites into resonance. This frequency is used to enhance high frequency signal content such as minute bearing anomalies and gear mesh issues.

Asynchronous Sampling Asynchronous sampling is the normal or default mode of measurement. It is when the individual samples of a time waveform are taken at a specific, constant, time interval. For example, if a time waveform includes 2048 samples, and has a period (total duration) of 2.0 seconds, then the sample interval is approximately 0.98 milliseconds.

$$\text{Sample Interval: } \frac{2.0\text{ s}}{2048\text{ samples}} = 0.0009765625 \times \frac{\text{s}}{\text{samples}}$$

$$\text{Sample Rate: } \frac{2048\text{ samples}}{2.0\text{ s}} = 1024 \times \frac{\text{samples}}{\text{s}}$$

When sampling asynchronously, the sample rate is fixed and does not change.

An FFT calculated from an asynchronously sampled time waveform results in a spectrum with a fixed maximum frequency (FMAX). The FMAX frequency is calculated based on sample rate and quality of the applied anti-alias filtering.

Axial A directional measurement in line with or parallel to the center line of the machine rotating shaft.

Bent Shaft If a shaft is bent, the permanent bow causes 1 *x* vibration. This bow is indicated when high 1 *x* vibration is not reduced to acceptable levels with balancing. Bent shaft is characterized by high 1 and 2 *x* axial vibration as well.

Blade Pass A fan or turbine has a known number of blades on its rotor. As the shaft rotates the alignment of the blades to the outlet varies. As each blade crosses the outlet it obstructs the air (or steam) flow as it does, which causes the pressure at the outlet to vary. This pressure variation repeats as each blade passes the outlet, effectively creating a pressure pulsation that occurs at a frequency equal to the number of blades times the speed of the shaft. This frequency is called the 'blade pass frequency'.

On a multi-stage fan or turbine, there is a different number of blades on each wheel (stage). Consequently, there are blade pass frequencies that are associated with each stage of the machine.

Elevated vibration at the blade pass frequency can indicate turbulent flow, due to either an obstruction or a problem with an input nozzle, or an alignment problem with the wheel.

Cavitation Cavitation occurs when the fluid intake of a pump is starved for liquid relative to the pressure at the discharge side of the pump. As the liquid tries to fill the void that is created by the vane or cylinder opening, the pressure on the liquid drops to below the boiling point of the liquid. Gaseous bubbles of the liquid form and travel through the pump. As the pressure begins to increase at the discharge, the pressure overcomes the gaseous bubbles and they collapse back to a liquid. The accelerometer picks up this collapse as random, high frequency vibration.

Eddy Current Probe A three part system consisting of probe, driver, and extension cable. This three part system is calibrated and sold as a set. Eddy current probes (ECP) measure actual movement of the object in mils of displacement.

FFT Fast Fourier Transform. The Fourier Transform is a mathematical function that breaks complex data sets into repeating components. In this case, repeating frequencies are identified in a complex set of data. When done with a computerized analyzer, the function is said to be a Fast Fourier Transform.

FFT Band An FFT spectrum is the typical result of an FFT vibration analyzer. The spectrum is a function of various frequencies along the X-axis and the amplitude of each frequency along the Y-axis of the function. An FFT band is a frequency or group of frequencies based on a low frequency limit and high frequency limit, set along the X-axis, and within the high and low limits of the frequencies filtered during the data sampling.

FMAX The highest frequency that is sampled by the FFT analyzer. FMAX is typically set a 3.25…3.5 times the highest expected vibration frequency from the machine train that is to be monitored.

gSE/Spike Energy A signal processing function that is designed to measure short duration pulses, or spikes, of vibration energy generated by impacts of rolling element bearing balls or rollers with microscopic indentions, or spalls, in the bearing races. gSE also measures other high frequency impacts such as rotor rub, insufficient lubrication, cavitation, steam and air leaks, fan turbulence, and others. See Spike Energy (gSE) Measurements for a more detailed discussion of gSE.

Harmonics Harmonics are exact whole number multiples of a forcing frequency, such as running speed. The more harmonics present in the spectrum, the more likely there is a rub or impact situation.

Inches per Second (ips) A measure of velocity. One ips equals one inch of travel in one second.

Integration Integration is a math function where a measurement amplitude in one unit of measure is 'integrated' to the next lower measurement unit. An example would be to integrate g's of acceleration from an accelerometer to inches per second of velocity. The velocity can be integrated to displacement. The reverse is not possible and only one integration of data is reliable. Integrating twice from g's of acceleration to displacement in mils is highly unreliable data and is not recommended.

L_{10} Bearing Life The L_{10} bearing life is an ISO standard that indicates the number of hours that 90% of a group of bearings must attain or exceed before initial stages of failure.

$$L_{10} = \frac{16,666}{RPM} \times \left(\frac{Rating_B}{Load_E}\right)^3 \times Hours$$

Where:

- $Rating_B$ is the dynamic load rating of the bearing in pounds.
- $Load_B$ is the actual radial and axial load that is incurred by the bearing in pounds.

Overloading a bearing with excessive loads, vibration induced loads, improper lubrication, and so on, all affect the bearing to the third power. Rotational speed has the same effect.

Looseness Also called mechanical looseness, is the condition where the entire mechanical structure can be loose from the base. Also the pillow blocks in a bearing structure can be loose from the machine, or there are improper fits in the rotating components of the machine such as bearings.

Mil/Mils A measure of distance. One 'mil' is 1/1000th of an inch (.001' equals 1 mil).

Misalignment Occurs when the driving and driven shafts in a mechanical train are not properly aligned. This misalignment could occur between a motor and pump for example. There are three basic forms of misalignment. Angular misalignment occurs when the shaft centerlines connect at an angle to one another. Offset misalignment occurs when the shaft centerlines are parallel, but not on the same centerline. Misalignment can also occur when a bearing is cocked on the shaft. It is also possible to have a combination of these factors occur. Misalignment is frequently associated with couplings and is indicated by high 1 and 2 x vibration amplitudes. 3x is also possible in some cases.

Natural Frequencies Natural frequencies are inherent in all objects. When a short duration application of energy is applied to the object, a hammer blow for instance, a series of natural frequencies can be excited causing the object to vibrate temporarily. An example of a natural frequency is the tone that is generated by a bell. In vibration, the term 'ring' is often used to describe a natural frequency that is excited by machine vibration or impact.

Oil Whip A condition where an oil whirl frequency coincides with a system natural frequency and the oil whirl becomes locked to the natural frequency. Oil whip remains at that frequency even if the machine speeds up. An example is start up. Oil whip can be destructive and is limited only by the bearing clearance.

Oil Whirl An oil film is present during normal operation of a hydrostatic bearing. The shaft typically is supported by a wedge of oil just in front of the load zone. If an outside force such as a sudden increase in speed of the shaft is applied to the shaft, the space that is vacated by the shaft during the event has additional oil pumped into it. Then it can actually drive the shaft. This new path can actually be a whirling configuration. If the whirling action does not return to normal, the resulting motion can become violent. Typically, any subharmonic frequency in the 40…48% of running speed range, and 50% of the bearing clearance or greater vibration displacement amplitude, is considered excessive and must be corrected.

Order An order is a speed unit that is 1*x* the running speed of the shaft. Orders of running speed are the unit that is typically used for variable speed machines.

Overall Vibration The measure of vibration through the entire spectrum from the high pass filter to the FMAX of the monitor.

Peak Used to describe amplitudes of vibration units where the maximum energy from vibration is displayed. Peak is used with units of velocity and acceleration.

Peak to Peak Used to describe amplitude where the total travel of the vibrating object is considered. Peak to Peak is used with units of displacement. Mils Peak to Peak, or mm Peak to Peak.

Radial A directional measurement perpendicular to the centerline of the machine rotating shaft. It can also be referred to as horizontal or vertical direction on machines with a horizontal shaft.

Resonance When a machine vibration frequency coincides either at or near a natural frequency of a machine component, the natural frequency can be excited into a resonant state. In this case, the actual vibration amplitude at the frequency can increase to from 10…30 to times greater than the original machine vibration amplitude. For this reason, resonance can be highly destructive.

RMS Root Mean Squared, A calculation resulting in a measurement amplitude that provides the square root of the average of a set of data samples that have been squared. RMS value is 0.707 times the peak value.

Synchronous Sampling When sampling synchronously time waveform samples are taken regarding shaft rotation, such as '32 samples per revolution'. In this mode, the speed of the machine determines the interval between samples. And the interval can change within a time waveform, or within one revolution.

The advantage to synchronous sampling is that it assures that the frequency of integer order multiples (1*x*, 2*x*, and so on) is always centered within their respective FFT bins. This centering assures the accuracy of the measurement. There is no bin leakage.

When synchronous sampling is applied the FMAX is defined in orders, the number of orders that are in the FFT, which is a function of the selected number of samples per revolution. For the 1444 Series Dynamic Measurement Module, the relationship is shown inTable 90.

Table 90 - Synchronous Sampling Example

Samples Per Rev	FMAX (Orders)
8	2.0
16	3.9
32	7.8
64	15.6
128	31.3

The FMAX, in Hz, for an FFT from any synchronously sampled TWF, is dependent on the speed of the machine at the time of the measurement. It can be calculated using the following equation.

$$\mathrm{FMAX(Hz)} = \frac{\mathrm{FMAX(orders)\ x\ RPM}}{60}$$

Vane Pass A centrifugal pump has a number of vanes on the rotor. Typically an odd number such as 5 or 7. As the impeller rotates, the alignment of the vanes to the outlet (volute) varies. As each vane crosses the outlet it obstructs flow as it does, which causes the pressure at the outlet to vary. This pressure variation repeats as each vane passes the outlet, effectively creating a pressure pulsation that occurs at a frequency equal to the number of vanes times the speed of the shaft. This frequency is called the vane pass frequency.

On a multi-stage pump, there is a different number of vanes on each impeller (stage). Consequently, there are vane pass frequencies that are associated with each stage of the machine.

Elevated vibration at the vane pass frequency can indicate turbulent flow, due to a possible obstruction or a problem with an input valve, or an alignment problem with the impeller.

Vibration Acceleration Acceleration measures the rate of change of velocity of the vibrating object/component, and is measured in g's of acceleration. Acceleration readings are typically used for higher frequency vibrations from sources over 5000 Hz. Conditions such as turbulence and cavitation are also much easier to detect with acceleration.

Vibration Displacement Displacement is the total travel of the vibrating object/component that is measured in mils (.001'), or in microns (Metric).These units of measure are native to Eddy Current Proximity probes, and are typically used in low frequency situations, typically speeds less than 10 Hz. (They are always used with proximity probes.)

Vibration Velocity Velocity is the speed of the vibrating object/component and is typically measured as peak speed or RMS speed, and the units are inches per second (ips) or mm/sec (Metric). Velocity measures are not highly frequency-dependent as displacement or g's acceleration, and are reliable between frequencies of 5…5000 Hz.

Notes:

Rockwell Automation Support

Use the following resources to access support information.

Technical Support Center	Knowledgebase Articles, How-to Videos, FAQs, Chat, User Forums, and Product Notification Updates.	https://rockwellautomation.custhelp.com/
Local Technical Support Phone Numbers	Locate the phone number for your country.	http://www.rockwellautomation.com/global/support/get-support-now.page
Direct Dial Codes	Find the Direct Dial Code for your product. Use the code to route your call directly to a technical support engineer.	http://www.rockwellautomation.com/global/support/direct-dial.page
Literature Library	Installation Instructions, Manuals, Brochures, and Technical Data.	http://www.rockwellautomation.com/global/literature-library/overview.page
Product Compatibility and Download Center (PCDC)	Get help determining how products interact, check features and capabilities, and find associated firmware.	http://www.rockwellautomation.com/global/support/pcdc.page

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