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the magazine for PdM & CBM professionals
june 2007

# Maintenance Under the Microscope

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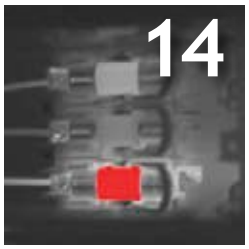
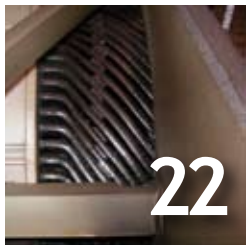










# Connecting the Dots

The feature article this month by Ian Barnard is a real eye opener.

I would venture to say that most people don't realize their entire suite of asset management strategies, including predictive maintenance and condition monitoring programs, gets dissected by the insurance industry. I didn't. To be honest, I had never put much thought into a connection between asset management and insurance. But now that someone has connected the dots for me, it makes perfect sense. The implications for maintenance and reliability are quite far reaching.

The companies that insure large industrial facilities willingly put themselves on the hook for an enormous amount of money, many times well into the billions of dollars. So, needless to say, those insurance companies are quite interested in the way companies they will be insuring manage their assets.

With premiums for insurance reaching into the millions dollars, companies can save significant amounts of money by having excellent maintenance programs and asset management strategies in place.

But the insurance connection goes further than just saving on premiums. Without a comprehensive, high quality asset management system, there is a chance the insurance companies will saddle a company with an unfavorable risk rating. This puts the viability of the entire company at risk. Not only will insurance be difficult to obtain, but banks will not finance companies that are not insured.

It's an excellent article and would be well worth the effort to see that people up the chain of command at your company get the opportunity to read it.

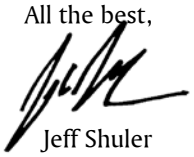
On another note, I recently returned from the 13th Annual Industry Advisory Board meeting of the Center for Intelligent Maintenance Systems. IMS is an industry/university cooperative research program based at the University of Cincinnati.

The Center is focused on frontier technologies in embedded and remote monitoring, prognostics technologies, and intelligent decision support tools. Their research pushes the envelope of predictive maintenance and is transforming the traditional maintenance practices from "fail and fix " to "predict and prevent" methodology.

You can get a glimpse of predictive maintenance's future and more information about IMS at their website www.imscenter.net

Thank you for reading. We hope you find something of value within these pages. If you have any questions, comments or suggestions that will make Uptime more useful to you, please let us know.

All the best,



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# Macro Management

# Examining The Hidden Relationship of Insurance, Asset Management and the Viability of Your Business.

*by Ian Barnard*

It may surprise you to learn that the global insurance industry is vitally interested in the Asset Management systems that are implemented at industrial sites like yours. While these systems are usually implemented to protect and manage plants and equipment, often the impact of these systems on the corporate financial model are overlooked. The type and effectiveness of the asset management system can affect the company's viability in terms of insurance coverage, which affects capital raising and risk profiling. This article discusses the operation of the insurance market and the impact that asset management systems have on both the insurer and the insured.

## Background

The insurance industry is a global business centred in New York, London, Zurich and the Bahamas. Worldwide, insurance premiums in 2004 exceeded $3 trillion. In the United States, which accounts for about 34 percent of the world's insurance business, premiums total in excess of $1 trillion. The insurance industry is generally viewed as three distinct segments, property/casualty, life and health. The property/casualty part of the industry provides insurance for cars, homes and businesses. The term "casualty" dates back to the time before the 1950's when property/casualty insurers were two distinct kinds of insurance companies, with casualty or liability insurers covering losses that resulted from casualties and property insurers covering damage to or loss of property.

Property/casualty insurance can be broken down into two major categories: commercial lines of insurance and personal lines. Personal lines, as the term suggests, includes coverage for individuals –- car and homeowners insurance. Commercial lines, which account for more than half of U.S. property/casualty insurance industry premiums, include the many kinds of insurance products designed for businesses.

Commercial insurance performs a critical role in the world economy. Without it, the economy could not function. Insurers essentially protect the economic system from failure by assuming the risks inherent in the production of goods and services. This transfer of risk frees insured companies from the potentially paralysing fear that an accident or mistake could cause enormous losses or even financial ruin. As a consequence, the Insurance industry's interest in asset management systems is very real and very focused.

Managing risk requires an in depth knowledge of the systems and processes utilised by a large variety of businesses. The North American Industrial Classification System identifies some 1,170 different industries according to the processes used to produce goods or services. In the U.S., there are some seven million business establishments, each of which employs one or more of these processes and each of which buys some kind of insurance.

AIG and all comparable insurance companies collect plant and system root cause failure analysis data, including type faults which may occur in established and new plants and equipment. This is the data used as reference material for this article.

## The Way the Industry Works

**Capacity** - Each insurance company has a finite level of risk (financial reserves) that can be allocated to a given project or industrial complex. This risk is termed Capacity, and the magnitude of the capacity varies with the size of the company in terms of its financial reserves, and the risk profile that the insurer currently has on its books. Companies vary their capacity in proportion to the level of risk already on their books in a given industry sector or geographical region. In order to secure the full insurance cover required by large businesses, insurance risk is spread over a number of insurance companies.

Every policy has a lead insurer, who will negotiate terms and conditions with the insured or their agents (brokers). The lead insurer takes the first and highest probability of risks for the site, and allocates a certain capacity to that risk, at a certain premium. Other companies are then invited to participate in the risk, and their involvement will be initiated if the losses for an event exceed a predetermined level. In this way, the insurance book is filled for a given client.

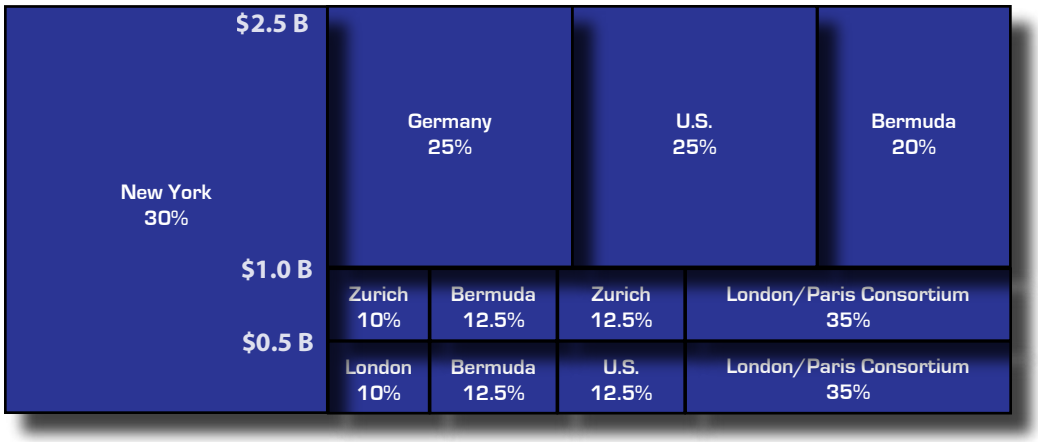


Figure 1 - Typical Insurance Placement Grid

The globalisation of the insurance industry can better be understood by examining Figure 1, which is a typical (simplified) portfolio of insurance providers for a major industrial asset.

The diagram depicts the levels of insurance cover and the companies that offer that cover. The Y axis depicts the total level of cover for the industrial plant, which in this case is $2.5B. The X axis is compiled from companies who agree to cover certain percentages of the risk up to defined limits. In this example, a consortium of companies will provide cover to this risk up to the first $500 M. A different consortium of companies will provide cover for the second $500M, taking the combined cover to $1B. Likewise, the final $1.5B cover will be provided by a third and different set of companies. Each company will charge a premium based on their level of exposure, and where they are placed in that cover, be it the first, second, third or higher tier. Obviously, the companies in the first tier of cover are exposed to the greatest risk, and so the price for this tier is higher than for subsequent tiers. The first New York company, who is providing cover over the whole risk, is the lead insurer. However, this company will also offset its exposure by utilising the services of one or more reinsurance companies, effectively "buying" more capacity and divesting some of its risk to third parties. Other members of each tier may also utilise reinsurance.

The point of this diagram is that there are a large number of decision makers involved in determining not only the cost of insurance to a particular project, but also deciding on their involvement in the risk. This means that, for a large asset, there are many people and organisations that need to be convinced of the soundness of the insured's Asset Management strategies. Unless all parties are satisfied with the risk, the facility will not be insured.

**Typical Plant Policy Structure -** Insurance for a large industrial plant is generally structured so that relatively minor losses are born solely by the company which owns the plant. For example, the typical structure for a power generation facility is that only losses above $5M are claimable, and interruptions to business of less than 45 days are not insurable. This fact alone has a significant impact on the way insurance companies view asset management systems. In general, from a loss viewpoint, asset management systems which protect small and/or low impact systems are not on the Insurance radar.

Premiums will generally run at between 1% and 2% of the insurable loss in politically stable economies of the first world. For a typical large power generation facility with a capacity of 2000 MW, the premium will be somewhere between $2M and $4M.

**Loss Probabilities** - Loss probabilities for a risk are calculated based on the likelihood of certain events. Following are the standard loss definitions with examples for a power station.

***NLE - Normal Loss Expectancy*** **-** This type of loss is based on a normal event that has a reasonably high likelihood of occurring in a plant. While the event will affect production, the scenario is reasonably common and in general is not covered by insurance given that the policy deductible will exceed the cost of rectification.

| Plant | Boiler | Turbine |
|---|---|---|
| Event | Mild Furnace Explosion | Steam Turbine Incident |
| Property Damage | $250,000 | $2.7M |
| Business Interruption (BI) | 5 Days | 30 Days |
| BI (% Production Affected) | 25% | 25% |
| BI Cost ($) | $250,000 | $1.5M |
| TOTAL ($) | $500,000 | $4.2M |

***PML - Probably Maximum Loss***
This type of event is not expected to occur under normal condition, but experience shows that it has a finite probability of happening over the lifetime of the asset. The losses associated with this type of event are large, and the values are calculated assuming that all installed protection and asset management systems function as designed to limit the impact of the event.

| Event | Low Boiler Water | Fire Resulting in Loss of Rotor |
|---|---|---|
| Property Damage | $25M | $36M |
| Business Interruption (BI) | 180 Days | 365 Days |
| BI (% Production Affected) | 25% | 25% |
| BI Cost ($) | $9M | $18M |
| TOTAL ($) | $34M | $54M |

***EML – Expected Maximum Loss***
This is the loss that can be expected to occur under the worst possible conditions for a plant and where the protection systems designed to minimise the loss fail completely. The loss is limited only by passive protection such as plant spacing. This is the largest of the possible losses, and the values associated with this type of loss are greatly affected by inherent risk mitigation such as building and processes separation etc.

| Event | Boiler Explosion | Fire Resulting in Loss of Rotor |
|---|---|---|
| Property Damage | $160M | $85M |
| Business Interruption (BI) | 30 Months | 30 Months |
| BI (% Production Affected) | 25% | 25% |
| BI Cost ($) | $45M | $45M |
| TOTAL ($) | $205M | $130M |

Despite the significant level of research and technical analysis that is applied to determine the risk for a given project, often the

decision by a particular insurance company to be involved (and effectively investing in) a project is not fully quantifiable. Insurance professionals are mainly financial specialists, and are not engineers. Traditionally, they were concerned with questions of political and market stability, natural perils such as flood and earthquake, facility fire protection and a host of other risks to a project or site. While the actual numbers generated statistically to evaluate a risk profile are important to a decision, there is a certain amount of qualitative analysis that occurs, and this is best described by intangibles such as the experience of an underwriter in a particular risk field. Not only do the numbers for a risk have to be right, but the "feel" of the risk has to be right.

In recent years, the profile of the commercial and industrial insurance industry has developed to include risks far removed from the traditional insurance cover.

## Insurance Types

There are almost as many types of insurance as there are insurance companies. By type of insurance we mean the types of risks covered. As stated previously, the types of insurance can be divided into Personal and Commercial. For our purposes, we will concentrate on the insurance that is concerned with commercial business assets.

**Property Insurance -** Traditionally, insurance cover was primarily related to the destruction of a facility or assets by either fire or natural disaster. This type of insurance is known in the industry as Property cover, and as the name suggests it relates principally to the replacement value of real property. Impacts here include political, geological, fire, explosion and other major issues that can destroy or severely damage the physical plant as a whole. The insurance industry is well versed in analysing and assessing the risks related to these issues, and various statistical tools are used to quantify the risks from these aspects of an operation.

**Boiler and Machinery Insurance (BM)** - This type of insurance, despite the name, relates to the risk of a plant being destroyed or rendered inoperable from any source. In effect, it is Reliability Insurance for the plant, and takes into account aspects such as premature failure of major systems, the destruction of plant from fire, design defects and a myriad of other causes. It is interesting to note that stupidity is insurable, but laziness is not. By this we mean that damage due to honest mistakes by operators or maintainers will be covered by the policy. Damage resulting from poor maintenance and inspection practices, plant overload, operation outside design parameters and a number of other inferior practices are not covered by these policies.

**Business Interruption Insurance (BI)** - BI Insurance covers the loss of profit risk to a project or site from an interruption to the supply chain. It is entirely possible for this type of insurance risk to exceed the risk from all other sources. For example, the destruction of a compressor in a petrochemical plant from a catastrophic rotor failure may require a complete compressor replacement costing approximately $1M. However, if the compressor is a purpose built machine, with a replacement time of say 6 months, the loss of profit to the organisation may exceed the cost of replacement by orders of magnitude. Business interruption insurance has a critical interest in the quality of the asset management systems applied to processes and businesses.

BI insurance does not include damage related to wear and tear, or plant and equipment that is not maintained and inspected to industry standard. This aspect has connotations for asset management systems designers, as the term "industry standard" has no global reference, and varies from industry to industry and country to country.

## Risk Ratings

Asset management system design and implementation has a crucial impact on both Boiler and Machinery and Business Interruption insurance. Insurance companies assess the standard of asset management applied to a particular site, and decide not only on the cost of such cover, but on the provision of such cover based on engineering reports compiled on the project.

Engineering reports are produced by qualified and experienced engineers who visit and review a site or project, and inspect the systems in place at the site to minimise risk. The systems review includes human and technical systems, and looks at both the design of systems and the level of effective implementation of the system. At AIG and in other major insurance companies, plants are categorised and rated as Excellent, Good, Fair and Poor risks based on a quantitative and qualitative assessment of the risk management systems implemented at the site. The headings used to evaluate the risks for an engineering study are: Location, Plant Layout, Fuel Storage and Handling, Process Control, Reliability, Maintenance, Inspection,

| Process Control | Grading |
|---|---|
| Control rooms & systems | Grade 0: Process control DCS separate from critical ESD and trip systems, hard wired or Triple Modular Redundancy system, input voting system.<br>Grade 1: As above, but DCS and ESD are linked and dependent.<br>Grade 2: Manually actuated automatic shut down of process unit, with satisfactory level of auto shut down of unit packages such as rotating equipment.<br>Grade 3: Mix of old and new systems but not integrated. Unmanned stations.<br>Grade 4: Local controls only |
| Boiler | Grade 0: Fully automatic combustion controls on all burners. Alarms interlocks. Drum level trips, 2 out of 3 voting system, separate level indication in control room ( CCTV). Double isolation valve on gas or oil feed.<br>Grade 1: Some manual controls on non critical processes<br>Grade 2: Strict purge sequence, drum level trip, burner management system<br>Grade 3: No burner management system<br>Grade 4: No burner management system, no combustion control, manual light up |
| Turbine | Grade 0 : Double overspeed protection system, remote condition monitoring, on-line protection facility. Fully automated governor control with stress control (ST). On line gas analysis for GT<br>Grade 1 : On line vibration measurement with trips and trending<br>Grade 2 : On line vibration and eccentricity with trips. Single overspeed protection system.<br>Grade 3 : On line vibration and eccentricity without trips.<br>Grade 4 : No condition monitoring system (ST). |
| Generator | Grade 0: Generator core monitoring, on line partial discharge monitoring, phase and balance protection, reverse power protection<br>Grade 1: As above but no online PD<br>Grade 2: See best practice page 23 (minimum practice). |

Figure 2 - Excellence Grades for Power Station Process Control Element

Operation, Fire Protection and Risk Management. The underlined topics are critically dependant on asset management technologies.

Figure 2 is a sample of grading standards for the Process Control element in a power station as determined by AIG. Grade 0 is excellent, Grades 1 and 2 are Good, Grade 3 is Fair, Grade 4 is Poor.

From these ratings, an overall risk management profile and rating is developed.

Facilities placed in the Excellent category have no trouble obtaining insurance cover, and will have the lowest premiums and the least stringent policy conditions. These companies will have significant bargaining power in the insurance market and may even be able to dictate their own terms to the global insurance market.

Industrial sites placed in the Good category will obtain insurance on terms that reflect the lower rating. Premiums are likely to be higher and insurance companies may impose restrictions and riders that limit the level of cover. In many cases, additional risk management and asset protection systems will be specified by insurance companies before cover is applicable. In some cases, these additional systems can cost millions of dollars.

Facilities in the Fair and Poor categories will find insurance difficult to obtain, and this will affect the viability of the operation as a whole. Banks will not finance companies who cannot obtain insurance. If insurance companies are at all interested in being involved in the facility, a whole range of management systems may be mandated prior to cover being negotiated, and the premiums for the insurance will reflect the poor risk profile.

## Plant Failures

From an asset management perspective, it is necessary to understand the root cause of plant failure in order establish effective condition monitoring and asset management strategies. AIG and other organisations have conducted considerable research into the reasons for plant failure. Not surprisingly, many plant failure studies into mechanical and electrical equipment failures reveal similar system failure root causes, which are indicated in Figure 3 and 4.

The data for Figure 3 has been taken from a number of industries of varying ages and capacity factors. In general, the data indicates that human intervention in the machine in the form of lubrication quality and contamination control is the primary factor in the longevity of mechanical plants. The data also indicates that lubrication analysis is the prime control method of determining mechanical plant condition, particularly in rotating mechanical plants.

Figure 4 on the following page depicts the systems within electrical plants that have been the principle failure cause for those plants. It is interesting to note that mechanical systems within electrical plants are the prime failure areas. The obvious connotation from this data is that the complete suite of both mechanical and electrical testing and condition determination regimes must be applied to all plant types. The complimentary nature of the condition assessment techniques requires that the systems used to record and report on plant condition need to be fully integrated, and contain data, analysis and prognosis from disparate CM tools to give a complete description of the plant condition.

With these failure modes in mind, to obtain an excellent rating from insurers the as-

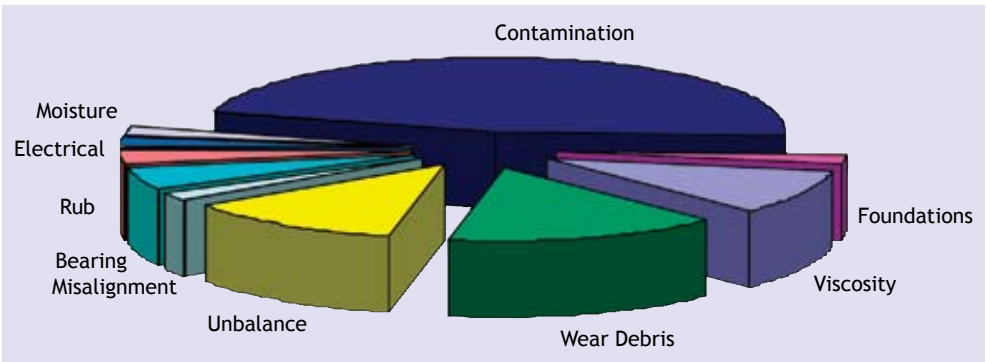


Figure 3 - Primary Causes of Mechanical Equipment Failure

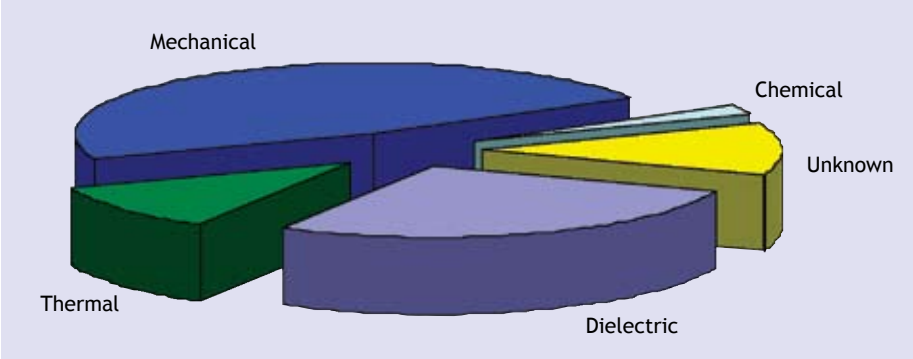


Figure 4 - Primary Causes of Electrical Plant Failure

set management and condition monitoring technologies that are required for a utility include:

Mechanical:

- Online vibration spectral analysis of main plant including turbines, generators, feed pumps.
- Online vibration measurement and trips on auxiliary plant.
- Spectro chemical and Microscopic wear debris analysis of all main lube oil systems.
- Infrared thermography of all main plant including turbine casings, boilers, auxiliary plant bearings.
- Eddy current testing of condensers
- Time of flight ultrasonics (TOFD)
- Eddy current & remote field testing (EC & RFT)
- Remote visual inspection (RVI)
- Replication
- Bore oxide ultrasonics
- Mini creep testing
- X Ray analysis

Electrical:

- Dissolved gas analysis (DGA)
- Dissolved moisture
- Oil quality (acid, interfacial tension, dielectric strength, dielectric dissipation factor)
- Particle count
- Dissolved metals and sulphur
- Furans
- Dielectric loss angle
- Insulation resistance
- Dielectric dissipation factor
- Insulation recovery voltage
- Partial discharge
- Time reflectometry
- Radio frequency emission
- Transfer function

Risk Assessment Tools:

- HAZOPS - Hazard & Operability Studies
- QRA - Quantitative Risk Assessment
- WRAC - Workplace Risk Assessment and Control
- RCM-RA - Reliability Centred Maintenance Risk Assessment

It should be carefully noted however that the existence of these systems at a site is **not** the determining factor for a risk management engineer. The critical aspect is how these systems are utilised in the asset management planning cycle. All of the above systems require analysis and prognosis development by humans, and the qualifications and experience of the people conducting the testing and performing the analysis are of crucial importance to the insurance industry.

## Impacts for Asset Management System Design

The insurance industry is both a friend and foe to the designers and researchers involved with asset management systems.

Firstly, the scale of the asset management systems from an insurance perspective is company and/or plant wide. The amount of insurance deductible charged means that companies effectively self insure for the first $5M, and cover their own business profit losses for the first 45 days. Asset management systems which protect plants at levels less than these figures are generally not important to the insurance industry. However, risks that are placed in the Excellent and Good ranges by the insurance Risk Consultant because of superior asset management systems can often help companies effectively negotiate the deductible and BI exclusion period.

Secondly, to obtain Business Interruption insurance, insurance policies mandate that an insured company follows a number of guidelines, including adherence to the equipment manufacturer's operations and maintenance schedules and compliance with industry normal practice for the asset. This has the effect of greatly limiting the scope for innovation to be applied in asset management techniques and methodologies from an insurance premium reduction perspective.

On the other hand, the insurance companies also mandate that traditional asset management and condition monitoring methods be applied. For example, the application of thermography, oil and vibration analysis, dissolved gas analysis, partial discharge testing, the full suite of non destructive metallurgical techniques and a host of others are all required by business interruption insurance providers to protect the assets.

The message here is that's its is necessary to utilise proven Condition Monitoring techniques to manage assets, but don't rely on anything too innovative and unproven.

Risk management engineers are usually assigned to a client for the duration of the insurance relationship. These people are the front line of the technical interface between the insurance portfolio developed for the client and the asset management processes implemented by the client. They are in a position to suggest additional asset management technologies that would benefit the client's business and reduce the risk of breakdown and business interruption.

The important point here is that a direct link between emerging asset management technologies and the potential market for those technologies exists through the insurance chain. This link has been largely unutilised by the asset management community. If asset management researchers and technology developers wish to get their work acknowledged and into the market, the insurance industry in general, and the risk management consultant in particular, can be a valuable ally, provided that the technology has demonstrable advantages to both the insured and insurer.

While it is not possible to quantify the effect that the installation of a particular technology at a site will have on insurance premiums, it is possible to move a risk from a Fair or Poor risk rating to a Good or Excellent rating, which will improve the company's insurance profile. In some cases, the implementation of additional risk management systems will determine the viability of the whole project.

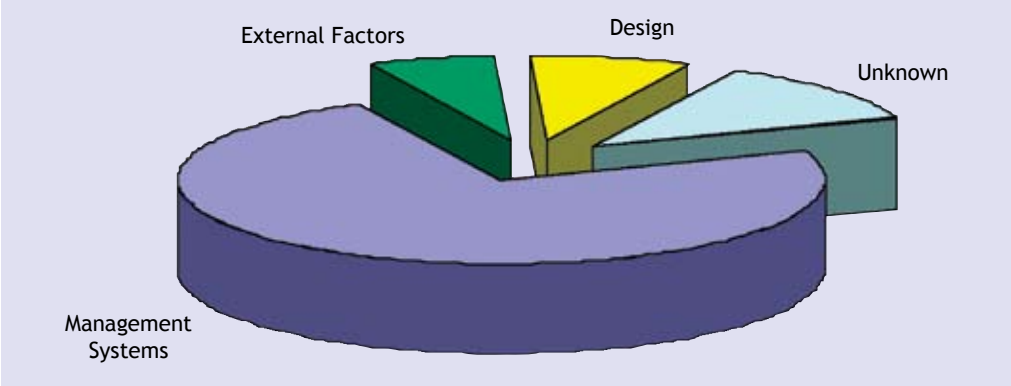


Figure 5 - Upper Level Systems Failure Analysis

## Human Element of Asset Management

Almost everyone who works in the Asset Management field knows of examples where effective asset management technologies have been implemented only to fail in the field due to a lack of integration with existing systems and processes. The assessment by insurers of asset management systems does not rely on the presence of installed technology, or the levels of capital investment by a company in asset management technologies. The assessment of the asset management standard of a plant is based solely on how the technologies are utilised to reduce the risk of plant failure.

The graph in Figure 5 depicts the major causes (from a systems failure viewpoint) of insurance losses experienced in the US in the power industry in 2005. While all these losses can be analysed to identify the aspects of the mechanical and electrical systems that failed, the losses have been further analysed to determine if a failure in asset management systems contributed to the level of loss incurred.

In Figure 5, Management Systems denotes all types of risk management systems, including HAZOP studies, plant inspections, condition monitoring systems and routine plant inspections. External factors include incidents that occurred outside the plant over which the plant management had no control.

The data indicates that approximately 75% of losses resulted from a failure to implement or sustain targeted and robust plant asset management systems.

The data was further analysed to ascertain the most appropriate loss prevention strategy that could have been implemented to reduce that particular risk of failure in the plant, which is shown in Figure 5.

This data indicates that approximately 50% of the loss values could have been avoided or reduced if the appropriate plant condition monitoring systems had been implemented. A further 30% of the losses could have been favourably impacted by the application of simple plant inspection regimes. It is interesting that only 6% of the losses incurred could not have been mitigated using technologies available today, and 8% of the system failures at the root of the loss is unknown.

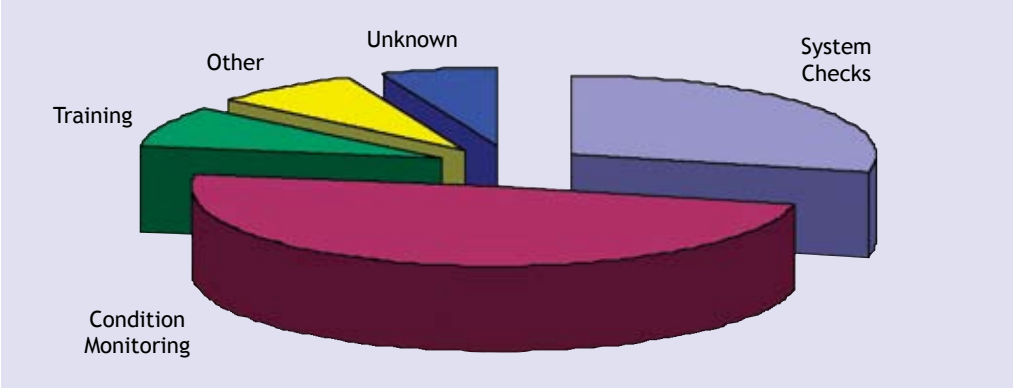


Figure 6 - Possible Loss Prevention Strategies

This data, plus large amounts of anecdotal evidence, suggests that it is not the development of additional asset management technologies that will have the greatest impact on the levels of losses in this industry. Rather, it is the development of systems which are fully integrated into the plant operations and maintenance process that will reduce the levels of plant downtime. This is not to say that the development of new techniques in plant condition monitoring are not important or required. Many companies and industries are in dire need of specialist testing techniques to enable product improvement and to increase reliability. However, Insurance companies are primarily interested in systems that deliver real reliability and availability returns on a plant wide basis.

## Conclusions

The information presented above includes some salient points for Asset Management system designers:

- Rightly or wrongly, the insurance industry has considerable influence on the types of asset management systems implemented in a commercial enterprise. This fact can be used to advantage by asset management system developers.
- Insurance underwriters rely almost completely on the information supplied by relatively few risk management engineers to determine the adequacy of the asset management systems installed at a facility.
- Generally, the insurance industry does not financially support the development of asset management technologies. The industry is principally geared towards the implementation of those technologies and systems which are proven to improve the reliability of plant and equipment.
- While it seems incongruous with the previous point, the insurance industry is vitally interested in technologies which reduce the risk of business interruption to plant and processes.
- Insurance company Risk Management Engineers can be a valuable aid in implementing innovative asset management and plant condition monitoring technologies.

*Ian Barnard started his career as an apprentice electrician at the tender age of sixteen years. It quickly became apparent that this career choice involved hard work so he managed to gain qualifications in electrical engineering and physics. Working for Pacific Power as a power station engineer and subsequently as a plant condition monitoring specialist, Ian was also a non executive director of the Centre for Machine Condition Monitoring at Monash University. After holding several management positions, Ian left Pacific Power International as General Manager/Operations after 23 years, and became a Senior Risk Management Engineer with American International Group. Ian is a keen sailor and is a qualified Yachtmaster Offshore Instructor.*

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# Don't Be (con)Fuse(d)

## Distribution Fusing and Infrared Thermography

*by Dave Sirmans*

In over three years with my current employer, I have yet to perform an electrical infrared (IR) scan where fuses weren't present in at least a few devices, and very few scans in which I didn't find an anomaly involving a fuse, or its associated hardware. There are literally dozens of different types of fuses in use in industrial and commercial electrical applications, with as many methods and types of hardware used for interfacing them to the circuit they are designed to protect.

Any thermographer who has performed an electrical IR scan has no doubt seen the infamous "loose fuse clip" diagnosis for an anomaly found in a fused circuit.

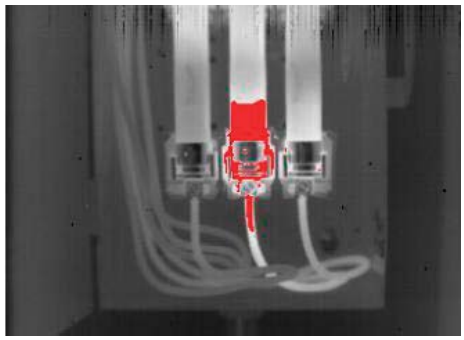

Figure 1 - Loose Fuse Clip?

Almost as often, however, there is another culprit, and sometimes several, responsible for abnormal heating in a fused connection. I know I have personally blamed load or thermal transfer for heating a fuse barrel, when something else was amiss.

Misapplication, incorrect installation and lack of familiarity with types and classes of fuses result in thermal anomalies presenting themselves within fused circuits. The same lack of understanding regarding proper fuse application also results in misdiagnosis of the anomalies when they are found by otherwise qualified and experienced thermographers. I have done it myself, I have seen my co-workers do it, and I have seen it in the reports of some of our competitors.

When performing an IR scan on fused devices, there are a number of factors to consider. Fuses aren't as complex as the Space Shuttle booster engines, the Rubix cube, or even online dating, but there are still a few things one must know about the types and ratings of fuses that contribute to how they heat as their load increases. We'll discuss this relationship in this article.

Depending on one's perspective, the word "fuse" may bring to mind any number of differing types of devices. Some of us remember the old style screw-in fuses that used to be common in residential electrical distribution. Then there are those cute little plastic ones my truck has, and the small glass ones my old truck had. And I am sure you can think of quite a few more. However, what we usually see in electrical Predictive Maintenance is different. I have personally seen large, elaborate limiters on medium voltage gear, all the way down to smaller, KTK type fuses in control panels. The one thing they have in common is that they all carry current, which, by its very nature, produces heat. I have seen thermal anomalies in all of these different types of fuses.

Main switchgear equipment, particularly if it's a bolted pressure switch, can often contain fuses. These fuses are the larger, bolt in type, and have a high current interrupting capacity because they can be exposed to high levels of current in the event of a fault. More often than not, these fuses may be time delay fuses. This simply means that they are designed to hold a fault current for a few seconds, rather than clear immediately upon sensing a current above their rating. This feature is important as it relates to IR, for reasons we'll discuss in a minute.

Downstream from main switchgear equipment, there are other distribution devices that may contain fuses. Typically, disconnect devices have them, such as those powering fan motors and HVAC units. These fuses may, or may not, be time delay types, depending on the design characteristics. They are almost always physically smaller than those found in main switchgear equipment, and will have lower current interrupting capacities.

Moving further downstream, we find midget fuses, as they are sometimes known. Type KTK and the like are the shorter, smaller fuses most typically found inside control panels. Again, they are smaller than the ones upstream, and sometimes have time delay characteristics. It's important to note that just because these types of fuses are usually carrying less current than a fused disconnect device or main switchgear equipment, that doesn't mean there will not be abnormal heating in them. My experience has shown that all too often,

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these smaller fuses might be overlooked entirely, and usually because no one expects a 3 amp fuse to get very hot. Well they can.

I mentioned time delay fuses earlier, and they are one of the types I find most often at the sites I visit with my trusty Mikron. The other two types I frequently see are Fast Acting, and Semi-Conductor. Here's where this all gets a little freaky...

Time delay is synonymous with slow-blow. Time delay fuses are usually applied in circuits where high current may be anticipated for short periods of time, without being the result of a fault. Motor applications are a common use for time delay fuses. These fuses will hold current levels above their rating for periods of time longer than a fast acting fuse would. For some insight on this, let's talk about the differences in their construction.

A fuse consists of an element, the insides, surrounded by a filler, most often a silicon based sand. The filler is enclosed in the body of the fuse, the part we see on the outside, which is typically fiberglass or a similar material. The element is soldered or

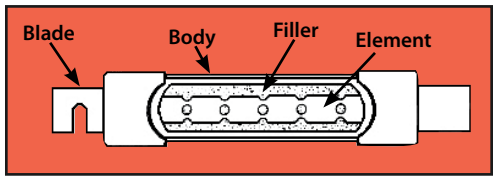


Figure 2 - Anatomy of a Fuse.

welded to the outside, conductive part of the fuse, whether they be blades or ferrules. In case you didn't know, the round ones are called ferrules. Sounds like the plumbing piece, but it's not the same. The element is a conductor, like a wire or bus section, except that it's calibrated, meaning that it's sized to withstand a certain amount of current, for a certain amount of time, before it overheats and ultimately melts. Those of us who have used a fuse that's too small in our car have

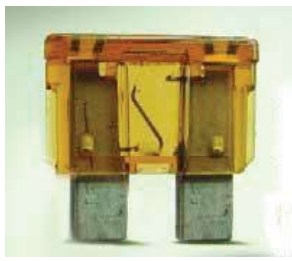


Figure 3 - Blown plastic fuse from a car.

seen the melted elements in either the little glass or plastic fuses.

In ordinary use, a fuse element will obviously heat. When the amount of current is below the fuse rating, the heat is transferred to the filler material, and dissipated into the surround air through the fuse body. The higher the applied current, the higher the surface temperature of the fuse body. Knowing this can alert a thermographer to circuits that are approaching overload, or circuits where the load is imbalanced. If the surface of one of the fuse bodies is higher than the other two, assuming three phase distribution, then something is not right. But what is it?

Earlier I mentioned differences in fuse types, and named a few. I'm not picking on anyone here, but there seems to be a school of thought among maintenance personnel that a fuse-is-a-fuse-is-a-fuse. This is simply not the case. If a particular type of fuse - for example, a semi-conductor fuse - is designed to hold X-amps for Y-milli-seconds, would it's element be constructed of the same type and density of material as, say, a time-delay fuse, designed to hold X+50 amps for Y+3 seconds? Now is the time to shake your

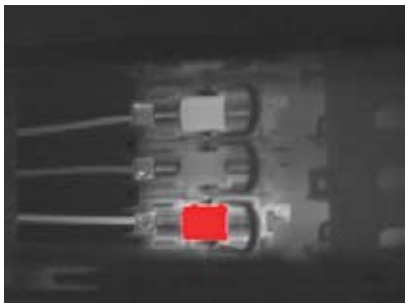
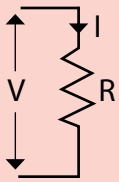


Figure 4 - What is the root cause of this anomaly in the fuse?

head. Not only would the element material be different, but the fuse body might also be different. If the element is designed to hold more current longer, or less current for not as long, will its electrical properties be different? Now you should nod your head.

Current squared times resistance is the formula for heat.




$$P = VI = \frac{V^2}{R} = I^2R$$

If all other circuit parameters remain constant, and resistance changes, temperature changes. This is Level I stuff, I know, but it applies to fuse application. Egg-headed (in a good way) engineer types who design fuses and power systems know this. That's why when you buy that disconnect, it says "Use Type whatever, Class thus-and-such fuses". To many a plant electrician, 100 amps is 100 amps. "This fuse has a hundred on it, it'll work". Not necessarily. And if that new kid on third shift put a semi-conductor fuse where a time delay fuse goes, when I look at it with my trusty Mikron, it's going to look different. And in my ignorance, I might call it a bad fuse clip. Don't laugh, I've seen it.

Also, when a fuse is exposed to currents above it's rating, but for a period insufficient to completely melt the element, the element can be damaged, or partially melted, and therefore not as conductive as it was before it was damaged. This means that it's resistance has changed, which again, means it will look funny in your camera. It's not a bad fuse clip, it's not "load". The fuse is damaged, and needs to be replaced. Or it might end up like Figure 5.

The bottom line is simple. We spend hours wielding these cameras around, and we've seen it all. Your span is set kind of low, you see a bright object in your FOV, and when you slow down and look, it's a fuse. They're

Figure 5 - A fuse that's seen better days.

supposed to be hot, right? Yeah, sure, phase C is hotter. You look closer, must be the fuse clip. You look a little more, and decide that phase C has a smidge more load than the other two. When John the engineer gets our report, he tells Joe the electrician, who whips out his clamp-on and checks. He calls you a nasty name, since you're long gone, and obviously didn't know what you were talking about, and goes to the break room for Vienna sausage and crackers. Three weeks later, that misapplied or damaged fuse clears when it shouldn't have, and Line 3 goes down. So much for the benefits of predictive maintenance.

When we see a fuse that's hot, we owe it to ourselves and our company or customer to apply some deeper knowledge. The manufacturers of these dandy little protective devices have reams upon reams of technical data on their websites to help us understand how these products behave, and how they should be used. For all we know, we might be the only guy who ever opens that panel except when something breaks. Take note of how different types and ratings of fuses look in your camera, and when you see a hot fuse, make sure it's not the wrong size, type or class. Sure, it's hot, but why?

Figure 6 - Don't assume that the fuse in place is the correct size, type and class.

*This article was originally delivered as a presentation at ThermalSolutions, one of the premier learning events in the world for infrared thermographers. For more info on the conference go to www.thermalsolutions.org*

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# What's in the Box?

## Evolution in Gearbox Technology and Lubrication Practices

*by Kevan Slater*

Robust and reliable, most gearboxes/reducers are designed to provide many years of service with typically very limited maintenance requirements. Continuous improvements in equipment design have resulted in the reduction of gear size while increasing gearbox efficiency. These improvements translate into higher power requirements, higher speeds, and higher stresses, all of which place added importance on the quality and performance of the lubricant. While incorrect operating temperatures, misalignment and resonant vibrations typically provide a slow path to component degradation, lubrication condition and lubricant contamination will lead to rapid and unnecessary wear generation and the eventual failure of the unit itself.

Unfortunately, even with all the effort and expense to achieve these increased efficiencies, the leading cause of premature failure in most gearboxes is still fluid contamination and/or the loss of essential lubricating properties. It is inevitable that rotating components in the gearboxes will eventually fail. However, numerous documented studies have shown that reducing contaminant levels and maintaining the physical and chemical properties of the lubricant will greatly increase the reliability and the service life of the components.

### Lubrication

Equipment designers/manufacturers (OEM) and operating manuals reflect the minimum lubricant requirements for operating their equipment. These requirements include the type of oil, the oil's specifications and typical operating temperatures. These requirements vary based on the three major classes of gears:

- Spur, bevel, helical, herringbone and spiral
- Worm
- Hypoid

The differences in gear teeth actions and materials influence the formulation and properties of the lubricants required; while the size, speed, load, and temperature of the bearings typically determine the viscosity requirements. During operation, the heat generated by metallic friction between the tooth surfaces and by fluid friction of the oil, will cause the temperature of the oil to rise. The final operating temperature is a function of both this temperature rise in the oil and the ambient temperature surrounding the gear case. Thus, a temperature rise of 90°F (50°C) and an ambient temperature of 60°F (15.6°C) will produce an operating temperature of 150°F (66°C). The same temperature rise at an ambient of 100°F (38°C) will produce an operating temperature of 190°F (88°C). For gear sets equipped with heat exchangers in the oil system, both the ambient temperature and the temperature rise are less important since the operating temperature of the oil can be adjusted by varying the amount of heating or cooling.

The lubricant is thrown from the gear teeth and shafts in the form of a mist or spray. In this atomized condition, it is exposed to the oxidizing effect of air. Bearing operating temperatures may be increased above normal by heat conducted to the bearing from a hot shaft, or spindle, or by heat radiated to the housing from a hot surrounding atmosphere. This high temperature will also increase the rate of deterioration due to oxidation. Sludge or deposits formed as a result of oil oxidation can restrict oil flow or result in an insulating deposit on the internal surface of the gear case to decrease cooling and cause further increases in the rate of oxidation. Allowing the gear case to become covered by material such as cement dust, wood chips, debris, or even multiple layers of paint will create an insulating effect allowing the internal temperatures to rise well above the OEM planned range.

The viscosity of the lubricant decreases drastically with this increasing temperature. Even though the lower viscosity reduces the churning losses somewhat, it also decreases the ability of the lubricant to fulfill its main function, namely to separate the components in contact by building an Elasto-Hydrodynamic (EDH) film. Without this separating film the components would score, wear out, pit and fail within a short period.

### Temperature Control

OEM's recommend an optimum operating viscosity for their equipment in order to perform and operate within the original design parameters. In many cases, maintaining an operating temperature that achieves the OEM recommended viscosity becomes the responsibility of the end user. The temperature of a gearbox in operation will increase until the heat balance of the internally generated heat, plus the external imposed heat, reaches equilibrium with the dissipated heat. If this heat cannot be dissipated by radiation through the

gearbox housing surfaces and through convection to the surrounding air, surrounding structures and components, then an alternate cooling system (heat dissipation) should be considered.

Heaters and/or coolers remain the standard methods of maintaining the designed system operating temperature of the lubricants. However, many gearbox/reducer manufacturers do not install them on the units as standard equipment. If supplied, heaters with regulating valves in water/oil heat exchanger and temperature switches for air/oil system components are used to control the operating temperature range of the lubricant. After initial set-up of these components, the condition can be monitored effectively for operation by the ability to maintain the correct operating temperature settings. Regulators that are not operating correctly or that are set improperly will result in an unacceptable operating viscosity range.

Monitoring, recording and trending temperatures of the lubricant is a critical link in maintaining lubricant and equipment health. If the gearbox is not thermo-coupled to a display panel or computer system, then a permanently installed thermometer or a non-contact temperature measurement can be obtained by using a portable style infrared instrument as shown in Figure 1. To assist in monitoring peak temperature measurements, tell-tale temperature strips can also be used to record temperature extremes.

Figure 1 - Infrared Thermometer

## Lubricant Volume

Typical design methods for applying lubrication to the system components application systems depend mainly on the running speed of the bearing. These methods can vary but usually follow these guidelines:

- Oil bath and splash systems - low and medium speeds.
- Circulating systems - medium speeds.
- Spray or mist - high speeds.

In oil bath or splash systems, the oil level in the gearbox is maintained so that the teeth of the bottom gear just dips into the oil. Alternatively, a pressure circulating system may be used in which oil is sprayed on the teeth close to the point of engagement and is re-circulated either directly from the bottom of the gearbox or by way of the oil tank.
In an oil bath or splash system, overfilling a gearbox sump can be just as damaging as under-filling. Overfilling may cause air entrainment and foam, overheated oil and leakage due to overflow. Over time, oxidation may occur due to increased temperatures and exposure to air. Marginal lubrication can also result in pitting because oil film does not spread the contact (cushion) over a sufficiently wide area. This can result in metal-to-metal contact in the load zone.

Level gauges and viewing windows allow for visual inspection of the fluid levels and oil condition (cloudy, dark, foaming etc.), which should be recorded and trended along with top-up activities. These results can be used to determine changing conditions and increased or decreased monitoring/testing activities.

## Contamination Control

A contaminant is any substance that enters a system and affects or interferes with the function of the system's fluid and/or the operation of its components. Solids, water and various gases (primarily air) entering or existing in a system can have mechanical or chemical interactions on the oil and/or the equipment. Fluids must be protected and monitored from such contaminants by a comprehensive contamination-control program. A comprehensive program is one that incorporates prevention of fluid contamination, removal of contaminants, and fluid-system condition monitoring.

Knowing the contaminants and their origin will provide clues as to how they might be excluded, removed, or their effects neutralized. Contaminants can be built-in due to manufacturing/maintenance processes, or they may enter a system while parts of it are open during construction or repair. Also, they can be internally generated as a result of system operation, such as wear debris, compounds of chemical reactions, or substances resulting from decomposition of the fluid or its additives. The most common entrance of contaminants from the atmosphere is either through breather caps, imperfect seals, or other unplanned openings during normal operation of the equipment. This would include the addition of fluid during initial fills or top-ups.

OEMs will provide their own equipment specific requirements for targets and limits of contamination in their maintenance manuals or service bulletins. In most cases, the end user will not obtain these targets unless a further investment into contamination control equipment is provided.

| Typical Gearbox/Reducer Requirements (OEM) | | |
|---|---|---|
| | Gearbox (OEM) | Bearing (OEM) |
| Filtration (ISO 4572) | ß25 =200 | ß 6 = 200 |
| ISO Particle Count (ISO 4406) | Max. 21/15 | Max. 14/11 |
| Water Concentration | Max. 0.05% (500 ppm) | Max. 0.05% (500 ppm) |

Table 1 - Typical OEM Requirements

## Breathers

As system temperatures or environmental temperatures change, the gearbox or bearing housings will have a movement of air. Whether the housing is inhaling or exhaling, the movement of air is always trying to equalize the temperature difference from the inside to the outside of the housing or in reverse direction. Exhaling in many cases carries a fine oil mist to the outside environment, while inhaling carries the industrial environment along with nature's environment into the gearbox or bearing housing. The area above the lubricant level but inside the housing is classed as "Headspace" and managing the quality of the environment in this headspace is one step in controlling the contaminants entering the lubricant.

Controlling contamination that enters the headspace environment can reduce and in some cases eliminate many of the potential root causes of lubricant failure. Most housings have a planned method or location for breathing; otherwise a build up of pressure would cause seals to fail resulting in external leakage of the lubricant. Since many of the breather systems on gearbox housings are either just a small hole in the cap or a very poor quality strainer style breather, they typically will not prevent the required contaminants from entering the system to maintain the OEM fluid requirements.

You must fully understand the environment around the housing in order to upgrade to a breather that will best control contaminant

ingress. For example, an effort to stop the ingression of water in a continuous hot dry environment will result in an investment with no financial return. And in an environment of high humidity and excessive airborne contamination, stopping only water ingression will not successfully stop the destructive powers of the airborne contamination.

Your goal is to provide maximum protection for your equipment in your unique operating environment. By conducting oil sampling and testing, and a thorough investigation of the surrounding environment, you will gather the information you need to select the breather that will provide the most economical and effective protection from the known contaminants reaching the headspace. In cases of extreme environmental contaminants or considerable air movement within the housing, there could be a need for an external bladder system (Figure 2). The chart in Table 2 lists some of the breather exclusion methods along with the contaminants that they restrict from entering the housing headspace.

At some point, all styles of breathers that are not maintained will become plugged with

Figure 2 - Expansion chambers prevent the exchange of air from the chamber to the surrounding atmosphere.

debris, resulting in an increase of internal pressure of the housing and the ultimate failure of the seals. Leaking seals in gearbox housings have become a common complaint in industry and the end result is typically the rework of the shafting material and the replacement of the seals due to the hard aggressive wear created by the abrasive particles attracted to the leaking lubricant. In too many cases the root cause (blocked or plugged breather) is not identified as the condition of the breather and is not monitored or replaced on a planned maintenance schedule.

## Filtration

Once contaminants are in the fluid, settling, filtration/separation and fluid replacement may reduce contamination. For settling to occur, a contaminant must have a density greater than the fluid transporting it. The lower the density of a contaminant particle, the more buoyant it will be in a fluid. Many gearbox manufacturers have designed the gearbox housing to allow the contaminants to settle out in areas that will not allow them to be redistributed into the system. Removal of these contaminants requires thorough flushing of the housing during the replacement of the lubricant. The best way to flush is to use compatible low-viscosity base oil, or a low-viscosity variation of the service oil that can be applied in a method that ensures that all the dead zones are cleaned and any debris is dislodged.

Maintaining the lubricant throughout its in-service life requires some form of filtration for the removal of accumulated contaminants. A properly designed high viscosity filtration system that will supply the correct flow rate to perform the function of removing the targeted contaminants in a reasonable time frame must be utilized. Re-circulating, kidney loop or auxiliary filtration, consists of a pump, filter, motor, cooler (if required) and appropriate hardware connections. Fluid is continuously pumped out of the reservoir, through the filtration system, through an incorporated temperature regulating system (if required), and back to the reservoir ensuring fluid conditioning regardless of the operating condition of the main system. These systems can be either portable or permanently retrofitted to the gearbox/reducer housing. The choice comes down to criticality of production (the need for reliability), safety and severity/penalty of failure.

Smaller, portable filter carts, and hand-held pump/motor/filter units are ideal for pre-filtering, flushing or transferring fluids into reservoirs. These off-line portable carts can be adapted to service many different gearbox/reducer housings in the same family of lubricants by adapting quick-connect self sealing fittings on the drain and fill ports. Care must be taken in selecting the right pump flow rate, filters, and conductor sizes to operate at higher viscosity gear oils. A properly designed filtration system will mini-

| Methods of Exclusion | Contaminants Excluded | | | | |
|---|---|---|---|---|---|
| | Solid Particles >50 Microns | Solid Particles <10 Microns | Solid Particles <5 Microns | AirBorne Water | Funes/Odor |
| Open Vent | | | | | |
| Metal Screen | X | | | | |
| Cellulase Paper | X | X | | | |
| Filter Pads | X | X | | | |
| Microglass Filter | X | X | X | | |
| Desiccant Silica Gel | | | | X | |
| 3M Filter/Desiccant Filter Gel | X | X | X | X | |
| Activated Carbon | | | | | X |
| Activated Carbon/3M Filter | X | X | X | | X |
| 3M Filter/Silica Gel/Activated Carbon | X | X | X | X | X |
| Expansion Bladder Systems | X | X | X | X | X |
| Note: All breathers must be properly sized and maintained | | | | | |

Table 2 - Methods of Exclusion

mize operating costs by reducing lubricant and equipment damaging contaminants while assisting in extending lubricant life and ultimately the mean time between failures.

To measure and trend the contaminants and the effectiveness of the filtration system, an oil analysis program should be incorporated into the planned maintenance program. To initiate this program, strategically located test ports should be installed to provide trouble-free, repetitive and representative sampling of the lubricant contaminants along with consideration of monitoring the health of the equipment. This sampling method should allow the equipment to be tested under its typical operating condition while being non-obtrusive and maintaining a safe sampling method for the technician.

## Summary

Temperature, pressure, contaminants, aeration, water, metal particles, and agitation all accelerate the oxidation process (destruction) of the lubricant. Combine that with the destructive nature of solid particles trapped between rolling elements, raceways and gear teeth, and emulsions created (lack of lubricating properties) by ingested water will ultimately result in the sacrifice of equipment performance and reliability.

Lubrication Management for Gearboxes/Reducers combines:

- Maintaining headspace quality,
- Operating within the designed operating conditions,
- Use of dedicated and/or portable filtration for contamination control,
- A competent oil analysis program for maintaining the specific physical, chemical and cleanliness standards throughout the life of the lubricant.

The end result will be increased company profits by maintaining the functionality of the equipment for production.

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# Electric Signature Analysis

## Voltage Waveform Provides Ability to Diagnose Multiple Fault Conditions

*by Howard W. Penrose, PhD, CMRP*

Over the past months, we have covered traditional and modern methods of testing electric motors both energized and de-energized. As time has progressed, the abilities of electrical testing for insulation degradation and weakness have improved, as discussed over the past two articles. However, the ability to detect dynamic faults in a machine remained primarily guesswork, experience and in the realm of mechanical vibration.

In the early 1980's, several different approaches were taken to look at the electrical signatures of rotating machines. One approach was to look at the electrical current, which became known as Motor Current Signature Analysis (MCSA) and one was developed by Oak Ridge National Labs for the detection of broken rotor bars in Motor Operated Valves (MOV's) in the nuclear power industry. This second method looked at both the voltage and current signatures and became known as Electrical Signature Analysis (ESA).

MCSA is primarily used by the vibration industry using special current probes which allow the vibration data collectors to take current input. This current is then converted from analog to digital, filtered and produced as an FFT (Fast Fourier Transform) spectra of amplitude versus frequency. ESA has been primarily used by the dedicated ESA instrument manufacturers and includes the voltage waveform as an input. The primary difference is that current tells the user what is from the point of test towards the load whereas voltage provides information from the point of test towards the supply. This allows the user to quickly determine where a particular signature exists.

In this article, we will discuss Electrical Signature Analysis and its application in AC induction motor circuits. ESA provides the capability of detecting power supply issues, severe connection problems, airgap faults, rotor faults, electrical and mechanical faults in the motor and driven load, including some bearing faults. It is important to note that the technology should not be considered a replacement for vibration analysis in mechanical analysis, but provides excellent data on motor condition from incoming power through to the rotor. From the bearings to the mechanical load still remains in the realm of vibration, in most cases.

### Fault Detection Using ESA

One of the original concepts behind the development of ESA was to eliminate the loss of instrumentation to test MOV's in the dangerous areas within nuclear power plants. The primary failure of these machines is the rotor which would overload and melt when limit switches failed. It was discovered that the rotor bar failure signature was unique enough that not only could the signature be quickly identified, but that condition values could be applied easily.

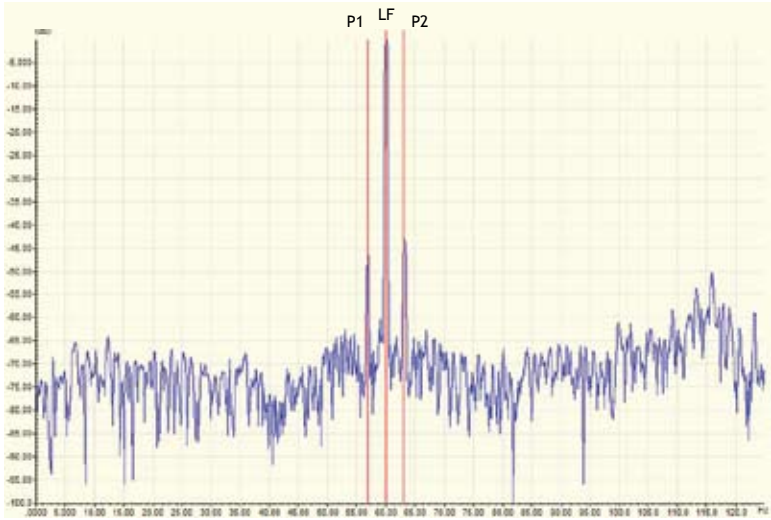


Figure 1 - Broken Rotor Bar Signature

When the Pole Pass Frequency sidebands (P1 and P2) of Figure 1 are compared to the values in Table 1, the condition of the rotor bars can be determined. However, in this case, the motor is 4,160 Vac and the data was taken from the Motor Control Center (MCC) Current Transformers (CT). The result can be a dampening effect on those peaks, resulting in the analyst needing to estimate the severity of the fault.

| dB | Rotor Condition Assessment | Recommended Action |
|---|---|---|
| > 60 | Excellent | None |
| 54-60 | Good | None |
| 48-54 | Moderate | Trend Condition |
| 42-48 | High Resistant Connection or Cracked Bars | Increase Test Frequency and Trend |
| 36-42 | Broken Rotor Bars Will Show in Vibration | Confirm w/ Vibration, Plan, Repair or Replace |
| 30-36 | Multiple Cracked/Broken Bars, Possible Slip Ring Problems | Repair/Replace ASP |
| < 30 | Severe Rotor Faults | Repair/Replace Immediately |

Table 1 - Rotor Bar Failure Levels

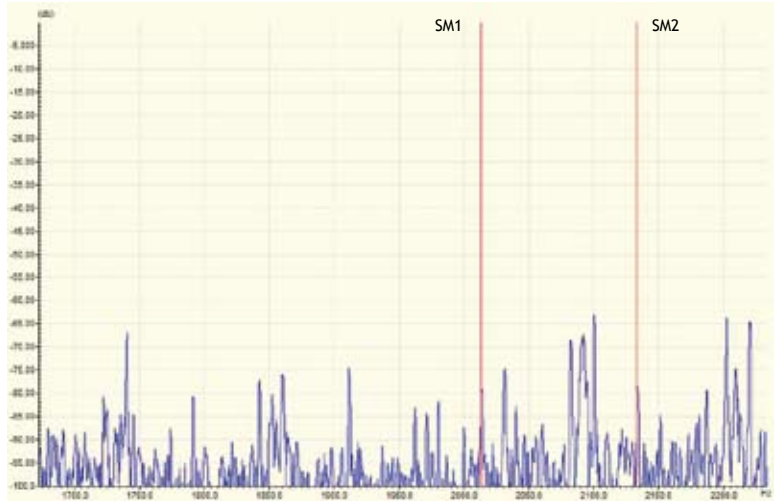


Figure 2 - Coil Movement Signature

Formula 1 - Pole Pass Frequency

$$2\left[\left(\frac{SS - RS}{SS}\right) * LF\right] = PPF$$

Where SS is Synchronous Speed, RS is Running Speed, LF is Line Frequency and PPF is the Pole Pass Frequency

Concerning most other faults detected with ESA, the number of rotor bars and stator slots in the design of the motor is necessary. Many of the ESA instrument manufacturers have built algorithms into their software which can assist the analyst in estimating either number. In figure 2, the motor is an 800 horsepower, 1785 RPM, 101 Amp, Louis Allis motor with 58 rotor bars and 72 stator slots. SM1 and SM2 are peaks related to the movement of the coil ends of the motor windings. As measured through the CT's, the values are about -78 dB which would be more severe if the current was measured directly. With an RPM of 28.793 Hz (1727.6 RPM), the stator mechanical (coil movement) frequencies would be the number of stator slots times the running speed plus and minus the line frequency. In this case, 2013.1 Hz and 2133.1 Hz, which relates to the fields passing through the coils' ends and interacting with the rotor fields.

| Type of Fault | Pattern (CF) |
|---|---|
| Stator Mechanical | CF = RS x Stator Slots<br>LF Sidebands |
| Rotor Indicator | CF = RS x Rotor Bars<br>LF Sidebands |
| Static Eccentricity | RS x Rotor Bars<br>LF and 2LF Sidebands |
| Mechanical Unbalance | CF = RS x Rotor Bars<br>LF Sidebands and 2LF Signals |
| Dynamic Eccentricity | CF = RS x Rotor Bars<br>LF and 2LF Sidebands with Running Speed Sidebands |
| Stator Electrical (Shorts) | CF = RS x Stator Slots<br>LF Sidebands with Running Speed Sidebands |

Table 2 - Electrical and Mechanical Faults

Excessive coil movement will cause fractures in the coils as they leave the stator slot. In the case of the 800 horsepower motor shown in Figures 3 and 4, this movement coupled with oil contamination caused the winding to fail where the windings leave the slot.

## Applications of ESA

ESA does have the capability of detecting some bearing failures and load related problems. With the ability of taking accurate data from the MCC or disconnect, a technician can take data on multiple machines from within a single MCC. This allows the user to evaluate equipment that is difficult, or dangerous, to access. Knowing the limitations of the technology will allow the technician to understand the risks involved in ESA detection in these applications.

In order for the technology to work, a torsional or radial force must occur within the stator airgap. The radial changes in the airgap effect the magnetic field and, as a result, the current. The small variations ride along the fundamental, or line, frequency which, when converted to an FFT, assists the technician in fault analysis. Major changes to the motor speed, rotor, torque, coupling and some loads will show as side bands around the line frequency while others will show as higher frequency signatures related to the number of rotor bars and stator slots. Bearings, however, will show in a similar fashion as in vibration analysis with a small change. As in vibration, bearing issues show as the running speed times the different bearing multipliers such as inner race, outer race, cage and ball-spin. The difference is

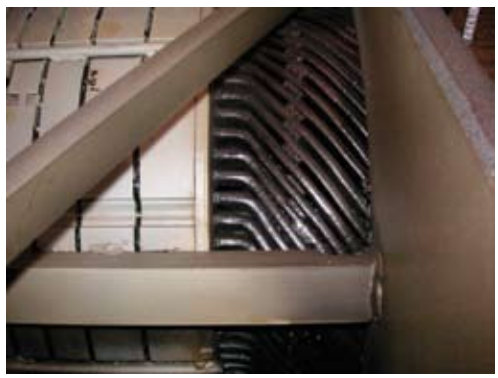

Figure 3 - Louis Allis 800 HP Stator

Figure 4 - Copper from Louis Allis Coil Failure

that in ESA, the signature will actually show as peaks +/- the line frequency.

The challenge is that the defect must cause enough of a change in the airgap in order to register in the current. The detection becomes less likely in situations where analysis is being performed through CT's and PT's. There are instances where a bearing is audible and the signature shows in vibration, but will not show in ESA.

Vibration related problems will be identified as a running speed peak sidebands around the line frequency current and one times the running speed in the demodulated current. However, while the demodulated current will show a potential problem, it takes the sidebands to determine the severity. The unbalance should be checked when the sidebands exceed -65 dB.

Alignment, sheave, fan, pump, and other component issues can be detected. However, ESA cannot always determine the exact nature of the problem that is detected. It can be used as a method for identifying that a problem exists before any additional testing or action is performed.

## Conclusion

Electrical Signature Analysis is a modern technology that can be used to identify faults that other technologies cannot, or have difficulty, detecting. Developed in the 1980's, ESA is only gaining ground within industry following the turn of the century. While the technology has the strength to easily detect rotor bar faults above any other dynamic test, it has limitations when it comes to mechanical faults. ESA can be used to test multiple machines from the MCC, itself, but the analyst must know the limitations of the testing technology.

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# The Great Debate

## An Alignment & OL2R Conversation Between Professionals

*by Brian Roy & John Piotrowski*

In the March issue, Uptime ran an article entitled *Diagnosis Unknown* by John Piotrowski. The article sparked an interesting exchange, debate if you will, between John and Brian Roy, an Reliability Specialist from British Columbia, Canada. The exchange is quite interesting and we thought it would be worth publishing to push the debate into more of an open forum. The exchange has been edited to fit into our magazine, but you can find it in its entirety on our website ad www.uptimemagazine.com/diagnosisdebate.htm and will be linked to www.maintenanceforums.com so you will be able to add to the discussion if you want to.

**Brian Roy:**
Mr. Piotrowski,
Interesting article. This complex test could have been done in a much quicker and easier way, with better results. Thermal growth is easily determined by installing dial indicators at strategic positions on the machine train and measuring the growth relative to a fixed point off machine directly upon startup and temperature stabilization. The fixed point can be selected from something nearby if you're lucky or it can be created quite simply by tack welding some plate to a beam or some other object not connected to the machine train. After the initial realignment is performed based on these thermal growth offsets, an Operating Deflection Shape analysis would be performed to determine how the machine is moving and where to focus any further efforts. The ODS can even be done prior to the thermal growth measurements for comparison purposes later. All this can be done in an 8 hour shift with some preparation and it would yield far superior results in a much more cost effective way. The rule of thumb for all troubleshooting efforts is to start with the simplest tests and solutions and work towards the more complex. On another note, OEM provided thermal growth offsets are NEVER correct and should always be verified and never taken for granted.

**John Piotrowski:**
Brian,
Thank you for reading and responding to the article in the March 2007 issue of Uptime magazine on the off line to running machinery movement survey on the motor - fluid drive - pump drive system. I would like to respond to each of the points in your correspondence.

In point no. 1, your suggestion on using dial indicators to measure thermal growth is most intriguing. Have you ever tried this? If so, what were your results?

Curious that you mentioned this as I have made two attempts to use dial indicators to measure as you say "thermal growth". I've never been very fond of the term "thermal growth" as it implies that all the movement that occurs in rotating machinery is due to temperature changes in the machine case and that machines will always grow or move upward. Neither of which is always true. We live in a three dimensional world so machinery will move in all three directions, not just upwards. Some machine will actually "thermally shrink", for example, refrigeration compressors. The term "thermal growth" also does take into account lateral or axial movement of machinery.

Around 1977, while working at a chemical plant, we were having some problems with a 32,000 hp steam turbine driving three barrel compressors. We suspected that the drive system was running in a misalignment condition and suspected that the shafts were moving after the unit was started up and running at full load. We had just finished replacing one of the compressors that had a bearing failure and I wanted to see if the steam turbine and the compressors were moving so I asked if I could weld a few pieces of angle iron to the soleplates near each bearing so I could place a dial indicator on top of each bearing to observe how far, how fast, and how much the bearings were moving upwards. I was not allowed to weld anything because the production department wanted to get the plant back up and running and I was unable to get the necessary paperwork done in time for the welding permits. So I welded a 1/2" thick plate to the bottom of each piece of angle iron and C-clamped each plate to the soleplates, and then set up the indicators on each bearing. Good enough for a quick test I thought.

It took about 24 hours to get the process up to full load conditions and this drive system was one of four pivotal drive systems in the process link. I remember standing outside on the compressor deck when the steam turbine started rotating the three compressors, eagerly running back an forth watching all eight indicators for any movement. Not much happened during the slow roll tests used so production could make sure the refurbished compressor was not going into tilt mode from a

vibration or temperature standpoint. After a 30 minute hold at 500 rpm, with all systems looking good, production starting opening the steam valve on its way to a final operating speed of 11,000 rpm. Fifteen minutes went by and none of the indicators moved, then after about 30 minutes, each indicator started moving. Most showed an upwards movement but two of them were moving downward. What? I decided to keep calm and just watch the experiment unfold and figure out what happens later. Now at 2500 rpm, production had a problem at another part of the process so they put everything in a temporary hold basis with the compressor and turbine drive steady at 2500 rpm. I noticed that the bearing housings were starting to vibrate a little and the indicator needles were swinging back and forth. No problem, I could just visually watch the sweeping needle and get an average position. Another hour passed and the speed of the compressor drive was still at 2500 rpm as I looked at the reed tachometer on the steam turbine. Cheez, I thought, what's going on? So I went down to the control room at ground level to get some first hand info on when we were going to get rolling up in speed. Two more hours went by and the problem at the other end of the process involved a stuck control valve that just got freed up and now seemed to be working properly. So production decided to continue increasing the flow rate through the system which meant that the compressor I was watching was on its way up in speed. I walked back up to the compressor deck and was crestfallen to discover that the needles on four of the eight dial indicators had vibrated off and were now useless. Thank goodness the indicators were still working on both turbine bearings and both bearings on the recently rebuilt compressor! I decided I should begin recording what the working dial indicators were measuring so I got my notepad and one by one looked at the dial indicators. The needle on the dial indicator at the exhaust end of the steam turbine was flapping back an forth 4-6 mils but the average sweep appeared to be 2 mils higher than when I started. The needle on the dial indicator at the supply end of the steam turbine (the hotter end) was also flapping back and forth 4-6 mils and much to my surprise showed that bearing had dropped 15 mils! The average dial indicator positions on the inboard and outboard ends of the compressor seemed to have an average reading of 4 and 6 mils respectively. As the speed of the drive system began to increase, so too did the flapping of the dial indicator needles. After 8 hours, every needle vibrated off but I still recorded the average dial indicator positions with the most surprising measurement of 28 mils of downward movement at the supple end bearing of the steam turbine. Since all of the needles had fallen off by this time, I decided to remove the angle iron supports.

I still remember what happened when I removed the angle iron support at the supply end of the steam turbine. I reached over and took hold of the angle iron about half way up and burnt the palm of my hand. The heat from the turbine transferred to the angle iron over the eight hour test. Couldn't see the heat, but I sure felt it. My guess is that it was about 180° F when I foolishly grabbed it. The next day I discussed the results of my experiment with my engineering manager who noticed the blisters on my hand. Embarrassed, I told him what happened. He didn't laugh but just thought for a minute and then he said "How do you know that what you measured wasn't the thermal growth of the angle iron itself?" At that moment, I realized that my reference position (i.e. the angle iron or any other object like a building column) wasn't really a thermally stable "fixed" reference point. The only way something like this would work would be to insure that the holding platform for a measurement sensor had to be thermally stable. Charlie Jackson of Monsanto also realized this which is why his reference stands are water cooled as I'm sure you are aware. I also realized that a dial indicator had a limited life span when subjected to vibration. A much more reliable sensor should be used, one that could still measure distance but not have to physically make contact with the object it is observing. Something like a proximity probe maybe? Water cooled reference stands with proximity probes is discussed in Chapter 16 of the third edition of the Shaft Alignment Handbook.

I would be most interested in hearing how you insure thermal stability in "The fixed point that can be selected from something nearby if you're lucky or it can be created quite simply by tack welding some plate to a beam or some other object not connected to the machine train". Also, with your field experience, how do you keep the needles on the dial indicators from vibrating off?

Incidentally, the second time I tried this was for another gentleman who suggested that this method would work. I told him of my experiment but he convinced me that putting a piece of rubber between the tip of the indicator and the bearing housing would take care of the vibrating needle problem. The results ended up the same way except this time I wore some thick leather gloves.

In point no. 2 you said: After the initial realignment is performed based on these thermal growth offsets, an Operating Deflection Shape analysis would be performed to determine how the machine is moving and where to focus any further efforts. The ODS can even be done prior to the thermal growth measurements for comparison purposes later. All this can be done in an 8 hour shift with some preparation and it would yield far superior results in a much more cost effective way. The rule of thumb for all troubleshooting efforts is to start with the simplest tests and solutions and work towards the more complex.

I'm really confused here. I thought ODS was the process of measuring vibration and phase at several points on a vibrating object (e.g. a drive system) then inputting the amplitude and phase angle data into a software program where one draws a "wire frame" simulation of the objects being measured. The software program then takes this data and runs an animation of the vibratory movement of the object showing how far and which way each point in the wire framed model moves.

What does this have to do with a positional change of the shafts from off line to running conditions? If you observe that the inboard bearing of a pump moves upward 25 mils, are you implying that it is vibrating up and down 25 mils? Are you implying that shaft misalignment will always show a 180 degree phase shift across the coupling? Are you saying that in an eight hour shift, you can:

1. Select several measurement points on a drive system to develop a meaningful wire frame model for the ODS software program.
2. Operate the drive system and collect the ODS data.
3. Input the data into the software to observe the running misalignment ODS simulation.
4. Shut the unit down, weld fixed references on beams or some other objects not connected to the machine train, and set up several dial indicators.
5. Start the unit up again and capture the measurements from the dial indicator to get the "thermal growth offsets".
6. Shut the unit down and align the drive train based on the "thermal growth offsets".
7. Start the unit up again and capture the same measurement points on the drive system now with the corrected alignment situation and input these points in the ODS software program and compare the before and after results.

I will have to witness this to believe it. Also, if the vibration increases after the alignment has been corrected, where are you going to "focus any further efforts"?

In your final point about OEM provided thermal growth offsets NEVER being correct:
I agree with one caveat. That assumes that

someone who works for the OEM actually knows what "thermal growth" means and they have actually conducted several accurate off line to running machinery movement surveys on their equipment and what it is connected to. Good luck finding them.
The primary reason why the Essinger Bar system was used for the study was because the plant had one and wanted to use it. They also wanted a secondary technique to verify the results of the Essinger Bar system so the BRTC system was selected since they had spare proximity probes and power supplies.
Again, I appreciate your reply and would seriously like you to answer the above questions I've asked. Also, once our correspondence is completed, would you allow me to send the entire correspondence to UPTIME so they could publish it in a future publication for the benefit of their readers? If you would prefer, I could indicate that your name be withheld by request.

**Brian Roy:**
Hi John,
Thanks for the comments, very interesting. Let me clarify some of my points and that may help with some of the confusion. Let's start with the Operating Deflection Shape analysis. What you described is correct except that you didn't mention the amplitude component of the ODS that is very important to consider or you're only looking at one part of the information. Misalignment can be diagnosed in several ways. On the run, you can take a vibration spectra and typically you will see the 2nd and 3rd harmonics of turning speed. This the TYPICAL spectral signature but doesn't apply to every case of misalignment. Another way is to acquire waveforms at points closest to both ends of the coupling. The waveform should show 5 or 6 rotation cycles. Misalignment usually shows up very clearly in time based analysis. A more powerful way to diagnose this is by taking phase readings in all directions on either end of the coupling. Since key phasors are rarely present unless you can stop the machine and apply one, cross-channel phase is the best option. This is where relative phase is measured between 2 accelerometers. Misalignment always causes a phase shift across the coupling, it is rarely 180 degrees but it is significant. In the same way, an ODS will show that phase shift in a graphical way. The amplitudes are grossly exaggerated and that allows you to visualize the movement of the machine at the animation frequency. Bearing clearances have nothing whatsoever to do with this. Even though the amplitudes are exaggerated, they are exact in relative terms. You will see the flexure at the point in the model where the two machines meet based on the extrapolation of nearby phase readings. Also, there could be other problems than simple misalignment. The ODS can give you powerful hints as to what could be the problem especially if you're good at visualizing dynamic movements. I've used this method for years with great success.

Now for the dial indicator method for measuring thermal growth. By the way, thermal growth is the accepted term and it refers to either an increase in dimension caused by an input of energy in the form of heat or a decrease in dimensions caused by the removal of energy from the cold. Blowers for example typically run cold on one end and hot on the other. Thermal growth considerations are critical in these applications. Dial indicators are a widely used and accepted method. In my 23 years of consulting, I must have used this method successfully dozens of times, if not more. As you said, we live in a 3D world and that has to be taken into consideration on any test you perform. The dials have to be installed in all 3 directions (horizontal, vertical and axial) at either end of the machine being measured. They must then be zeroed at the MIDPOINT of the indicators travel. This way you can see the true movement no matter what the direction. If the machine is vibrating too much to get proper readings from the dial indicators, then you have a significant problem with your machine that has to be addressed first. In such a case, I would shut the machine down, if that's possible, and perform an initial laser alignment to get it closer and hopefully this will allow the installation of the dials. At the same time, I'd perform a bump test, since the vibration was that significant, to ensure it is not in a resonant condition. If you can't shut the machine down, that's when the ODS will give good value as an initial test as it will help you determine, in conjunction with basic vibration data, what the issue is that's causing this excessive vibration. There will be a phase shift across the coupling if there is misalignment, it is rarely exactly 180 degrees but is usually significant, and it is easily seen in the ODS. The amplitudes are exaggerated for visual purposes but are very exact in relative terms. The ODS shows that phase component in a graphical way that's easy to interpret. An ODS on a motor and pump takes me about two or three hours to perform and display in the software. It's a simple test, the longest part is drawing the diagram in the software and you get quicker as you use it a few times. All this can be done in an 8 to 12 hour shift but that obviously depends on the tradespeople you're provided and the surrounding conditions.

Finally, the purpose of my response was simply to provide another method, which I believe is simpler and more effective. Just for background, I have been a reliability consultant for 23 years now. I have consulted worldwide and have solved many complex problems. I've also taught courses throughout North America. Everything I've talked about here, is field proven many times over. As for publication, I don't mind but I would request that the company name be withheld as these are my opinions and it would be presumptuous on my part to say that I am representing the company's views in this particular matter. Anyway, I hope this helps to clarify my thoughts in this matter and good luck to you in your future endeavors.

**John Piotrowski:**
Brian,
Thank you so much for responding to my correspondence. What a pleasure it is to know that someone is actually trying to measure off line to running machinery movement besides myself and trying to relate the dynamic behavior of misaligned equipment to the vibration patterns that are exhibited when rotating machinery is forced to operate in this destructive mode. I think this correspondence is very important because I know a lot of people who are confused about diagnosing shaft misalignment using vibration analysis or ODS or any other type of NDT technique and somebody needs to clarify this ataxia once and for all. Again, I would like to respond to each of the sections in your most recent correspondence.

In the first part of your first paragraph you said: Let's start with the Operating Deflection Shape analysis. What you described is correct except that you didn't mention the amplitude component of the ODS that is very important to consider or you're only looking at one part of the information.

I thought I did mention that both amplitude and phase data was inputted into the ODS database.

In the second part of your first paragraph you said: Misalignment can be diagnosed in several ways. On the run, you can take a vibration spectra and typically you will see the 2nd and 3rd harmonics of turning speed. This the TYPICAL spectral signature but doesn't apply to every case of misalignment.

I agree with some caveats. I have taken vibration spectral data on misaligned machines. Sometimes there are running speed components only, sometimes there are twice running speed components only, sometimes there are

3X, 4X, and higher multiples of running speed, and sometimes there is a mixture of all of the above.

Question 1. Why are misalignment vibration signatures so varied? Why doesn't misalignment always show up at 1X, 2X, and 3X?

Question 2. If you were to measure the vibration on a well aligned machine, shut it down and purposely misalign it, start it back up and measure the vibration again, what would happen to the amount (i.e. overall amplitude) of vibration that you would see? Would the overall vibration go up, go down, or stay the same? If possible, explain why it would go up, go down, or stay the same.

In the third part of your first paragraph you said: Another way is to acquire waveforms at points closest to both ends of the coupling. The waveform should show 5 or 6 rotation cycles. Misalignment usually shows up very clearly in time based analysis.

Question 3. The "points closest to both ends of the coupling" would be taken where ... on the inboard bearings in the radial direction or axial direction?

Question 4: How does misalignment usually show up very clearly in the time domain? What does the time domain signal look like?

In the fourth part of your first paragraph you said: A more powerful way to diagnose this is by taking phase readings in all directions on either end of the coupling.

In this case I'm assuming that a once per rev timing device is already installed on the machine that you are monitoring (e.g. a piece of reflective tape and a phototach or a proximity probe observing a keyway or a notch). If so, I'm not clear on your statement "...taking phase readings in all directions on either end of the coupling".

Question 5: Would I place a vibration sensor (I assume a seismometer or accelerometer) on both inboard bearing housings (i.e. on the bearings on both sides of the flexible coupling) in the vertical, lateral, and axial directions?

In the fifth part of your first paragraph you said: Since key phasors are rarely present unless you can stop the machine and apply one, cross-channel phase is the best option. This is where relative phase is measured between 2 accelerometers.

I'm not totally clear on your statement "... where relative phase is measured between 2 accelerometers".

Question 6: Where are the accelerometers placed? In the axial direction, 180 degrees apart (i.e. 3 and 9 o'clock) on one bearing on one of the machines then again in the axial direction, 180 degrees apart (i.e. 3 and 9 o'clock) on the other bearing on the other machine? Or is it in the axial direction, 180 degrees apart (i.e. 12 and 6 o'clock) on one bearing on one of the machines then again in the axial direction, 180 degrees apart (i.e. 12 and 6 o'clock) on the other bearing on the other machine? Or is it in the radial direction (i.e. 3 and 9 o'clock), 180 degrees apart on one bearing on one of the machines then again in the radial direction (i.e. 3 and 9 o'clock), 180 degrees apart on the other bearing on the other machine? Or is it in the radial direction (i.e. 12 and 6 o'clock), 180 degrees apart on one bearing on one of the machines then again in the radial direction (i.e. 12 and 6 o'clock), 180 degrees apart on the other bearing on the other machine? Or is it none of the above?

In the sixth part of your first paragraph you said: Misalignment always causes a phase shift across the coupling, it is rarely 180 degrees but it is significant. In the same way, an ODS will show that phase shift in a graphical way. The amplitudes are grossly exaggerated and that allows you to visualize the movement of the machine at the animation frequency. Bearing clearances have nothing whatsoever to do with this. Even though the amplitudes are exaggerated, they are exact in relative terms.

I'm not totally clear on your statement "Misalignment always causes a phase shift across the coupling, it is rarely 180 degrees but it is significant." In this case, I again assume that there is some sort of once per rev signal being inputted to a vibration analyzer. I also assume that the analyst would place the sensor on one machine, get the amplitude and phase angle then put the sensor on the other machine and get the amplitude and phase angle there. My understanding of phase is that it is a once per rev timing reference. I'm not sure but I think the way phase works is as follows:

1. The vibration analyzer watches and remembers the period of time between each once per rev pulse.
2. The vibration analyzer then divides this time period into 360 parts (i.e. degrees of rotation).
3. The vibration analyzer then looks at the vibration signal itself (i.e. the waveform) and looks for the voltage to cross from a positive to a negative voltage (or vice versa).
4. The vibration analyzer then measures the period of time from when it sees the once per rev pulse to when the voltage goes from positive to negative, divides that time period by the period between the once per rev pulses and multiples by 360. For example, if a shaft is rotating at 3600 rpm, the time period for one revolution is 16.6 milliseconds. If the period of time from when it sees the once per rev pulse to when the voltage goes from positive to negative is 8.3 milliseconds, the phase angle would be 180 degrees.

Question 7: Where do you place the vibration sensor on each machine? In the axial or radial direction and on which bearing(s) or position on the housing does the sensor get placed?

Question 8: If vibration from shaft misalignment shows up at 2 or 3 times running speed, what does phase mean if it's related to a once per rev signal?

In the seventh part of your first paragraph you said: You will see the flexure at the point in the model where the two machines meet based on the extrapolation of nearby phase readings. Also, there could be other problems than simple misalignment. The ODS can give you powerful hints as to what could be the problem especially if you're good at visualizing dynamic movements. I've used this method for years with great success.

Question 9: I do not understand what you mean by "...the flexure at the point in the model where the two machines meet". Would you explain this more clearly?

In the first part of your second paragraph you said: Now for the dial indicator method for measuring thermal growth. By the way, thermal growth is the accepted term and it refers to either an increase in dimension caused by an input of energy in the form of heat or a decrease in dimensions caused by the removal of energy from the cold. Blowers for example typically run cold on one end and hot on the other. Thermal growth considerations are critical in these applications. Dial indicators are a widely used and accepted method. In my 23 years of consulting, I must have used this method successfully dozens of times, if not more. As you said, we live in a 3D world and that has to be taken into consideration on any test you perform. The dials have to be installed in all 3 directions (horizontal, vertical and axial) at either end of the machine being measured. They must then be zeroed at the MIDPOINT of the indicators travel. This way you can see the true movement no matter what the direction.

I realize that thermal growth is the generally accepted term. I'm trying to get people to stop using it and begin using a more accurate term ... off line to running machinery movement (often abbreviated as OL2R). The mechanisms for OL2R movement are not always thermal and the direction for the movement is not always growth. I sincerely hope that dial indicators are not widely used and the generally accepted method for measuring OL2R movement. Every time I've tried it, it

has failed miserably. Remember what I said in my first correspondence, about my maintenance manager asking me, after I had burned my hands, if I was sure my "fixed" reference point was really fixed.

Question 10: Do you still have the results from the dozens of times you used this method? If so, would you be willing to share the results of some, if not all, of your data?

Question 11: Your statement "The dials have to be installed in all 3 directions (horizontal, vertical and axial) at either end of the machine being measured." Do you mean both ends or just one end. If so, which end do the indicators get put on, the inboard or outboard end?

Question 12: When you used dial indicators and reference platforms, how do you know that what you measured wasn't the thermal growth of the reference platform itself?

In the second part of your second paragraph you said:
If the machine is vibrating too much to get proper readings from the dial indicators, then you have a significant problem with your machine that has to be addressed first. In such a case, I would shut the machine down, if that's possible, and perform an initial laser alignment to get it closer and hopefully this will allow the installation of the dials.

This is really interesting. I would have done just the opposite, that is, I would have used dial indicators to do the alignment and laser to do the OL2R survey.

Question 13: If you are using dial indicators to do OL2R surveys why aren't you using them to do the off line alignment?

I think that we all need to take our measurements in a consistent pattern so our results agree. If I'm taking measurement one way and you are taking them another way then we will never agree.

Please forgive me for these thirteen questions but I believe this is very important because if I'm confused, I believe there is a possibility that other people may also be confused. In fact I think that hundreds of vibration analysts are confused.

Case in point. In the same issue of UPTIME, Jason Trantler talks about misalignment and phase (page 43). What he describes seems to be different from what you are describing. Another case in point, in the February 2007 issue of Pumps&Systems magazine, Dr. Lev Nelik submitted an article entitled "Pump-to-Motor Alignment: Why 0.002-in and Not 0.020-in?". In it he talks about asking a group of sixty engineers who attend a presentation he gave at a Vibration Institute session the following question:

"Will a dial indicator aligned pump last 10 times longer than a straight edge aligned one? Will its vibration be 10 times lower? 5 times lower?"

No one in the audience could answer the question. Later in the article he describes an experiment he did where he purposely misaligned a piece of machinery and measured the vibration before and after. The results he saw were similar to what I have seen when I have done several similar experiments. If you would, send me your answer to question 2 before you see what happened to him.

I will await your answers. Please let me know if you would agree to sending this to UPTIME and see if they would be interested in publishing our correspondence.

**Brian Roy:**
Great discussion points John, and I totally agree with your statements on misalignment and how it's diagnosed and sometimes misunderstood. Your questions are very relevant so I'll just go through them one at a time.

Question 1. Why are misalignment vibration signatures so varied? Why doesn't misalignment always show up at 1X, 2X, and 3X?

Brian Roy:
The reason that it doesn't always show up in the typical way has to do with the fact that the misalignment is not necessarily the only vibration source in the system. This is the exact reason that I always caution students against blindly using those diagnostics charts (the main one being the Technical Associates of Charlotte one by Jim Berry) to determine faults. These charts are based on the premise that there is only one fault component in the system at a time, which is rarely the case. Another thing that can influence the spectral signature is the machine construction itself. Ambient conditions, mass, stiffness and damping, while components of natural frequencies can also affect how vibration propagates and is dissipated through the system. Coupling issues can also change the spectral signature, as can base conditions. What if the base the motor is sitting on has got broken anchor bolts that aren't visible? The base looseness may show up as 1x, 2x, 3x turning speed which would tend to make a person think this was misalignment. However, phase readings across the coupling would eliminate misalignment as the true culprit as there would not be any significant phase shift in this case. Also, the noise floor underneath the turning speed multiples in the spectrum would tend to have a raised noise floor in the case of looseness. The waveform would also look different, more random. Induced vibration from nearby machines can also muddle up your spectrum. There are many reasons why misalignment doesn't always show up in the typical manner, that's one of our main challenges as machinery diagnostic specialists. We have to bring more to the table and use whatever tests we can think of to get more information to help us make the best determination.

**John Piotrowski:**
I agree that there can be more than one problem occurring at the same time on a drive system. I don't believe that Jim Berry from Technical Associates of Charlotte implied that his charts were to be interpreted based on the premise that only one problem can exist on a machine. Jim has done an exemplary job with his charts and I see them everywhere. The first attempt at vibration diagnostic charts I remember seeing was composed by John Sohre several decades ago. Jim Berry expanded on that idea and I think it has been very helpful to me and to a lot of people. Jim will attest to the fact that he and I do not see eye to eye about his misalignment vibration patterns.

As I, and most other experienced vibration analysts, understand is that other problems may be present that could account for the multiples of running speed such as looseness but the general consensus of most vibration analysts that I know is that shaft misalignment will appear in the spectrum as running speed or multiples thereof. You then went on to reiterate that phase analysis is the best indicator of misalignment but you never really answered why misalignment shows up at running speed or multiples thereof.

In Chapter 2 of the 3rd edition of the Shaft Alignment Handbook entitled "Detecting Misalignment on Rotating Machinery", in section 2.2.10 there are several examples of known vibration signatures of misaligned rotating machinery with different flexible couplings as shown in figures 2.34 through 2.39. The vibration spectral patterns shown in these figures were taken from case histories and information from other people who had done misalignment studies (Jerry Lorence, Daniel Nower, David Dewell, and Steve Chancey).

Although the author of the Shaft Alignment Handbook didn't explain why the vibration spectral patterns were different, here is my take on this...

The vibration that occurs on misaligned rotating machinery is quite possibly caused by the

mechanical action that is occurring in the flexible coupling as it attempts to accommodate the misalignment condition. Since different flexible couplings are designed to accommodate misalignment differently, the dynamic forces that occur will be distinctive, producing the different responses that are seen.

In all the years that I have been studying vibration analysis, very few people have ever mentioned anything about the dynamic forces that occur in a flexible coupling trying to accommodate a misalignment condition. Yet I continually hear that unbalance is caused by a dynamic force that is produced by a heavy spot as it whirls around once every revolution. I also hear about the dynamic forces of gear teeth as they mesh or when a ball rolls over a spall in a bearing raceway producing a pulse/force as it drops into the hole and comes bouncing back out. I have asked the same question to hundreds of other people over the years and they, as well as you, have yet to give me a scientific explanation of why this happens. Forget about phase, answer the question someone, anyone, please. The answer ... "I heard it from a vibration expert or at a vibration course." doesn't count.

I know this as a fact of physics:
Vibration can only be caused by dynamic forces. A machine that is not rotating does not produce vibration despite the fact that it may be out of balance and have damaged gears or bearings. Not until rotation starts will the dynamic forces become present. Then, and only then, will the vibration occur.

I also know this as a fact of physics:
There a two basic types of forces, dynamic forces and static forces, both of which are present in every drive system. Some of the forces that act on our bearings and rotors are dynamic and some are static. The total amount of force on our bearings and rotors is a combination of both static and dynamic forces.

If I see a vibration signal where a three times running speed component is present, to me, for that to happen, three forces must occur every time the shaft rotates around one revolution. For example, a jaw type coupling (e.g. a Lovejoy) typically manifests a misalignment condition with a strong three and six times running speed components as shown in figure 2.34 (in the Handbook). Isn't it curious that there are three "fingers" on each of the two coupling hubs. Under a misalignment condition, each time the shaft rotates around once, each "finger" on each coupling hub produces a pulse that carries through the shafts to the machine casing and the bearing housings where we measure the vibration with a sensor. Three fingers, three pulses, three times running speed. Is this just a coincidence? But why is there a six times running speed component? I guess because there are six total "fingers" (three on one hub and three more on the other hub) where each alternating finger is spaced 60 degrees apart. Six times sixty equals 360, degrees that is, in one rotation. Other types of flexible couplings (e.g. gear, rubber tire, flexible disk) exhibit different vibration spectral patterns and I could go on about how I think each design emanates the dynamic forces that cause the vibration patterns but this dialog is going to be long enough so I'll stop here.

**Brian Roy:**
Question 2. If you were to measure the vibration on a well aligned machine, shut it down and purposely misalign it, start it back up and measure the vibration again, what would happen to the amount (i.e. overall amplitude) of vibration that you would see? Would the overall vibration go up, go down, or stay the same? If possible, explain why it would go up, go down, or stay the same.

If the machine was running well and you induced a misalignment, then I would tend to think that the overall vibration would go up as you are generating running speed multiples that were not previously present and causing forces to be exerted at the coupling. It is absolutely impossible to guess that by misaligning the machine that the overall vibration will go up by X% or that the 2nd multiple will appear but not the third etc. There are modeling techniques and known case studies that could help you take a guess but you can never anticipate exactly what will happen. If the same thing happened in the same way every time a fault is induced in a machine, we would be out of business as an on-line system could do our job more efficiently that we could.

**John Piotrowski:**
I ask this question in every one of my shaft alignment training courses and have also asked it at several presentations at conferences where I have been asked to speak about this subject. Although I do not have an accurate count, my sense is that 70% of the people believe that the vibration will go up on a drive system if you misalign it.

Here is my answer to that question:
If you misalign a drive system, the vibration may go up, go down, or stay the same but in the majority of cases the vibration levels will go down.

Yup, exactly the opposite of what the majority of people say will happen. In Chapter 2 of the Shaft Alignment Handbook, sections 2.2.7 and 2.2.8 show examples of controlled tests where vibration data was taken under misalignment conditions and then again after the alignment was corrected to acceptable tolerance levels. These tests show that the vibration will go down if you misalign a machine.

I am in the process of publishing an electronic book tentatively entitled the "Turvac Field Service Files" where I have 12 examples of actual case histories where this is true. Right now there are 50 case histories in the book taken from about 60% of my field service reports dating from 2000 through 2007 and the current page count is just shy of 800 pages. If I go back to 1979 and put all of my field service reports in this book, it may tip the scales at 3000+ pages. I literally have at least a hundred case histories where the vibration amplitude levels were lower on a drive system running under a misalignment condition, then after shutting the unit down, correcting the misalignment, starting the machine back up and measuring the vibration again, the vibration went up (making it look like I did the job wrong). I found it extremely hard to believe that I was the only person seeing this until I realized that the majority of people who measure vibration on rotating machinery are not the same people who align the equipment. Talk about a disconnect in communication, there's a big one. Several top notch mechanics I know think that their vibration analysts are fruitcakes for this and many other reasons. Wait a minute here, why would vibration decrease with increasing misalignment?

The author has given me permission to reprint a section from Chapter 2, to explain this phenomena:

*When two or more shafts are connected together by some flexible or rigid element where the centerlines of each machine are not collinear, the forces transferred from shaft to shaft are acting in one direction only. These forces do not change their direction as an imbalance condition will. If a motor shaft is higher than a pump shaft by 50 mils, the motor shaft is trying to pull the pump shaft upwards to come in line with the motor shaft position. Conversely, the pump shaft is trying to pull the motor shaft downward to come in line with the pump shaft position. The misalignment forces will begin to bend the shafts, not flutter them around like the tail of a fish.*

*Static forces caused by misalignment act in one direction only which is quite different than the dynamic forces that generate vibration. Under this pretense, how could misalignment ever cause vibration to occur? If anything, misalignment should diminish the capacity for motion to occur in a rotor / bearing / support system.*

Get it? If you take a balance training demonstrator and set it on top of a table and start it up with a huge amount of imbalance in it, the demonstrator could actually start bouncing up and down on the table. Before it vibrates off the table and crashes on the floor, if you grab a hold of the unit and push down hard enough, the unit will stop bouncing around. If, during this experiment, a vibration sensor happened to be taking a measurement in the vertical direction on one of the bearings, the vibration amplitude would drop the instant you push down preventing it from falling off the table. Why did the vibration go down? Because the dynamic forces from unbalance went away? Absolutely not. But by applying a static force to the drive system, you diminished it's capacity to move freely. If you think I'm lying, try the experiment and let me know what happens.

I know this as a fact of physics: Vibration sensors measure motion, not force. We are totally incapable of measuring the amount of force generated in a bearing using any type of vibration sensor. The best electronic sensor I know of to directly measure force is a strain gauge. Do you know anybody who has strain gauges buried inside their machine at or near the bearings to measure force? I do. Ask Dr. Wes Hines at the University of Tennessee who directed an experiment in 1997 where they misaligned two different drive systems using four different types of flexible couplings subjecting each to fifteen different misalignment conditions. A landmark study that was not received very well by vibration analyzer manufacturers and some laser alignment system distributors. So much for telling the truth.

Here's something to ponder, if the total amount of force in a bearing comes from both static and dynamic forces, for all rotating machinery on our planet, what average percentage of the total force is due to dynamic forces and what percentage is due to static forces? Is it 50% dynamic and 50% static, or 20% static and 80% dynamic, or is it 80% static and 20% dynamic? I don't think anyone can answer that question, but in my humble opinion, if it's the third choice, then we're looking at the wrong thing if we are using vibration as the sole determiner for detecting misalignment or any other destructive mechanism that is damaging our rotating machinery.

**Brian Roy:**
Question 3. The "points closest to both ends of the coupling" would be taken where ... on the inboard bearings in the radial direction or axial direction?

Exactly. At the inboard bearings in both radial and axial directions. When doing any diagnostics on a machine, I'm a strong believer in taking all data in all directions whenever possible. There's no such thing as too much data as long as the time constraints are respected.

**John Piotrowski:**
OK, that clarifies it, I think.

*Remember, this informational and entertaining discussion/debate continues online, complete with several case studies and a few laughs. Please go to the following link at our website:*

*www.uptimemagazine.com/diagnosisdebate.htm*

# Now Hear This

## Testing With Ultrasound Takes More Than Just Listening

*by Jim Hall*

When trending bearings, you should not never, if at all possible, depend solely the sound of the bearing. Why? Because, in all likelihood, your hearing and my hearing are not the same. Hey, I grew up with Lynyrd Skynyrd...literally. The late Allan Collins was married to my second cousin, and we used to throw for quarters against the wall at the Burger Hutt. You didn't grow up in those days without first replying to everyone by asking, "What did you say, I couldn't hear you?" Nothing on the radio or the "reel-to-reel" sounded good in those days unless it was over 100 db's.

It's a fact, men 45-64 years of age suffer from hearing loss more than individuals over the age of 65.[1] I also believe that with today's electronic culture of iPods and other MP3 players, more of the younger men/women will suffer hearing loss sooner rather than later. Add to this the day to day high noise levels within our plants, and hearing loss among the population will only get worse.

If you haven't had your hearing tested, you should do it now. I am a big proponent of getting your hearing tested before it is too late. Companies that provide an Airborne Ultrasound Level I Certification Class do have hearing standards. Usually, this means corrected hearing of >20 dBHL[2] at 500 Hz to 4 kHz.

I will not certify anyone that cannot hear the low-frequency sounds through the headphones when taking the practical exams for certification. There is not an ultrasonic instrument made today on which you can trust the analog or digital reading alone to analyze a bearing, electrical discharge or an air leak. Knowing what the ultrasonic receiver is translating through the headphones is absolutely imperative.

I had a customer one time that manufactured a rubber tire breakdown machine, the kind used to remove and/or install the tire of a wheel or rim. This company had several stations on their assembly line where the machine would be tested for leaks. One day I was asked to visit the plant to make sure the ultrasound instrument they had purchased was functioning properly. Many of the Quality Assurance inspectors were complaining that too many of these machines were leaving the plant with leaks on the pneumatic air system.

I got to the plant and found several of the final assembly inspectors using the ultrasound instrument with NO headphones on their heads. I immediately walked over and saw several of the men and woman talking to each other while they vigorously moved the ultrasound instrument side to side, up and down for leaks. At one point I dragged my foot across the floor to create a lot of white noise. One of the inspectors immediately aimed the instrument towards my feet thinking there was a leak in that direction.

Care must be taken also when using a contact probe with a digital instrument set to capture a peak reading. The instant you make contact with the bearing surface the contact could slip causing the instrument to capture a higher than normal reading. If the person couldn't hear the sound through the headphones, they would store the higher reading. Subsequent readings would then record a lower reading, and a good bearing may be replaced or scheduled for replacement too soon.

Using an ultrasound instrument with headphones, but not being able to hear the low-frequency sound within the headphones is really the same as not having a set of headphones on your head. You cannot distinguish between background noise and or competing ultrasound.

### Bearing Analysis

What does this have to do with you? Well if you cannot hear what has been converted to a low-frequency in the headset, how are you going to assess a bearing for wear


Figure 1 - Airborne ultrasound instrument being used for bearing analysis.

## How Hearing Effects Your Bearings


Figure 2a - Shows a reading of 21.3 when taken with a sensitivity of 80.


Figure 2b - Shows a reading of 27.6 when taken with a sensitivity of 70.

If you happen to have a substantial hearing loss and use an instrument like this, you may find yourself increasing the sensitivity to hear the bearing. However, as you increase the sensitivity and record or log the reading, you may, in fact, have recorded a decrease in decibels. This means you could allow a bearing to remain for several weeks in a failure mode instead of scheduling a removal/replacement. Hopefully, the bearing does not fail.

Note: There is nothing wrong with the ultrasonic instrument or readings. This issue is caused by a hearing loss of the user. Increasing the sensitivity helps him/her hear the bearing prior to recording the reading and scheduling for removal or replacement.

and tear? How are you going to hear an intermittent click, meaning a ball defect? Or, a uniformed click, typical of an outer race problem? Can you tell the difference between a dry bearing and a well lubricated bearing?

Unless, you have had an extensive amount of hours with a set of headphones on your head and using an ultrasonic receiver, I doubt if you are going to diagnose the condition of most bearings. Very few predictive maintenance programs are allowing you the time to do so.

Most ultrasound receivers being used today have data-logging capabilities. If you do not know how to use the data-logger, get familiar with how to set-up routes and trend your bearings.

I recently visited a coal mine with well over 68,000 bearings. Now, no one person is going to take a reading on all those bearings, and trending this amount of bearings is not advised anyway. For this number, scanning the bearings with an airborne scanner (flexible wand) from a set distance and angle is the preferred method. This particular predictive group needs to figure out what their target decibel is for this type of bearing.

Bearing lubrication is another application which demands that someone be able to hear a subtle change in the amplitude or noise level and be able to interpret whether a bearing is dry or over-lubricated.

### Electrical Discharge

Let's say you are a technician or infrared surveyor whose task it is to listen to an electrical switchgear panel prior to opening. Imagine if you missed the sound of electrical discharge such as destructive corona or active arcing. Even though you have your personal protective equipment in accordance with NFPA 70E, it sure would be nice to know if something is going on inside the switchgear panel before you open the panel. Remember, your infrared imager does not see corona under 240 kV. Corona is detected by the ultrasonic receiver at 1000 volts (1 kV) or higher. I read an article several months back in Uptime entitled, "See the Heat Before You Get Burned". I want you to hear the discharge before you get burned.

Figure 3 shows you why scanning closed electrical panels periodically during the year could in fact detect those anomalies that would only be found yearly when the panels are opened for an infrared scan. Arcing was heard inside this panel. Once opened, a visual inspection revealed the lower fuse and the deteriorated condition of the insulation, as well as the discoloration of the blade at that end of the fuse.

The ability to hear subtle changes when scanning electrical panels could mean the difference in someone getting injured or not or an unscheduled electrical shutdown instead of a planned outage.

### Steam Trap Diagnostics

Troubleshooting steam traps using airborne ultrasound can be fairly simple. Ultrasound can help technicians identify problems with most traps, if they know how the trap works internally and cycling times, if applicable. I have had several calls from technicians using ultrasound receivers that mention the new Armstrong stainless steel inverted bucket

Figure 3 - Arcing was heard inside this panel when it was closed, providing an early warning.

Figure 4 - Armstrong Stainless Inverted Bucket Traps within circle.

traps and how they have not been able to hear the trap working. These are traps that are "consumables," they are not repairable. Once diagnosed as not functioning correctly, these traps are removed and disposed of. Figure 4 shows the traps mounted on a universal mount with two bolts, making them easy to replace.

Here are a couple of more reasons why relying solely on the sound to make diagnoses is not a good idea. First, depending upon what instrument you are using, sound quality may be not be very good. And secondly, whether or not the instrument has frequency tuning or filtering that eliminates most background or competing high-frequency sounds makes a big difference in what you hear.

## Conclusion

I know that some of you are not going to like being told to get a hearing aid. I prefer to think of my hearing loss as "selective hearing". I hear what I want to hear. My father had two hearing aids and I know for sure that when using the ultrasound receiver and listening to same bearing on the same motor, he could hear better with the headphones on than I could.

How well do you hear? Be honest with yourself. Airborne ultrasound is a maintenance practice for which good hearing is essential. You must be able to hear to practice it well.

*Jim Hall is the president of Ultra-Sound Technologies, a "Vendor-Neutral" company providing on-site predictive maintenance consultation and training. UST provides an Associate Level, Level I & II Airborne Ultrasound Certification. Jim is also a regular provider of on-line presentations at ReliabilityWeb.com and is a contributing editor for the new UPTIME Magazine. Jim has provided airborne ultrasound training for several Fortune 500 Companies in electrical generation, pulp & paper, petro-chemical and transportation (marine, automotive, aerospace). A 17 year civil service veteran, Jim served as an aerospace engineering technician for Naval Aviation Engineering Service Unit (NAESU) and with the Naval Aviation Depot Jacksonville Florida (NADEP). Jim is also president of All Leak Detection, LLC an underground leak detection company.*

All photos provided courtesy of Ultra-Sound Technologies, Woodstock, GA.


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1. Laboratories web based article: http://www.nwhealth.edu/healthyU/stayHealthy/ear2.html
2. dBHL Decibels Hearing Level (audiology)

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# Measuring Shock Pulse

## Another Approach to Front Line Condition Monitoring

*By Lou Morando*

Downtime in a paper mill or any 24/7 facility is very expensive in maintenance costs, but even more so in the impact to profit because of production loss. In this article we will explain the Shock Pulse Method, why it's a good choice for frontline vibration measurement and show the resulting savings that the Hallsta Mill in Sweden realized from utilizing it as the primary component of its Condition Based Maintenance Program. Hallsta personnel determined these calculations when they looked at the number of incidents in which they were able to perform maintenance in a planned shutdown, instead of taking the equipment (and production line) down.

Condition monitoring should always start with a list of machine faults, specific for each machine. Only if you know exactly what you expect from the monitoring method, can you apply it efficiently and correctly. Otherwise, there is a danger that you will simply be collecting data. And data is of no use unless it is converted to useful information that you can act upon to realize your true goal of maintaining plant equipment in good working order.

When we look at the rotating component that gives us the most concern, it usually comes down to the bearing. I think it's fair to say that 70-80% of rotational problems are bearing related. Whether the causes are due to under or over lubrication, contamination, installation faults, secondary forces or just plain fatigue, we need to know the operating condition of bearings most frequently. So it's very important to determine the best technique for identifying your particular bearing problems. The other rotational problems certainly need to be identified as well, so again, choose the most cost effective, efficient technique to accomplish that.

How do you run a cost efficient, effective Condition Based Monitoring program? Start by selecting the appropriate technique for the application and for the type of answers needed. As a general rule, you can apply the 80/20 rule in many facilities. That is, around 80% of equipment needs to be monitored without the need of spectral data and large amounts of data collection. You could then utilize spectrum analyzing only on the equipment that needs it. For those pieces of equipment that are so critical that periodic monitoring is not enough, then continuous monitoring needs to be considered.

The Shock Pulse Method (SPM) is the front line technique the Hallsta paper mill chose to quickly manage input from its 800 rolls, with 4000 machines and 16,000 measurement points. With 8 inspectors, they need a quick method to know whether bearings need to be greased or not, or that damage is present and needs to be monitored more frequently.

### What is Shock Pulse?

What we loosely call 'machine vibration' is a very complex form of movement that has many different causes and that can be described and measured in many different ways. Vibration exists in all machines with moving parts, because some of the force, which makes the machine work, is directed against the machine structure and tries to shift it from its position. Thus, vibration is normal up to a degree, and all machines are constructed to withstand a certain amount of vibration without malfunctions. In order to use vibration monitoring to diagnose machine condition, we have to:

- Find a suitable way of measuring vibration, and
- Decide what normal vibration is and what excessive vibration is for any particular machine.

All vibration measurement starts with a time record, a registration of vibration over a length of time. A transducer converts the movement into an electric signal, which an instrument quantifies, displays and stores. The signal can then be evaluated in terms of 'good' or 'bad'.

One way of looking at vibrations is to define the type of force, which causes it. Most industrial machines are rotating, so the main force is rotational, operating on masses which are imperfectly balanced. This accounts for approximately 99% of the total vibration energy. Rotational forces are continuous and cyclic – the force does not stop (while the machine is running under power) and the movement is repeated once per revolution of a part. About 1 % of machine vibration is due to shock. Shock forces are not continuous but can be repeated, either at regular or irregular intervals. The remaining small amount of vibration, about 0.1 %, is attributed to frictional forces.

Even bearing damage can be detected through vibration analysis. A bearing produces a group of peaks in the vibration spectrum, caused by the rolling elements passing, at different speeds, over the inner race and the outer race, and by spinning around their axis. A further peak is caused by cage rotation. Given the small mass

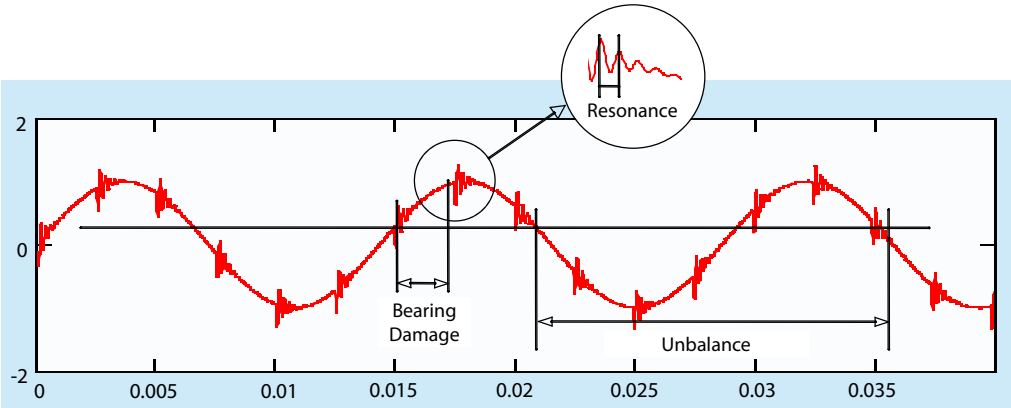


Figure 1 - Transients superimposed on wave from shaft rotation.

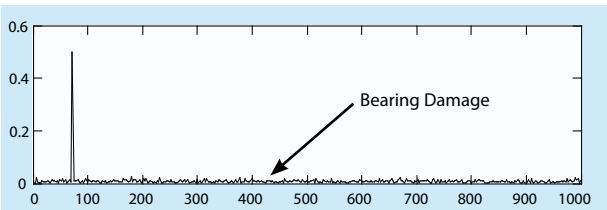


Figure 2 - Ball Pass Frequency lost in spectrum noise.

of the bearing in relation to the large mass of the machine, these peaks normally have very low amplitudes and many times are difficult to pick up with a spectrum before there is severe damage.

A shock pulse transducer contains a reference mass (m) and responds with a dampened oscillation when hit by a shock wave. Attached to the reference mass is a piezoelectric crystal which produces a voltage when compressed by the movement of the reference mass. This voltage is proportional to the amplitude of the oscillation and thus to the energy of the shock wave. The principle is the same as used in accelerometers for vibration measurement. There is, however, an important difference.

When a mass is excited at its resonance frequency, it will oscillate with much greater amplitude than at any other frequency. For vibration measurement, one normally stops measuring far below the resonance frequency of the transducer. On the other hand, shock pulse meters are mechanically and electrically tuned to operate exclusively at their resonance frequency of 32 kHz (fm), where the resulting signal is strongest. This gives us a very sensitive transducer for shocks only, but which will not react to "normal" machine vibration frequencies.

When a ball hits a damaged area in the raceway, it produces a shock wave. Shock waves are "transients" or short-lived waves starting with relatively high amplitude that quickly dampen out. In a time record displayed by an oscilloscope, these transients are often clearly seen, superimposed on the continuous wave produced by shaft rotation (see Figure 1). When the distance between transients is constant and corresponds to the ball pass frequency, this is clear evidence of bearing damage.

In the spectrum, however, peak amplitude is determined by the energy contents of the vibration at any given frequency. In relation to the energy at the shaft frequency, the energy of the shocks produced by the damaged bearing can be negligible. Thus, the ball pass frequency line has low amplitude and is easily lost among the "noise", as sown in Figure 2.

In the area around the resonance frequency, we can record a time signal, which clearly shows the transients produced by the damaged bearing. Each shock is a single event, but is also repeated at a regular rate, the interval being the time between one ball passing the damage and the next. The signal is treated by rectifying (which cuts off the negative amplitudes) and by enveloping (which produces well-defined peaks). Figure 3 illustrates this process.

The enveloping technique used by vibration analysis attempts, by manipulating the signal, to make shocks visible and measurable in the frequency domain, simply because frequency analysis is the general technique used to detect machine faults. The main strength of the Shock Pulse Method is its specialization on shock detection. The

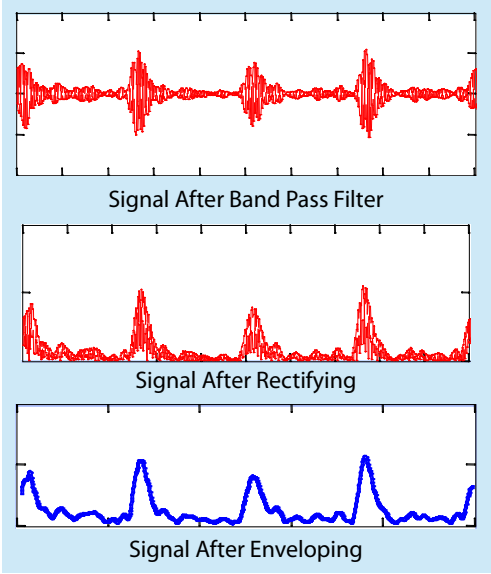


Figure 3 - Illustration of effects on signal after band pass, rectifying and enveloping.

transducer and measuring instrument are designed to measure the magnitude of shocks directly in the time domain. All generations of shock pulse meters give readouts of both the magnitude of the peaks (maximum value dBm) and of the signal level between peaks (carpet value dBc). Together, these two values can be directly translated into bearing condition information by utilizing the bearing bore diameter and rpm.

## What is the Shock Pulse Method?

Many years ago SPM took the Shock Pulse technology and developed it into the Shock Pulse Method. Through actual testing in bearing test labs, empirical data was de-

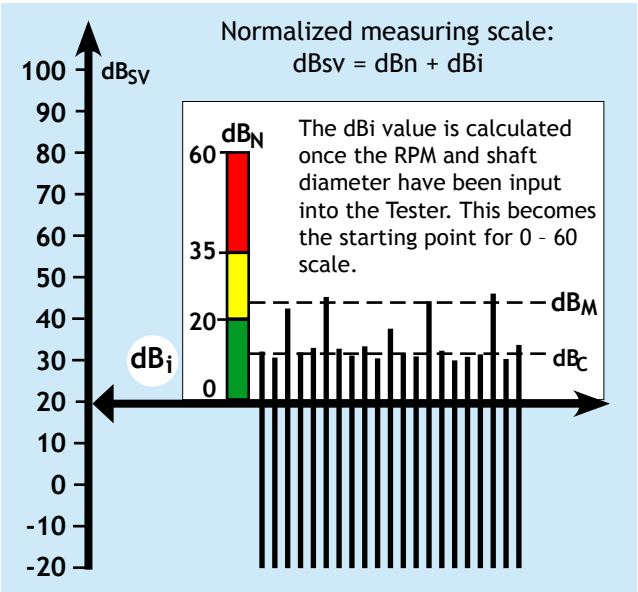


Figure 4 - Shock Pulse Method Evaluation Range/Scale

veloped by using the bore diameter and rpm. With this info a dBi value is determined, which positions the normalized condition color alarm scale onto the dynamic range of the shock pulse transducer. This enables users to utilize a standardized alarm scale, regardless of the rpm or bearing bore diameter (see Figure 4). The dBm is the maximum value, the measured value of the strongest pulses detected during the measuring interval. While the bearing surfaces are undamaged, the difference between dBm and dBc (decibel level Carpet) is small. A high dBm and a large difference between dBm and dBc are caused by surface damage or foreign particles between rolling element and raceway.

Due to the sensitivity of the Shock Pulse Method, bearing lubrication condition is measurable through the signal monitored as dBc. The dBc is measured in the time wave signal of the shock pulse transducer. The filtered transducer signal reflects the pressure variation in the rolling interface of the bearing. When the oil film in the bearing is thick, the shock pulse level is low, without distinctive peaks (green area, Fig 5). The level increases when the oil film is reduced, but there are still no distinctive peaks (yellow area, Fig 5). Damage causes strong pulses at irregular intervals (red area, Fig 5).

In 2002, SPM expanded the SPM Method by performing an FFT on the same 32 kHz signal utilized, which resulted in a more in-depth analysis capability. By identifying the different bearing frequencies (symptoms) we can now see the matches of those frequencies within the SPM Spectrum. Likewise typical symptoms such as imbalance or looseness can also be introduced for more accurate pattern recognition. The x-axis of the SPM Spectrum is scaled in Hz. The y-axis is in SD (Shock Distribution unit). The amplitude in the SPM spectrum should be used in conjunction with the SPM values. A new damage can cause high SD readings and an older more severe damage can have lower SD values. Primarily the SPM Spectrum is used for pattern recognition. It is known, but not quantified, that the delta (difference between high peaks and average level) in a spectrum is related to the bearing status.

Figure 6 shows a typical Shock Pulse Bearing Condition chart. The x-axis represents the time frame. The Y-axis is signal strength intensity divided up as a Green-Yellow-Red condition code. As explained

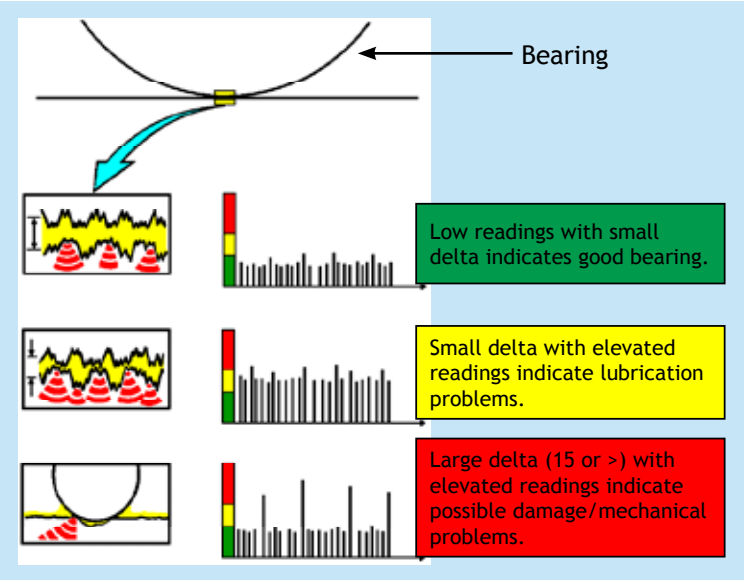


Figure 5 - Shock Pulse Method Readings and what they mean.

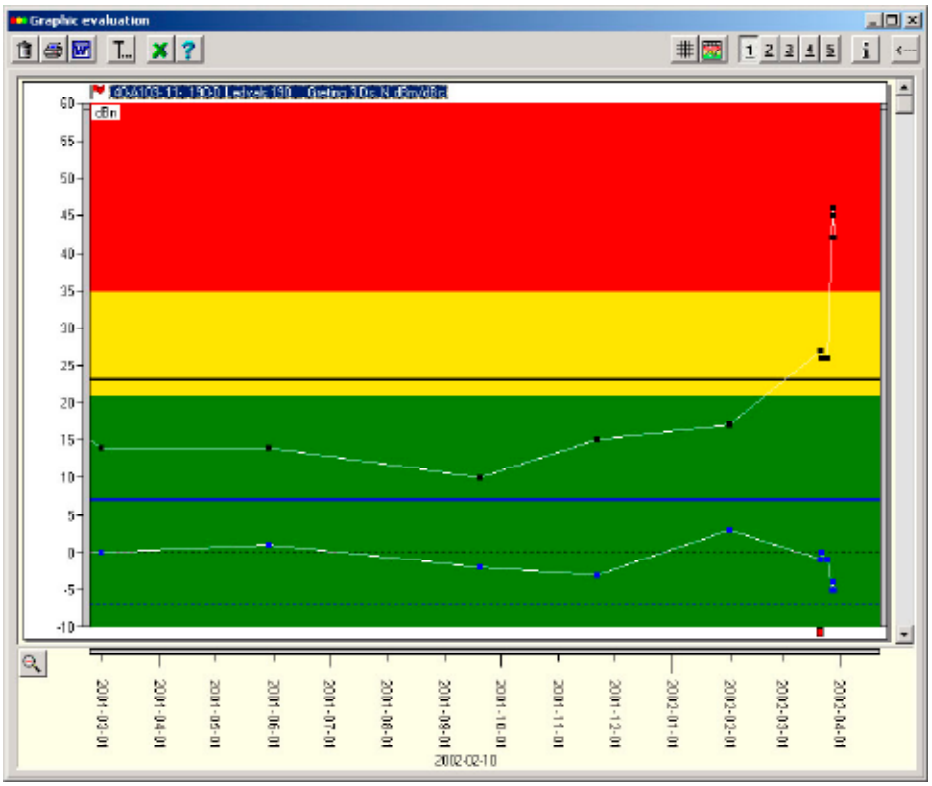


Figure 6 - Typical SPM Chart showing the dBm and dBc of a Felt Roller at a Paper Mill.

earlier the Alarm level is determined by the shaft diameter and RPM that is programmed into the instrument and/or the PC software. That defines the baseline and from there the Green/Yellow/Red divisions are further defined. On the chart we see the readings in the Green zone until about March 2002. Then they take off into the Yellow then the Red zones. Plus we see the development of a large delta (dBm-dBc), which also indicates bearing damage in progress.

Refiners are critical pieces of equipment in the paper making process. They are the part of the process that breaks down the cellulose fibers, helping them stick to one another in the paper web. With a series of rotating serrated metal disks, refiners "beat" the pulp for various lengths of time depending on its origin and the type of paper product that will be made from it.

Figure 7b on the next page is an on-line history identifying a bearing in the RED zone. It identifies damage in progression, bearing replacement and then new lower readings as a result of a new bearing. This was accomplished using only the shaft diameter and rpm. A subsequent SPM Spectrum (Figure 7c) on the same location identifies the problem area as the inner raceway. The pattern displays as an inner race defect with sidebands.

If the philosophy of front line Condition Monitoring is utilized, shock pulse measurements would be utilized as the first stage of identifying anomalies. Because the shock

Fig 7a – Refiners in a Paper Mill

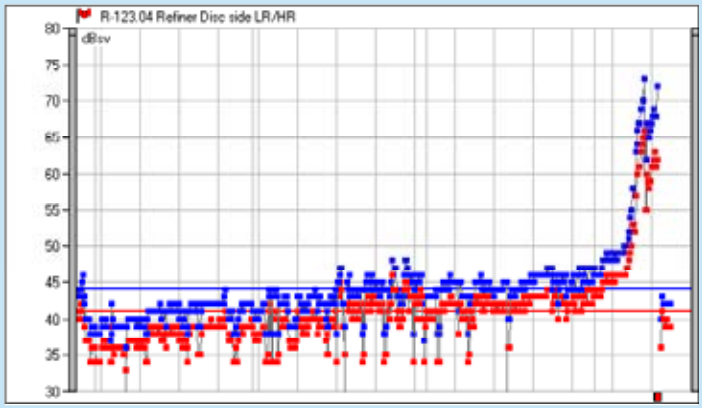


Fig 7b – SPM on-line history charting bearing condition.

Fig 7d – Damage in Inner Raceway was confirmed.

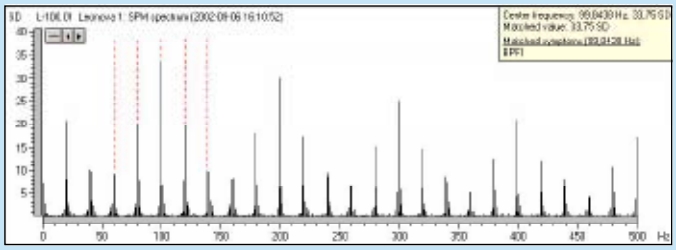


Fig 7c – SPM Spectrum on Refiner Motor

pulse transducer is "seeing" only the bearing signal, it makes the analysis of bearing condition easier to see, and provides an earlier call. With this technology, when the shock values rise and the delta (difference between dBm/dBc) increases over time, that is a prime indicator that bearing damage is in progress. And by using the SPM Spectrum, you can clearly identify bearing problems from secondary signal sources. The matching of symptoms (bearing components) makes the decision making process smoother.

In the SPM Spectrum (Fig 7c) we do an FFT on the unique Shock Pulse signal that is developed only from the compression waves being generated by the operating bearing. The individual frequencies, or symptoms, are predefined, and we simply match the symptoms with the signal patterns of the components that caused the Shock Pulse Method to go into the red. The software identifies the matches and the Y axis (shock distribution scale) identifies which symptom is generating the most shocks. Between the SPM Method identifying the bearing and the SPM Spectrum identifying the bearing component with the greatest shock saturation, the bearing call can be made more easily.

Remember the mill in Hallsta, Sweden that utilizes the SPM Method and the SPM Spectrum. They produce over 785,000 tons of magazine, book, office and newsprint paper per year. Hallsta personnel compiled data from 1993-99 on over 2326 pieces of machinery in their facility, and their average warning time is shown in Figure 8. Because

of the extended warning before failure, 100% of replacements were able to be completed during a scheduled shutdown. They calculated that this worked out to be an $800,000 contribution to profit per inspector, or $6,400,000 in total.

They have a 95% confidence level for the average prewarning times. When the bearing condition first goes from green to yellow and lubrication correction does not reverse the trend, these values represent the average warning time for bearing replacement. With this knowledge, they now use these average values to determine the corrective action and the time to replace.

The main arguments for CBM are the considerable cost reductions achieved by reducing the time it takes to make a necessary repair. A planned replacement means less waiting time and less repair time. When you also add the cost for secondary damage and lost production resulting from a breakdown, it is easy to understand why an effective CBM program costs far less than a run to failure philosophy.

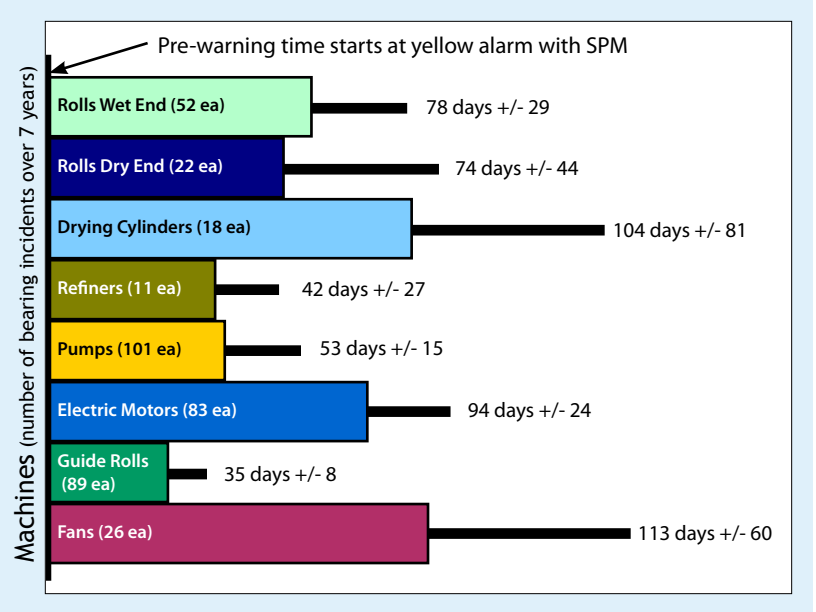


Figure 8 - Results from seven years of data on pre-warning times.

*Lou Morando is the Manager Director for SPM Instrument, Inc. in the U.S. He is a Connecticut native who graduated from the University of Connecticut with a Bachelor's of Science degree in Chemical Engineering. His early career consisted of the design and manufacture of heat transfer equipment and pressure vessels for the Nuclear Power Industry with a progression into the sales of this equipment for seven years. Lou also has 11 years experience in the design and manufacture of large tonnage compression and refrigeration systems using rotary screw compressors. He has held the managerial positions of Refrigeration Sales Manager and then Plant Manager of Industrial Refrigeration Division. Lou currently has 22 years of experience in the sales and application of Predictive and Preventive Maintenance Systems and Products for bearing and machine condition monitoring. He has written and presented numerous papers on the Practical Approach to Machine Condition Monitoring at many national and international conferences.*

*Set it and forget it oil condition monitoring. Sounds pretty good, doesn't it?*

# Vectron's ViSmart

Technological advances have made in-line fluid monitoring more feasible and cost effective than ever before. Vectron's ViSmart is a line of new breed acoustic wave sensors that can give you real time information on viscosity and temperature which are key parameters in determining your lubricating fluid's health. Karem Durdag, Vectron's Director of Business Development took some time to answer our questions about acoustic wave technology, the ViSmart sensors and in-line fluid monitoring in general. Here are his thoughts on the technology and where it will lead...

Vectron's ViSmart Sensor

*First, why don't you briefly explain in layman's terms how acoustic wave sensing in general works?*

The semiconductor chip is excited using transducers that are etched on the chip itself and vibrated at a very high frequency (160 MHz or 5 MHz). When it vibrates, it generates a sound wave through the thickness of the chip and across the surface of the material from the input transducer to the output transducer. Because of the vibration, the molecules of the fluid (oil) move and when they move, they absorb the sound wave energy. The difference in the input and output values is directly related to the viscosity-density product. The viscosity is measured instantly as soon as the fluid touches the sensor surface. It's important to keep in mind five things that Vectron has done to advance the technology so that we offer a reliable, robust commercial product:

1. The transducers are on the opposite side of the sensor surface, so all the electronics are completely and hermetically sealed from the sensor surface side. This enables the entire surface to be completely immersible.
2. There is a proprietary hard coat on the sensor surface itself which is scratch proof, abrasion resistant and chemically inert. This results in the sensor surviving in very harsh and particulate laden environments.
3. The temperature range is large (-25° C to 125° C), the viscosity range is wide (1 to 10,000 cP depending on model) and the accuracy is high (1-5% depending on conditions and fluids).
4. Because there are no moving parts it is immune to flow (as long as the fluid touches the sensor surface it will measure viscosity) and vibration (has passed 100G applications).
5. By taking advantage of standard semiconductor packaging and manufacturing practices it's cost-effective and embeddable for real-time, in-line applications.

*Now why don't you tell us what ViSmart sensors bring to the table?*

We know what maintenance professionals in the oil condition monitoring community need from a viscosity sensor. We focus on bringing trouble free, automatic viscosity measurement within reach for any facility. All of our sensors meet what we think are the fundamental requirements for a viscosity sensor:

1. The sensor has to be very robust, cost-effective and extremely easy to install.
2. It's operating range has to be wide enough to handle a multitude of oils and environments.
3. It has to be accurate enough to provide performance as a lab tool and repeatable enough to perform as all the other sensors we are used to working with (pressure, temperature and flow).

In additional to the fundamentals, what the ViSmart sensor accomplishes is to provide the customers with the following:

1. It is durable and reliable. It's been used in extremely challenging applications from down-hole drilling (extremely high temperature and pressure) to deep space (extreme high reliability).
2. It's scalable and cost-effective because it's made on a semiconductor fabrication line. This results in a solid-state viscometer that is commercially available to all customers.
3. Because it's small (our half inch threaded bolt version is only the size of a quarter), it can be installed in existing mechanical interfaces with all the communication protocols present.
4. The sensor can measure hydraulic, turbine, gearbox, engine and both synthetic and multi-grade oils and provide temperature data.
5. Of course, measuring relative changes from a baseline is something the sensor does everyday at our customer sites. We can also create correlation function on an oil-to-oil basis for 99% accuracy, if needed. Finally, because the sensor itself is made from semiconductor wafers, we can achieve % repeatability in the sensors.

*You make high shear and low shear models of ViSmart. Please explain the difference, and what applications they are best suited for.*

The high shear sensor operates at 160 MHz while the low shear sensor operates at 5 MHz. The differences are that the high shear has very wide operating range (3 to 10,000 cP) and measures the viscosity of extremely thick fluids (in the excess of 500 cP at room temp). During its operation, it will impart shear thinning to the oil. We can also work with the customer to determine if measuring relative viscosity changes or correlating to lab measurements is a desired solution. The accuracy levels on this sensor range from 5% to 10% depending on the application. The applications it is best suited for are:

a. Process fluids in the coatings industry (ranging from paints to inks to slurry to adhesives and specialty applications).
b. Very viscous oils such as those used in wind turbine gearboxes.

The low shear is specifically targeted to the oil condition customers that need to measure the viscosity of oil from 1 cP to 500 cP with accuracy levels of 1%, and there is significantly less shear thinning imparted to the oil. We can easily report the data correlated to the lab tools or use it for relative measurements. The applications it is best suited for are: mobile applications (on & off-highway vehicles, including marine), fixed asset (power generation, turbine, gearboxes), plant-wide asset (rotating equipment), hydraulics and petroleum.

*Are there any conditions (types of liquids, harsh environments, etc.) where ViSmart sensors should not be used?*

We recommend that the sensor not be used with hot (in excess of 60° C) concentrated sulfuric acid or sodium hydroxide. Also, given the performance and resolution range of sensors, we recommend not using the sensor for liquids that are 1 cP or less in viscosity.

*Please explain the data collection process of the sensors and how they interface with the user's data system.*

The signal generated is a voltage. Using converters and signal processing we can provide 4-20 mA outputs and other standard protocols (for example, we offer USB connectivity) for ready interface to any host control system. We are currently working on our next generation sensor that has CANBUS and DeviceNet capability.

*In-line, real-time fluid monitoring seems like the most efficient method to gauge the health of your fluids. Why doesn't every company that uses fluids employ in-line fluid monitoring?*

Well, for a long time the ideal solution was not available from a size, cost and reliability point of view. This has resulted companies continuing to use traditional methods. Given the new advances in technology and products, there is a learning cycle and a level of comfort that needs to be established. We are confident that as more people realize the availability of these kind of commercial products, more and more companies will utilize in-line fluid monitoring.

*What are the three top reasons a company should consider investing in ViSmart sensors?*

First - It provides a portable and embeddable instantaneous, real-time, in-line measurement of viscosity that is a key parameter in process control and oil condition monitoring.
Second - It delivers the tool to optimize costs and make objective real-time decisions, whether process or equipment related.
Third - It is flexible enough for customization, allowing our customers to differentiate themselves from their competitors.

*What is time frame a company can expect for a return on their investment in ViSmart sensors?*

It varies, but our on average customers report a return on investment on the order of 3 months or less. The key aspect in this determination is if the customer already has a strong database of cost information of capital equipment costs, day-to-day maintenance costs and scheduled down-time. The sensor enables cost optimization in all of those regards and can be readily assessed. For applications, where the sensor is used for process control, cost of elimination of waste and reduction in trouble-shooting is also factored in. And finally, for applications where the sensor is embedded, the cost return is measured in extension of the life cycle of the equipment asset, readiness of the asset for operation on a 24/7 basis and prevention of catastrophic failure.

*Give us a success story or two from companies that are using ViSmart sensors now.*

One great success story is an install in a marine diesel engine where the customer using the ViSmart sensor to monitor whether the right oil is being used or not, amount of fuel dilution (Fig 1) and if the viscosity of oil is within spec across the entire temperature range as it gets used (Fig 2). The sensor has been used 24/7 for 6 months in this challenging shipboard environment. The customer has been very satisfied with the results.

*How can interested people get more information about ViSmart sensors?*

Our community members can visit www.visensor.com to get all the information they need and call Dan McCormick at 603-578-4077 to ask for in-depth information.

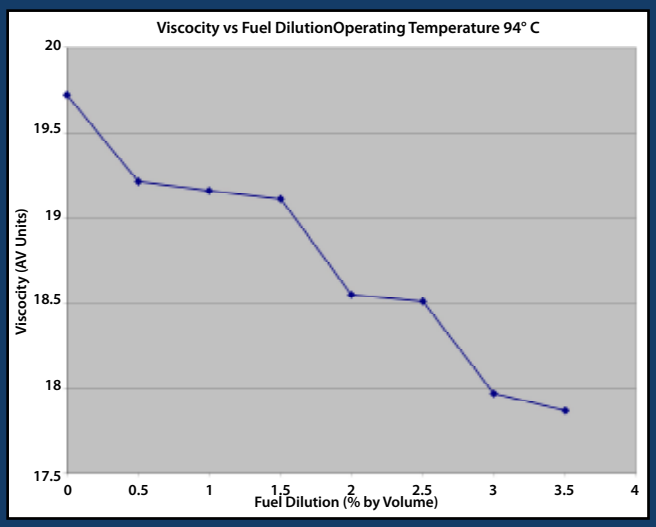


Fig1 - Viscosity change as a function of fuel dilution

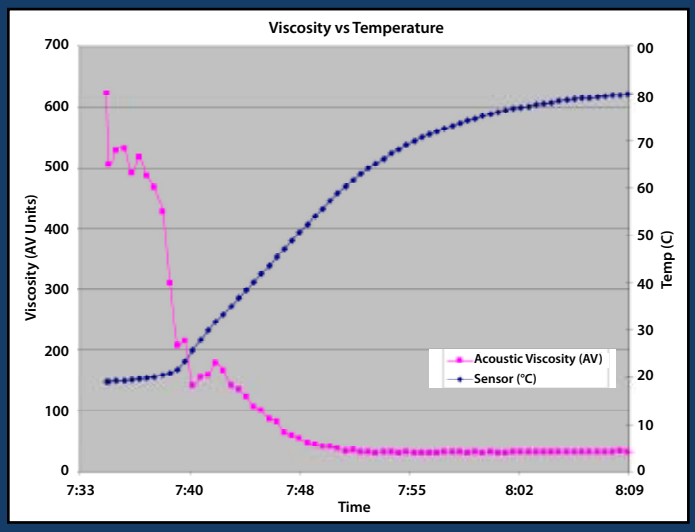


Fig 2 - Viscosity change vs Temperature

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The Kittiwake Viscotube is a device suitable for quick, simple testing of hydraulic oils, lubrication systems and fuel oils. The Viscotube is light, easy to use and offers a high degree of accuracy for an unlimited number of tests, while its price ensures an effective return on investment after only a few tests. The standard unit of measure for oil is in centistokes at 40C. With the Viscotube, the samples can be tested on-site for later input into the viscosity calculation software that is provided with the set. Weighing only 0.5 kg, the Viscotube can easily be carried to the sampling points throughout the plant, site or ship.

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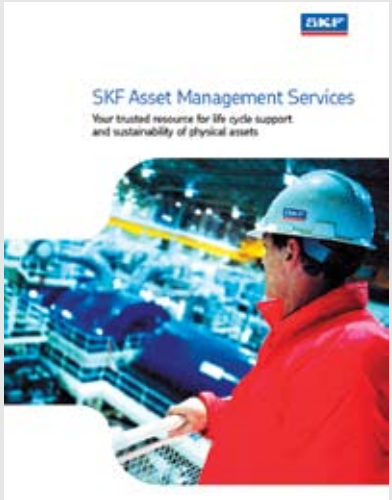


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Hawk IR International has announced the launch of its Infrared Safety Centre (ISC) devoted to the training and certification of infrared thermographers operating within explosive atmospheres. ISC incorporates COMPEX, DSEAR and ATEX modules within a Level 1 HazEx course in addition to infrared training techniques developed to increase thermographer awareness and competency. ISC will provide HazEx Thermography training for both trained safe area thermographers and newcomers. The five day HazEx training course incorporates 50% safety and 50% thermography with two independent exams. Successful students will be awarded HazEx Thermography Certification along with ATEX Competent Person status.

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