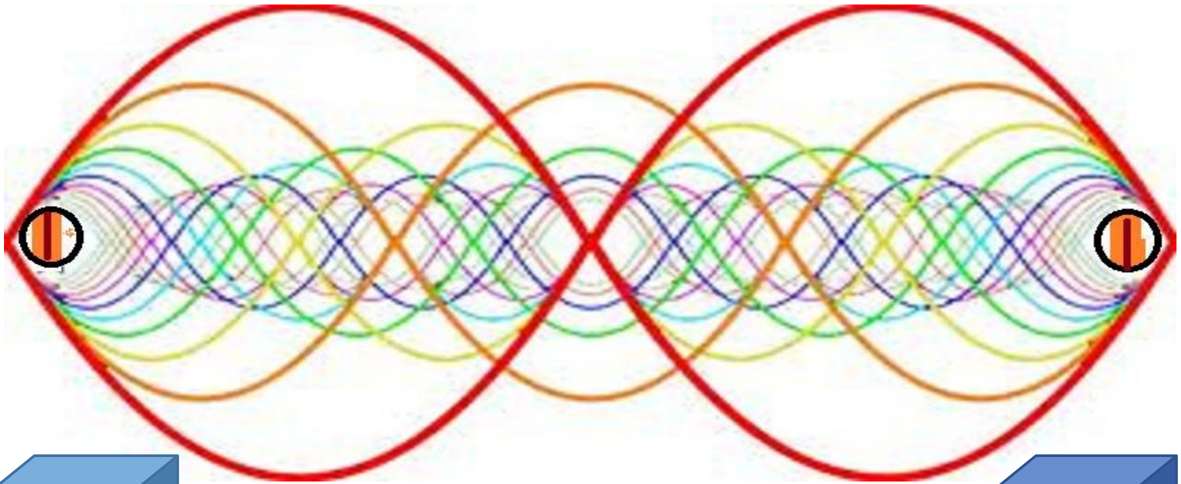


Q4 2021

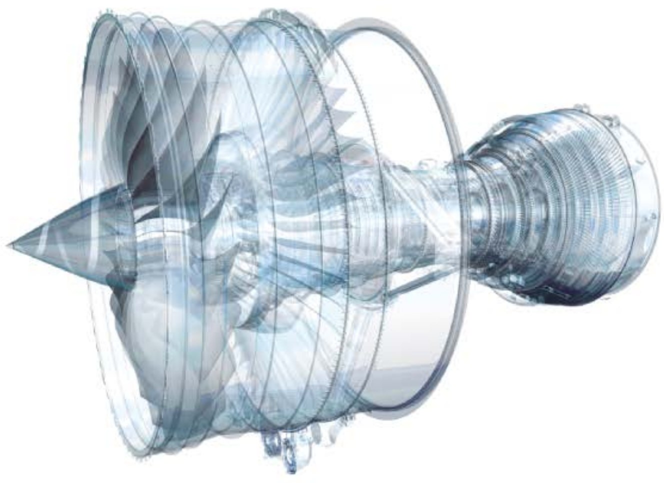
Resonance

Quarterly Magazine of Sumitek Solution

Sumitek
Turbo-machinery Solutions

What is stored for you

1. MEET THE EXPERT
2. HOW MUCH DO YOU REALLY KNOW ?
3. TECHNICAL ARTICLES



HOW MUCH DO YOU REALLY KNOW ?

We hope that you have started preparing YOUR book of knowledge !!!

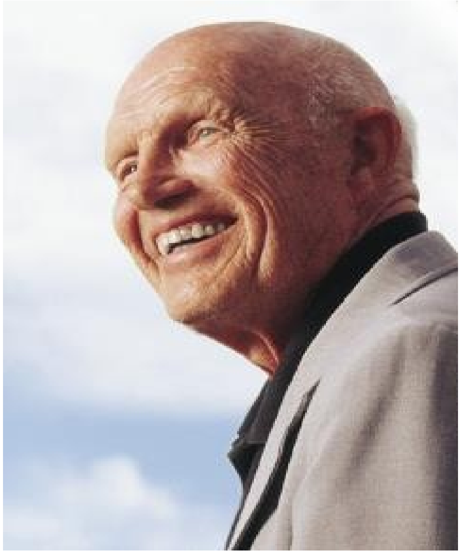
Those who had purchased our custom made training material and If you have any questions left unanswered by you , ask us .
We shall be happy to answer for free those questions ,once you are our premium member . Contact us for premium membership .

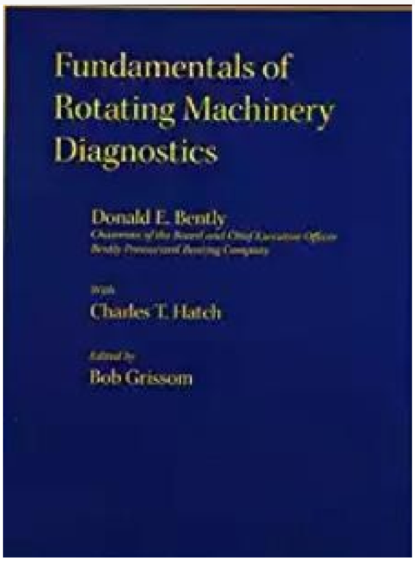
These questions shall help you to prepare for job skill enhancement , job interview and attain superiority over your peers !!!

Meet the Expert

DONALD BENTLY

Founder of Bently Nevada Corporation now known as GEMCS





BENTLY
Nevada

In 1981, Bently established a pure research organization called Bently Rotor dynamics Research Corporation (BRDRC). BRDRC's objective was to conduct rotor-dynamic research, furthering the knowledge of rotating machinery behavior, modelling techniques, and malfunction diagnostic methodologies. Its mission was considered complementary to Bently Nevada, with BRDRC focused on understanding how machinery behaved, and Bently Nevada focused on understanding and building instrumentation to measure machinery behavior.

BRDRC made a number of important contributions to the field of rotor dynamics such as a better understanding of fluid-induced instabilities, advanced models for understanding shaft crack behavior, insight into rubbing malfunctions between stationary and rotating parts, and enhancement of the rotor dynamic equations via introduction of a new variable lambda (λ) which denoted the fluid circumferential average velocity ratio and more accurately modelled hydrodynamic effects. Bently was personally responsible for many of these developments, publishing his work under the auspices of BRDRC

Compressor curve variation during site acceptance test

In a recent site acceptance test of turbo-compressor unit , it was found that the shape and position of the centrifugal compressor curves varied from the design performance curve. These compressor curves were taken from the compressor as tested curve and contractual data sheet, and then used to calculate polytropic head and actual inlet flow for 12 test data points, at over 4 different compressor speeds. These values were further verified using software (HYSYS) calculation.

The performance points were plotted on the design performance curves. In addition, the fan laws were used to calculate additional 'design' speed curves, at compressor speeds equal to those used to gather the performance data. This allowed a direct comparison between the measured performance points and the calculated design speed curves. The plot is shown below -

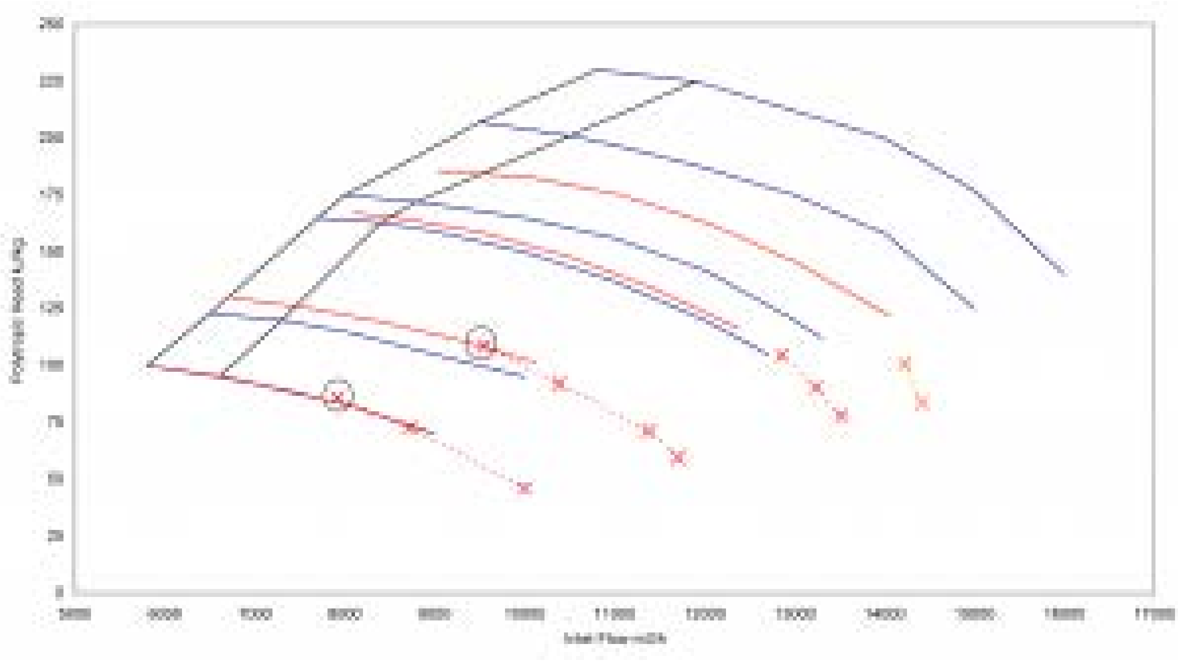


Figure 1 compressor curve interpolation

Where

- Blue lines – Design constant speed curves from built performance curves at 70%, 79%, 89%, 92%, 100% and 105% speeds respectively
- Black lines – Design Surge Control Line (SCL) and Surge Limit Line (SLL) from built performance curves
- Red solid lines – Calculated constant speed lines at 72%, 80%, 90% and 95% speeds
- Red Crosses, connected by Red dashed line – Performance data points from on-site performance testing at 72%, 80%, 90% and 95% speeds

- Circled Crosses – Performance points with a Compressor controller DEV of almost zero, expected to be on the SCL.

Interpretation of data

The performance data at site correlated well with the design compressor curves. This was indicated by a DEV value on the compressor controller of 0.05 (i.e. almost 0) for two of the points taken (circled). This raised doubts regarding the correct set-up and positioning of the Compressor controller SCL.

Compressor controller surge control line set-up

The set-up of the compressor controller SCL was investigated, particularly the calculation of flowrate from the V-Cone flowmeter device pressure drop (deltaP) reading. It was seen that for a given deltaP, the calculation of mass flowrate by the compressor controller did not correlate with the calculated mass flowrate by process engineering department.

For example, flow measuring device (FMD) delta P of X bar at X bara suction pressure:

Compressor controller equated this to Y kg/h

Process engineering department equated this to Z kg/h

Root cause of discrepancy:

It was found from the compressor controller manual that the system calculates flowrate from the pressure drop across the flowmeter device, and it is dependent on a constant A, as per the equation:

$\Delta P = (Q_s/A)^2 \rho_s$

Where

Q_s = Suction flowrate

ΔP = pressure drop across the flowmeter device

ρ_s = Suction gas density

A = FMD (Flow Meter Device) A constant

FMD A constant is calculated using a number of factors and coefficients specific to the flowmeter device sizing, as well as the service conditions:

$A = N \times F_a \times C_F \times Y \times E \times \beta^2 \times D^2$

This equation enables the conversion of pressure drop across the device into flowrate in the compressor controller software. However, the values used in this formula as stated in the compressor controller manual, differed from the values stated on the flow measurement device data sheet. It was, therefore, suspected that compressor controller could have been supplied with incorrect flowmeter device sizing data.

To correct this error, an inaccurate constant ‘A’ was being used within the compressor controller software, and would subsequently lead to the incorrect calculation of flowrate by the compressor controller. This led to the compressor controller SCL being set up significantly to the right of the design SCL.

Corrective action

The compressor manufacturer and the compressor controller software supplier, along with the site process engineering department support, worked out the embedded errors in the calculations and resolved all discrepancies in the calculation of flowrate.

First, the nameplate of the flowmeter device in the field was reviewed to ascertain the correct device sizing. This data was used to calculate and check the compressor controller ‘A’ constant and ensure that the correct ‘A’ constant is used within the compressor controller software. The software will need to be updated in the field as required.

The compressor controller testing was then repeated to correctly set the Surge Control Line and shaping of compressor curve.

Configuring lubrication systems

Centrifugal compressors have lubrication systems designed as per API 614 consisting of an oil reservoir, pumps, filter, cooler and oil lines. The programmable logic controller (PLC) starts the auxiliary AC power-driven lubricating oil pump. Pressure at the oil header creates starts the drive. If the header pressure is low, the standby oil pump starts and maintains oil pressure during operation. Oil is pumped to the lubricated parts (radial and thrust bearings). If a speed increaser gear is present, they are lubricated with the same oil as the bearings (Figure 1).

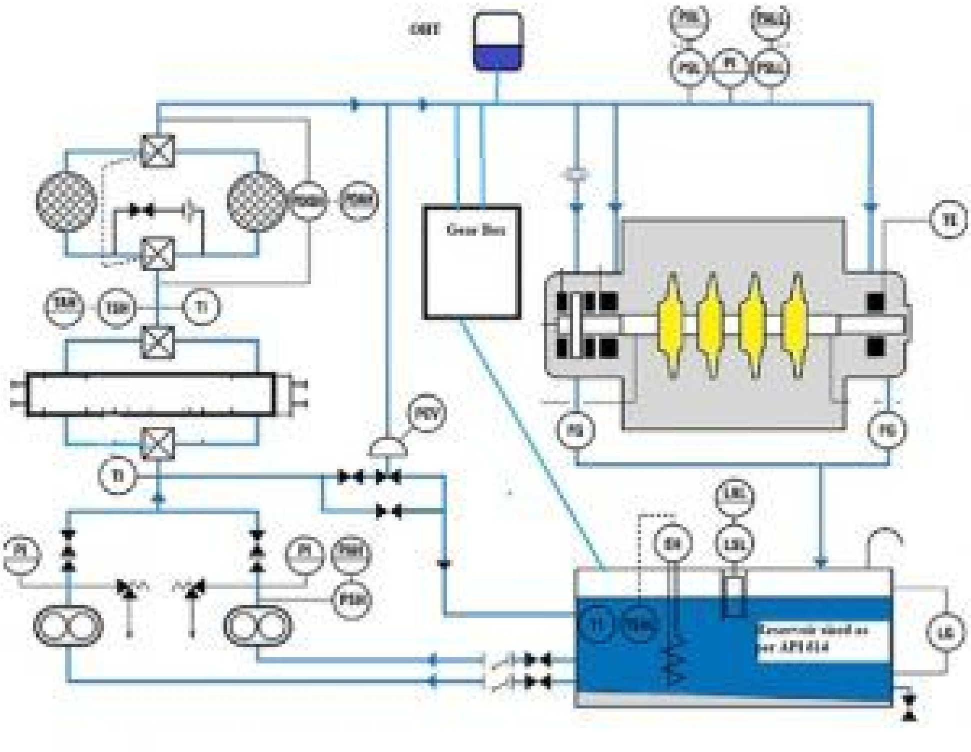


Figure 1: Generic lube oil system diagram for a centrifugal compressor

Some end users require two pumps that have different types of drive or power sources. When both are driven electrically, they have to be connected to separate supply feeders in order to maintain constant and uninterrupted operation should a feeder fail. However, it is not required for oil coolers to have such redundancy.

In some compressors with step-up gearboxes, the main oil pump is required to be driven mechanically from the gearbox, and the auxiliary pump operates during the start-up and run-down phases of the compressor train. As API 614 calls for the oil reservoir to be away from the skid with a slope requirement, shaft-driven mechanical pump performance may be hampered by net positive suction head-related (NPSH) air bubble issues (Figure 2).

Figure 2: API 614 compliant lubricating oil system

Sometime a screw pump connected to a gearbox can cause rotor-dynamic problems due to high overhung weight. Shaft-driven gear pumps are mostly used to maintain oil supply during coast down in the case of an electrical power outage as compressors need lubrication until their rotating shafts come to a complete stop.

A conventional approach is to maintain lubrication for a fixed period of time following a shutdown request. Based upon an estimate of the time for a rotating shaft to stop, this approach, does not always ensure that the shaft receives enough lubrication. Therefore, a rotating shaft and its corresponding bearings may be physically damaged if it rotates after that fixed period. To avoid possible issues, therefore, a well sized run-down tank is a better alternative.

Another big lubrication challenge is avoiding bearing damage during reverse rotation. This can occur during coast down, gas blow down, or short circuiting of high-pressure gas into a standby compressor .Most of the time, radial bearing pads have an offset pivot design while thrust bearings have a tapered land-thrust design that has a significantly lower load-carrying capacity in the reverse direction. The compressor designer can specify center-pivoted pads in bearings and bi-directional seals. But regardless of the design, rotor train inertia must be higher than gas inertia. This cannot be guaranteed for a high-pressure compressor if it has a low rotor weight. In that case, due diligence is required to overdesign the overhead tank OHT to meet the duration of reverse rotation, and to create a short circuit between the suction and discharge lines with a class IV sealing valve (open when the compressor is not in operation) or a fast-acting recycle valve.

If the compressor is a single operating unit with no standby, it is advisable to start the lube oil pump first before beginning blow down of the compressor loop. The duration of operation of the lube oil pump is dictated by API 521 as regards the blow down rate of loop. A non-slam check valve at the discharge side is usually installed although a clause of API 521 states that a single check valve cannot be regarded as a reliable layer of protection against reverse flow.

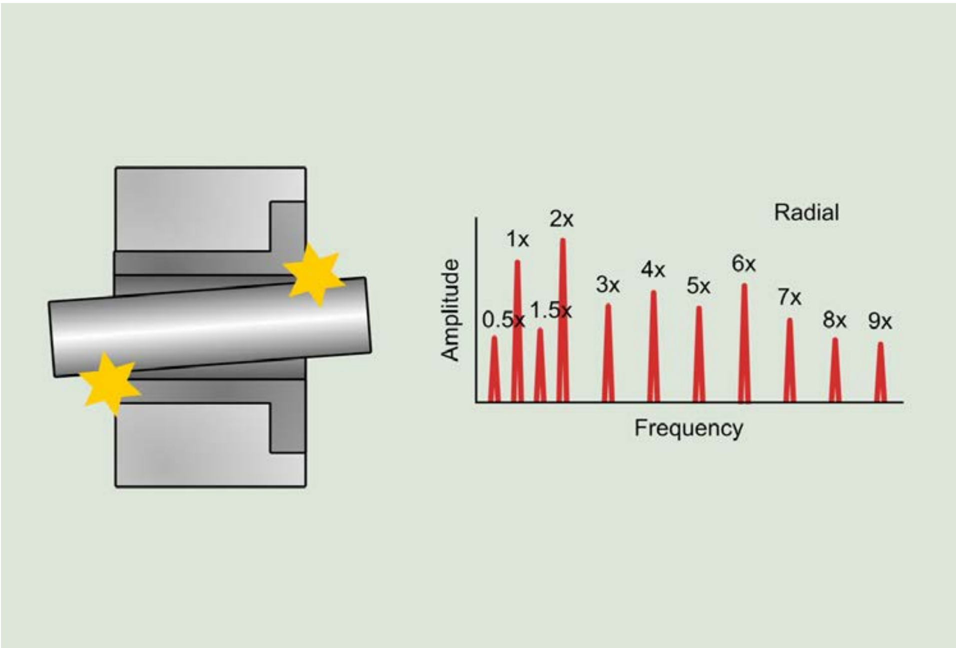
Another option is to provide a reverse rotation detector which can give a command to start a DC pump or the installed auxiliary AC motor driven pump. The reverse rotation detector can be set to arm a digital tachometer connected to a microprocessor or PLC. The tachometer should be suitable for low RPM detection. The RPM signal from the tachometer is transmitted to the PLC which can the start the auxiliary or DC pump. All of the above points must be taken into account while placing an order for a turbo-compressor package.

Failure Prevention

LOOSENESS IN MACHINERY – NOTES

Rotating element looseness

This type of looseness is characterized by a spectrum containing many frequencies related to the rotating frequency of the rotor, being able to distinguish: harmonics (1x, 2x, 3x, etc.), subharmonics (0.5x) and half harmonics (1.5x, 2.5x, 3.5x, etc.). They generally manifest more clearly in the radial directions of measurement (horizontal and vertical) than in the axial direction.



As the wear increases, the frequency spectrum changes, allowing a classification of the severity of the looseness into the following four groups:

- Incipient looseness: presence of the first four or five harmonics of the rotating speed, showing less amplitude the higher harmonics.
- Potentially serious looseness: increase of the amplitude of the first harmonic of the rotating speed and appearance of very low amplitude half harmonics.
- Serious looseness: increase of the amplitude of the harmonics and half harmonics of the rotating speed.

- Severe looseness: it is characterized by presenting half harmonics, harmonics and subharmonics of the rotating frequency of the rotor, so that a wide energy band is observed in the spectrum.

Structural looseness

These refers to looseness associated with the mechanical non-rotating elements of the machine: bench fixings, joints between pipes, bearing casings, etc. Normally, it manifests more clearly in the radial measurement directions than in the axial ones, with the presence in the frequency spectrum of several harmonics of the rotating speed of the shaft. The comparison of the amplitude of the 2x and 3x harmonics with respect to the frequency at 1x provides an indication of looseness severity, so that an amplitude of these harmonics above 50% of the amplitude of the peak at 1x, points to a greater severity of the looseness. The reason why this kind of looseness manifests in frequency spectra with several harmonics of the rotating frequency (1x, 2x, 3x, 4x, etc.) is illustrated in the following figure. The rotor presents a slight unbalance as a driving force of the looseness due to loosening of the fixings between the bearing supports and the bench. In the four stages of the figure we can see that as the unbalance heavy spot rotates to complete a full revolution, there are four forces or impulses, indicated in the four stages, two of which are due to the unbalance and the other two to the return of each of the sides of the support to the bench. This will give rise to several harmonics of the rotating frequency in the vibration spectrum.

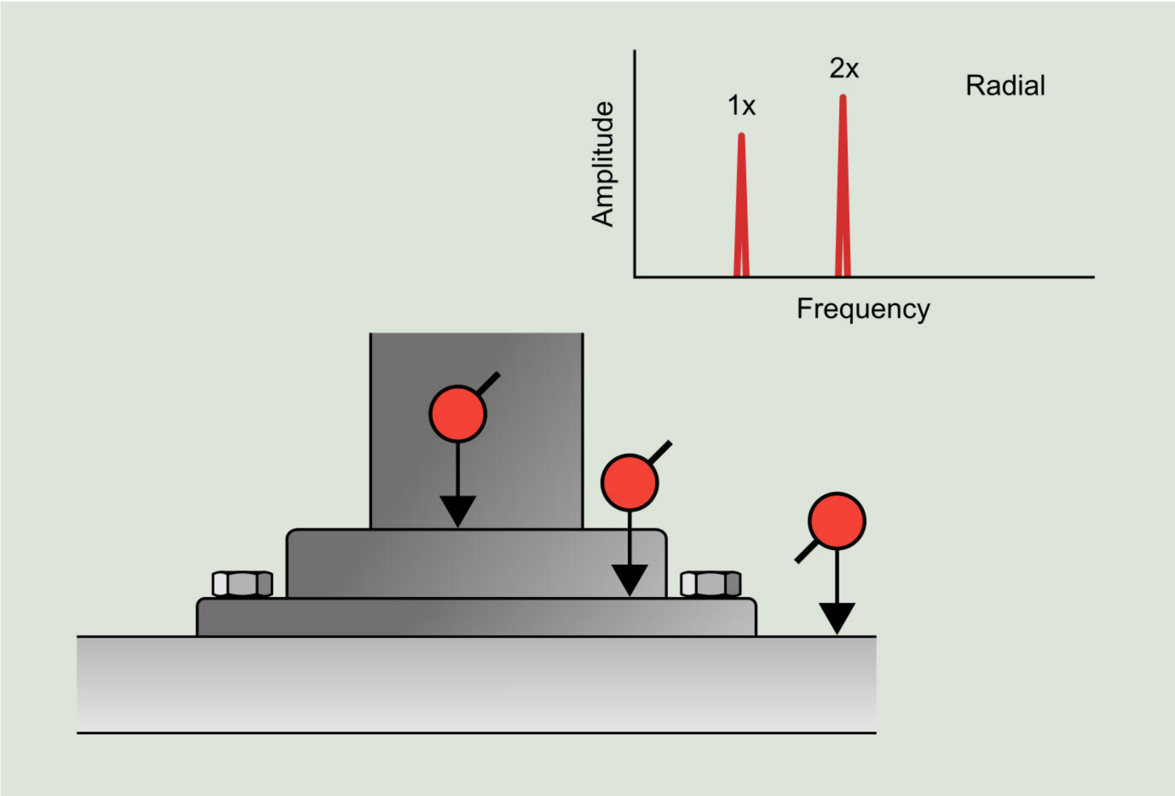


Figure 2: Structural looseness

To identify and locate whether or not looseness is present between two structural elements, such as a motor foot and the bench, due to loosening of the bolts or breakage of the anchor, there is a very simple method that consists of taking vibration amplitude and phase readings, in all possible measurement directions (axial, horizontal and vertical) in the two elements under analysis. The data obtained can give us two possible well differentiated results, according to which we will determine with total reliability the presence or not of looseness between them:

- If the spectra obtained in the same direction for the two elements shows similar amplitudes and in addition the phase readings are identical then this is indicative that there is a good union between the two elements.
- On the other hand, if the amplitudes of the harmonic peaks of the rotating frequency in the same measurement directions for the two elements are different, and in addition there are important differences in phase between them, then we can confirm the existence of looseness between both elements.

Do it in Right way

1. Right goals	• Having clearly defined and achievable goals that may evolve over time
2. Right People	• Having the right people in the right roles with the right training.
3. Right leadership	• Inspiring continuous improvement.
4. Right tools	• Having the right tools and technology to help reach the goal.
5. Right understanding	• Equipment audits, reliability and criticality audits, FMEA, maintenance strategies, etc.
6. Right data collection	• Collecting the right data at the right time to detect anomalies, defects or impending failures.
7. Right analysis	• Turning data into defect or fault diagnoses.
8. Right reporting	• Turning data into actionable information and getting that information to those who need it at the right time and in the right format.
9. Right follow up and review	• Acting on reports, reviewing and verifying results, benchmarking, auditing and improving, etc.
10. Right processes and procedures	• Tying together: people, technology, information, decision making and review.

Shock Pulse Monitoring also known as SPM is a patented technique of predictive maintenance by measuring vibration and shock pulses of bearing in motors and to identify their condition and operating life before the next overhaul procedure.

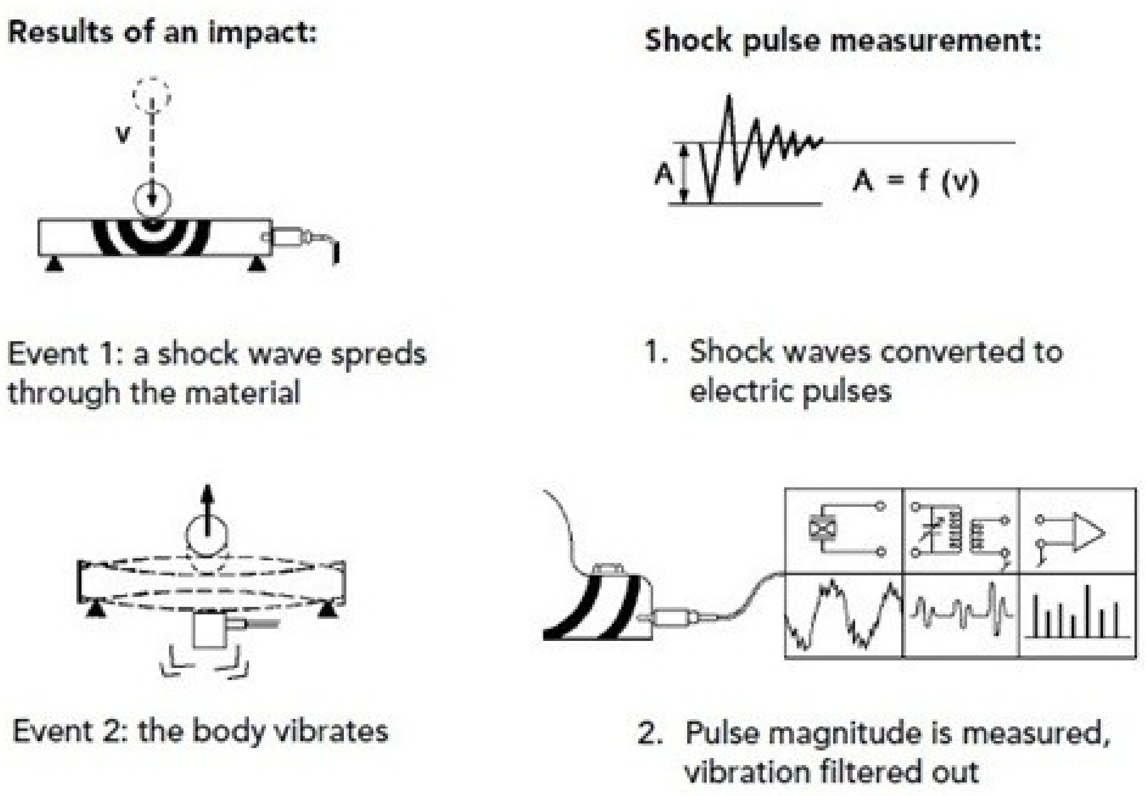
It was introduced in 1969 and is now a known Condition Monitoring method for monitoring of machines like electric motors using roller bearing.

Difference between SPM and other Vibration Techniques

Vibration analysis has been used for motor predictive maintenance for many years. Traditionally marine engineers have been using a listening rod to listen to the sound and ascertain the condition of rotating machinery. Nowadays to avoid premature overhaul of motors many companies are supplying vibration analysis pens and SPM instruments on board ships. However there is a difference between vibration analysis and Shock Pulse Measurement.

When two metal surfaces contact each other while in motion, an impact occurs and a shock wave develops, which travels through the metal. The shock wave is in ultrasonic range and is around 36 KHz. This shock wave is utilized in the SPM.

As impact continues the metal flexes and is compressed and recoils. This second phase is called vibration. The frequency of this vibration is dependent on stiffness, shape, mass and the dampening property of the material. In SPM this phase of collision i.e. vibration is filtered out as it is depending on the structure and the material of the machine. Thus the inaccuracies that are frequently encountered by hand held vibration analysis pens is not here, especially in machinery with flexible mountings and working in vicinity of other vibration prone machinery.



Another difference between SPM and other vibration measurement techniques is that in SPM the transducers respond and resonate to a frequency of 36 KHz only, which ensures a calibrated response and accurate measurement to the shock pulses.

How does SPM Works?

Whether new or old, any bearing generates shocks in the interface between the loaded roller element and the race way. Initially these shocks or vibrations are subtle and hardly felt till already damage is done, but these are captured routinely by the SPM machine which tells about the condition of the bearings, the state of lubrication and the maintenance interval required. This type of monitoring and maintenance based on this evaluation is called as Condition Monitoring System or Condition Based Monitoring.

The amplitude of shock pulses measured by the SPM meter is due to the following factors:

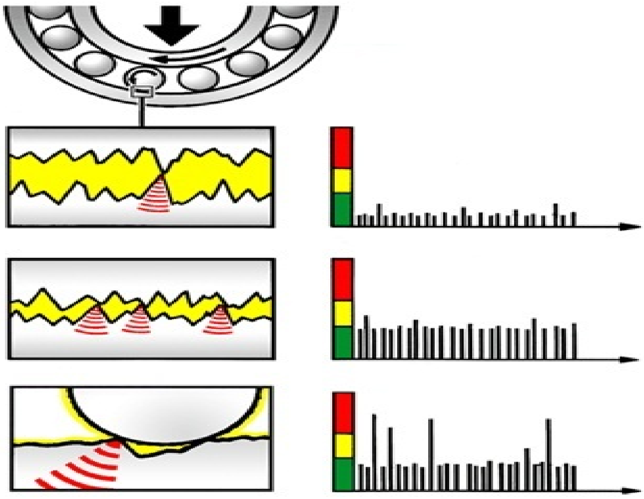
- Rolling velocity which is a function of speed or rotation and size of bearing.
- The thickness of the oil film, which in turn depends on preload and the quantity of oil supplied as well as the viscosity of the oil.
- The alignment of the system. That means between the prime mover and the load.
- Other mechanical factors like roughness of the raceways, the stress and damages.

The shocks are received by the SPM transducer which then gives an output signal proportional to the magnitude of the shock felt. The SPM meter measures the shock pulses per second and then lowers its threshold so that two amplitude levels are discovered, first the decibel carpet value of 200 shocks per second and secondly, the maximum level of incoming shock under 2 seconds.

The decibel carpet value gives an indication of the condition of the lubrication and the peak value gives the extent of bearing damage.

The peak value can be ascertained by increasing the threshold value till no signal is received. In this equipment the noise generated due to the rolling velocity is negated by entering the shaft diameter and RPM of the motor. This gives an accurate condition assessment of the machine being monitored.

The amplitude of the shock is a function of the rolling element and the instrument measures the absolute value and subtracts from it the expected shock value from a good bearing at similar speed. This gives us an indication of the bearing operating condition.



There are three condition zones namely Green for Good Condition, Yellow for Caution and Red for Damaged condition. The peak value measured by the operator gives the status of the machine and the zone it belongs to.

SPM is not influenced by the size, speed, design, background noise or installation of the machine

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