An Engineer‘s Guide

Making Maintenance Matter

Optimizing plant availability using

- Laser shaft alignment
- Vibration analysis
- Dynamic balancing
- Wear debris analysis

An Engineer‘s Guide

to shaft alignment, vibration analysis, dynamic balancing and wear debris analysis

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Introduction

The purpose of producing this handbook is to provide basic information and guidelines for the implementation of good shaft alignment, vibration analysis and dynamic balancing practice for standard rotating machines systems.

Laser alignment, dynamic balancing and condition monitoring are essential components of a viable maintenance strategy for rotating machines. In isolation, each strategy helps to reduce unexpected machine failure. But taken together, they form the hub of a proactive maintenance strategy that will not only identify incipient problems, but also extend machine operating life considerably.

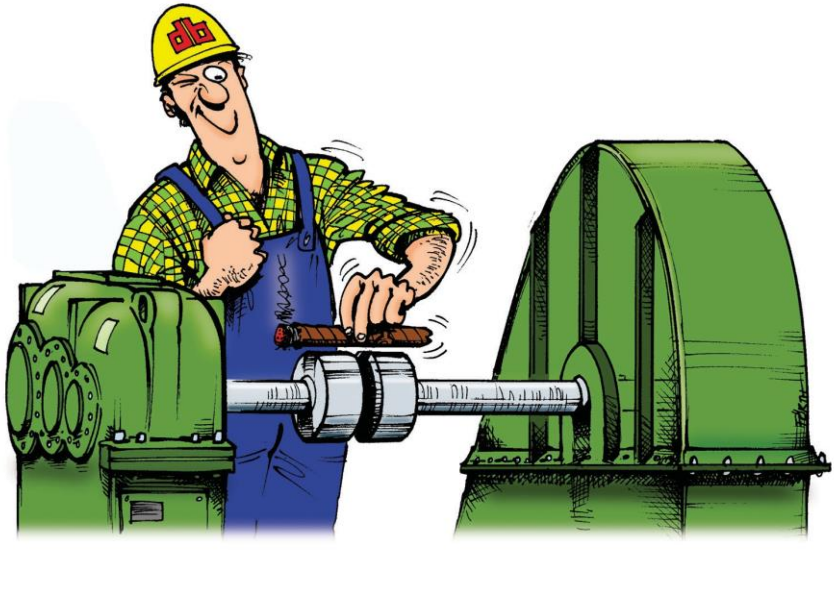
In each section of this handbook, we have used one or two examples of the available methods for measuring the required parameters. We do not suggest that the methods illustrated are the only ones available. For any- one wishing to pursue further, the subjects covered here, a bibliography of some of the available literature is to be found at the end of this handbook.

PRUFTECHNIK are specialists in the alignment and monitoring of rotating machines. We have accumulated substantial practical knowledge of these subjects over the 40 years of our existence. In so doing we have produced many handbooks covering individual subjects and systems. This handbook is a distillation of this accumulated knowledge. Each section has a brief overview of the latest PRUFTECHNIK systems that address the specific ap- plications concerned.

We hope that this information is presented in a clear readable form, and will provide readers new to the subject, a platform to successfully apply profitable maintenance practice on their plant.

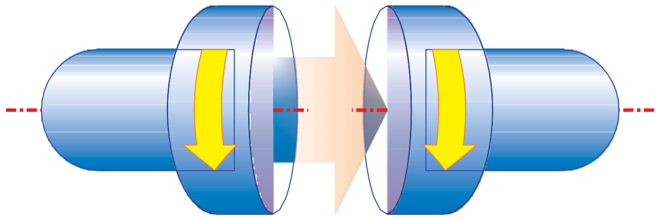
Section 1

Shaft Alignment



What is shaft alignment?

Shaft alignment is the process whereby two or more machines (typically a motor and pump) are positioned such that at the point of power transfer from one shaft to another, the axes of rotation of both shafts should be collinear when the machine is running under normal conditions.



As with all standard definitions there are exceptions. Some coupling types, for example gear couplings or cardan shafts, require a defined misalign- ment to ensure correct lubrication when operating. Note the following important points in the above definition:

At the point of power transfer:
All shafts have some form of catenary due to their own weight, hence not straight. Therefore the location where the alignment of the two shafts can be compared is only at the point of power transfer from one shaft to the next.

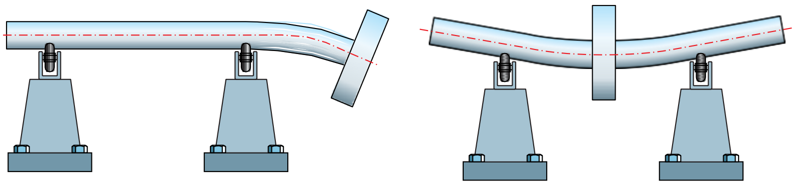
The axes of rotation:
Do not confuse “shaft alignment” with “coupling alignment”. The cou- pling surfaces should not be used to measure alignment condition since they do not represent the rotation axis of the shafts. To save manufacturing costs coupling surfaces are often only rough machined or in some cases not machined at all.

The accuracy of the fit of the coupling on the shaft is unknown: Rotating only one shaft and using dial gauges to measure the opposing coupling surface does not determine the axis of rotation of both shafts.

Under normal operating conditions:
The alignment condition can change when the machine is running. This can be for a number of reasons including thermal growth, piping strain, machine torque, foundation movement and bearing play. Since shaft alignment is usually measured with the machines cold, the alignment con- dition as measured is not necessarily the zero alignment condition of the machines.

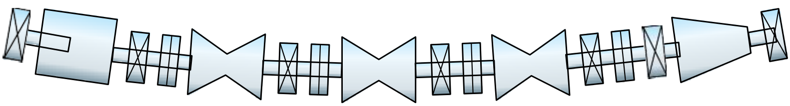
Alignment condition should be measured whilst turning the shafts in the normal direction of rotation. Most pumps, fans and motors etc. have ar- rows on the end casing showing direction of rotation.

Machinery catenary:
The amount of shaft deflection in a machine depends upon several fac- tors such as the stiffness of the shafts, the amount of weight between overhanging supports, the bearing design and the distance between the supports.



The natural bending of shafts under their own weight

For the vast majority of close coupled rotating machines this catenary bow is negligible, and therefore for practical purposes can be ignored. On long drive machine trains, e.g. turbine generators in power generation plants or machines with long spacer shafts e.g. cooling tower fans or gas turbines, the catenary curve must be taken into consideration.

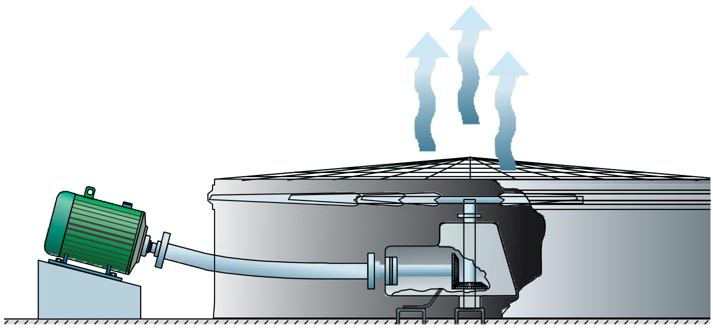


Machinery catenary

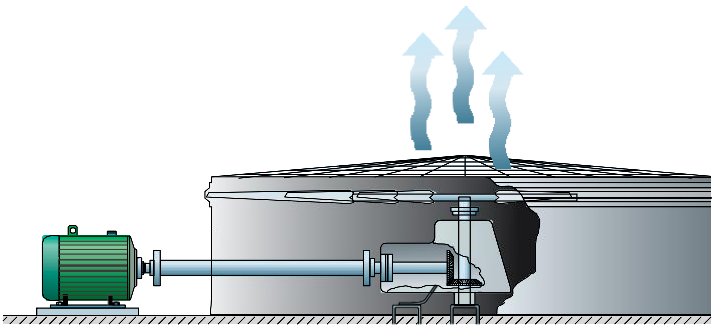
In a steam turbine for example the shafts are usually aligned to each other better than 1/100th mm, but the mid point of the center shaft could be as much as 30 mm lower than the two end shafts.

Operation above critical speed?

When a very long, flexible shaft begins to rotate, the bow of the shaft tries to straighten out, but will never become a perfectly straight line. It is im- portant to understand that the axis of rotation of a shaft could very possibly run on a curved axis of rotation. In situations where two or more pieces of machinery are coupled together with one or more of the shafts rotating around a catenary shaped axis of rotation, it is important to align the shafts so that they maintain the curved center-line of rotation.



Drive shaft operation below critical speed:
Align machine couplings to spacer couplings

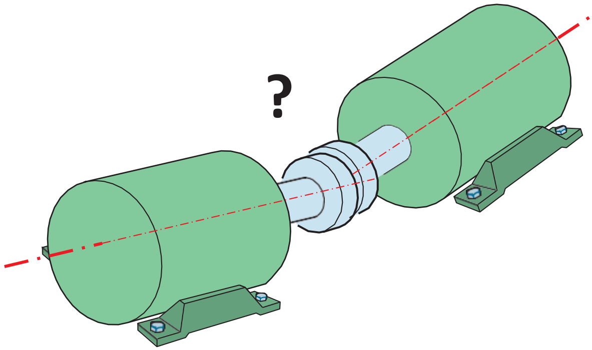


Drive shaft operation above critical speed:
Align machine couplings to one another ignoring spacer.

Expressing alignment

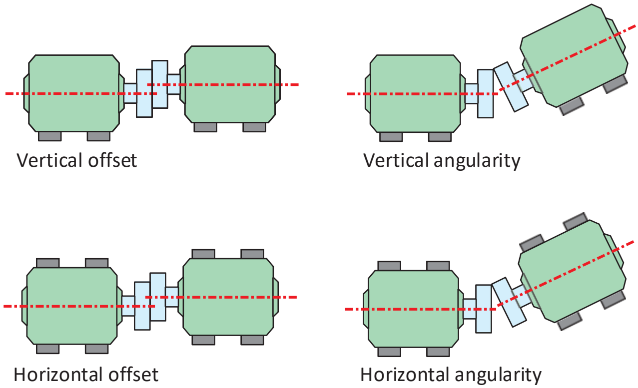
Alignment parameters

Since shaft alignment needs to be measured and subsequently corrected, a method of quantifying and describing alignment condition is necessary.



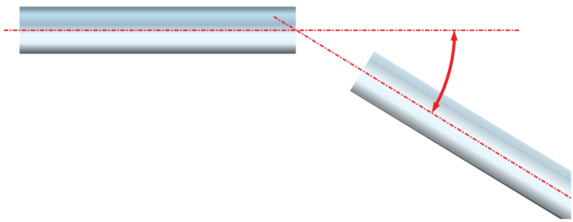
Traditionally alignment has been described in terms of dial indicator readings at the coupling face or position values at the machine feet. The measured values from both of these methods are dependent upon the dimensions of the machines. Since there are many different methods for mounting dial indicators (e.g. reverse indicator, rim and face, double rim) the comparison of measurements and the application of tolerances can be problematic. Additionally, rim indicator readings show twice the true offset, and sign reversals must be observed depending on whether the indicator measures an internal or external, left or right coupling face or rim.

A more modern and easily understandable approach is to describe machine alignment condition in terms of **angularity** and **offset** in the horizontal (plan view) and vertical (side view). Using this method, four values can then be used to express alignment condition as shown in the following diagram.



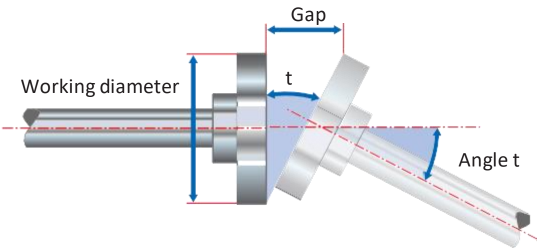
Angularity, gap and offset

Angularity describes the angle between two rotating axes.



Angularity can be expressed directly as an angle in degrees or mrads, or in terms of a slope in mm/m or thous/inch. This latter method is useful since the angularity multiplied by the coupling diameter gives an equivalent gap difference at the coupling rim.

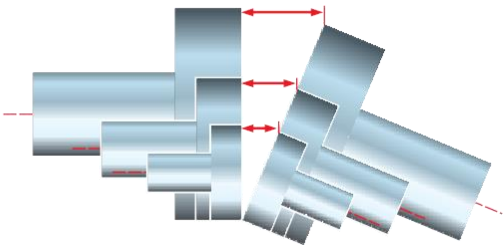
Thus the angle is more popularly expressed in terms of GAP per diameter. The gap itself is not meaningful, it must be divided by the diameter to have meaning. The diameter is correctly referred to as a "working diameter", but is often called a coupling diameter. The working diameter can be any convenient value. It is the relationship between gap and diameter that is important.



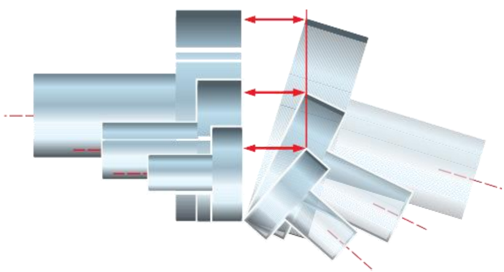
Relationship of angle, gap and working diameter.

A 6 inch (152.4 mm) coupling open at the top by 0.005 inches (0.127 mm) gives an angle between shafts axes of 0.83 mrads.
For a 10 inch working diameter this corresponds to a gap of 0.0083 inches. For a 100 mm working diameter this corresponds to a gap of 0.083 mm.

Note: 1 mrad = 1 thousandths of an inch per inch 1
mrad = 1mm per meter

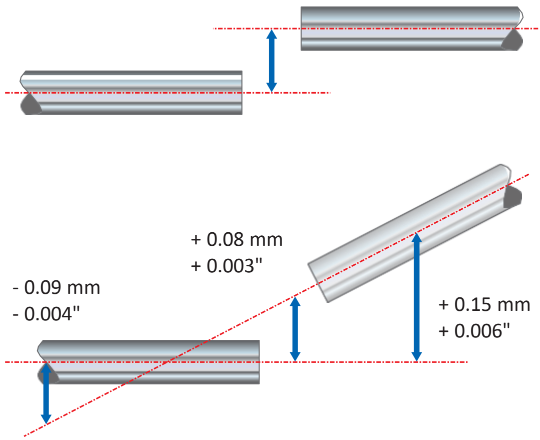


same angle – different gap



same gap – different angle

Offset describes the distance between rotation axes at a given point. Offset is sometimes incorrectly referred to as parallel offset or rim misalign- ment. Shaft rotation axes are however rarely parallel, and the coupling or shaft rim has an unknown relationship to the shaft rotation axes.



As shown above, for the same alignment condition, the offset value varies depending upon the location where the distance between two shaft ro- tation axes is measured. In the absence of any other instruction, offset is measured in mm or thousandths of an inch at the coupling center. This defi- nition refers to short flexible couplings. For spacer couplings, offset should be measured at the power transmission planes of the coupling.

Short flexible couplings

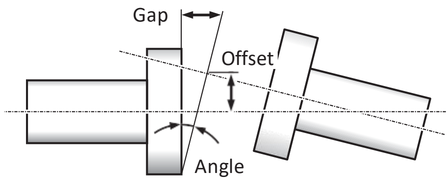
For ease of understanding, we define short flexible couplings when the ax- ial length of the flexible element or the axial length between the flexible el- ement is equal or smaller than the coupling diameter. Machines with short flexible couplings running at medium to high speed require very accurate alignment to avoid undue loading of the shafts, bearings, and seals.

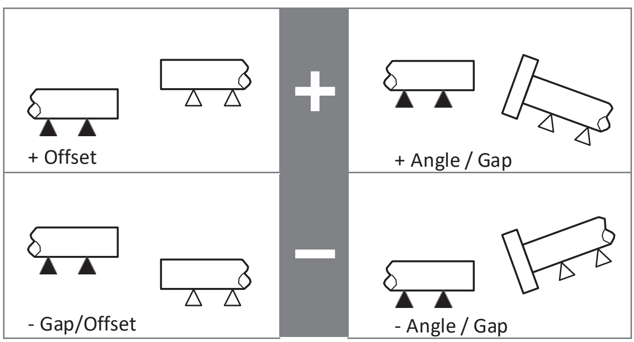
Since the alignment condition is virtually always a combination of angu- larity and offset, and the machine has to be corrected in both vertical and horizontal planes, four values are required to fully describe the alignment condition.

- Vertical angularity (or gap per diameter)
- Vertical offset
- Horizontal angularity (or gap per diameter)
- Horizontal offset.

Unless otherwise specified, the offset refers to the distance between shaft rotation axes at the coupling center.

The sketch below shows the notation and sign convention.





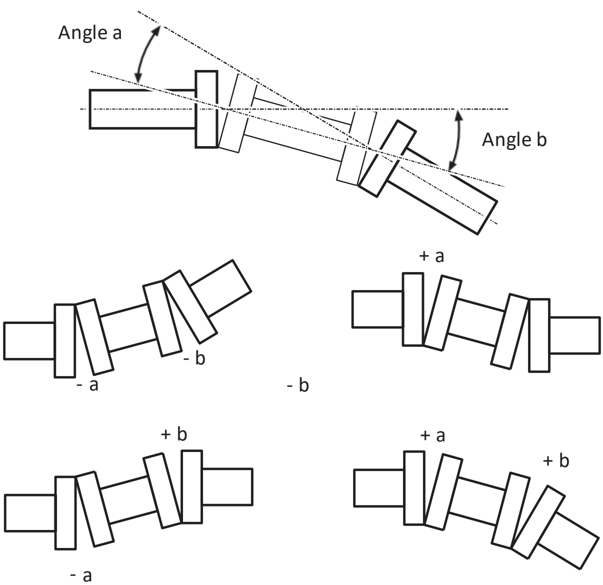
Spacer shafts

Spacer shafts are usually installed when significant alignment changes are anticipated during operation of the machine, for example due to thermal growth. Through the length of the spacer shaft, the angular change at the spacer shaft end remains small even when larger machine positional chang- es occur. The alignment precision for machines fitted with spacer shafts that have flexible elements at each end is not as critical as for machines that have short flexible couplings installed. Four values are required to fully describe the alignment condition.

- Vertical angle a
- Vertical angle b
- Horizontal angle a
- Horizontal angle b

Angles are measured between the spacer shaft rotation axis and the respec- tive machine rotation axes.

The sketch below shows notation and sign convention



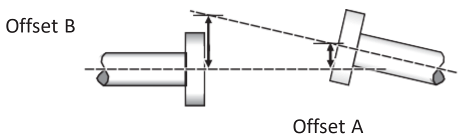
Offset B – offset A

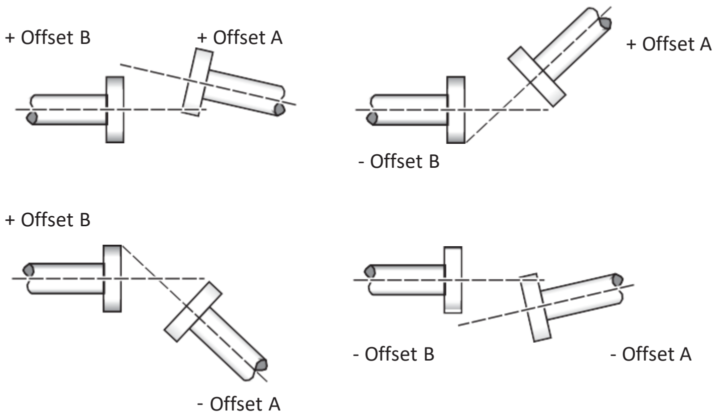
As an alternative to the 2 angles a and b the alignment can be specified in terms of offsets.

- Vertical offset B
- Vertical offset A
- Horizontal offset B
- Horizontal offset A

The offsets are measured between the machine shaft rotation axes at the location of the spacer shafts ends. This is similar to reverse indicator alignment.

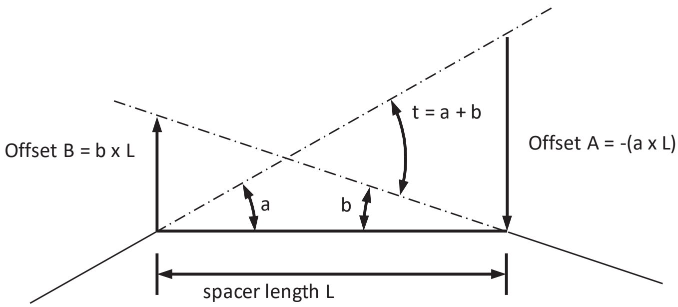
The sketch shows the notation and sign convention.





Relationships

By studying the diagram below a clearer understanding of the relationship between the various offsets and angles will be obtained.



How precise should alignment be?

Alignment tolerances for flexible couplings

The suggested tolerances shown on the following pages are general values based upon over 20 years of shaft alignment experience at PRUFTECHNIK and should not be exceeded. They should be used only if no other tol- erances are prescribed by existing in-house standards or by the machine manufacturer.

Consider all values to be the maximum allowable deviation from the align- ment target, be it zero or some desired value to compensate for thermal growth. In most cases a quick glance at the table will tell whether coupling misalignment is allowable or not.

As an example, a machine with a short flexible coupling running at 1500 rpm has coupling offsets of -0.04 mm vertically and +0.02 mm horizontally. Both these values fall within the “excellent” limit of 0.06 mm.

Angularity is usually measured in terms of gap difference. For a given amount of angularity, the larger the diameter the wider the gap at the cou- pling rim. The following table lists values for coupling diameters of 100 mm or 10 inches. For other coupling diameters multiply the value from the table by the appropriate factor. For example, a machine running at 1500 rpm has a coupling diameter of 75 mm. At this diameter the maximum allowable gap would be; 0.07 mm x 75/100 = 0.0525 mm.

For spacer shafts the table gives the maximum allowable offset for either 100 mm or 1 inch of spacer shaft length. For example, a machine running at 6000 rpm with 300 mm of spacer shaft length would allow a maximum offset of; 0.03 mm x 300/100 = 0.09 mm at either coupling at the ends of the spacer shaft.

Rigid couplings have no tolerance for misalignment, they should be aligned as accurately as possible.

Suggested alignment tolerance table					
	rpm	Tolerance – [mm]		Tolerance – [mils]	
Softfoot	any	0.06		2.0	
		Acceptable **OK**	Excellent	Acceptable **OK**	Excellent
Short flexible couplings					
Offset	600			9.0	5.0
	750	0.19	0.09		
	900			6.0	3.0
	1200			4.0	2.5
	1500	0.09	0.06		
	1800			3.0	2.0
	3000	0.06	0.03		
	3600			1.5	1.0
	6000	0.03	0.02		
	7200			1.0	0.5
Angularity (gap difference at coupling edge per 100 mm diameter or per 10“ diameter)	600			15.0	10.0
	750	0.13	0.09		
	900			10.0	7.0
	1200			8.0	5.0
	1500	0.07	0.05		
	1800			5.0	3.0
	3000	0.04	0.03		
	3600			3.0	2.0
	6000	0.03	0.02		
	7200			2.0	1.0

Suggested alignment tolerance table					
	rpm	Tolerance – [mm]		Tolerance – [mils]	
Spacer shafts and membrane couplings		Acceptable OK	Excellent	Acceptable OK	Excellent
Offset (per 100 mm or per 1“ spacer length)	600			3.0	1.8
	750	0.25	0.15		
	900			2.0	1.2
	1200			1.5	0.9
	1500	0.12	0.07		
	1800			1.0	0.6
	3000	0.07	0.04		
	3600			0.5	0.3
	6000	0.03	0.02		
	7200			0.3	0.2
		Tolerance – [mrad]		Tolerance – [mrad]	
Angularity (mrad)	600			3.0	1.8
	750	2.5	1.5		
	900			2.0	1.2
	1200			1.5	0.9
	1500	1.2	0.7		
	1800			1.0	0.6
	3000	0.7	0.4		
	3600			0.5	0.3
	6000	0.3	0.2		
	7200			0.3	0.2

Note

For industrial equipment the amount of misalignment that can be tolerated is a function of many variables including rpm, power rating, coupling type, spacer length, design of coupled equipment and expectations of the user with respect to service life. Since it is not practical to consider all these variables in a reasonably useful alignment specification, some simplification of tolerances is necessary.

Tolerances based on rpm and coupling spacer length were first published in the 1970s. Many of the tolerances were based primarily on experience with lubricated gear type couplings. Experience has shown however that these tolerances are equally applicable to the vast majority of non-lubricated coupling systems that employ flexible elements in their design.

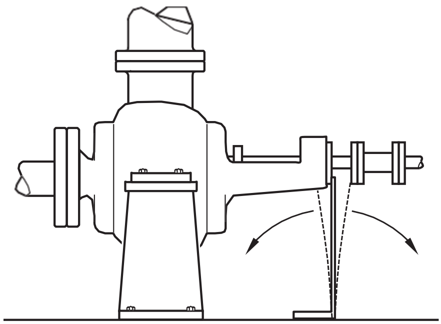
In the previous table “acceptable” limits are calculated from the sliding velocity of lubricated steel on steel, using a value of 12 mm/sec for allowable sliding velocity. Since these values also coincide with those derived from elastomer shear rates they can be applied to short flexible couplings with flexible elements.

“Excellent” values are based on observation made on a wide variety of machines to determine critical misalignment for vibration. Compliance with these tolerances does not however guarantee vibration free operation.

Troubleshooting

Coupling strain and shaft deflection

New readings do not agree with moves just made?
When performing an alignment whether using dial indicators or laser optical systems, sometimes the readings following an alignment adjustment does not agree with the corrections made. One possibility is that coupling strain is bending the shaft, the machine mounts or the foundation. This has frequently been noticed particularly on pump sets which have a front "steady" mount as shown in the following sketch.

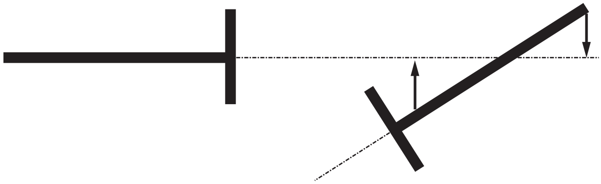


In this application the flexible coupling element is radially quite rigid and can influence the alignment measurement. In this situation we advise splitting the coupling element to free the measured alignment from such external forces.

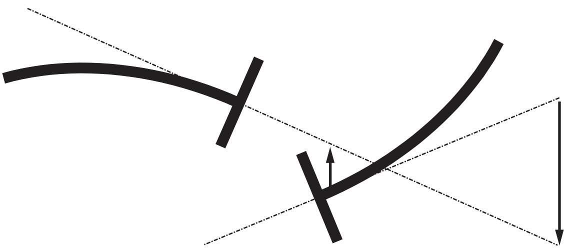
If not accommodated the net effect of influences such as noted above is that the new alignment is not only wrong but quite often has been made in the opposite direction to the required alignment correction.

In extreme cases coupling strain imposed by the newly aligned machines can bend shafts during operation. In most cases this bending will be mini- mal but sufficient to effect the measured axes of shaft rotation.

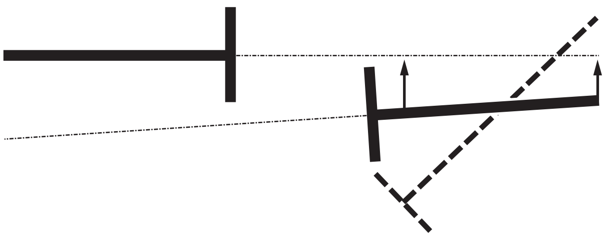
The following sketches illustrate the potential problem



This is the alignment condition with shafts uncoupled



This is the measured alignment with the shafts coupled.
Projected centrelines of rotation are shown



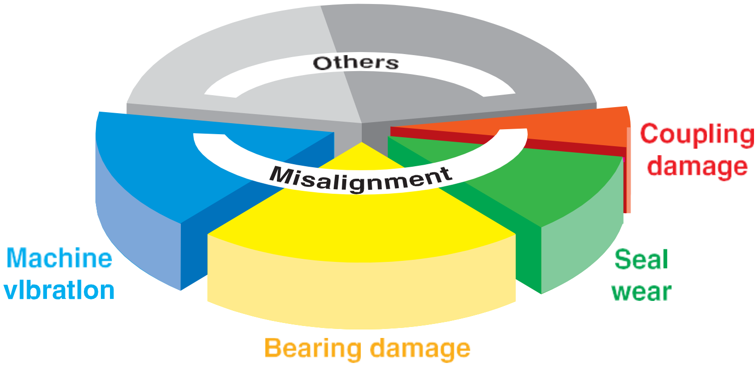
The moves are made as measured.
There is less strain on the coupling now and the shafts will be properly aligned at the next attempt

Causes of machine breakdown

Couplings can take misalignment?

An often quoted comment is "...why bother to align the machine when it is fitted with a flexible coupling designed to take misalignment?"

Experience and coupling manufacturers maximum misalignment recom-mendations would suggest otherwise. Anecdotal evidence suggests that as much as 50% of machine breakdowns can be directly attributed to incor- rect shaft alignment.



It is true that flexible couplings are designed to take misalignment, typically up to 10 mm or more radial offset of the shafts. But the load imposed on shafts, and thus the bearings and seals increase dramatically due to the reaction forces created within the coupling when misaligned. For example a 445 mm vulcan coupling designed for a maximum of 6 mm radial offset at 600 rpm produces a reaction force of 1.2 kN per mm of radial offset.

Anti-friction bearings

Bearings are precision manufactured components designed to operate with clean lubrication and constant but restricted operating temperatures. Com-ponents manufactured within 0.005 mm accuracy are:

- Not able to withstand operating for long periods at elevated tempera-tures caused by misalignment.
- Not able to withstand contamination caused by mechanical seal failure which has allowed ingress of dirt, grit, metallic elements or other ob- jects.

- Not manufactured to operate for long periods with misalignment imposing axial shock loads on the carefully machined and honed components.

In addition to the damage imposed on the bearings through the misalignment itself, when mechanical seals fail, bearings have to be removed from the shaft assembly, sometimes re-fitted or in most cases replaced. Removal and re-fitting in itself can cause bearing damage. Most pump manufactur- ers and repairers recommend that when repairing damaged pumps, bear- ings should always be replaced irrespective of apparent condition, since it is easy to miss minor damage to the bearing that will progressively worsen after re-fitting.

Mechanical seals
Seal wear increases due to shaft loading when shafts are misaligned. Pump seals are a high cost item often costing up to a third of the total pump cost. Poor installation and excessive shaft misalignment will substantially reduce seal life. Manufacturers have addressed the problem of poor installation practice by the introduction of cartridge type seals which can be installed with little or no site assembly. Seals however have precision ground and honed components with finished accuracy of 2 microns (0.002 mm). They do not tolerate operation in a poorly aligned condition, face rubbing, ele- vated temperatures, and ingress of contaminants quickly damage expensive components. Seal failure is often catastrophic, giving little or no pre warn- ing. The resultant plant downtime, seal replacement costs, pump repair costs and bearing replacements makes seal failure due to misalignment an expensive and unnecessary problem.

Machine vibration
Machine vibration increases with misalignment. High vibration leads to fatigue of machine components and consequently to premature machine failure.

The accumulated benefits of precision shaft alignment

The benefits that accrue from adopting good shaft alignment practice be- gin with improved machine operating life thus ensuring plant availability when production requires it. Accurately aligned machinery will achieve the following results.

- Improve plant operating life and reliability
- Reduce costs of consumed spare parts such as seals and bearings
- Reduce maintenance labor costs
- Improve production plant availability
- Reduce production loss caused by plant failure
- Reduce the need for standby plant
- Improve plant operating safety
- Reduce costs of power consumption on the plant
- “Push” plant operation limits in times of production need
- Obtain better plant insurance rates through better operating practice and results

Symptoms of misalignment

It is not always easy to detect misalignment on machinery that is running. The radial forces that are transmitted from shaft to shaft are difficult to measure externally. Using vibration analysis or infrared thermography it is possible to identify primary symptoms of misalignment such as high vibra- tion readings in radial and axial directions or abnormal temperature gradi- ents in machine casings. Without such instrumentation it is also possible to identify secondary machine problems which can indicate inaccurate shaft alignment.

- Loose or broken foundation bolts
- Loose shim packs or dowel pins
- Excessive oil leakage at bearing seals
- Loose or broken coupling bolts
- Some flexible coupling designs run hot when misaligned. If the coupling has elastomeric elements look for rubber powder inside the coupling shroud
- Similar pieces of equipment are vibrating less or have longer operating life

- Unusually high rate of coupling failures or wear.
- Excessive amount of grease or oil inside coupling guards.
- Shafts are breaking or cracking at or close to the inboard bearings or coupling hubs.

Good shaft alignment practice should be a key strategy in the maintenance of rotating machines. A machine properly aligned will be a reliable asset to the plant. It will be there when it is needed and will require less scheduled (and unscheduled) maintenance. In a later section we will review some spe- cific case studies that will show how shaft alignment will deliver substantial cost benefits to operating plants. The next section of this handbook how- ever will review the various methods of shaft alignment that can be used to deliver good installed machinery alignment.

Alignment methods and practice

There are a number of different methods whereby acceptable rotating machine alignment can be achieved. These range from an inexpensive straight edge to the more sophisticated and inevitably more expensive laser systems. We can condense these methods into three basic categories,

- Eyesight – straightedge and feeler gauges
- Dial indicators – mechanical displacement gauges
- Laser optic alignment systems.

Within each category there are a number of variations and options. It is not the intention here to evaluate all of these options. We will concentrate on the most widely used methods in each category.

Preparation is important

The first preparatory step toward successful alignment is to ensure that the machine to be aligned may be moved as required; this includes vertical mobility upwards (using proper lifting equipment, of course) and downwards, should the machine require lowering, as is frequently the case. This can be achieved by inserting 2 to 4 mm (0.08" – 0.16") of shims beneath the feet of both machines on initial installation. We recommend shimming both machines initially so that changes in the foundation condition may later be compensated, if need be.

Horizontal positioning of machines is best performed using jack bolts or a simple 'machine puller' tool or hydraulic equipment, all of which allow fine control of movement in a slow, gentle and continuous manner. Methods such as hammers make exact positioning more difficult and can damage machines (e.g. by causing chatter marks on bearings).The resulting vibra- tion could displace the alignment system during the Live Move function and therefore lead to less accurate monitoring of correction positioning.

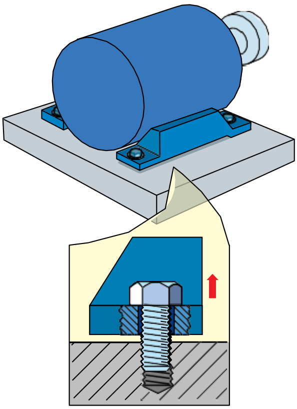
Machine installation guidelines

The installation of machinery such as a pump, gearbox, compressor or oth- er plant machinery require some general rules to be followed.

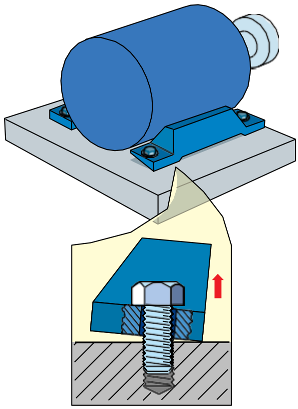
- The driven unit is normally installed first, and the prime mover or motor is then aligned to the shaft of the driven unit.
- If the driven unit is driven through a gearbox, then the gearbox should be aligned to the driven unit and the driver aligned to the gear box.
- Basic checks should be carried out to determine the accuracy of the ma- chine couplings, i.e. check for "run-out" (concentricity and squareness to the shaft centerlines) of coupling halves using a dial indicator, if pos- sible. Out of "true" coupling halves can cause out of balance problems.
- Preparation of the machinery baseplate and machine mounting surfac- es, feet, pedestals and similar foundation is of paramount importance. Otherwise, successful alignment may not be easily achieved.
- Clean, dress up and file any burrs from mounting faces and securing bolt holes.
- Have quality shims available to align precisely and effectively.
- Before assembling the shaft alignment system/ instrumentation to the machines, take a few minutes to look at the coupling/shaft alignment. Remember, your eyes are your first measuring system.
- Check that the pump/motor set is sitting square to the baseplate (soft foot check), and correct as required – see following pages.
- Keep shims to a minimum – if possible use no more than a maximum of 3 shims under machinery feet/mounts.
- Correct alignment as required to ensure that, when the machinery is running, the machinery shafts are centered in their bearings and are aligned to manufacturers tolerances.
- Always check manufacturers alignment figures prior to commencing work. Temperature growth may require specific "cold" alignment off- sets.
- Ensure that any pipework attached to machines is correctly supported but free to move with thermal expansion.

Measurement and correction of soft foot.

An essential component of any successful alignment procedure is the determination and correction of soft foot. Just as a wobbly chair or table is an annoyance, a wobbly machine mount causes alignment frustration. The machine stands differently each time an alignment is attempted, and each set of checking measurements indicate that the machine is still mis- aligned. Additionally when the machine is bolted down strain is placed upon the machine casing and bearing housings. Essentially there are two types of soft foot as illustrated in the sketch below.



Parallel soft foot



Angular soft foot

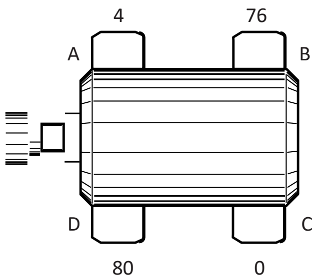
Parallel soft foot indicates that the baseplate and machine foot are parallel to each other allowing correction by simply adding shims of the correct thickness. Angular soft foot is caused by the machine feet forming an angle with one another. This situation is more complex to diagnose and to correct. One solution is to use tapered shims to fill the angular space between the baseplate and the foot. A more drastic but long term solution is to remove the machine and grind the machine feet flat.

Soft foot measurement

One of the following techniques may be used to determine soft foot prior to commencing alignment:

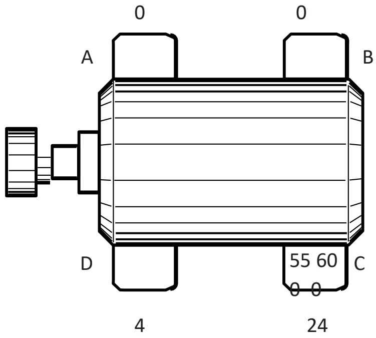
- Using a dial indicator mounted on a magnetic foot; position the indica- tor above one of the machine feet, zero the indicator and then loosen off the machine foot. Record any change in the indicator reading. Tight- en the machine foot down. Repeat this for all machine feet.
- Using a set of feeler gauges; loosen one machine foot at a time, mea- sure the gap that appears below the loosened machine foot and record this. Tighten the machine foot and move to the next foot.
- Using a laser alignment system; loosen one machine foot at a time. The alignment system records the amount of foot lift at each foot. Tighten the machine foot before proceeding to the next foot.

Having determined the amount of soft foot present as indicated below, it is possible to make adjustments to the machine according to the soft foot condition diagnosed.

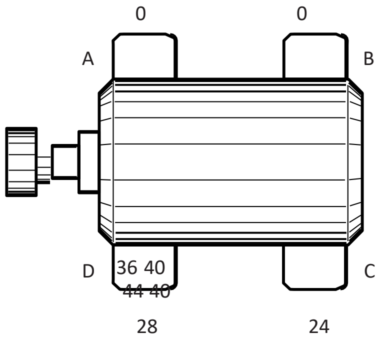


This example shows classic soft foot problems with a rock across feet B and D. It is tempting to shim both feet to eliminate the rock but this would be a mistake. The best solution would be to shim only one foot 80/100 mm and to recheck all four feet.

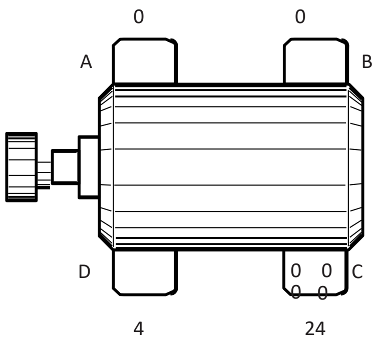
Many additional soft foot problems may be found including bent feet strain imposed by pipe work or “squishy” foot cause by too many shims under the machine feet. Some examples are shown in the following sketches.



Soft foot example: bent foot - step shim at foot C and recheck all feet



Soft foot example: pipe strain – relieve external forces



Soft foot example: squishy foot – re shim all feet with max 3 shims and recheck

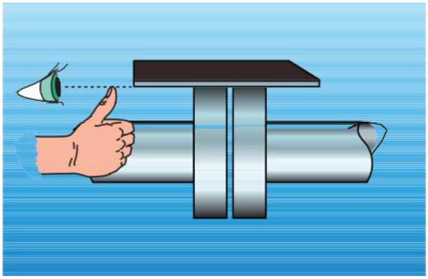
When eliminating soft foot follow these steps.

- Check all four machine feet. Any foot showing over 0.08 mm is to be corrected as appropriate.
- Examine the largest soft foot (or two largest if having the same value) with feeler gauges to determine the type of soft foot. It never hurts to examine the other feet as well, but concentrate on finding and fixing the largest problem first.
- Correct the condition diagnosed by shimming only one foot if any.
- If all feet are within tolerance commence the alignment.

Alignment methods – eyesight

The straightedge

This method of shaft alignment was common practice in many plants, provided a flexible coupling was used. It was considered good enough to eyeball the alignment and bolt the machine down. The system is certainly cheap and equipment is readily available.

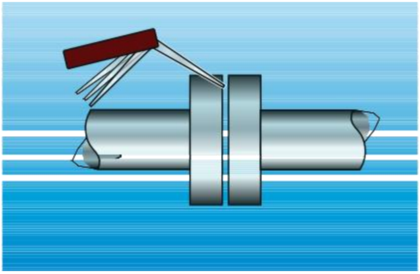


The corrective values for the machine feet were usually estimated accord- ing to the experience of the engineer carrying out the alignment. Most often corrections at machine feet need to be repeated on a trial and error basis before the "eyeball" alignment condition was completed. Even then there is no certainty that the completed alignment was correct. Since the resolution of the human eye is limited to 0.1mm, alignment accuracy is correspondingly limited. Additionally without having carried out extensive checks on the fitting accuracy of the coupling on the shaft, no direct cor- relation between the completed alignment and the actual alignment of the machine shafts can be made.

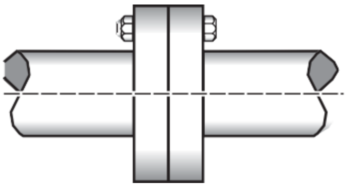
At best this alignment method can be described as coupling alignment not shaft alignment as defined earlier.

The feeler gauge

Although classified here as an "eyesight" method of shaft alignment the feeler gauge method under certain circumstances and for some machines can be perfectly acceptable. In the installation and alignment of turbine sets where the coupling half is an integral part of the rotor shaft and has no flexible elements, it is possible for a skilled turbine engineer to align the two coupling halves very accurately. (As noted in the section on alignment tolerances, no allowance for offset or gap is permissible on these “solid” type of couplings).



Using the feeler gauge or a vernier caliper the engineer accurately measures any gap between the coupling halves. Jacking oil is then used to rotate the shafts together through 180 degrees and the “gap” is then checked again (with the jacking oil off). This procedure is then carried out for the horizon- tal alignment measurements.

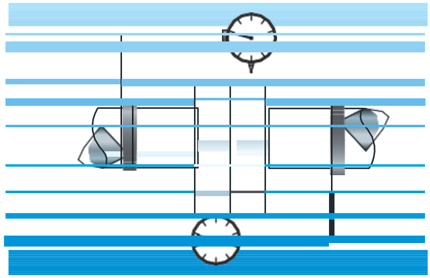


Readings are usually graphically plotted to establish alignment condition and any necessary corrections that are required. In some cases engineers will rotate one shaft through 180 degrees and take additional readings. These readings are then averaged to eliminate any possible shaft machining errors. The averaged readings form the basis for the alignment graph.

On machines that employ flexible elements in the coupling design, the use of feeler gauges is beset with the same limitations as the straightedge method and can only be described as coupling alignment.

Alignment methods – dial indicators

The use of dial indicators for the vast majority of shaft alignment tasks where a flexible coupling element is used represents a substantial step for- ward in accurate shaft alignment methods.



There are a number of dial setups that can be used to effect the alignment of machines, this section will review some of these. There are a number of factors that the engineer should take into account before embarking on a dial indicator alignment task.

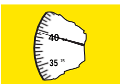
Indicator bracket sag

This should always be measured before actual alignment read- ings are taken – no matter how solid the bracket appears (see section on measuring sag).

Internal friction / hysteresis

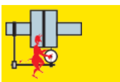
Sometimes the gauge has to be tapped in order for the indica- tor needle to settle on its final value.

1/100 mm resolution



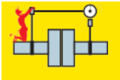
Up to 0.005 mm rounding error may occur with each reading. This may be compounded several times in a full set of readings.

Reading errors



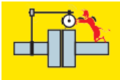
Simple errors occur when dials are read under difficult condi- tions and severe time constraints.

Play in mechanical linkage



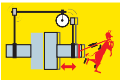
Slight amounts of play may go unnoticed but will produce large reading errors.

Tilted dial indicators



The gauge may not be mounted perpendicular to the measurement surface so that part of the displacement reading is lost.

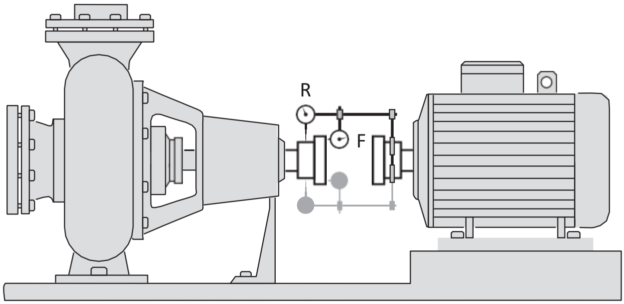
Axial shaft play



This will affect face readings taken to measure angularity unless two axial gauges are used.

Rim and face method – by trial and error

The interpretation of shaft alignment readings using dial indicators, taking factors such as bracket sag into consideration requires an elementary understanding of maths and geometry. In some cases these skills are limited and a rough trial and error procedure is used where bracket sag and shaft float are ignored. Additionally only one shaft is rotated during the measure- ment adding errors to the alignment caused by coupling run-out and shaft bending.



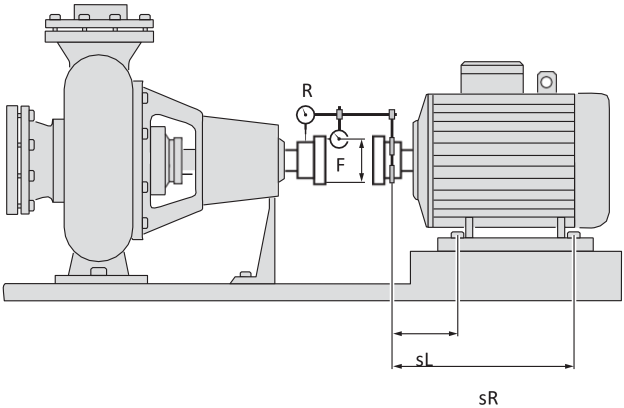
The above sketch illustrates the scenario. Rim and face indicators clock the fixed machine coupling. Indicators are zeroed at 12 o'clock and the shaft of machine to be moved is rotated through a half turn to the 6 o'clock position. The foot nearest the coupling is raised (or lowered) by an amount equal to half the rim indicator reading. Shims are repeatedly placed under the foot furthest from the coupling until the face indicator readings do not change as the shaft is rotated.

Similarly the indicators are zeroed at the 3 o'clock position and rotated to the 9 o'clock position for the horizontal correction.

It is usually easy to spot when this procedure is used as there are often a number of thin shims under the rear feet of the machine. Usually this trial and error procedure results in significant misalignment errors at the cou- pling transmission planes. Where possible, this method should be discour- aged in favor of other dial or laser methods of alignment.

Rim and face method – by calculation

The measuring device for this type of alignment is a dial indicator. The dial hand indicates or points to increments marked on the dial face. As the foot is pushed into the body, the dial hand rotates clockwise. The number of indicator marks that the hand moves is equal to the distance that the foot was pushed into the body. When the foot travels out from the body the dial hand similarly indicates the travel distance. The dial count is positive when the foot travels in and negative traveling out.



Rim and face alignment takes its name from the positions of the indicator feet during measurements. A traditional indicator setup is shown above. Once mounted, the two shafts are rotated together and the dials are read at 12:00, 3:00, 6:00 and 9:00 o'clock positions.

Formulas for calculating alignment corrections

For such setups, the MTBM alignment at the plane of the indicator foot is as follows:

- VO = (R6 - R0 - RS) / 2
- VA = (F6 - F0 - FS) / dia
- HO = (R9 - R3) / 2
- HA = (F9 - F3) / dia

Where:
VO / VA = Vertical Offset / Angularity HO
/ HA = Horizontal Offset / Angularity

R0 = Rim reading at 12:00 o'clock position R3
= Rim reading at 3:00 o'clock position R6 =
Rim reading at 6:00 o'clock position R9 = Rim
reading at 9:00 o'clock position F0 = Face
reading at 12:00 o'clock position F3 = Face
reading at 3:00 o'clock position F6 = Face
reading at 6:00 o'clock position F9 = Face
reading at 9:00 o'clock position
dia = Diameter of the circle travelled by face indicator foot RS
= Sag of rim indicator
FS = Sag of face indicator*
s = Span from measurement plane (rim indicator foot) to machine foot (front or back) – this value can be positive or negative

Clockwise is determined looking along shaft from MTBM towards STAT.

- Shim = (VA) (s) - VO
- Shim = (F6 - F0 + FS)(s) / dia - (R0 - R6 + RS) / 2
- Move = (HA)(s) - HO
- Move = (F9 - F3)(s) / dia - (R3 - R9) / 2

If the dial indicators are set to zero at 12:00 and then read at 6:00, the shim calculation becomes:
- Shim = (F6 - FS)(s) / dia - (R6 - RS) / 2

Positive result means add shims. Negative result means remove shims

If the dial indicators are set to zero at 3:00 and then read at 9:00 the Move calculation becomes:
- Move = (F9)(s) / dia - R9 / 2

Positive means move toward 3:00. Negative means move toward 9:00

The Shim and Move calculations must each be done twice; once for the front feet, and once for the back feet.

Indicator reading validity rule.

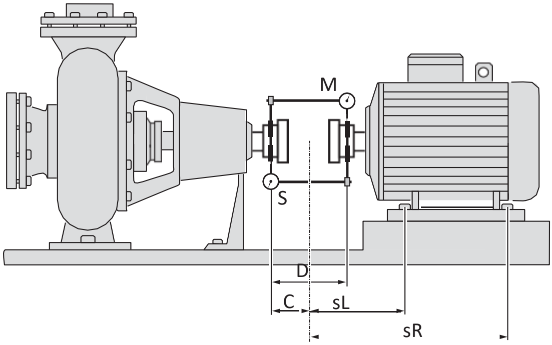
The sum of the 3 and 9 o'clock readings should equal the sum of the 12 and 6 o'clock readings. This applies to both radial and face readings.

Sag

A major source of error in the above procedure is the sag of the spanner bar. This error can affect the shim amounts to such an extent that the ma- chine will be grossly misaligned. To compensate for this sag, measure it and then add the sag reading (it can be positive or negative) to the 6:00 o'clock readings (see the previous formulas).

Reverse indicator method – by calculation

The reverse indicator method of alignment is the most advanced dial indicator alignment method. It is recommended by the American Petroleum Institute (API 686) as the preferred dial indicator alignment method



Reverse indicator alignment takes its name from the positions of the indicators opposing one another on the opposite coupling halves. A traditional indicator setup is shown above.

Once mounted, the two shafts are rotated together and the dials are read at 12:00, 3:00, 6:00 and 9:00 o'clock positions.

Formulas for calculating reverse indicator alignment

For such setups the misalignment at the coupling center is as follows:

- VO = (S6-S0-SS)/2 - (S6-S0-SS +M6-M0-MS)C/2D
- VA = (S6-S0-SS +M6-M0-MS)/2D
- HO = (S9-S3)/2 - (S9-S3+M9-M3)C/2D
- HA = (S9-S3+M9-M3)/2D

Where:
VO / VA = Vertical Offset / Angularity HO
/ HA = Horizontal Offset / Angularity

S0 = Left rim reading at 12 o'clock S3
= Left rim reading at 3 o'clock S6 =
Left rim reading at 6 o'clock S9 =
Left rim reading at 9 o'clock

M0 = Right rim reading at 12 o'clock M3
= Right rim reading at 3 o'clock M6 =
Right rim reading at 6 o'clock M9 =
Right rim reading at 9 o'clock
D = Distance between left and right indicators
C = Distance between left indicator and coupling centre SS
= sag of left rim indicator (*)
MS = sag of right rim indicator (*)
sL = Distance from the coupling center to front feet of right machine sR = Distance from the coupling center to back feet of right machine. (*) these values can be positive or negative

The corrections at the right machine feet can be calculated as follows

- Shim front feet = (VA * sL) - VO
- Shim back feet = (VA * sR) - VO

Positive result means add shim, negative result means remove shim.

- Move front feet = (HA * sL) - HO
- Moe back feet = (HA * sR) - HO

Positive result means move towards 3 o'clock, negative means move to- ward 9 o'clock.

If the dial indicators are set to zero at 12 o'clock and then read at 6 o'clock the shim calculation are as follows:

- Shim front feet = (S6 - SS + M6 - MS) (C+sL)/2D - (S6 - SS) / 2
- Shim back feet = (S6 - SS + M6 - MS) (C+sR)/2D - (S6 - SS) / 2 Positive result means add shim, negative result means remove shim.

If the dial indicators are set to zero at 3 o'clock and then read at 9 o'clock the move calculations are as follows:

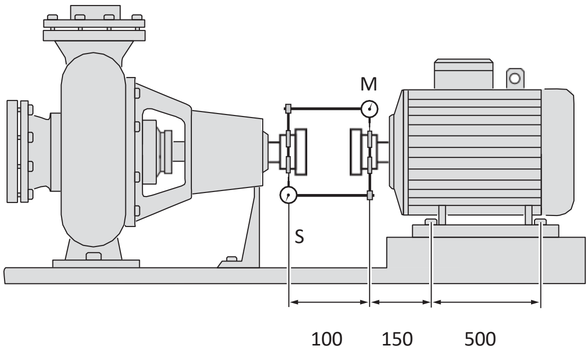
- Move front feet = (S9 + M9) (C+ sL)/2D - S9/2
- Move back feet = (S9+M9) (C + sR)/2D -S9/2

Positive result means move towards 3 o'clock, negative means move to- ward 9 o'clock.

For sag calculations see following section.

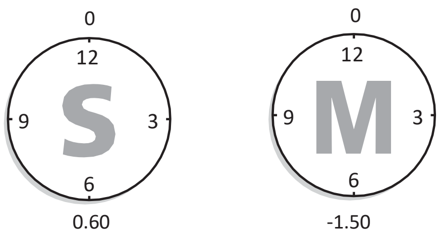
Reverse indicator method – by graph

The calculations shown in the previous section can be daunting for many engineers and unlike with rim and face measurement, trial and error corrections are not possible. To avoid mathematical computation, a graphical solution can be used to resolve the alignment condition and the necessary shim and move corrections.



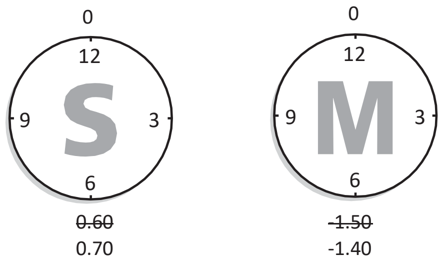
The sketch above shows a typical reverse indicator configuration with the right machine as the machine to be moved (MTBM). Both indicators are zeroed when the indicators are in the 12 o'clock position. The direction of view is from the MTBM to the stationary machine.

The shaft is rotated through 180 degrees in the direction of normal shaft rotation. Dial values are read and noted; as an example assume the follow- ing readings were taken.



All values shown in mm

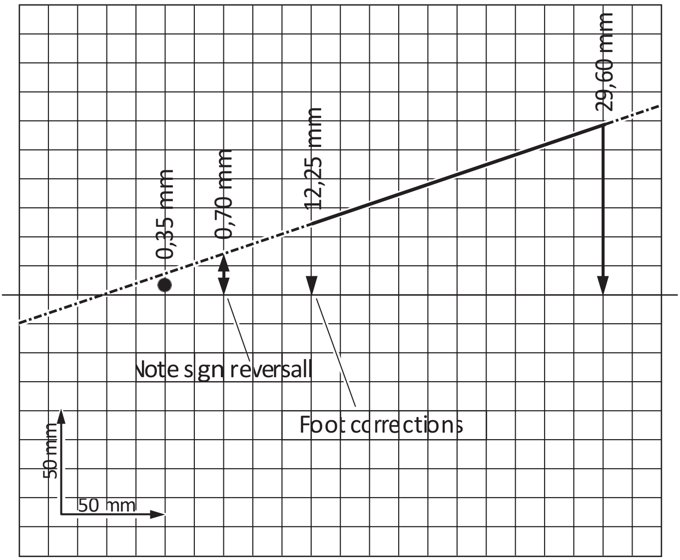
The indicator bracket sag was -0.10 mm. The total indicator readings (T.I.R.) after correction for bracket sag are thus:



T.I.R. values must be divided by 2 to determine the true shaft offset values in the planes of the dial indicators.

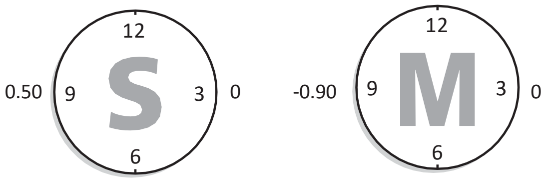
Offset S = +0.70 / 2 = +0.35 mm
Offset M = -1.40 / 2 = -0.70 mm.

These offsets are then plotted on the graph as follows.



Both indicators are then zeroed with the indicators at the 3 o’clock posi- tion. The shaft is rotated through 180 degrees in the direction of normal rotation. The readings are noted. On returning the shafts to the 3 o’clock position the indicator readings should return to zero.

Assume the following readings were taken:

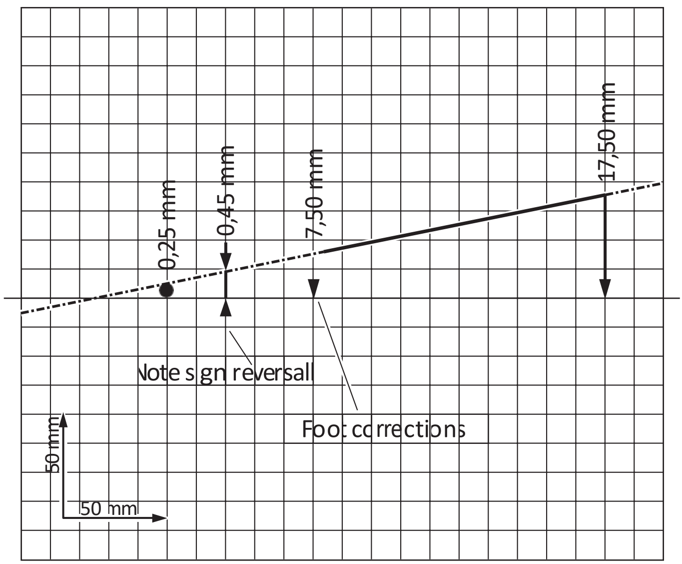


Sag correction values are not applicable to horizontal readings.

T.I.R. values must be divided by 2 to determine the true shaft offset values in the planes of the dial indicators.

Offset S = +0.50 / 2 = +0.25 mm
Offset M = -0.90 / 2 = -0.45 mm

These offsets are then plotted on the graph as follows



Vertical and horizontal shim corrections are shown on each graph. The corrections assume that the alignment should be 0.0/0.0 in vertical and horizontal planes. Any manufacturers figures or computed figures for thermal expansion should be accommodated in these shimming corrections or in the original dial indicator readings.

Indicator bracket sag measurement

To measure sag, mount the entire measurement fixture (brackets, bars and indicators) onto a piece of straight pipe. Adjust the fixture until the brackets are the same distance apart, as they will be, when they are mounted on the actual machinery. Likewise position the indicators as near as possible to the way they will be set on the machinery. With the indicators held at the 12 o'clock position, zero the dials. Rotate the pipe until the indicators are at 6:00 o'clock. Read and record the dial indicators (the rim indicator will be a negative value, the face indicator may be positive or negative but should be close to zero).

Alignment methods – Laser shaft alignment

Shaft alignment by laser became popular in the mid 1980s when PRUFTECHNIK introduced OPTALIGN the world's first commercially available computer-aided laser shaft alignment system. Despite its then relatively high price, the system quickly gained a market popularity with engineers and companies across a wide spectrum of process industries world wide.

OPTALIGN offered many significant advantages in effecting quick and accurate alignment of coupled rotating machines. Since the introduction of the first system, developments in laser and microprocessor technology has allowed a new generation of laser systems to be developed which offer the user simple-to-understand, menu led, systems that can be used for virtually any shaft alignment task irrespective of complexity or size.

As we have seen in the previous sections that there are a number of import- ant considerations to be taken into account when using mechanical meth- ods of shaft alignment. Additionally, calculations of alignment corrections can be complicated and error prone. None of these considerations apply to the laser method of shaft alignment. Access to precision shaft alignment and the benefits that this brings (see following section) are readily available when laser shaft alignment is used on plant.

A summary of the advantages offered by laser systems are shown here:

- Precision alignment with no manual input of data and no graphical or numeric calculations to perform
- Graphic display of alignment results at the power transmission planes of the coupling, and shim and adjustment corrections at the machine feet
- No mechanical fixtures - no bracket sag
- No need to disassemble the coupling to effect an alignment
- No need to take readings at predetermined locations such as 12;00, 3;00, 6;00 and 9;00 o'clock positions; results can be obtained with less than 90 degrees of shaft rotation
- Data storage and print out of results for report generation of alignment condition
- Certified calibrated accuracy of the laser system to comply with ISO 9000 requirements
- Universal bracket systems which cover all types of alignment applica- tions; no need for special "Christmas tree" brackets for long spacer shaft measurement

- Menu driven operator system allows use by a wide range of engineering skills and disciplines
- Live dynamic display of vertical and horizontal corrections during alignment corrections
- Inbuilt go/no go alignment standards for analysis of alignment accuracy

Having identified some of the benefits and advantages that can be obtained by using a laser alignment system to carry out shaft alignment, it is important to establish the functionality of the alignment system that will suit the users requirements. There are a number of systems available and a number of manufacturers who offer laser alignment systems.

As a minimum the system you choose should have the following capabilities:

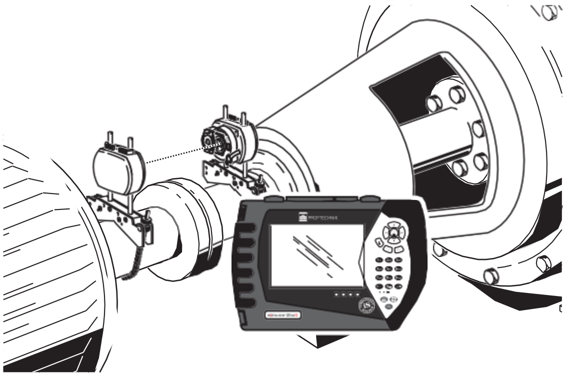
- **Certified calibration to a traceable standard:** There is no point purchasing a system for accurate shaft alignment that cannot have its measurement accuracy certified.
- **High accuracy and repeatability:** Poor accuracy simply results in wrong correction values. High repeatability means that fewer measurements are required to acquire sufficient data to calculate accurate results.
- **Rugged, water, shock and dust proof**: A rugged enclosure means outdoor use in wet conditions is not a problem. Rugged instruments with a guaranteed seal of approval like the IP standards (65 and 67) let you continue working even in adverse conditions.
- **Measurement resume capability:** Resume allows you to easily re-start an alignment in progress after an interruption, or at the start of a new day, the user won't have to input dimensions or targets again. Even measurement results are saved. Data will never be lost.
- **Measurement extend capability:** The ability to extend the dynamic range of the laser detector system ensures that no matter the misalignment being measured, the laser system will cope with the alignment task. Static detector systems will not allow measurement of gross misalignment on long or intermediate spacer shafts whatever the stated size of the detector plane (see later notes).
- **Interchangeable static feet:** The ability to vary static feet allows the engineer maximum flexibility and the ability to deal with bolt bound feet

on the MTBM without the need for re-measuring or complex calcula- tions; all possible alternatives of machine move can be shown.

- **Assortment of brackets:** A wide range of brackets means that mea- suring equipment can be fitted even to the most awkward of machines with speed and ease.
- **Tolerances (TolCheck):** Built in verification of alignment tolerances save time and effort. No time is wasted on unnecessary machine moves. Automatic tolerance check shows when excellent or acceptable align- ment has been reached.
- **Report generation directly from the box:** Direct reporting means faster reporting to any printer with the serial number, date and time, and operator name printed on the report, allowing full compliance with ISO 9000 traceability requirements for example.

Laser systems basic operating principles

Essentially there are two types of laser systems: one that uses a single beam projected onto either a detector or on to a reflector that returns the beam to the laser detector; the other type of system uses two lasers each with inbuilt detectors. The former single laser system is a patented system used exclusively by PRUFTECHNIK, the two laser systems are employed by all other system suppliers.



The single laser system as shown above has a number of advantages that have been incorporated to improve system versatility and usability.

Measurement extend capability – only one laser datum means that it is possible to dynamically extend the detector range of the system to incorporate gross misalignment – see later explanation.

Split alignment capability – one laser allows alignment of machines that have no spacer or coupling in place, each machine can be rotated independently. This is particularly useful when large spacer couplings or fluid couplings are used, when aligning large machines such as turbines, or when one or both machines cannot easily be rotated.

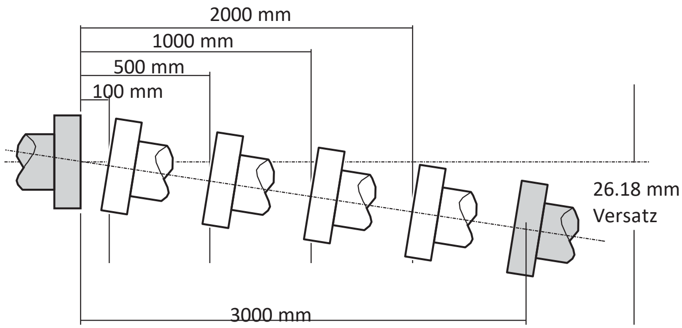
Single cable technology – only one (or no) cable is required. This is particularly useful on long spacer shafts such as cooling tower drives where long cables can influence alignment measurements by becoming entangled during measurement.

Only one laser to adjust – on long spacer shafts or large machines, set up is much easier with only one fixed datum position to adjust.

Measurement extend capability explained

Why would it be useful to be able to extend the range of the detector plane on a shaft alignment system; surely it would be better to have a larger de- tector area? Well, yes, theoretically it would be useful to have a static detec- tor plane of 500 mm. But the system would be unusable simply because of size and weight. An ideal compromise is to dynamically extend the detector plane if it is required. This keeps the system to a minimum size and weight and therefore maximizes the system's use in difficult to access areas.

Taking as an example a cooling tower drive with a spacer shaft coupling of 3000 mm. The offset between the driver and driven shafts can be substan- tial even with only a small angular offset between the shafts.



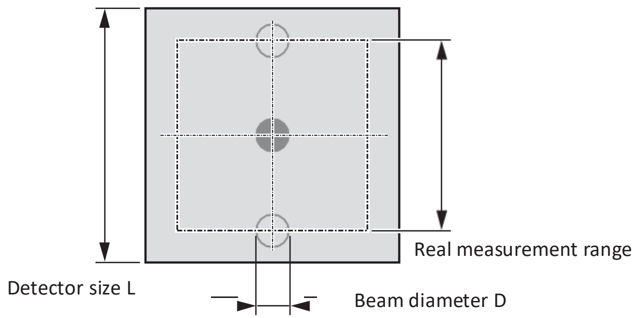
This above sketch illustrates the limitations imposed by long spacer cou- pling lengths.

Taking as a simple example a coupling setup with an angular misalignment between the couplings of 0.5 degrees would mean that over a simple short coupling length of 100 mm an offset of 0.87 mm between coupling centerlines would occur. This offset could be comfortably measured by any laser system.

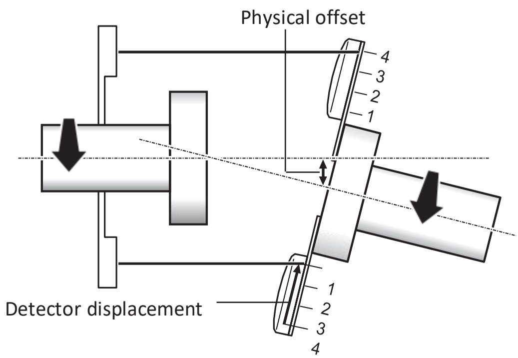
If the distance between coupling faces increases to 500 mm the centerline offset becomes 4.36 mm, a value out side the range of most static laser detector systems. Now increase the distance to 1000 mm. The offset be- comes 8.72 mm. As the coupling spacer gets longer so does the offset until at 3000 mm a massive 26.18 mm offset occurs. This with only a 0.5 degree angle between the shaft centerlines.

This large offset can only be measured by an extendable detector range since a static detector area of approximately 60 mm is required to accom- modate this offset. The reason for such a large detector can be explained as follows:

The working area of the detector is less than the physical detector surface. For example, if the detector area is 20 x 20 mm, and the laser beam is 4.0 mm diameter then the maximum useful measurement range is 16 mm as shown below.

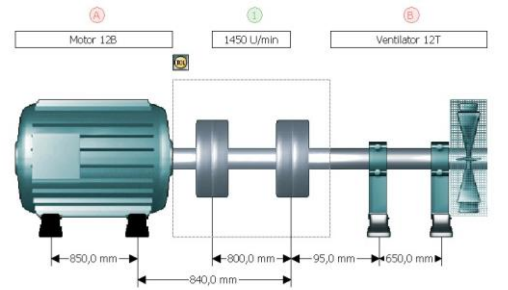


To be able to measure an offset a system detector range has to be twice the offset. As with a dial gauge, the laser sensor measures twice the physical offset of the two shafts as shown below.



To measure a physical offset of 2.0 mm we need a detector measurement range of 4.0 mm.

A measurement extend capability you may say is all very well if you measure cooling tower drives or other long spacers but when the maximum spacer measurement is less than 1 meter, why is this ability to extend the detector range important? An example of the benefit of this ability to extend the detector range is illustrated here with a real application. A motor / fan drive was measured, as shown below:



The coupling spacer length was 800 mm.
Measured offset and gap per 100 mm spacer shaft length:
Vertical = 0.00 offset 0.72 mm gap
Horizontal = 0.00 offset 1.05 mm gap

To facilitate this measurement it is necessary that the detectors are able to measure an offset of 8.40 mm. This requires a detector range of 20.8 mm. This is derived from the following formula:

Measured offset x 2 + beam diameter,
(8.4 mm x 2) + 4 mm = 20.8mm.

Depending on the specific requirements encountered in day-to-day alignment tasks, the ability to extend the system detector range could be the sin- gle most influencing factor in choosing a measurement system. Regardless of laser alignment system chosen, the operating plant will gain substantial benefits as illustrated in the following case studies.

Case studies

Laser shaft alignment cuts energy costs

A project to determine the extent to which shaft misalignment influenced the power consumption of the plant was set up as a graduate student proj- ect at a major UK chemical processing plant. The study was conducted over a six-week period in a controlled environment that accurately reflected the normal operating conditions across the plant.

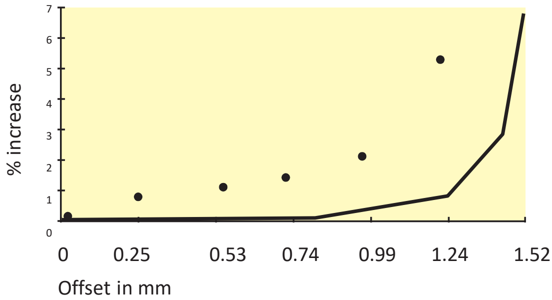
A 7.5 kW pump rig on a redundant plant was used for the investigation. Before the project commenced the pump and motor were removed to the workshop where new bearings were fitted, and both units were rebalanced to eliminate any external factor that could distort the project results. Plates and jacking bolts were attached to the motor base plate to allow fine ad- justments in alignment condition. The pump set was installed to circulate water through a closed loop of piping with the motor running at 3000 rpm (+/- 1% due to variations in load condition). The pump and motor were initially installed with the alignment recorded as 0.00 gap and offset in the vertical and horizontal directions. The system was run in this condition for a number of days with current drawn being measured at the distribution board every few hours. During the course of the trial period, the alignment of the machines was adjusted and at each misalignment interval run for a set period with current drawn measured at regular intervals.

Across the site the two principle types of coupling installed were “pin” and “tyre” couplings. In order to obtain a reasonable picture of potential sav- ings that could be made on the plant both types of coupling were installed with the same amount of misalignment/current measured on each coupling type.

The results of the study are shown in the following graphs. Offset mis- alignment affected power consumption more than angularity; angular misalignment affected power drawn by “pin” type couplings more than “tyre” couplings. The components of misalignment are additive irrespective of whether the misalignment was vertical or horizontal.

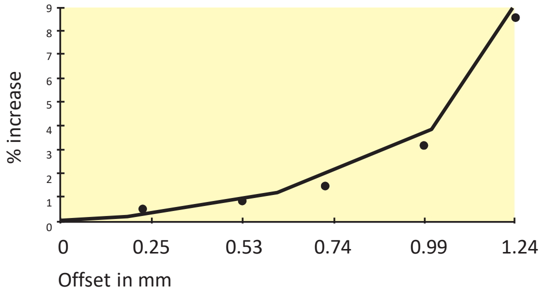
Effects on power consumption

Pin coupling at 3000 rpm



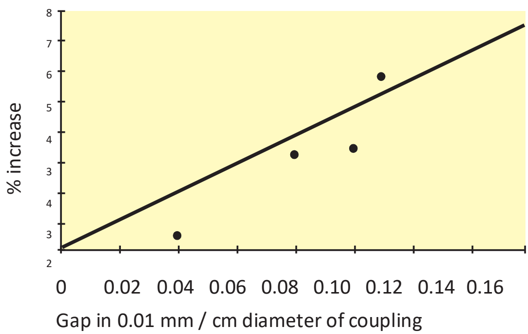
Offset	**%**
0.03 mm	0.0
0.25 mm	0.7
0.53 mm	1.0
0.73 mm	1.3
0.99 mm	2.0
1.24 mm	5.2
1.24 mm	6.6

Tyre coupling at 3000 rpm

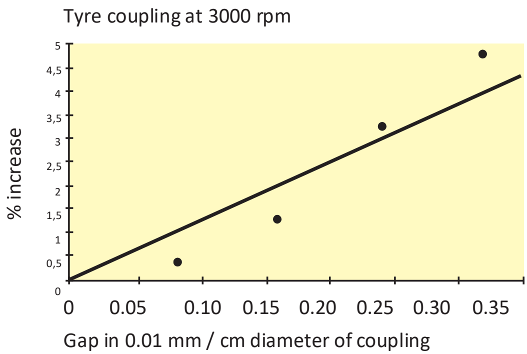


Offset	**%**
0,03 mm	0,0
0.25 mm	0.7
0.53 mm	1.0
0.74 mm	1.3
0.99 mm	2.8
1.24 mm	8.5

Pin coupling at 3000 rpm



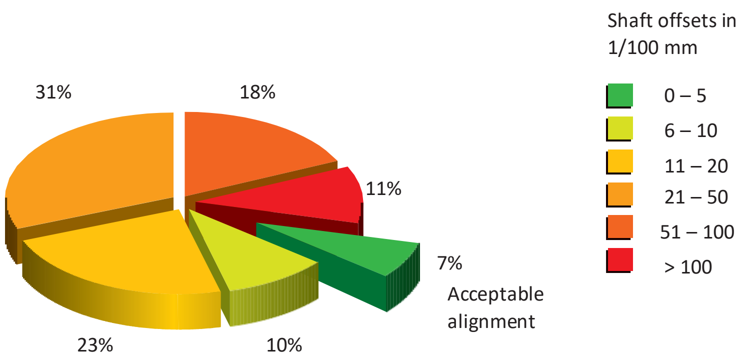
Gap	**%**
0.004 mm	0.3
0.008 mm	3.1
0.010 mm	3.3
0.011 mm	5.7
0.016 mm	7.8



Gap	%
0.008 mm	0.3
0.016 mm	1.2
0.024 mm	3.2
0.032 mm	4.8

The conclusion from the project was a site wide recommendation to align machines to within an offset tolerance of 0.15 mm and an angularity tolerance of 0.05 mm per 100 mm of coupling diameter.

To estimate the potential cost savings that could accrue from this new site standard a random sample of machines were measured to estimate the extent of misalignment that existed on the plant. The pie chart below illustrates the findings of this survey.



Shaft offsets in 1/100 mm at the coupling center for a sample of 100 machines operating at 3000 rpm.

Only 7% of machines measured were within the recommended site align- ment standard.

Using the pie chart a representative median offset of 0.35 mm was estimated as a reasonable figure for calculating the potential power saving on the plant. Given that the power consumption for the rotating equipment on the plant was in the range of 30 Megawatts, the following estimate of power saving that could be achieved was:

Assuming electricity rates of 0.10 Euro per kWh and a conservative % pow- er reduction of 0.75%:

30,000 kW * 0.75% * 0.10 Euro / kWh = 22.5 Euro per hour or
197,100 Euro per year.

Laser shaft alignment improves pump reliability

Substantial plant operating improvements were achieved following the introduction of a comprehensive pump alignment and monitoring program at a major acetate chemical plant in Derbyshire.

The production process requires materials to be mechanically moved around the plant from process stage to process stage. Some 260 pumps are used on this plant. It is therefore vital that both duty and stand-by plant is reliable and available. Maintenance was very much a firefighting exercise. The plant engineer at the time persuaded management of the need to take a more pro-active view of pump performance maintenance and monitoring. Using PRUFTECHNIK laser alignment systems and condition monitoring equip- ment a coordinated plan to improve plant performance was introduced.

In the preceding years there had been an estimated 120 pumps repaired per year at an annual cost of some £98,000. The calculated mean time between failure (MTBF) of these pumps was 10 months.

By applying a combination of laser alignment of newly refurbished machines and alignment of installed machines when time permitted plus routine plant condition monitoring together with a comprehensive review of installed components such as seals, bearings and gaskets, the plant be- gan to see significant savings on maintenance of the all important pump systems.

The program, now well established, has returned substantial dividends. Plant reliability has improved to more than 46 months MTBF and routine pump repairs have been drastically reduced.

Calculated savings are now in excess of £80,000 per annum, and since the beginning of the program is in the order of £450,000.

A comprehensive plan of action was used by engineers to achieve these extraordinary savings on the plant. The key factors include:

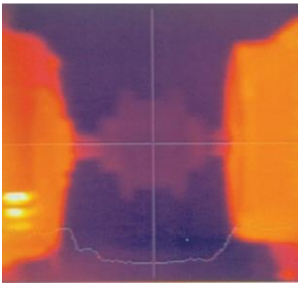
- Engineers and managers commitment to the program
- Patience
- Laser alignment
- Condition monitoring
- Training
- Root cause analysis
- Careful mechanical seal selection
- Careful bearing selection
- Partnerships with suppliers
- Improved piping design and installation
- Considered pump selection
- Advanced lubrication systems selection

Laser shaft alignment improves bearing and seal life

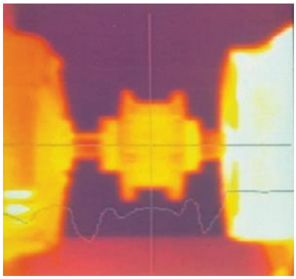
A study was conducted by Infraspection Institute in the USA to evaluate the affect of misalignment on key machine elements such as bearings, seals and couplings.

In a series of tests, misalignment was introduced into a pump – motor set. At each new misalignment interval thermographic pictures were taken to identify the degree of temperature rise on key components.

The tests were conducted across a wide variety of flexible coupling types. Without exception all couplings, bearings and machine housings (and therefore seals) showed significant temperature rise. The graphics below show the affect of misalignment on components when the machine set was aligned to +/- 0.05 mm and when the misalignment was increased to +0.5 mm.



Aligned to 0.05 mm



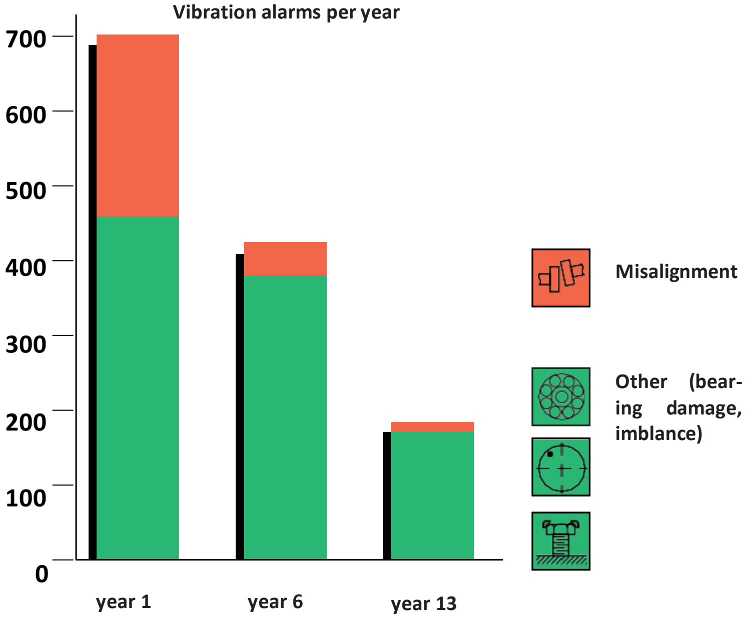
Misaligned to 0.5 mm

Not only was the flexible element of the coupling shown to heat up, but the machines themselves also develop elevated temperatures particularly around the bearing housings. Neither bearings nor seals are designed to operate at the elevated temperatures (caused by misalignment) for pro- longed periods of time. An inevitable result of their operating in these con- ditions is premature failure and reduction in machine operating life.

Laser shaft alignment reduces vibration alarms

A major UK petroleum refinery adopted laser shaft alignment as a standard policy for all coupled rotating machinery. OPTALIGN® and later the ROTALIGN® from PRUFTECHNIK were the preferred systems. They also monitored the incidents of vibration alarms and how, if at all, the use of laser shaft alignment would help reduce this. Alarms were broken down into problems caused by "misalignment" and "other" problems such as bearing damage, unbalance and mechanical looseness.

The graph provided by the company shows clearly that a substantial reduction in alarm violations was achieved, with those of alignment related problems all but eliminated altogether.



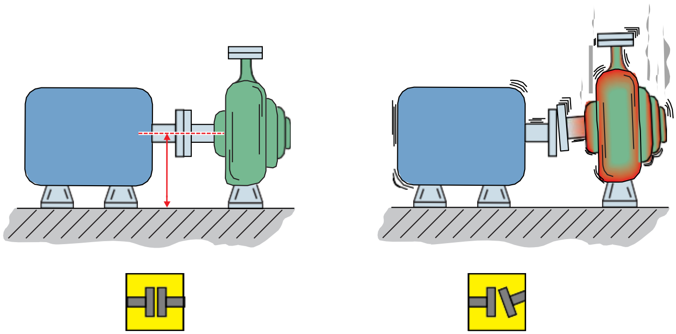
PRUFTECHNIK laser alignment systems

ROTALIGN® touch is a truly versatile laser shaft alignment system, de- signed with simplicity of operation as its first principle. The system has many innovative features including:

- Tablet-like capacitive touchscreen with 3D display
- Extremely durable with strengthened glass screen and tough housing
- High measurement quality with intelliSWEEP
- Live Move with acoustic assistant
- Voice recognition for hands-free operation
- Inbuilt mobile connectivity: RFID, Wi-Fi, Bluetooth and integrated cam- era
- Free cloud space and alignment software
- Live Trend to monitor continuously relative machine positional changes
- Vibration Spot Check to measure overall values
- intelliPASS and intelliPOINT measurement modes for uncoupled shafts

Thermal expansion of machines

In most cases in this handbook, we have considered only the cold align- ment conditions of rotating machines. However, for larger machine sets and for equipment that operates at elevated temperatures on one compo- nent of the machine sets it is necessary to consider the effects of expansion (or contraction) on the alignment condition of the machines. There is little point in accurately aligning a machine set at cold if this alignment condition will change at the normal operating condition of the machine set.



There are a number of ways of establishing the final alignment or operating alignment condition.

- Manufacturers of machines should be able to provide thermal offset information
- Imperial calculation based on coefficient of thermal expansion for spe- cific materials per unit length of center line height per degree of thermal change (see following page)
- On line measurement of cold to hot alignment condition using contact or non contact alignment measurement instruments

Estimating or calculating the effective alignment position change is by no means a simple operation. On complex machine systems such as compres- sors where there are a number of machine elements, each with varying temperature gradients, simple thermal growth calculations become very complex. In these cases on-line measurement of the machine components is usually necessary.

In these cases laser systems such as PRUFTECHNIKs PERMALIGN® system is an ideal tool. Systems such as PERMALIGN® must be designed for long term operation in difficult conditions. Often, the very act of mounting the equipment onto a turbine or compressor operating in excess of 300 degrees C will mean that the measurement system needs to be cooled to avoid dam- age or inaccurate thermal growth readings.

Thermal expansion is not however the only cause of machine position change. Many elements can impinge on the accuracy of the final result; examples included:

- Thermal expansion of bearing supports
- Changes in radial or axial forces
- Changes in oil film thickness on bearings
- Changes in foundation or base plate supports
- Changes in piping forces

Thermal growth calculations
If the direction and extent of growth are known, the machines may be purposely misaligned such that they grow into place, resulting in good alignment condition during normal operation.

PRUFTECHNIKs laser alignment systems contain a special function designed especially to incorporate such alignment target values. The most readily available target specifications for cold alignment are generally obtainable from machine manufacturers. Where this information is not available, the following calculations will assist in establishing thermal growth.

$$dL = L * a * dT$$

where

dL = thermal expansion
L = height centerline to base of machine
a = coefficient of thermal expansion of material
dT = change in temp fromambient

Example:

- A pump with liquid at 150 ºC
- Base to center height 700 mm
- Ambient temp 10 ºC
- a (cast iron) = 0.0000059

dL = L * a * dT
dL = 700 mm * 0.0000059 * (150-10)
= 700 mm * 0.0000059 * 140 = **0.578 mm**

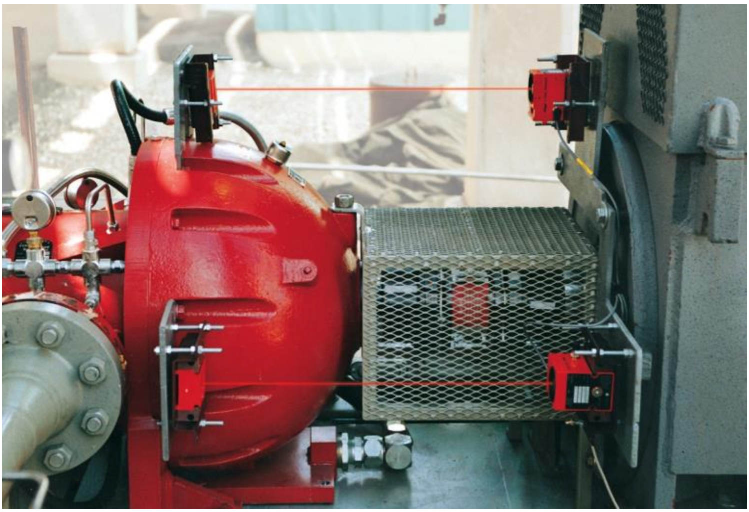
Some advanced laser alignment systems such as ROTALIGN Ultra perform these calculations for you.

PRUFTECHNIK alignment monitoring systems

PERMALIGN® is the ideal system for permanent or temporary monitoring of machinery positional changes. It measures and monitors continuously, and in real time, the alignment changes of rotating machinery during op- eration. It may be used for permanent monitoring or for just the necessary time to determine the positional changes from cold to hot or vice versa.

All four alignment parameters – vertical offset, horizontal offset, vertical angularity and horizontal angularity are monitored simultaneously using two monitors.

- Laser-based positional change monitoring system with graphical display
- Impact of temperature changes minimized through cooling of measure- ment components
- Impact of vibrations minimized by adjusting sampling time
- Monitors multi-element machine trains
- Live Monitoring software application for monitoring analysis (ALIGN- MENT CENTER)
- Intrinsically safe option available



Alignment of pulleys and sheaves

The use of flexible belt drives represents a significant percentage of all industrial power transmission applications, particularly when the speed of the driver and driven shafts are different or where shafts have to be widely separated. There are obviously a number of design factors that preclude the use of flexible belts. But where appropriate, such drives offer an efficient and economic design solution with some useful advantages over other means of power transmission. The advantages include:

- Overall economy
- Cleanliness
- No need for lubrication
- Low maintenance costs
- Easy installation
- Damping of shock loads
- The ability to be used for variable speed power transmission between widely spaced shafts

The power that is transmitted by a belt during operation works on the rim of a pulley, therefore the belt on a pulley drive system must be tight enough to prevent 'slip' during operation. The forces that work during operation are not uniform around the entire belt length. There is always a tight side tension and a slack side tension. The difference between these pulls is often referred to by manufacturers of belts systems as the effective or net pull. This effective pull is applied at the rim of the pulley and is the force that produces work.

The design of pulley systems and the selection of correct belt design and application is by no means simple. The number and variety of belt types is a testament to this. Manufacturers of belts and pulleys do not produce a wide range of different types and styles just to be different from their competitors. Vee belts, flat belts, wedge belts and synchronous belts all have their specific design applications and application criteria. Within each design section there are numerous different configurations, cross sections and operating criteria.

It is not the brief of this section to consider either belt design or selection. Listed in the bibliography at the end of this book are a number of useful guides to belt selection and design.

It is however useful to note the following basic design criteria.

- The maximum center distance of pulleys should be around 15 times that of the pitch of the smallest pulley, and should not exceed 20 times the pitch of this pulley. Distances greater than this require tight control of belt tension because a small amount of stretch will cause a large drop in belt tension, creating slip and power transmission inefficiency.

During operation a flexible belt experiences three types of tension as it rotates around a pulley.

- Working tension (tight side – slack side)
- Bending tension
- Centrifugal tension

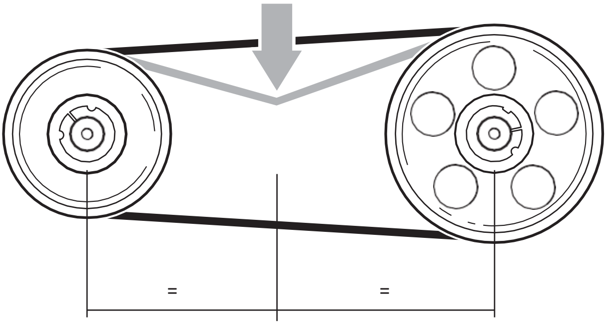
Belts are designed to withstand these working operation states (provided that pre-selection of the belt meets the operating criteria). The design life of the belt will be met and usually exceeded provided that no other forces other than the above act upon the belt during its operating life. Forces such as misalignment and loose or overtightened belt tension are killers of flexible belt drives. Useful operating life can be reduced by as much 80% by poor pulley alignment. In addition to belt wear, pulleys, bearings and seals are also damaged by inattention to basic installation requirements.

Belt tension

The required tension of newly installed belts is virtually always specified by the belt manufacturers. This specification should always be followed meticulously. If there is no specification then a guide to belt tension can be applied as follows:

Tension load =

Distance (cm) between the axes of the driver and driven shafts x 1.0 mm



After tensioning and alignment re-start the machine. After a running pe- riod of 48 hours the tension on the new belts should be re-checked and re-tightened to correct any mid span deflection that exceeds the tension specification.

It is advisable to use a custom designed belt tension testing device for accurate and repeatable measurement of belt tension. Periodic checking of each belt drive will quickly identify any drives that need tightening (or loosening) before incipient damage to the belts and other components cause premature failure.

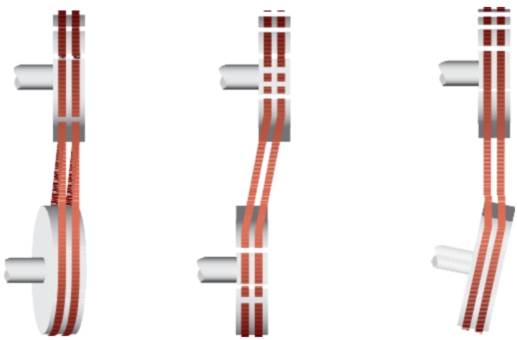
Pulley alignment

By far the most common and damaging installation error that occurs on belt drives is that of misalignment of the driving and driven pulleys. This is not usually due to carelessness on the part of the installer. It is more often due to a lack of suitable tools with which to carry out the required align- ment. For many years, at best, a tight wire or straight edge were the only available tools with which to do the job.

Both methods rely entirely on the installers eyesight to ensure that the alignment is correct. Neither method has any measurements documented. Both rely upon the installer adjusting the driven pulley until the faces or grooves of the driven pulley touch the surface of the straight edge or tight wire. The driven pulley is then rotated half a turn and then rechecked and adjusted. The measurement is then repeated until the pulleys appear to be in line. No angularity or inaccurate mounting of the reference line is measurable. The system is purely an estimate of the alignment of the two pulleys.

Types of pulley misalignment

There are three basic parameters that describe pulley misalignment. These are **vertical angularity**, **horizontal angularity** and **axial offset**. These conditions usually occur in any or all combinations of alignment condition.



Vertical angularity Axial offset Horizontal angularity

Pulley run-out

In addition to correct alignment of the pulleys, the run-out errors of the pulley should also be measured and corrected. The two types of run-out - rim (radial) and face (axial) should be corrected until they meet tolerance before final alignment of the pulleys takes place. If this is not corrected the effect could be that the belts slacken off at one position and then snap into tension at the opposite position. This continuous snapping action if not corrected quickly wears out belts and bearings.

The tolerance for radial or rim run-out on high speed pulleys (1500 rpm and higher) should not exceed 0.12 mm (0.005 inches) total indicator reading (T.I.R.) on average, and may be increased up to 0.24 mm (0.01 inches) on slower pulley drives.

The tolerance for axial or face run-out should not exceed 0.05 mm per 100 mm (0.0005 per inch) of pulley diameter for high speed pulleys, and may be increased up to 0.1 mm per 100 mm (0.001 per inch) of pulley diameter for slower pulley drives.

Check that the offset from the pulley mounting face to the groove is the same for both pulleys.

The pulley or machine manufacturers tighter tolerance recommendations should be followed where possible. Start by checking for radial run-out. If unsatisfactory, check for shaft run-out. If excessive run-out is present on the shaft it may be bent and must be replaced before radial run-out on the pulley is checked again. If no shaft run-out is detected replace the pulley instead. If the pulley is mounted on a tapered shaft bushing, inspect and clean the bushing both inside and out to ensure proper seating.

Now check for face (axial) run-out and if necessary correct it by reposition- ing the pulley on its shaft. When run-out is in tolerance proceed to install new belts.

Drive belt fitting

- Clean pulleys of all foreign matter with a stiff brush, not a wire brush, as this can damage the surface of the groove walls.
- Use the “go-no-go” profile gauges that can be obtained from the belt manufacturer to ensure the pulley condition is acceptable for the fitting of new belts.
- Replace any pulleys with worn, chipped or cracked groove surfaces.
- Install new belts on the pulleys so that the slack sides of all belts are on the same side at either top or bottom of the drive.

DO NOT UNDER ANY CIRCUMSTANCES install belts by prying them onto pulleys by any forcible method. Belts should be fitted by hand pressure only.

- De-tension the motor to allow this fitting to be effected without undue pressure.
- In the case of multiple belt drives when replacing belts, even where only one belt appears worn, all belts should be replaced together.
- Only belts from the same manufacturer should be combined together, preferably a factory matched set.

After replacing belts it is worth examining the replaced belts for noticeable defects such as cracking or glazing. The condition of the belt is a good indication of the type of installation problems. Uneven wear or cracking on belt sides are a good indication of misalignment. Glazing on the contact surface of the belts indicate slipping and therefore poor belt tension.

Checking soft foot

Having mounted the belts and positioned the driver and driven units in their approximately correct position, a check on the soft foot condition of the movable machine (usually the driver) is useful. Undetected soft foot can lead to distortion of the machine frame when bolted down, causing dam- age to bearings, seals and higher than acceptable vibration on bearings.

To check for soft foot use feeler gauges under each machine foot in turn or a dial indicator mounted on a magnetic base. Loosen each foot in turn; measure any rise in the loosened foot and record it. Tighten the foot down and proceed to the next foot.

Having established the type of soft foot (if any) shim as necessary and re-check each foot. As a guide, no reading of soft foot should be larger than 0.05 mm (0.002 inches).

Alignment tolerances

Having completed the soft foot check the drive is ready for alignment Whichever system is employed for this, be it tight wire, straight edge or laser system (a brief description of this type of alignment will follow), the alignment should be as accurate as possible.

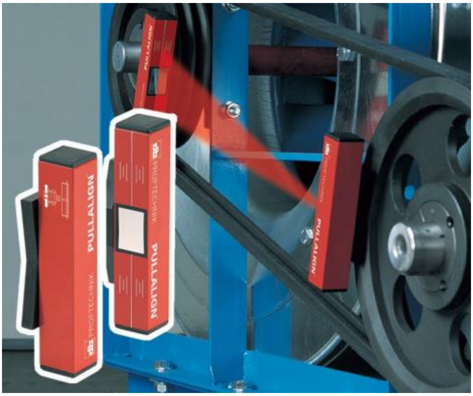
The nominal recommended tolerance for belt drives is **0.5 degrees**. Most major belt and pulley manufacturers specify this value. Better tolerances can be achieved if the alignment procedure is carefully followed. The table below converts the tolerance from degrees into offsets in mm per 100 mm and in thousandths of an inch per inch.

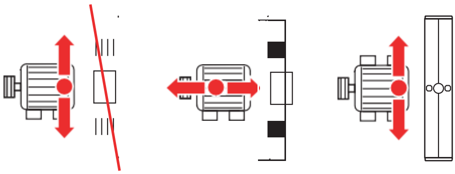
Tolerances conversion table		
Angle of offset misalignment	**Offset mm / 100 mm**	**Offset thous / inch**
0.1°	0.18	0.00175
0.2°	0.35	0.00349
0.3°	0.52	0.00524
0.4°	0.70	0.00698
0.5°	0.87	0.00873
0.6°	1.05	0.01047
0.7°	1.22	0.01222
0.8°	1.40	0.01396
0.9°	1.57	0.01571
1.0°	1.74	0.01745

Note: Values between 0.1° and 0.5° fall within recommended tolerances.

PRUFTECHNIK belt pulley alignment system

PULLALIGN® is tailor-made for the job as it is easy to use and only requires a single operator. Due to its versatile design and strong magnets, the units mount onto virtually any pulley face or ring gear.

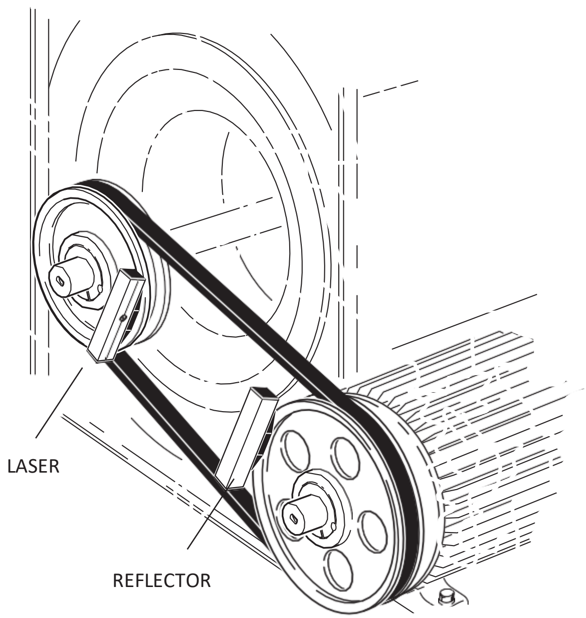




Vertical Angularity Axial Offset Horizontal Angularity

- Displays offset and angular misalignment
- More accurate and efficient than wires and straightedge
- Setup is quick and requires no training
- Prolongs belt and pulley life
- Reduces vibration and belt noise
- Reduces downtime, manpower needed and energy costs

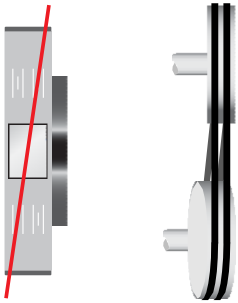
PULLALIGN® comprises two compact measurement units, a laser and a reflector system both with magnetic mounts to enable mounting onto pulley faces. Each unit is marked with a series of graduation lines to enable fine adjustment of the movable machine in order to correct vertical and horizon- tal angular misalignment and axial offset.



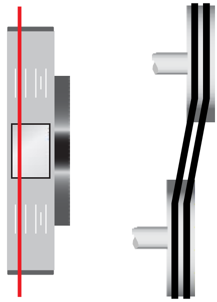
The laser sender transmits a laser line onto the reflector mounted on the opposite pulley (the laser sender should be mounted on the stationary machine). Depending on alignment condition the laser line will be shown clearly on the reflector and also will be transmitted back to the laser sender. The reflector indicates any vertical angularity that is present and simultaneously shows the amount of axial offset. The laser sender shows the horizon- tal angular condition of the drive.

The diagrams on the following page show typical displays of misalignment condition.

Laser line on the reflector unit

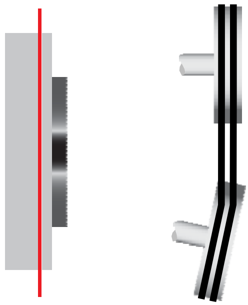


Observed vertical angularity



Observed axial offset

Laser line on the laser unit



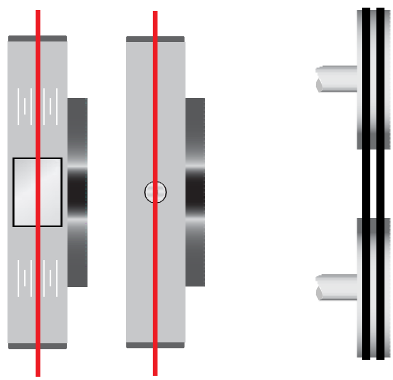
Observed horizontal angularity

Correction procedure

- Correct vertical angularity by shimming the movable machine – this can often be achieved by shimming (or removing shims) from the rear feet of the movable machine only. The corrections can be viewed on the re- flector during adjustment.
- Correct horizontal angularity by adjusting the movable machine laterally. This can be viewed on the laser sender during adjustment.
- Correct offset by adjusting the movable machine axially, this correction can be observed on the reflector unit whilst adjustment is in progress.

By following the three steps described above the alignment of the pulleys should be quickly effected. When you are satisfied that the alignment is correct, it is then necessary to properly tension the belts in accordance with the manufacturers tolerances. Leave the PULLALIGN® system in place during tensioning of the belt, this will give a clear indication of any changes to the alignment condition of the drive. If adjusting tension has changed the alignment condition make adjustments as required by following steps described earlier.

Laser line on the reflector and laser units following alignment correction



Section 2

Vibration Analysis

Condition Monitoring

Most people involved in plant maintenance have heard of Condition Monitoring (CM). By definition CM means to periodically view machine oper- ating condition and when necessary respond to any changes in machine condition. CM can be carried out by a number of “maintenance functions”; visual inspection, wear debris analysis, thermographic analysis and vibration analysis are the most popular methods. In this handbook we are concerned with the use of vibration analysis to measure, monitor and analyze machine condition.

Later we will look at vibration measurement techniques and explain some of the basic parameters and terminology that is used for machine condition measurement. First, it is useful to briefly look at the benefits that a CM regime can bring to an operating plant.

In 1988 the DTI reported that companies who have implemented a CM program on their plant on average spend 25% less on maintenance of the plant than companies who have no CM program (DTI Boardroom report on maintenance in British Industry 1988).

Given that a moderately sized UK plant will spend £250,000 annually on plant maintenance a saving of £62,000 plus additional savings on produc- tion, power and ancillary activities represents a very good return on the investment in a CM program.

If the returns on an investment on a CM program are so good why doesn’t every plant have a system in operation? Most often the answer to this lies in a lack of understanding of what is required for CM on the plant, and on a fear that the cost of implementing a system and running it will be more than the return on cost that the system will realize.

Condition monitoring essentially means that the machines on plant get a “regular health check.” This is usually taken in the form of periodic vibra- tion measurements. These measurements are compared to a standard or “known” operating condition. In the case of vibration this standard is usu- ally an ISO norm or in some cases an on plant standard or manufacturers recommendations. By making a comparison between current condition and “ standard or known” condition an evaluation can be made as to whether the machine operating condition has changed. Depending on the extent of change the machine condition can be further investigated or monitored more frequently to detect further changes. The key tool in this CM concept is to trend collected data and respond to changes in the trend. The objec- tive is to intervene before the machine fails catastrophically.

The "Nuts and Bolts" of a CM system

Having identified the basic principle of a CM regime, what are the costs of getting a program up and running? The answer is, not much. A system can cost as little as a few hundred pounds for a simple portable "point and shoot" product that is touched onto the machine and gives a reading of vibration severity, usually RMS vibration. This reading can be manually recorded on a chart. Readings are then compared over time to identify a change in measured operating condition.

VIBSCANNER – a typical low cost portable CM system

A development of this is to collect machine condition data and input this into a PC program to automatically establish and trend machine condition data. Further developments allow analysis of machine condition via algo- rithmic calculations (FFT analysis) to determine specific machine condition defects. From portable systems you can progress to on-line monitoring sys- tems which provide round the clock measurement and alarming of machine condition.

As system complexity and analysis capabilities expand so does system cost. Normally the criticality of machines to the operation of the plant will determine the level of monitoring system required on the plant. It does not follow, however, that a company who spends £20,000 on a CM system gets a 20 times better system than a company who spends £1,000, or will get 20 times better CM results.

Take the CM route that is most comfortable. Match the system to the plant and to the skills available. Don't go for the most expensive system option just because a salesman says it is the best, it may be, but, it may not be appropriate to the plant.

If the plant has many process critical machines on-line monitoring may need to be considered. If the process plant is small, start simple; routine vi- sual inspection and simple overall vibration readings will suffice. If in doubt, irrespective of plant size, start simple. Provided the system implemented is capable of expansion and can grow as CM requirements grow, you need not worry about more complex analysis capabilities.

Trend the data collected – most systems (even the most expensive) trend the data and only analyze when a problem occurs. If necessary outsource the analytical expertise required.

Implementing a CM program

First and foremost, implementing a CM regime means that you have to know your plant. A basic understanding of the way the machines behave and the way they should behave is essential. This doesn't mean extensive initial investment in sensors, expert analysis or highly skilled personnel. Information from ISO standards, machine suppliers and past plant operating experience will often provide the information required to initially establish how the plant **should** behave. How the plant **actually** behaves can be established by a combination of techniques including vibration measurement, thermography, oil analysis and operator experience of the plant.

One of the keys to running an effective CM regime is the investment that the plant management is prepared to make in ensuring operators are skilled in using the systems they employ, and that on going training is available to maintain operator skills.

Implementing and maintaining a CM regime doesn't have to be a full time job; but is does require commitment and regular monitoring routines via some form of data acquisition, storage and regular review.

Returns on CM investment

The most effective prevention of machine breakdown is a combination of regular data acquisition, trending analysis, root cause analysis, and machine operator awareness. Attention to changes in operating condition, a leaking seal, an increase in overall vibration, a change in machine operating temperature or even an increase in operating noise will notify an alert operator or engineer of a potential problem. It doesn't mean you shut the plant. What it should mean is that you investigate further, eliminate variable pro- cess changes and then increase frequency of monitoring of the machine to establish the rate of change in operating condition. A rapid rise requires intervention quickly before plant failure, a slow rise means that you can plan a convenient future time for intervention. This is one of the principle benefits of implementing CM on your plant.

- It allows your engineers to plan plant shutdown, order spares, and get the right personnel available to carry out the shutdown work.

Everyone who has been involved in maintenance or production using process plant will have experienced a "sudden" plant failure. A bearing on a pump for example seizes; the catastrophic result is a mechanical seal is destroyed; product spills; a shaft or coupling is destroyed. If you can reduce or eliminate these "sudden" failures and intervene before catastrophic failure, only the failing component (in this case the bearing) has to be replaced.

- CM will help prevent ancillary plant damage thus the cost of replace- ment parts will be lower, and time to effect repair will be less – plant will restart quicker.
- Labor costs can be reduced by focusing the work force on problem areas.
- A proven CM routine should enable you to negotiate better plant insurance rates.
- You can "push" process machines harder to gain extra production if needed whilst monitoring plant condition.
- You can reduce or eliminate routine machine shutdown.
- You can build production reserves prior to a forecast machine shutdown in order to eliminate production losses.

The common denominator of all these benefits is cash. Improved process plant availability, and reduced maintenance costs effectively mean a more profitable production plant.

Having reviewed the options available and the CM strategies that you can adopt, you should sit back and ask what you really want out of a plant improvement scheme. You may just want a quiet life, or to improve machine reliability or you may want to improve plant-operating profitability. What you want will dictate what you are prepared to spend and commit to CM. Whatever the reasons for the investment, CM will repay you long after the cost of equipment has been capitalized and written off in your accounts.

Vibration analysis

Vibration data has high information content

Vibration measurements contain a lot of useful information that will help determine the health of the machine for example:

- It provides information for safe machine operation.
- It can detect that the condition of a machine has changed.
- It can be used to diagnose the cause of change.
- It can be used to classify the condition of a machine.

Vibration measurement is normally a non intrusive measurement procedure and it can be carried out with the machine running in its normal operating condition.

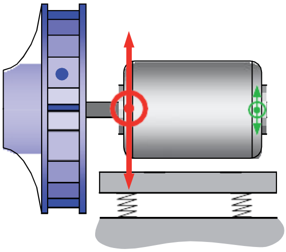
Basic parameters

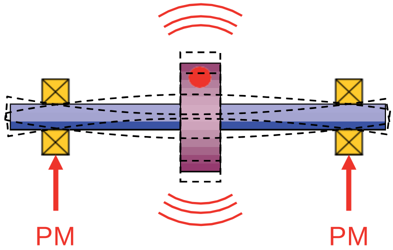
Vibration is an effect caused by machine condition. Vibration is simply the oscillation about a reference point (i.e. a shaft vibrates relative to the casing of a piece of machinery and a bearing vibrates relative to a bearing hous- ing.) Vibration exists when a system responds to some internal or external excitation and can be broken down into 3 basic types.

The amplitude of vibration depends on the magnitude of the excitation force, the mass and stiffness of the system and its damping. Vibration oc- curs because we are not able either to build a perfect piece of machinery or to install it perfectly. If we could build a perfect piece of machinery, the cen- ter of mass of the rotating element would be located exactly at its center of gravity. When the center of mass and center of gravity do not coincide the rotor has a heavy spot and some degree of unbalance. This unbalance pro- duces a vibration proportional to the amount of weight of the heavy spot. Additional sources of vibration are machine tolerances, machine structure, bearing design, loading and lubrication, machine mounting and rolling and rubbing between moving parts.

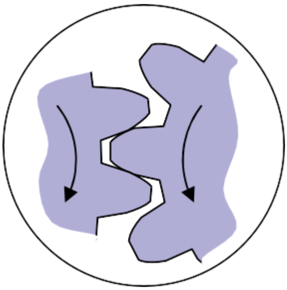
The analysis of vibration requires an understanding of the terminology used to describe the components of vibration.

Free body vibration

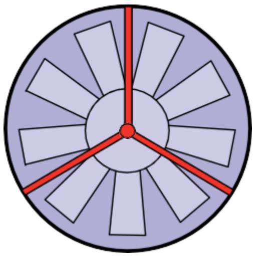




Meshing and passing vibration

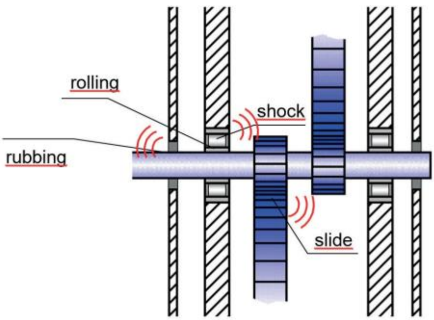


Gear mesh vibration



Blade pass vibration

Frictional vibration



Frequency

Frequency is the cyclic movement in a given unit of time. The units of frequency are:

- rpm = revolutions or cycles per minute.
- Hertz (Hz) = revolutions or cycle per second.

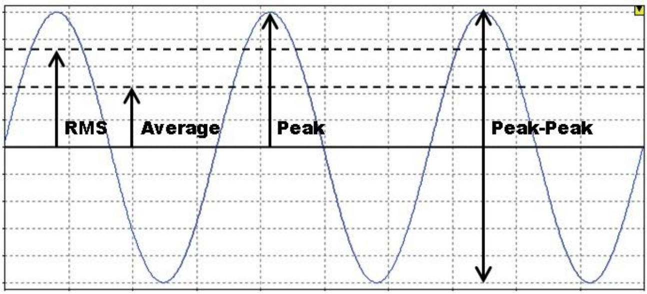
These are related by the formula: F = frequency in hertz = rpm/60.

Amplitude

Amplitude is the magnitude of dynamic motion of vibration. It is typically expressed in any of the following terms:

- RMS (Root Mean Square); Zero to Peak; Peak to Peak.

The sketch below illustrates the relationship of these three units of measurement associated with amplitude.



Amplitude, whether expressed in displacement, velocity or acceleration is generally an indicator of severity. Since industrial standards of vibration severity will be expressed in one of these terms, it is necessary to have a clear understanding of their relationship. Care must be exercised to note the "type" of amplitude measurement when comparing machinery vibration to industry standards.

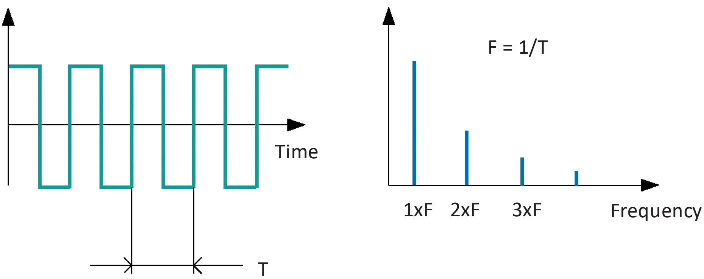
Fundamental Frequency

Fundamental frequency is the primary rotating speed of the machine or shaft being monitored and usually referred to as the running speed of the machine.

You will also see the fundamental frequency referred to as 1 x rpm, or as Hz; using as an example an 1800 rpm motor this would be 30 Hz (1 x 1800/60). The fundamental frequency is important because many machin- ery faults such as misalignment or unbalance occur at some multiple of the fundamental frequency. For example misalignment at 1 x fundamental frequency.

Harmonics

These are the vibration signals having frequencies that are exact multiples of the fundamental frequency (i.e. 1 x F, 2 x F, 3 x F etc.).



Displacement (D)

Displacement is the actual physical movement of a vibrating surface. Dis- placement is usually expressed in mils (thousands of an inch) or microns. When measuring displacement, we are interested in the Peak to Peak displacement which is the total distance from the upper limits to the lower limits of travel.

Velocity (V)

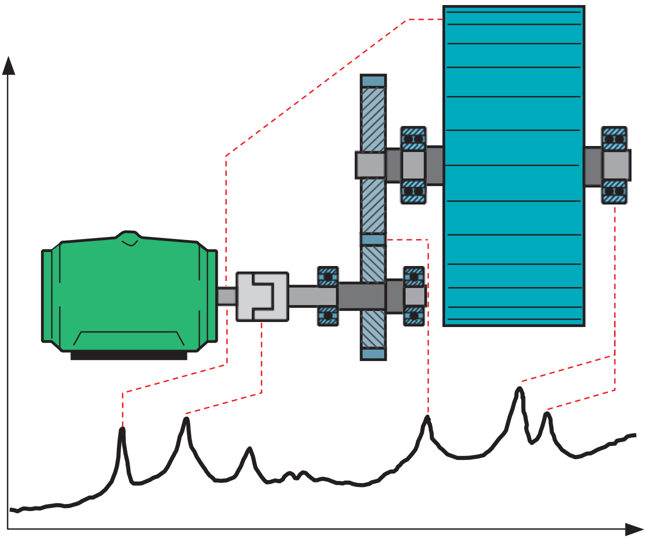
Velocity is the speed at which displacement occurs. We define velocity as the rate of change in the relative position. Velocity is usually measured in mm/sec RMS, or inches/sec RMS.

Acceleration (A)

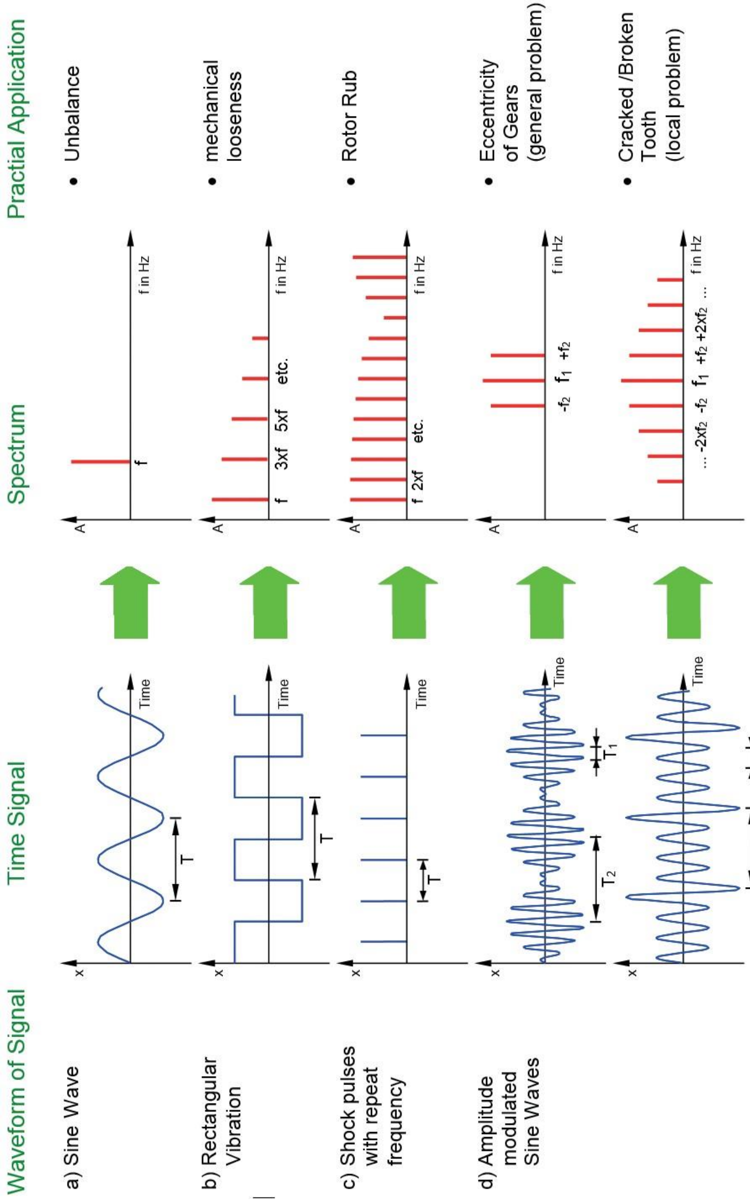
Acceleration is the rate of change of velocity. This we can simply define as the change of velocity in a period of time or change in rate of velocity. Ac- celeration is usually measured in m/s^2 or in g's of gravitational force.

Vibration frequency spectrum

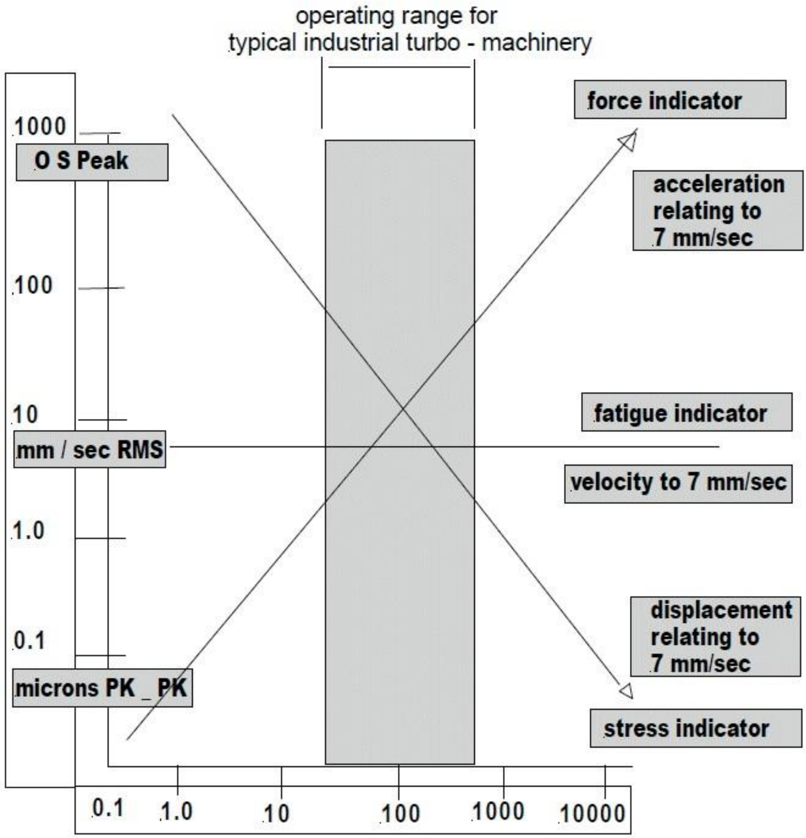
Machinery vibration consists of various frequency components as illustrated below. The amplitude of each frequency components provides an indica- tion of the condition of a particular rotating element within the machine.



The advantage of frequency spectrum analysis is the ability to normalize each vibration component so that the complex machine spectrum can be divided into discrete components. This ability simplifies the analysis of mechanical degradation within the machine. The following chart illustrates typical signal forms for various machine components.


Waveform of Signal
Time Signal
Spectrum
Practial Application
a) Sine Wave
b) Rectangular Vibration
c) Shock pulses with repeat frequency
d) Amplitude modulated Sine Waves
x
Time
T
T1
T2
A
f in Hz
f
3xf
5xf
etc.
2xf
-f2
f1
+f2
... -2xf2
+2xf2 ...
Unbalance
mechanical looseness
Rotor Rub
Eccentricity of Gears (general problem)
Cracked /Broken Tooth (local problem)

Relationship between displacement, velocity, amplitude, frequency Variation in the values of velocity and acceleration with frequency is ex- tremely important, for it forms the basis for vibration severity criteria, provides guidelines for selecting the variable which will be most represen- tative for a particular purpose, and explains how failures can occur without warning if the wrong variable is monitored. This variation is best illustrated by plotting displacement and acceleration verses frequency at a constant velocity amplitude of 7 mm/s as shown below.



Note that velocity appears to be a valid indicator of condition across the entire range of frequencies. This is the main reason why vibration is used as the prime indication of mechanical condition.

The relationship between displacement, velocity and acceleration also provides the best indication of which parameter should be measured to assess condition. The diagram clearly shows that when examining the low frequencies around or below the running speed of most machinery, displacement or velocity measurements are likely to produce the best quality signal. On the other hand, phenomena such as bearings resonance's at 5 – 10KHz and above, are best measured in terms of acceleration.

FFT (Fast Fourier Transform)

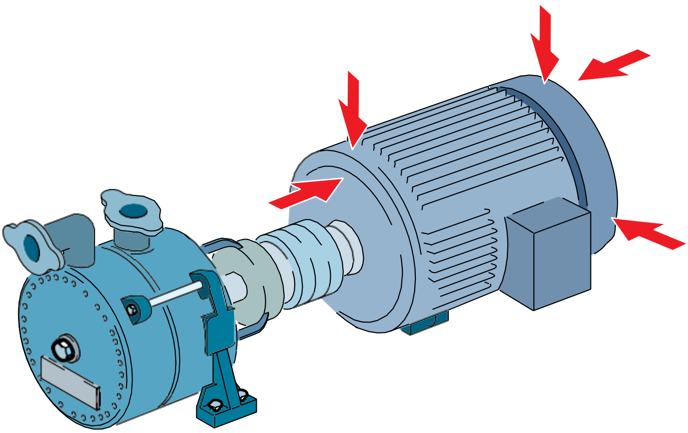
FFT is predominantly the most used tool in analysis of spectral data with respect to vibration analysis of machine components. Fourier transform is a mathematical operation which decomposes a time domain function into its frequency domain components.

Transducers

Mounting location

The position and the manner in which data is collected is very important to a successful vibration monitoring program. In order to properly diagnose a fault, data must be collected in the right plane and must be repeatable. Some faults show the high amplitude in the radial direction and some in the axial direction.

Measure location should usually be on exposed parts of the machine that are normally accessible and that reflect the vibration of the bearing hous- ing. Vertical and horizontal mounting directions are the most usual trans- ducer locations for horizontally mounted machines, any angular position is acceptable provided that the location reasonably represents the dynamic forces present in the machine. For vertically mounted machines the location giving the maximum vibration reading should be used as a future monitor- ing reference point



The data collection points should be clearly marked to ensure that data is collected at the same point every time. (Frequently, measurement studs are permanently fixed to the machine ensuring reproducibility of measurement location). When analyzing a machine for changes, the analysis can be inac- curate if data is not collected at the same point each time.

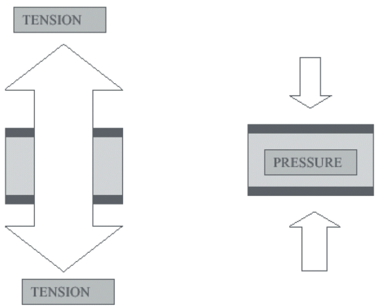
Measurements should be carried out when the rotor and main bearings have reached their normal steady state operating temperature, speed, load, voltage and pressure. Where machine speeds vary measurements should be taken at all conditions at which the machine operates for a prolonged period.

Transducer design

There are a variety of instruments (transducers) that will convert actual mechanical movement (vibration) into electrical energy.

The industrial accelerometer

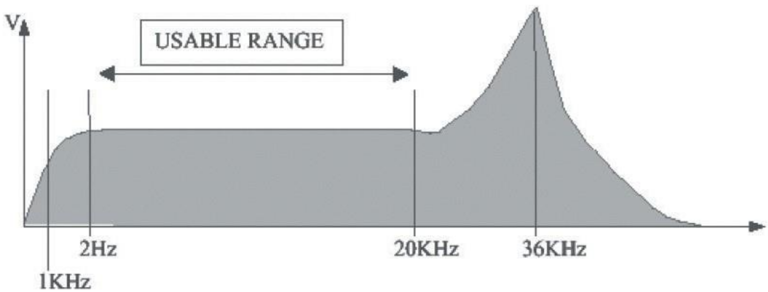
Accelerometers are the most widely used transducer in routine vibration monitoring programs. A typical accelerometer contains a piezoelectric crystal element which is pre-loaded by a mass of some type and the entire assembly is enclosed in a rugged protective housing. The piezoelectric crys- tal produces an electrical output when it is physically stressed by either a pressure or tension effect as shown below.



The variable vibration force exerted by the mass on the crystal produces an electrical output proportional to acceleration. Accelerometers have a broad frequency range, typically from 2 Hz to 10 KHz. The accelerometer is also easily mounted using either a stud, a magnet, an adhesive or by hand-hold- ing it onto the machinery surface. Accelerometers also have good tempera- ture and environmental responses and are usually of a rugged construction.

Accelerometer frequency response

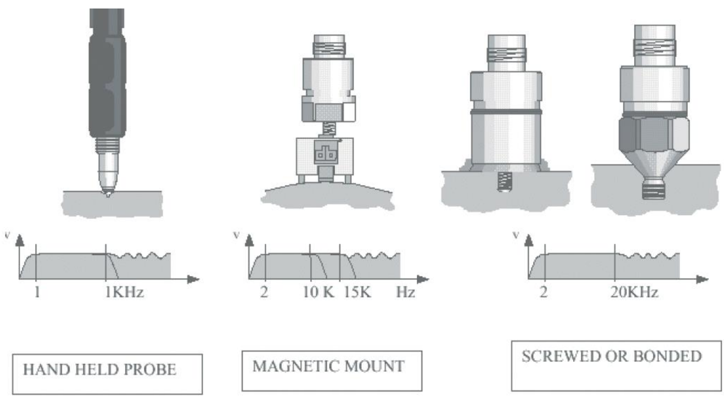
Each accelerometer has a usable frequency range and response curve typically as shown below. If the data to be collected is outside the frequency range shown on the response curve, an accelerometer having the correct response should be chosen. As a general rule, the smaller the accelerome- ter, the higher the usable frequency range.



Typical accelerometer response for threaded or bonded transducers

Accelerometer mounting

The mounting of an accelerometer plays a significant role in its frequency response. Shown below are four different types of mounting methods for transducers, all of which are used in vibration analysis programs.



The screw or stud mounted unit with the proper accelerometer has a frequency response of around 20 KHz. The epoxy mount (glue mount) has approximately the same response. The permanent magnet mount has a frequency response of approximately 5 KHz, while the hand held unit is typically around 1.5KHz. The more rigid the transducer contact with the machine the better the frequency response and hence the better the reliability of the vibration reading.

Signal processing

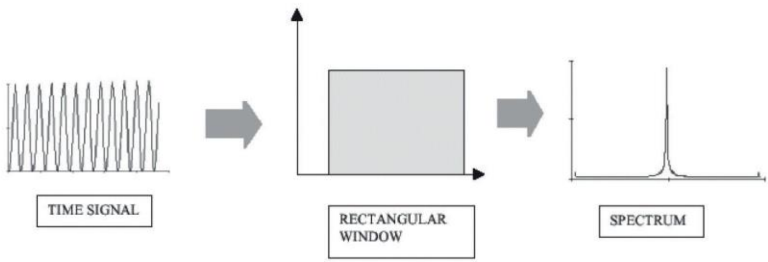
The raw data collected by the transducers must be enhanced to provide useful information. For vibration data this raw data must be "conditioned" to prevent errors. Typically such conditioning includes:

- Filtering to remove unwanted or spurious signals
- Amplifying to enhance the resolution of low energy signals
- Data averaging to remove spurious data
- Conversion to frequency domain (FFT)

To assist with these filtering techniques many analyzers provide a number of "window" functions which, depending on the type selected, will assist with analysis of data.

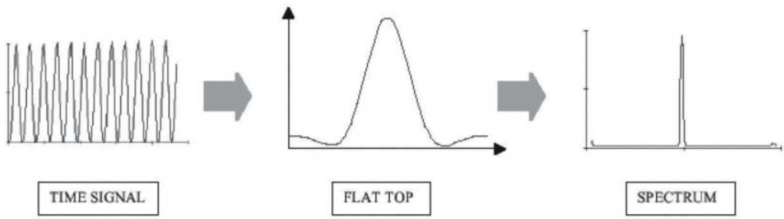
Rectangular window

This provides for higher inaccuracy in the amplitude domain but with a greater accuracy in the frequency domain. A practical use for this window is for transient process e.g. bump tests to identify natural component frequencies.



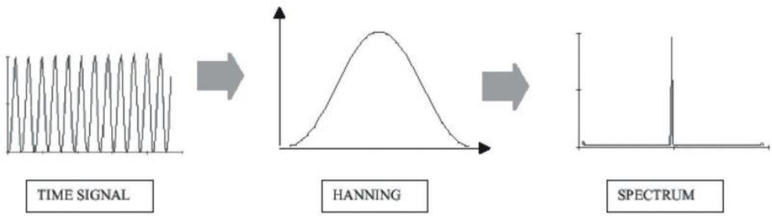
Flat top window

This has the highest accuracy for the amplitude domain but higher inaccuracy for the frequency resolution domain.



Hanning window

This is the standard window for most vibration analysis, it has the best accuracy for the frequency resolution domain, but with higher inaccuracy in the amplitude domain.



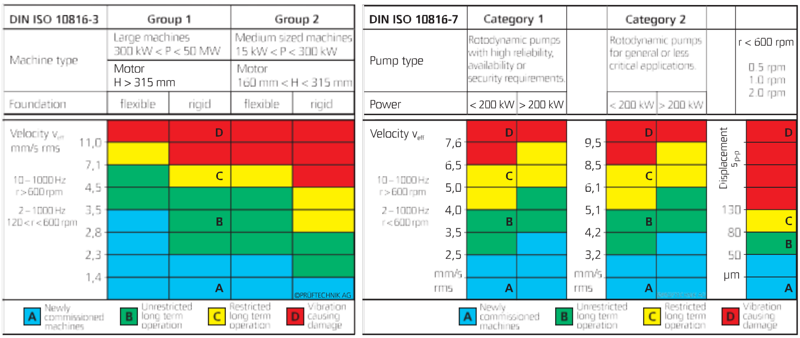
Fault detection

As has been discussed, the main objective of a vibration monitoring pro- gram is the detection of incipient machine failures. The methodologies associated with fault prediction usually involve comparing current vibration information with a vibration description of that machine or a similar ma- chine in satisfactory operating condition. This comparison is made by two methods:

- Comparison to industrial standards – ISO 10816-3-7
- Comparison to a previously measured reading

ISO 10816-3-7

The ISO 10816-3-7 is the current standard for the evaluation of “standard” rotating machine operating condition. Issued in 2009 it covers “large and medium size industrial machines with nominal power rating above 15 kW and nominal speeds between 120 rpm and 15000 rpm”. In addition pumps are added as a specific category for consideration. This range covers most rotating machines and can therefore be used as a good guide for in-situ operating condition. The chart below illustrates the standard and allows a quick comparison of actual against standard operating condition.



Variations will inevitably occur when comparing these standards to actual machine operating condition. Machines should not however be con- demned because of variations in readings without first considering other potential reasons for the difference in readings.

The chart on the previous page shows 4 zones of vibration severity ranging from good to unacceptable.

- Zone A – the vibration of newly commissioned machines would normal- ly fall into this zone.
- Zone B – machines with vibration within this zone are normally considered acceptable.
- Zone C – machines with vibration in this zone are normally considered unsatisfactory for long term continuous operation. Machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial operation. (It is advised to increase the frequency of vibration monitoring during this operating period.)
- Zone D – vibrations within this zone are normally considered to be of sufficient severity to cause damage to the machine.

The use of these zones and the numerical values ascribed to them are not intended as an acceptance standard for machine manufacturers and customers but the values do help to establish alarm and warning criteria for a routine condition monitoring program. If machines are found to be operating at vibration levels consistently above the nominal values shown in the standard, investigation as to the cause should always be carried out.

Comparison to previous readings

This method is the most widely used method of identifying changes in machine operating condition. Most commonly referred to as trending, comparison quickly shows the machine operator or manager if the machine condition has changed, and by how much, and in what period of time. Trended graphical measurement values combined with ISO alarm limits give clear visual warnings of machine condition change and with some software packages, can be used to predict likely run to failure intervals in order to schedule remedial repair. (Graphic predictions of machine trends should always be considered as a guide to likely failure and not as a definitive measurement of machine failure intervals.)

Fault mode analysis

There is a great deal of literature on diagnostic techniques employed for various types of equipment. Although we will not discuss specific diagnos- tic methods in this handbook, you will find that a great deal of information must be readily available to execute an effective vibration monitoring pro- gram. Once the vibration monitoring program flags a machine as potentially having a mechanical problem, the following questions must be answered.

- How severe is the problem?
- What is the problem?
- When must the machine be taken out of service for repair?

Machinery diagnostics using vibration analysis provides information that addresses these questions.

The machinery diagnostics technique viewed here is based on a technique known as “fault mode” analysis. This technique utilizes the fact that spe- cific mechanical events, such as unbalance, misalignment, looseness, bear- ing defects, aerodynamic and hydraulic problems, and gearbox problems usually generate vibration frequencies in specific patterns. The frequency, amplitude and pattern of the peaks in a vibration spectrum can be a telling indication of the type of problem being experienced by the machine. The principles of “fault mode” analysis include:

- Measurement of mechanical faults such as unbalance and misalignment generate mechanical vibration in a well defined frequency pattern.
- Comparing the vibration levels and vibration spectra on similar types of machines will help establish the severity and cause of a vibration prob- lem.

The following chart summarizes specific machinery faults and their vibra- tion patterns.

Possible cause	Dominant frequency	Direction	Comment
Imbalance	1x rotational frequency	Radial for dynamic im-balance; possible axial	Vibration amplitude proportional to imbalance & rpm – causes severe vibration to occur
Misalignment	2x rotational frequency	Radial & axial	Severe axial vibration 2nd harmonic, best realigned using a laser alignment system.
Bearing defect	High frequency vibration	Radial & axial loaded	Use bearing enveloping diagnosis or shock pulse to determine damage severity
Machine foundations	Typically at one or more nat-ural frequency (transient vi-bration)	Radial	Natural resonant frequency of foundation or ma-chine base plate
Belt vibrations	Rotational frequency and multiples thereof	Radial	Additionally recommend strobe to combine ma-chine rpm and belt speed to check for belt slip- page
Blade pass vibration	Number of vanes or blades x the fundamental frequency	Radial	Vibration frequency represented by the number of blades x the shaft rpm
Electrical	Line frequency, 50 Hz (UK) 60 HZ (USA) and multiples thereof	Radial & axial	Side bands may also occur at multiples of the ro-tational frequency. Vibration stops when power is turned off
Gear mesh defect	Gear frequency equal to the number of teeth x the rpm of the gear	Radial & axial	Sidebands occur from modulation of the gear teeth meshing vibration at the rpm e.g. the output shaft speeds of the gearbox
Resonance	Natural component frequency	Radial & axial	A components natural frequency coincides with an excitable frequency

Imbalance

Vibration caused by imbalance occurs at a frequency equal to 1 rpm of the imbalanced part, and the amplitude of vibration is proportional to the amount of imbalance present.

Normally, the largest amplitude will be measured in the radial (vertical or horizontal) directions.

Misalignment

Generally, misalignment can exist between shafts that are connected with a coupling, gearbox or other intermediate drives. Three types of misalign- ment are:

- Angular – where the center line of the two shafts meet at an angle
- Offset – where the shaft center lines are displaced from one another
- A combination of angular and offset misalignment

A bent shaft looks very much like angular displacement, so its vibration characteristics are included with misalignment.

Misalignment, even with flexible couplings, have two forces, axial and ra- dial, which result in axial and radial vibration. The significant characteristics of vibration due to misalignment or a bent shaft is that it will be in both the radial and axial directions. For this reason when axial vibration is greater than one half of the highest radial measurement (horizontal or vertical), then misalignment or a bent shaft should be suspected.

All misalignment conditions will produce vibration at the fundamental (1 x rpm) frequency components since they create an imbalanced condition in the machine. Misalignment will sometimes produce vibration at the second harmonic (2 x rpm).

Loosness

Mechanical looseness can be caused by loose rotating components or loose machine foundations.

Mechanical looseness causes vibration at a frequency of twice the rotating speed (2 x rpm) and higher orders of the loose machine part. In most cases, vibration at the fundamental (1 x rpm) frequency will also be produced.

Bearing problems

One of the results of damage to rolling element bearings is that the natural frequencies of the bearing components are excited by the bearing defect. The resonant vibration or “ringing” occurs at frequencies between 2 KHz and 60 KHz.

This vibration is most effectively measured at a level of acceleration in units of g’s Peak. Vibration is measured by the machinery monitoring system as a HFE (High Frequency Energy) measurement and gives an effective indication of the condition of rolling element bearings. Based on field experience, the shock pulse technique works well on motors and other quiet equipment. Care must be taken when using the technique on pumps and gearboxes, where flow, cavitation, and tooth meshing can produce impulses which interfere with and mask the impacts produced by bearing defects.

Rotational frequencies related to the motion of the rolling elements, cage and races are also produced by mechanical degradation of the bearing. These frequencies are dependent on bearing geometry and shaft speed and can be found typically, in the 3 – 10 x rpm range and because of these reasons the “enveloping” method is the most widely adopted method of viewing specific bearing defects. This method of bearing condition evalua- tion and that of shock pulse analysis of bearing condition are reviewed in the following pages.

Aerodynamic and hydraulic problems

Normally associated with blade or vane machinery such as pumps or com- pressors, aerodynamic and hydraulic vibration is created by an unstable or unbalanced condition within the machine.

In most cases this will produce a vibration at the fundamental frequency (1 x rpm) of the machine and blade pass/vane frequency components.

Gearbox problems

Gear defects or faulty gears produce low amplitude, high frequency vibration. The vibration is predominantly at gear mesh frequency. Gear mesh frequency is calculated as follows:

Gear mesh frequency (GMF) =
Speed of output gear x Number
of teeth in output gear

Example:
52 tooth gear running at 90 rpm (90/60 = 1.5 Hz).
GMF = 52 x 1.5 = 78 Hz.

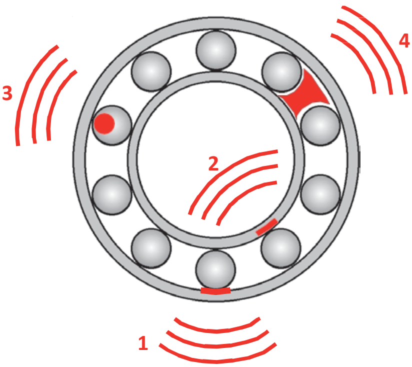
Most gear problems exhibit vibration at the gear mesh frequency, and may be summarized as follows:

- Imbalance – predominant at the 1 x rpm of the gear.
- Misalignment – predominant at the 1 x rpm and 2 X rpm; may excite GMF.
- Pitch line run out – predominant at GMF with 1 x rpm sidebands.
- Faulty gear teeth – predominant at GMF with sidebands at 1 x rpm of faulty gear.

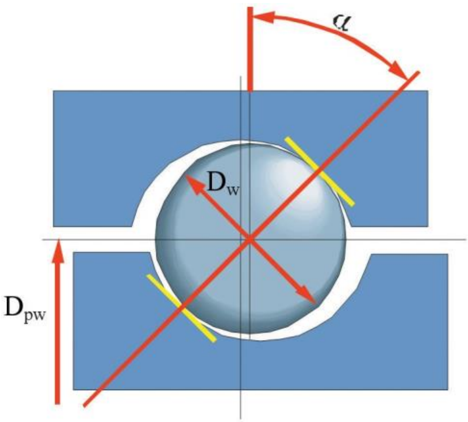
Basic theory of enveloping

When a bearing defect exists in a rolling element bearing the vibration signature will show high frequency vibration generated each time a damaged roller or damaged race make contact. These repetition rates are known as the natural bearing defect frequencies. In any rolling element bearing arrangement there are four types of element defect frequency.

- Ball Pass Frequency Outer Race – BPFO
- Ball Pass Frequency Inner Race – BPFI
- Ball Spin Frequency – BSF (Rolling element defect)
- Fundamental Train Frequency – FTF (Cage defect)



1 = Outer ring defect
2 = Inner ring defect
3 = Rolling element defect
4 = Cage defect



a = contact angle
D_w = Rolling element diameter D_{pw}
= Bearing pitch diameter
Z = Number of rolling elements
n = Shaft speed in rpm

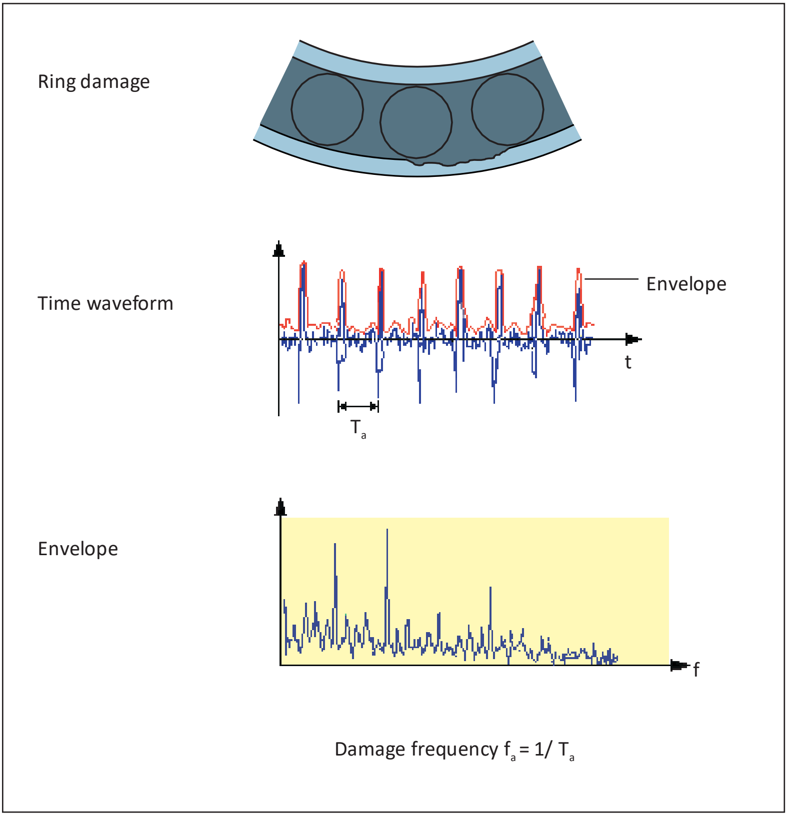
Formula for calculating bearing defect frequencies:

- BPFO $= Z * n / (60 * 2) * (1 - (D_w / D_{pw} * \cos (a))$.
- BPFI $= Z * n / (60 * 2) * (1 + (D_w / D_{pw} * \cos (a))$.
- BSF $= (D_{pw} * n) / (D_w * 60 * 2) * (1 - [D_w / D_{pw} * \cos (a)]^2)$
- FTF $= n / (60 * 2) * (1 - (D_w / D_{pw} * \cos (a))$.

Example: Pass frequencies.
Bearing type SKF 6211, operating speed 2998 rpm

Dimensions	Defect frequencies	
D_{pw} = 77.50 mm	BPFO = n / 60 * 4.0781	= 204 Hz
D_w = 14.29 mm Z	BPFI = n / 60 * 5.922	= 294 Hz
= 10	BSF = n / 60 * 5.239	= 264 Hz
a = 0	FTF = n / 60 * 0.4079	= 20 Hz

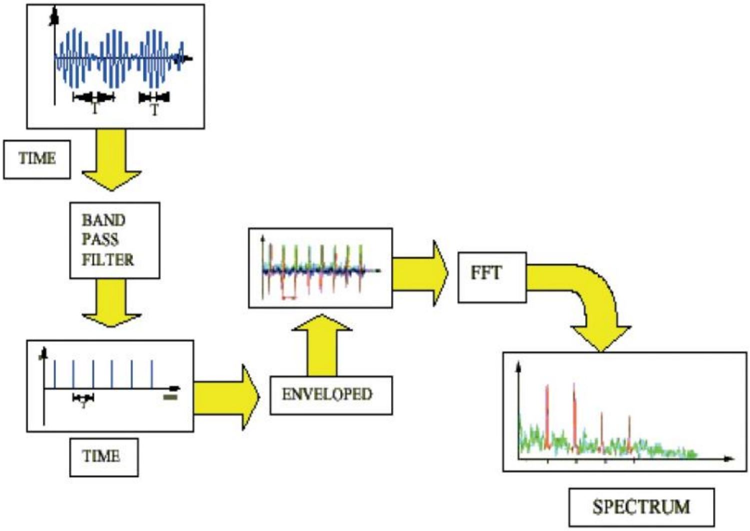
Enveloping is essentially a 2-stage process; the first stage is a band pass filtering of the time waveform. The filtering process results in a series of spiky peaks when enveloping is applied to extract the repetition rate relating to the bearing defect and its harmonics as shown in the following frequency spectra. Since healthy rolling element bearings may exhibit vibration at the natural frequency of the bearing components, it is very important to mea- sure accurately the severity of bearing deterioration.



To measure the severity of a defect in an enveloped spectra the following must be undertaken:

- Measure the amplitude of the specific component in dB (decibels) above the carpet value shown in the spectra.
- The spectra above identifies the carpet and peak values.
- Experience tells us that when a bearing starts to deteriorate a peak to carpet difference of around 10 dB can be seen. This does not mean that bearing breakdown is imminent. As deterioration continues to show a difference of around 15 dB between peak and carpet levels, the bearing should be monitored more closely and preparation made at some point to strip down the machine for repair.

- When the defect amplitude is 20 dB or greater immediate action should be initiated to repair and or replace the bearing.
- The carpet level of the bearing should not be used as a stand alone method of monitoring bearing condition but should be used in conjunc- tion with another trending technique such as shock pulse measurement.



The above diagram shows the steps involved in obtaining an envelope spectra for a bearing.

A common sense approach

Most vibration problems respond well to a logical, systematic approach. A list of suitable steps towards firstly defining and secondly solving problems is given below:

Raw data

- Where is the vibration level highest on the machine and in which direction?
- Is the vibration present in associated machinery and pipework or is it at highest levels on the bearing houses?
- Do changes to the process and lubricating the bearings radically change the vibration response?
- Does the trend show a roughly exponential growth with time?
- How does the machine feel and sound in comparison with similar machines elsewhere?

Diagnostics

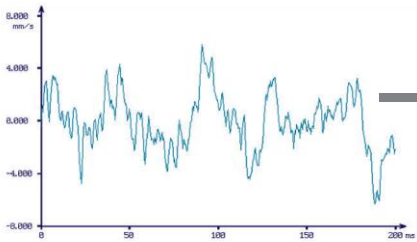
- Is this a new machine or one which has recently been worked on? If so, what could have been assembled wrongly and does this tie up with the raw data?
- Which frequencies are dominant on the spectrum? Do these occur at gear meshing frequencies or low order multiples of shaft speed?
- On rolling element bearings and gearboxes, how does the enveloped spectrum appear?
- How quickly is the machine deteriorating and hence how soon does it need to be repaired? (This includes consideration of capital worth of equipment, downtime and maintenance costs.)
- Vibration is usually highest at the point of maximum damage unless a resonant condition exists.
- Vibration is usually the response of a machine to a fault so the only way to stop the vibration is to find the source not the response.
- It is always necessary to build up several items of evidence before diagnosing a fault. The weapons available are HFE, spectra, envelopes, temperature and sound. For each fault the interaction and evidence from these will be different.
- Vibration is a physical phenomenon and as such can be defined by phys- ical means.

Solutions

- Where a fault has occurred on previously sound equipment it should be clear from the steps suggested above where the likely problem lies. Having defined the problem, the best course of action should be clear.
- If the fault is on new or recently serviced equipment it may be unclear where the problem lies. Is something resonant? Is there a defect in the installation? Is there a basic design error? The solution to a problem of this type should be achieved in a logical manner. Try one solution at a time (starting with the most likely) taking new sets of data at each step. The best solution will gradually emerge.
- Try to explain all the responses in relation to the damage found once the problem is solved.

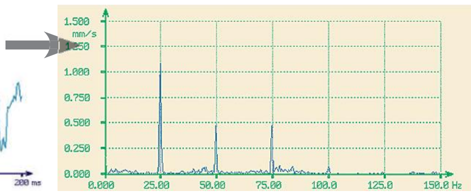
FFT analysis

As the name of this short overview suggests the focus of the rest of the section concerns the measurement and subsequent analysis of vibration measurements taken using an FFT analyzer. In effect what we will be doing with this FFT analyzer is taking a time signal analysis and applying a calculation to convert the measured signal into a series of peaks on a standardized graph. Each peak represents an amplitude of vibration and a frequency. Using these two parameters it is possible to see in a relatively simple way the magnitude of vibration and to identify the root cause of the highest vibration signatures.



Time waveform

Frequency analysis

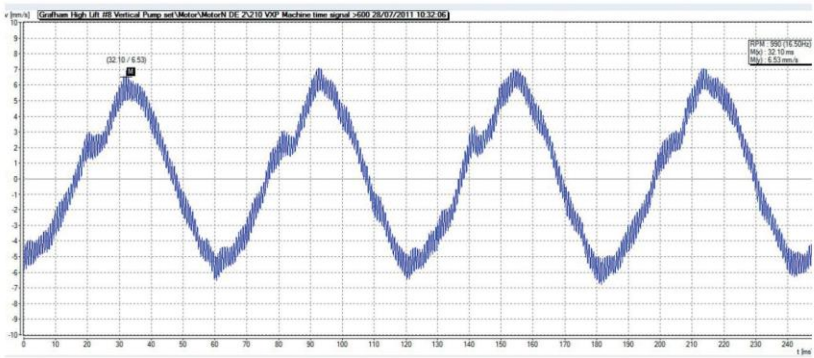


Machine vibration spectrum

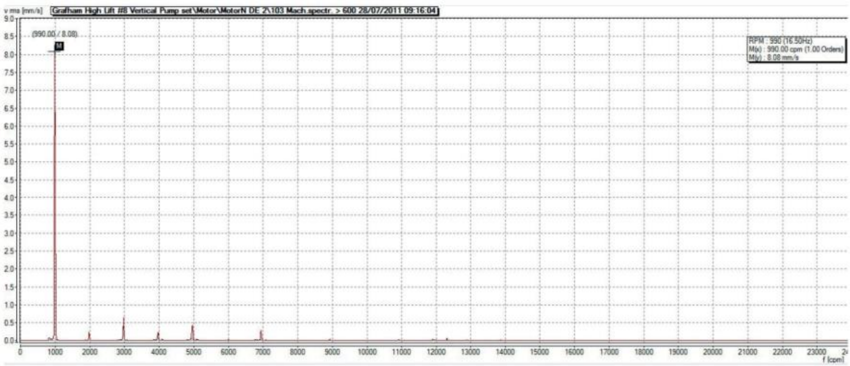
Different factors such as misalignment, unbalance, mechanical looseness can be identified because each factor exhibits its highest vibration amplitude at a different frequency. As an example, misalignment usually exhibits its highest amplitude of vibration at 2 x rotational frequency whereas unbalance exhibits its highest vibration amplitude at 1 x rotational frequency. As we will see later however nothing in the world of vibration analysis is as straightforward as this simple example. However, the principle stands we convert a complex (to understand) time signal analysis into a more understandable FFT vibration plot.

Not only do different factors, such as imbalance or misalignment, exhibit different frequencies of vibration amplitude, so do different components such as bearing or gears. As a general rule (for this initial examination of FFT analysis), machine components have higher frequency vibration characteristics than does the gross mechanical defects such as misalignment, imbalance, looseness or other complete machine operating parameters.

The following two graphs where a time signal for a machine operating in an unbalanced condition is converted to an FFT graph clearly shows the unbalance in an understandable form as a defined frequency – in this case in cycles per minute (cpm) or rotations per minute (rpm) of the machine. An explanation of this important frequency conversion follows.



Time waveform of an unbalanced machine at 1000 rpm.



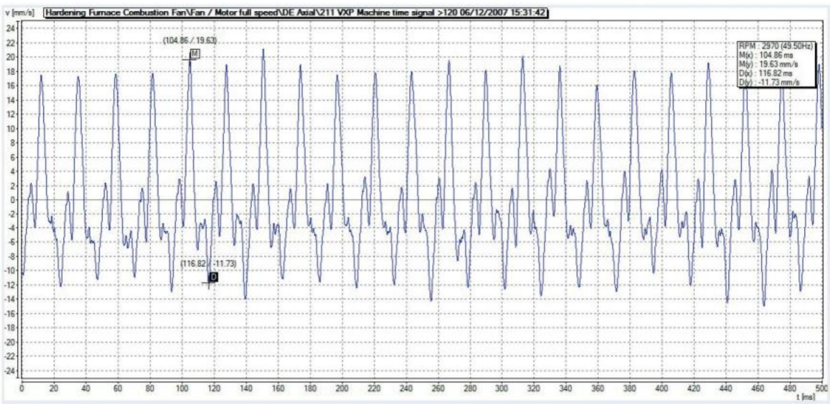
Converted FFT spectrum of the above time waveform.

In the previous time waveform graph the horizontal scale is time in milliseconds and the vertical scale is velocity in mm per second. In the spectrum, the horizontal scale is in cycles per minute (cpm) whilst the vertical scale is again velocity in mm/sec.

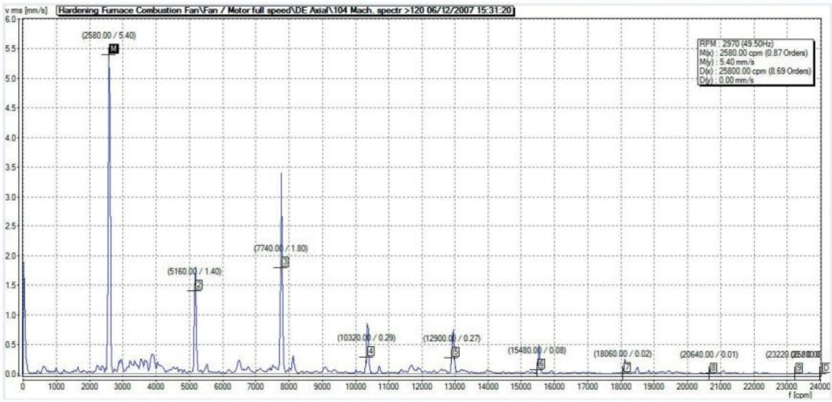
A note about horizontal scaling in FFT graphs will help at this point. There are two conventions that are equally valid in scaling on an FFT graph. Either cycles / rotations per minute (cpm / rpm) or Hertz (Hz) are used. Cpm / rpm is simply a term for the rotating speed of the machine, whilst Hz is the fre- quency of the machine, simply put it is the cpm / rpm divided by 60.

The spectrum clearly shows a large vibration “spike” at 1000 cpm, which is the fundamental frequency of the machine running at 1000 rpm. This “spike” is described as being at 1x the fundamental frequency of the ma- chine, smaller “spikes” are present at 2x, 3x, 4x and 5x the fundamental frequency. If the horizontal scale had been in Hz the “spikes” would have been in the same position but the scale would have shown the first “spike” at 16.6 Hz (1000/60).

The above is a good example of the ability of an FFT conversion to illustrate specific machine problems in a simplified format. Similarly for another com- mon machine fault (misalignment), the conversion to FFT works just as well (as below).



Time waveform of a misaligned machine set at 1000 rpm.



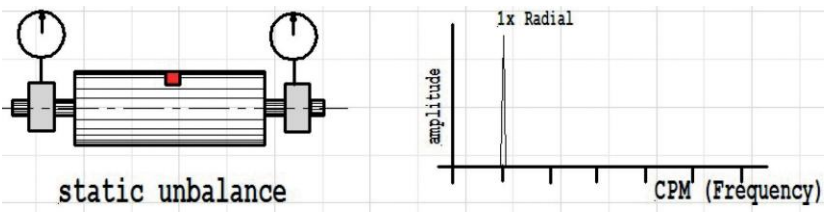
Converted FFT spectrum of the previous time waveform.

It is perfectly possible to produce a basic reference chart for common machine faults which will help the person new to FFT analysis with diagnosis of machine faults, but at this point it is more helpful for future understanding of the subject that we continue with an explanation of the basics.

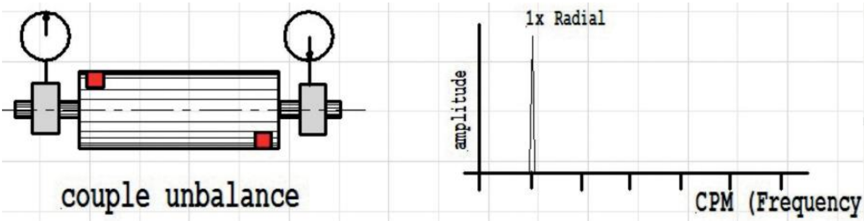
For example why does unbalance exhibit its highest frequency at the fundamental frequency? In fact, the explanation is pretty simple: When an out of balance rotor is spinning, the out of balance force is thrown towards the vibration sensor at its greatest velocity once every revolution of the machine.

Some commonly encountered types of unbalance

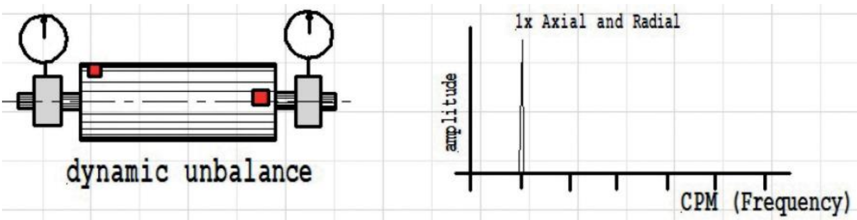
Static unbalance: Also referred to as forced unbalance, this occurs when a heavy spot is located at the mid point between the bearings. This form of unbalance is most common in rotors that have a short length compared to its diameter.



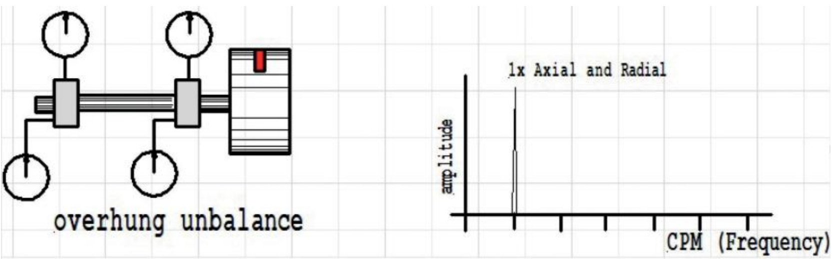
Couple unbalance: Where unbalance forces are 180 degrees out of phase on the same shaft, the 1x frequency spike is always present and dominates the FFT spectrum. Correction requires balance weights to be in at least 2 planes.



Dynamic unbalance: Sometimes referred to as quasi-static unbalance, this is the most commonly encountered form of imbalance. It occurs, when the rotational axis of the shaft and the weight distribution of the rotor do not intersect. This is effectively a combination of static and couple unbalance.



Overhung unbalance: This exhibits radial and axial vibrations – always at 1x frequency, radial signals are the result of influence from shaft bending effects caused by the unbalance load. Axial readings may vary and appear unsteady during the measurement.

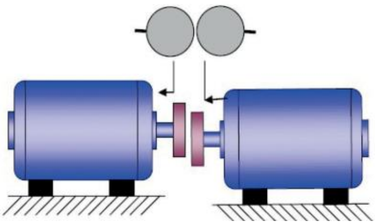


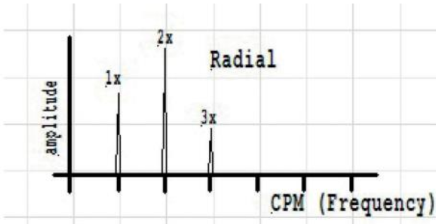
Misalignment

Misalignment alongside unbalance is perhaps one of the most commonly found causes of high vibration in coupled rotating machines such as pumps and other standard machine trains. Unlike unbalance it does not offer a clear frequency spike at one single frequency. Instead misalignment can be identified by having its highest frequency amplitudes at 1x and 2x cpm and with smaller harmonic frequency spikes at 3x cpm up to and including 7x cpm.

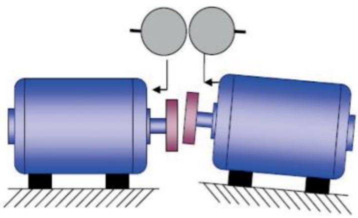
The commonly encountered parameters of misalignment

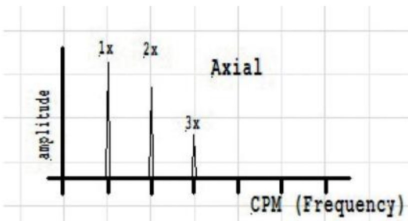
Parallel offset: This exhibits high radial vibration, 180 degrees out of phase with 1x, 2x, 3x and 4x radial vibration prominent with 2x being the dominant vibration amplitude. Vertical and horizontal parallel offset exhibit the same frequency pattern.



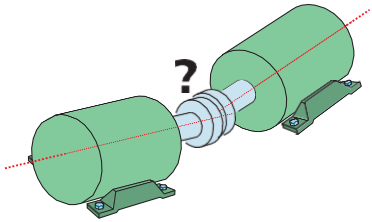


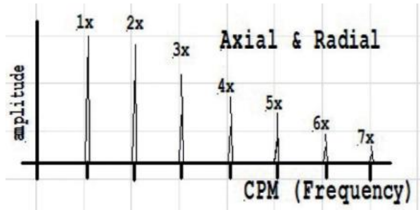
Angular misalignment: This exhibits high axial vibration, 180 degrees phase change across the coupling, 1x, 2x, 3x axial vibration with 1x and 2x dominant (either can be the dominant amplitude). Vertical and horizontal angularity exhibit the same frequency pattern.





General misalignment: This is the most commonly encountered misalignment, it is a combination of parallel and angular misalignment. Axial and radial measurements both show the major frequency components with 1x and 2x the dominant frequencies with harmonics showing up to 7x cpm.





Rolling element bearings

Of all components in a rotating machine probably the most common component associated with vibration analysis is that of the rolling element bearing. The basic construction of the bearing comprising four components; namely an **outer race**, a ball or roller **cage**, **balls** or **rollers** and an i**nner race**, means that its fault diagnosis is considerably more complex than that of many components. Each component has a distinct fault signature on the FFT spectrum. In order to identify the signature faults emanating from the bearing, it is necessary to indulge in a series of mathematical equations to establish the fundamental frequencies of each component as follows.

Inner Race Ball Pass Frequency:
- BPFI $= Z * n / (60 * 2) * (1 + (D_w / D_{pw} *\cos(a))$

Outer Race Ball Pass Frequency:
- BPFO $= Z * n / (60 * 2) * (1 - (D_w / D_{pw} *\cos(a))$.

Ball Spin Frequency:
- BSF $= (D_{pw} * n) / (D_w * 60 * 2) * (1- [D_w / D_{pw} * \cos(a)]^2)$

Fundamental Train Frequency:
- FTF $= n / (60 * 2) * (1 - (D_w / D_{pw} *\cos(a))$

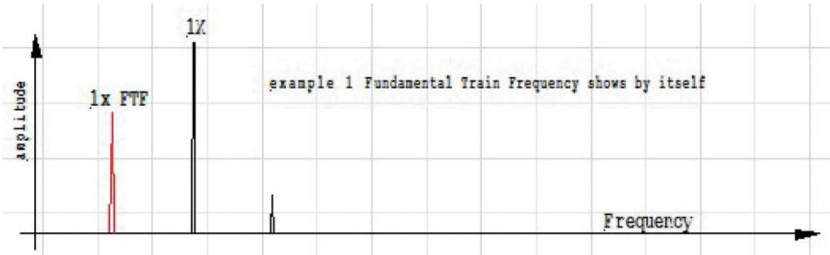
where

a = contact angle
D_w = Rolling element diameter
D_{pw} = Bearing pitch diameter
Z = Number of rolling elements
n = Shaft speed in r.p.m.

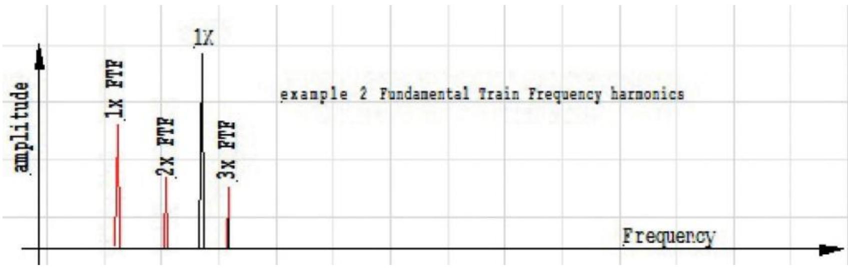
The number of balls and dimensions of the bearing can be obtained from the manufacturers catalogue of the bearing. Additionally, advanced FFT analyzers, such as the VIBXPERT II, have built into the support software the fundamental frequencies of the bearings. This is obtained by simply typing into the software the manufacturers bearing model number. Needless to say, this saves considerable time in setting up and analyzing bearing condi- tion. There is no shortcut, however this information is required in order to properly diagnose bearing condition using FFT spectra.

Cage fault components in velocity spectrum

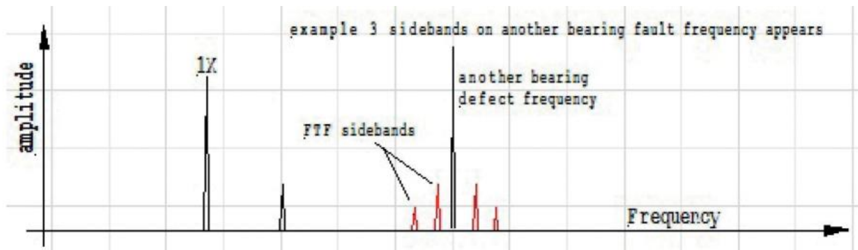
To separate the rolling elements is the chief function of the bearing cage, permitting safe operation at a variety of operating speeds. The cage reduc- es rolling element sliding, contact and wear. Cages facilitate uniform load distribution by the elements in the bearing but carry no load. Cage faults can appear in the velocity spectrum in different forms based upon the bear- ing fault condition.



Fundamental frequency of cage alone



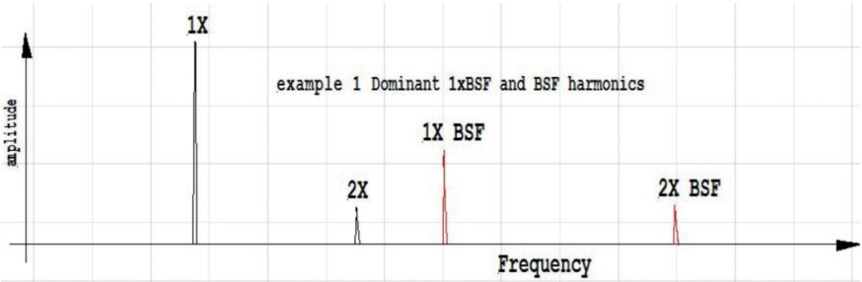
Fundamental train frequency and harmonics



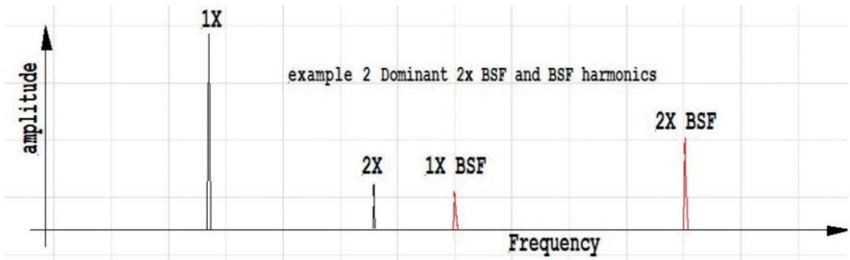
Another bearing fault frequency plus sidebands thereof

Roller / Ball fault components in velocity spectrum

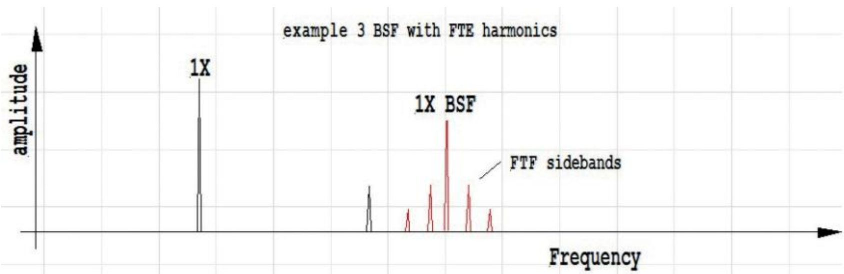
The function of a ball bearing is to connect two machine members that move relative to one another in such a manner that the frictional resistance to motion is minimal. In most applications one of the members is a rotating shaft and the other a fixed housing. Separating these are the balls or rollers which are in effect the load carrying component of the bearing. Defects or damage to these rolling elements are shown in the FFT spectra.



Dominant ball spin frequency and harmonics (2x BSF).



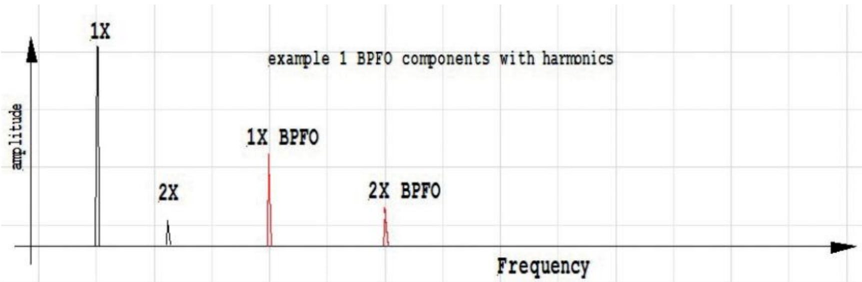
Dominant ball spin harmonics (2x BSF).



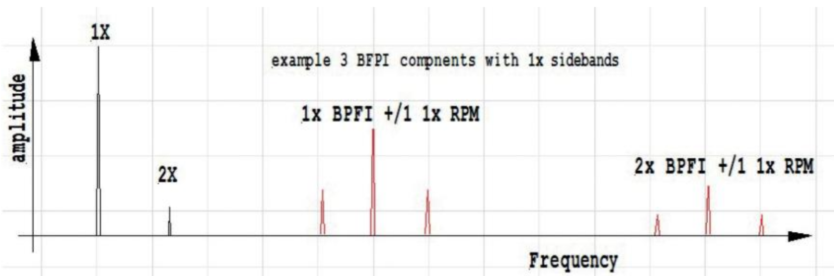
Ball spin frequency with fundamental train frequency sidebands.

Inner and outer race fault components in velocity spectrum

The inner and outer races of bearings are of course two separate bearing components. Each race is a ring with a groove where the balls rest. The groove is usually shaped, so the ball is a slightly loose fit in the groove. Thus, the ball contacts each race at a single point. However, a load on a small point would cause extremely high contact pressure. In practice, the ball deforms (flattens) slightly where it contacts each race, and the race also dents slightly where each ball presses on it. Depending on load, fitting, ingress of wear debris particles and lubrication, the inner or outer races can become permanently damaged. This damage shows up in the FFT spectra adjacent to the fundamental component frequencies calculated earlier.



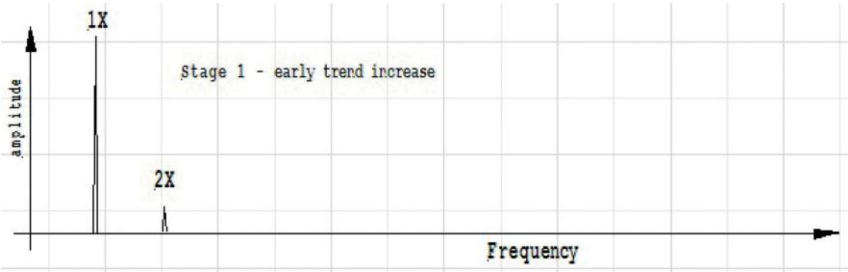
Outer race damage spectrum with harmonics.



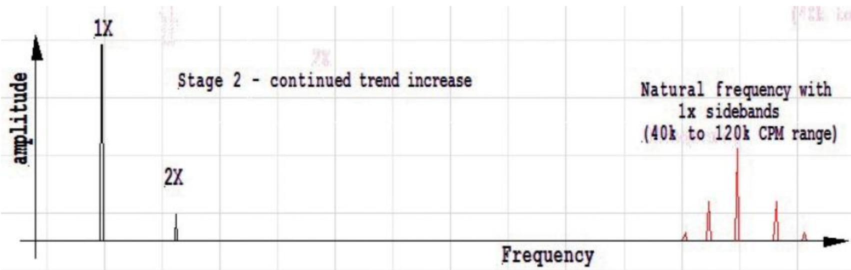
Inner race damage spectrum with sidebands.

Bearing fault progress stages

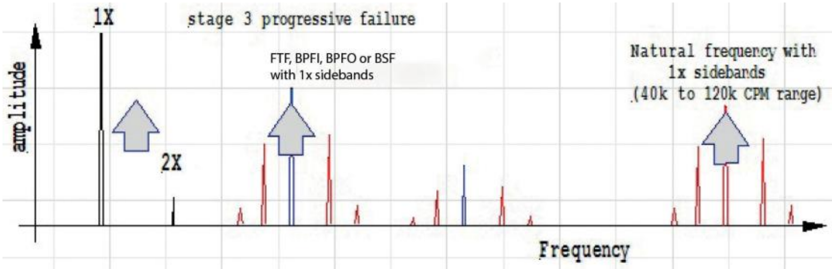
The foregoing examples of bearing component failure are an example of the type of spectra that you can expect to find on a damaged bearing. It is more probable however that when analyzing a bearing there will be progressive damage occurring which progresses over time to a position where intervention is necessary. As discussed earlier, progressive damage is highlighted in a trend graph of the measurements. It is helpful, however, to be able during this trending to identify cause of damage and implement corrective action in order to extend the life of the bearing. At stages in the trend graph it is therefore useful to view the bearing FFT spectra. Four typical failure stages are illustrated in the following examples.



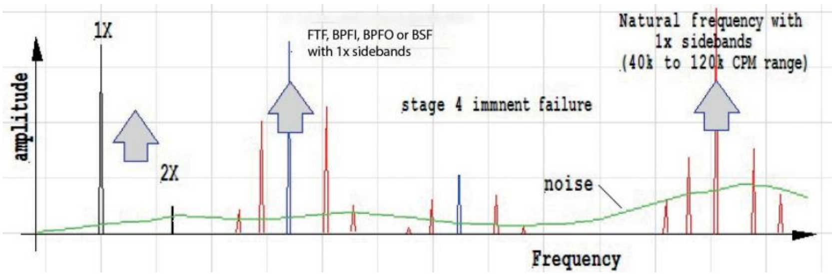
Early trend increase spectrum.



Continued trend increase spectrum: intervention stage.



Quickly increasing trend damage: intervention recommended.



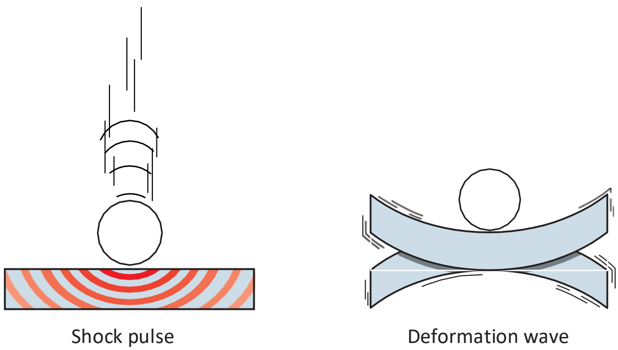
Rapidly increasing trend: failure imminent.

It more or less goes without saying, that you should try and intervene be- fore you arrive at a stage 4 spectra. Of course it is not always possible to shut down production critical equipment. In such circumstances you should rely on the rate of increase and the amplitude of the trend graph as well as the measured spectra. BUT always be aware that catastrophic bearing failure usually does not begin and end with just the bearing failure. Bent shafts, coupling damage and a whole range of additional major component failures could result, in failure to intervene when an obviously damaged bearing has been identified.

Shock pulse evaluation

There is one non-FFT method widely used to assess the operating condi- tion of anti-friction bearings. It is one of the most successful and popular techniques available and deserves mention in this FFT section, if only for the fact that it is a valuable first line of analysis for small bearings, where overall condition is used as the defining criteria for stopping a machine and changing bearings. It is that of **shock pulse evaluation**.

Shock pulses are a special type of vibration which must be clearly distinguished from ordinary machine vibrations. The actual shock pulse is the pressure wave generated at the moment when one metallic object strikes another. The bulk of the impact momentum, however, acts to deform the target object, which then oscillates at its natural frequency. This vibration ultimately dissipates, primarily as heat, due to internal friction (material damping).

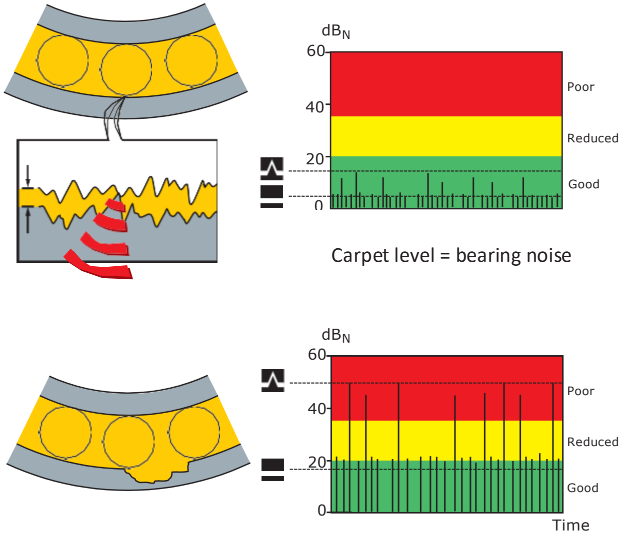


Shock pulses occur during bearing operation when a rolling element passes over an irregularity in the surface of the bearing race. Of course, there is no such thing as a perfectly smooth surface in real life. Even new bearings emit a signal of weak shock pulses in rapid succession. This 'carpet level' rises when the lubrication film between rolling elements and their races becomes depleted.

A defect (pit or crack) on the surface of a rolling element or bearing race produces a strong shock pulse with up to 1000 times the intensity of the carpet level. These irregular peaks (the 'maximum value'), which stand out clearly from the background level, are ideal indicators of bearing damage.

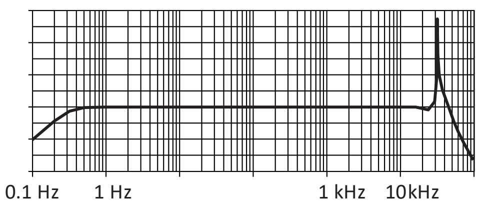
Shock pulses propagate within a much higher frequency range than that of ordinary machine vibration, and their energy content is much weaker. Therefore, the accelerometer used for shock pulse measurement has a resonance frequency (approx. 36 kHz) that lies precisely within this range.

Shock pulse diagrams for good and damaged bearings



Signal peaks above carpet level. Signal strength is shown in decibels (logarithmic scale) for clarity. 60 dB represents a change of 1000x.

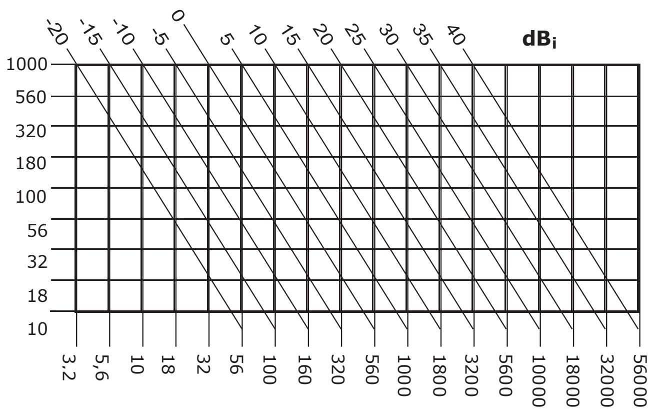
This means, that in this high frequency range of particular interest for bear- ing condition evaluation, the transducer is especially sensitive to the shock pulse signal – even when far more energetic machine vibration occurs at lower frequencies (for example, due to unbalance or shaft misalignment) or from adjacent machines. And since high frequency signals tend to dissipate rapidly, very little interference is encountered from adjacent bearings.



Evaluating bearing condition

Just as with other condition evaluation methods, the shock pulse technique reaches its conclusions via certain defined parameters. These are influenced by factors such as bearing size, rpm, signal damping and lubrication. Shock pulse readings generally should be compared with 'signature' readings, taken when condition is known to be good, or normalized to take these factors into account.

Over the years reliable normalization methods have been developed based upon extensive measurements, to calculate the effect of bearing size and rpm on shock pulse readings of new, perfect bearings. The normalized signal level (dB_n) calculated for an actual bearing allows its condition to be rated directly as 'good', 'reduced' or 'poor'.



Normalization graph plots bearing diameter over machine speed

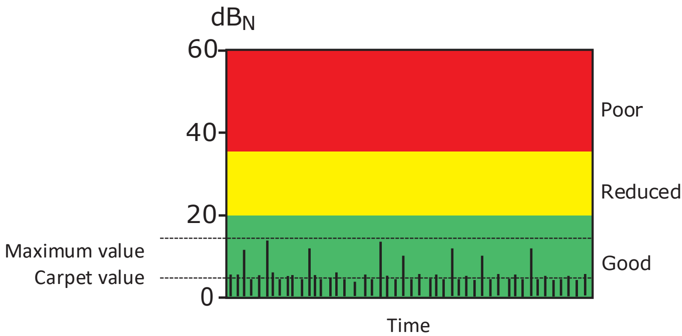
Two normalized parameters are used to determine bearing condition.

- The **carpet value** indicates deteriorating or poor operating condition (e.g. caused by insufficient lubrication, shaft misalignment or improper installation).

Damaged bearing elements, in contrast, generate individual shock pulses of greater intensity.

- The resulting **maximum value** is a direct indication of bearing operat- ing condition.

Specific types of damage can be recognized not only from the absolute sig- nal amplitude, but also from the difference to the carpet level and the pat- tern of pulses. Comparison with typical shock pulse diagrams often shows clearly where the problem lies (e.g. ‘lubricant contamination’).

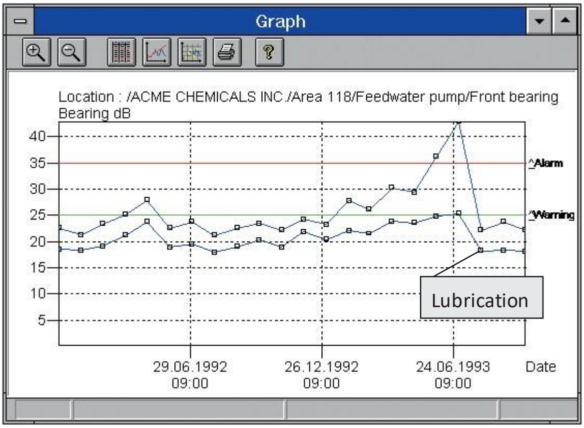


Shock pulse comparison chart

The above diagram shows a typical shock pulse graph where bearing con- dition over time has changed. The upper line of the graph is the normalized overall dB_n measurement of bearing condition and the lower line is the carpet level measurement dB_c. As can be seen, normalized and carpet levels trend much the same in a good-operating-condition bearing. A developing problem is indicated when the level trends begin to separate, for example by an increase in the dB_n (upper) trend. An increase in the dB_c (carpet) level indicates a potential lubricating film breakdown which can often be resolved by lubrication.

The dB_c trend provides important information regarding lubrication, mount- ing and loading condition of the bearing. It is related directly to the fluid film thickness at the rolling component interface.

The dB_n trend gives information on irregularities in the bearing surface which give rise to single shock pulses at random intervals. These high val- ues give a good indication of damage already done to the bearing and the overall condition.



Typical shock pulse trend graph

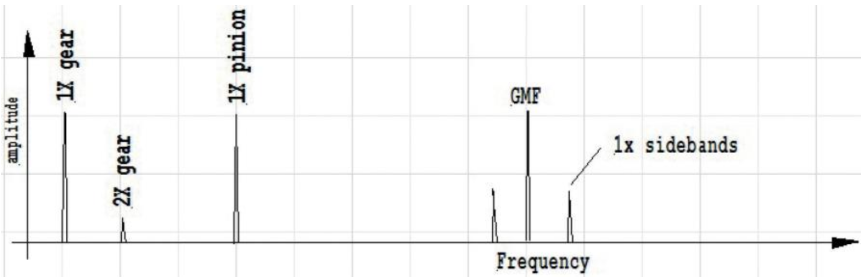
If after lubrication, the dB_c falls below alarm levels and stays there, it is a good indication that intervention in the form of lubrication has brought the bearing operation back within acceptable operating limits. An increase in dB_n over dB_c trends indicates a more fundamental problem unlikely to be improved by lubrication.

Gear and gearbox faults

Gearboxes and more specifically gear meshing and wear problems occupy a significant section in the analysis of rotating machinery. Complex gear- box design, planetary gear systems and the fault analysis thereof can be a daunting task for the condition monitoring engineer no matter what level of training or experience. It is not the plan of this publication to delve too deeply into the analysis of gears and the complex fault analysis that can be required. Instead we will look at a typical spectrum of a gear system in a good condition and at three common and relatively easy to detect prob- lems:

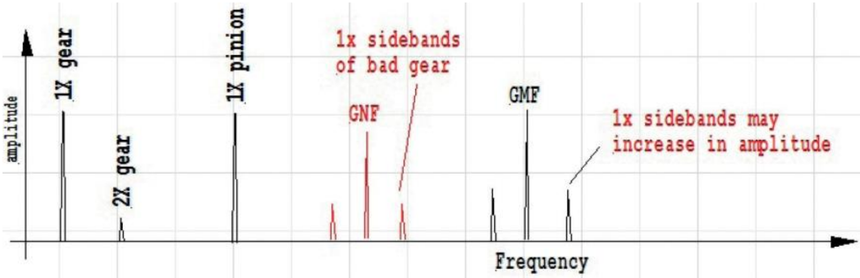
- Excessive gear tooth wear
- Excessive loading on gear teeth
- Mechanical misalignment between gears

The following four spectra graphs illustrate the basic FFT spectra you would expect to see given these problems.



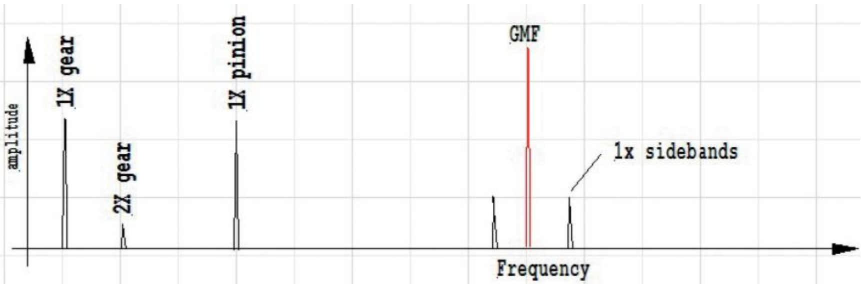
Expected spectrum of a gear in a good condition

This spectrum is showing 1x and 2x frequencies of the gear plus 1x the frequency of the pinion and the Gear Mesh frequency (GMF) with 1x side- bands. There are no gear natural frequencies showing and all peaks are of low amplitude.



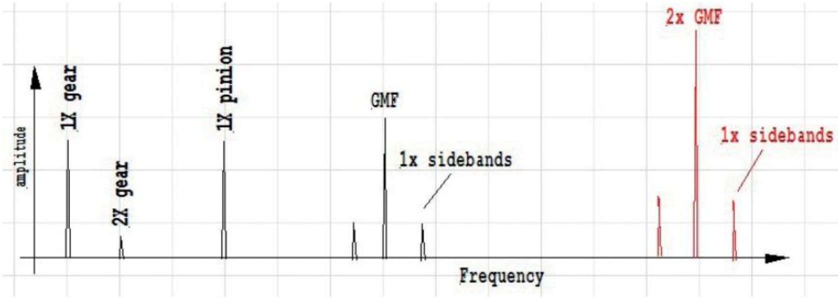
Spectrum of a gear assembly with worn teeth

The above spectrum shows tooth wear which excites the gear natural frequency with 1x sidebands of the bad gear. Gear Natural Frequency (GNF) sidebands may also increase in amplitude. Other frequency components are similar to what you would see in a good gear spectrum.



Spectrum of a gear assembly with excessive loading

High loading of gear teeth will show an increase in the Gear Mesh Frequen- cy amplitude (GMF) and very little change in sideband amplitude.

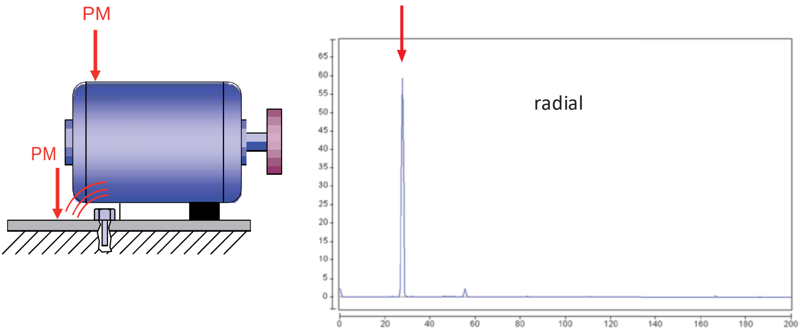


Spectrum of a gear assembly with excessive misalignment

In this example, gear misalignment excite 2nd harmonic of the Gear Mesh Frequency (GMF) and even 3rd harmonic in some cases. The frequencies 1x and 2x are lower compared to 2x GMF.

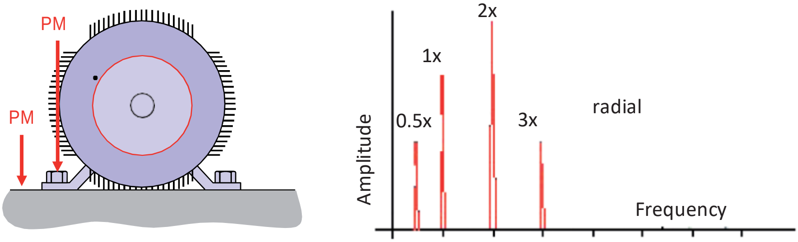
Looseness fault analysis

Looseness of components, poor structural integrity, cracked or broken holding down bolts and poor foundation condition can cause untold problems for the engineer analyzing FFT data. It is often difficult to establish from a spectra since it often manifests itself as another common fault such as unbalance or even misalignment. The users EYES are often the key to unlocking looseness FFT diagnosis. Always look for any obvious problems such as a broken foot. This makes the diagnosis much easier. Below are three typical looseness spectra.



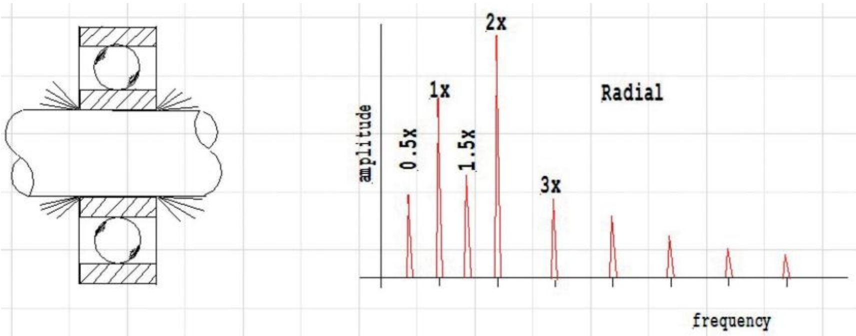
Structural looseness

Structural looseness is caused by a weakness of machine components, typically machine soft foot, baseplate or foundation distortion, or even poor baseplate design creating flexing of the baseplate. This manifests itself as a strong 1x component measured in the radial direction and is therefore frequently misdiagnosed as a form of static unbalance.



Mechanical looseness caused by loose holding down bolts

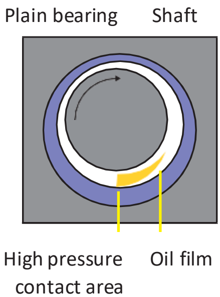
This type of looseness is often easily identified by observation, loose bolts or cracked or broken machine feet, identified by 0.5x 1x, 2x, and 3x cpm, measured in the radial direction.

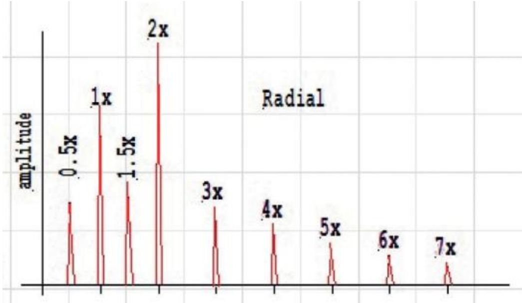


Looseness caused by poor fit components

Component fit looseness can be caused by poor fitting of bearings or excessive clearance of fan impellers on the shaft. There may be a phase change from one measurement to another, generating numerous harmon- ics 1x, 2x, 3x; and 0.5x, 1.5x, 2.5x may also be present in the spectra.

Journal bearing wear will also show up as a form of mechanical looseness.



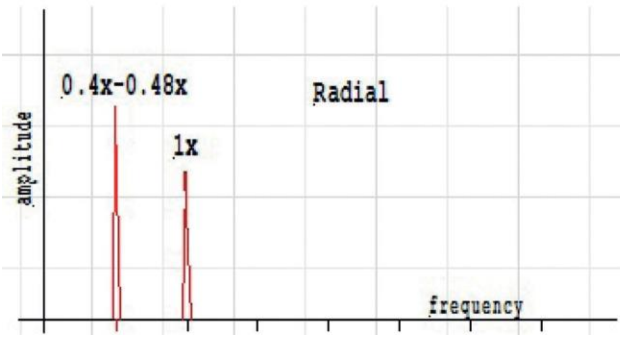


Journal bearing wear

Vibration readings to establish journal bearing wear can be taken from the shaft or the bearing housing. The FFT spectrum generated is very similar to component looseness with 1x component plus many harmonics in the radial vibration spectrum. In very severe cases, peaks may also show at 0.5x, 1.5x etc. The most definitive way to establish actual journal wear is via oil analysis and wear particle debris analysis.

Journal bearing wear is frequently caused by **oil whirl** which occurs when a lubrication wedge cannot form in the high contact areas of the assembly but instead whirls around the bearing. This leads to direct metal to metal contact between shaft and bearing which quickly wears out the bearing.

Sub-synchronous components between 0.4x and 0.48x appears in the spectrum with an unstable amplitude.



Journal bearing oil whirl

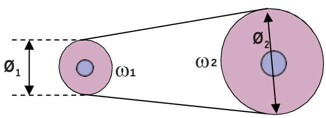
Belts and pulleys

During operation a flexible belt experiences three types of tension as it rotates around a pulley.

- Working tension (tight side – slack side)
- Bending tension
- Centrifugal tension

Belts are designed to withstand these working operation states, provided that pre-selection of the belt meets the operating criteria. The design life of the belt will be met and usually exceeded provided that no other forces other than the above act upon the belt during its operating life.

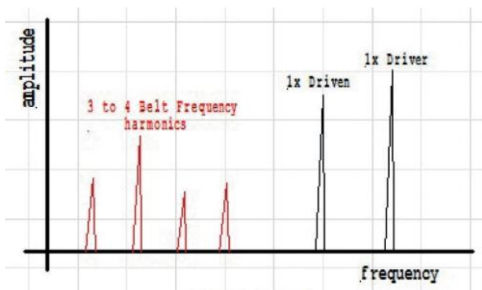
Forces, such as misalignment and loose or over tight belt tension, are killers of flexible belt drives. Poor pulley alignment can reduce useful operating life by as much 80%. Furthermore, not paying adequate attention to basic installation requirements leads not only to belt wear but may also damage pulleys, bearings and seals. FFT analysis can identify many of the “belt drive killers”. The following example provide an insight into what you should look for when analyzing belt drive spectra.



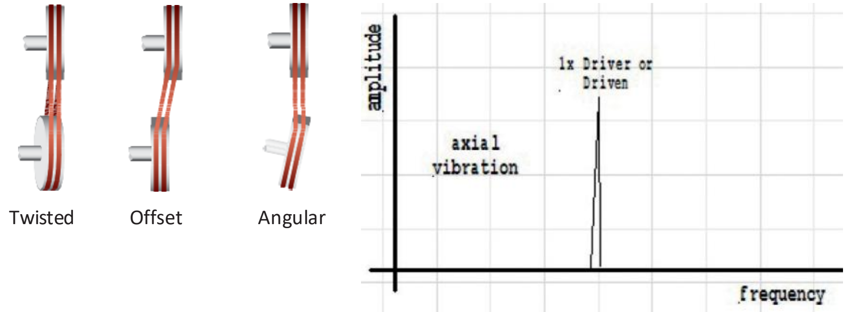
$\omega_1 * \Phi_1 = \omega_2 * \Phi_2$

$f_B = \pi * \omega_1 * \Phi_1 / l_B$, where

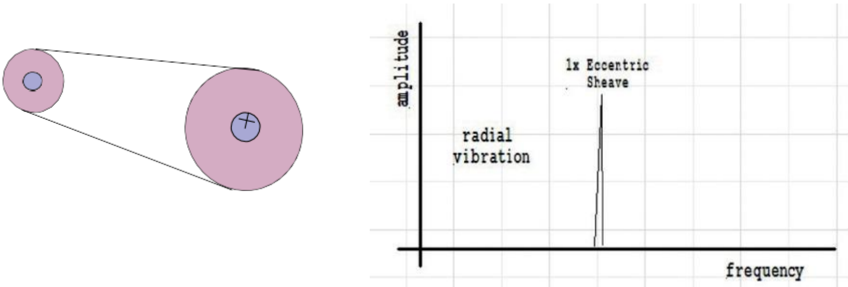
f_B: Belt frequency
l_B: Belt length



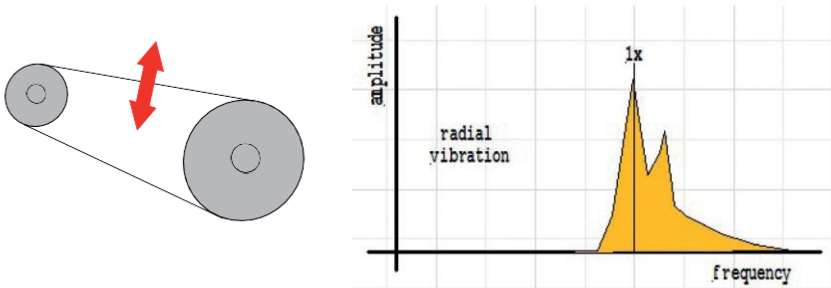
Worn, loose or mismatched belts



Misaligned belts



Eccentric sheave



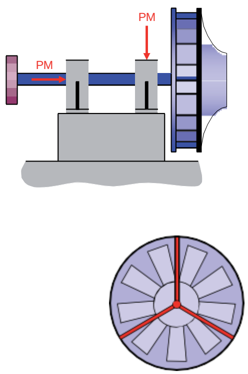
Belt resonance

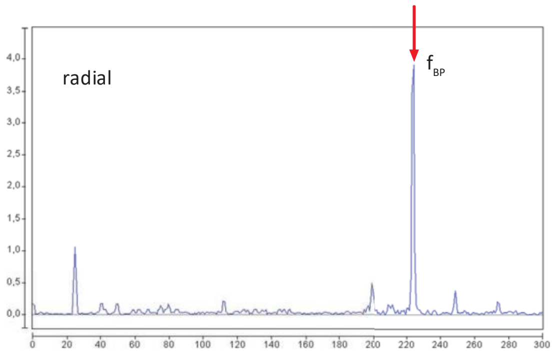
Belt resonance occurs when the belt natural frequency approaches or coincides with the drive or driven frequency. To correct this, change the belt tension to change the belt natural frequency.

Fans

Most fans we encounter in standard process applications are either ax- ial-flow propeller type or centrifugal. Such fans are prone to an uneven build up of debris on the blades, particularly when handling particle laden air in applications such as car manufacturing paint shops. Particle build up leads to fan unbalance. FFT analysis will quickly show up such operational problems. Fans are also frequently belt driven which also gives rise to typical belt drive FFT characteristics as described in the previous pages.

If however during the course of analysis, unbalance and misalignment can be eliminated from elevated FFT readings on a fan, then the likelihood is that there is some form of mechanical damage that the fan has suffered. This could vary from an extreme problem such as a missing blade to a cracked or chipped blade tip. FFT investigation of this problem requires that the analyst knows the blade pass frequency (f_{BP}) of the fan. It is not a complicated calculation, it is simply the number of blades multiplied by the rpm of the fan itself.





$$f_{BP} = B_n * N$$

B_n = Number of blades or vanes N
= Rotor speed in rpm

Example calculation of a fan with 9 blades and 3 support struts on the intake housing:

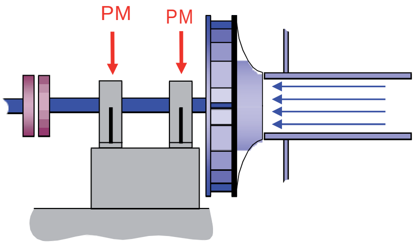
3 struts on intake	$x = 3$
9 blades	$B_n = 9$
Rotor speed	$N = 600$ rpm

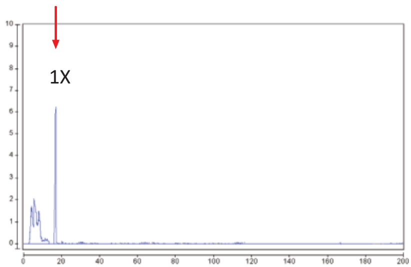
$$f_{BP} * x = N * B_n * x = 600 * 9 * 3$$

Characteristic frequency = 16,200 cpm

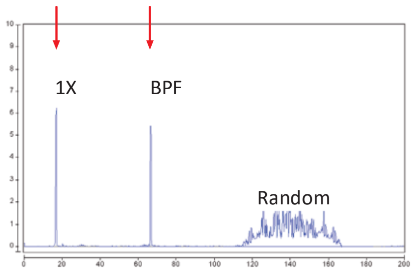
Aerodynamic and hydraulic forces

This is a problem that can affect both fan and pumps assemblies and is associated with fluid and air movement through the structure. The problems can be classified as either **cavitation** or **turbulence**. As can be seen in the following examples, both offer distinct FFT spectra.





Turbulence FFT spectrum



Cavitation FFT spectrum

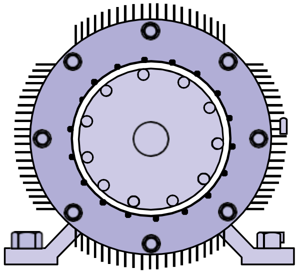
Electric motors

Probably the most common equipment in any process plant is the electric motor. Almost everything is driven by them, and therefore, as a distinct item of equipment, we should consider the electromechanical faults that can arise from their operation. Of course, since they are invariably coupled to another component (fan, pump, etc.), they are subject to the same basic component faults such as unbalance, misalignment, looseness, bearing problems etc,(covered earlier in this section). However, electric motors are rather complex and suffer from electromechanical problems which are dis- tinctly unique to themselves.

Essentially motors consist of a stator (laminated steel sections wound by copper wire or bars) and a rotor (again a laminated steel section wound by copper wire or bars). The rotor is supported on bearings and is separated from the stator by an air gap of a given dimension. Two main types of mo- tors are synchronous and induction motors. They differ in that the synchro- nous motor has a permanently magnetized rotor which is rotated by the stator “dragging” it round by magnetic attraction. The induction motor is different in that the rotor is not a permanent magnet but an electromagnet. This is determined by the design of the rotor with rotor bars embedded into the laminations. The rotor bars are connected to each other at each end with a continuous copper ring. The induction motor works by magnetic repulsion rather than attraction as with the synchronous motor. Both how- ever experience similar electromagnetic faults in operation.

If you suspect a motor has electromagnetic problems a first useful step is to disconnect it from the driven component and carry out an FFT analysis with the motor running alone. Electric motor problems can be roughly classified as below.

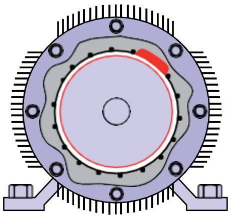
- Stator eccentricity
- Rotor eccentricity
- Rotor problems
- Loose connections

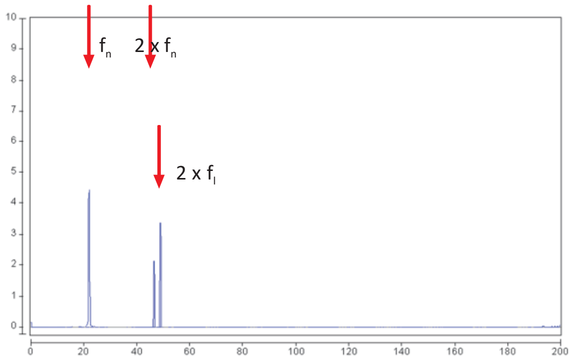


Stator eccentricity

Caused by loose iron (laminations), shorted stator laminations, and soft foot.

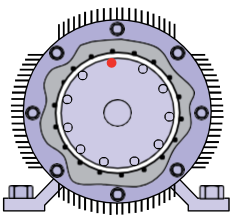
High 1x f_n and 2x f_n signals; 2x f_l (twice line frequency) without sidebands. Radial measurements predominant. High resolution settings should be used for measurement.

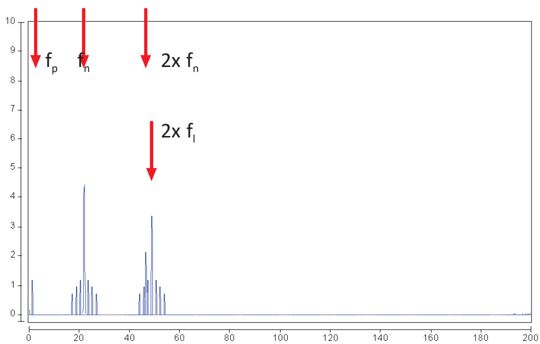




Rotor eccentricity

Caused by rotor offset, misalignment and poor base; f_p, 1x, 2x and 2f_l signals; 1x and 2f_l with sidebands at f_p (pole pass frequency). Radial mea- surements predominant. High resolution settings should be used for mea-
surement.





Key parameters

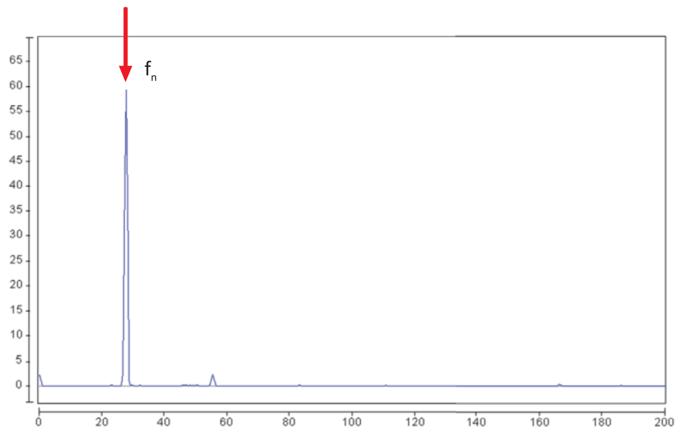
Twice line frequency vibration:	$2{*}f_l$
Bar meshing frequency	$f_{bar} = f_n * n_{bar}$
Synchronous frequency:	$f_{syn} = 2{*}f_l / p$,
Slip frequency:	$f_{slip} = f_{syn} - f_n$
Pole pass frequency:	$f_p = p * f_{slip}$

where

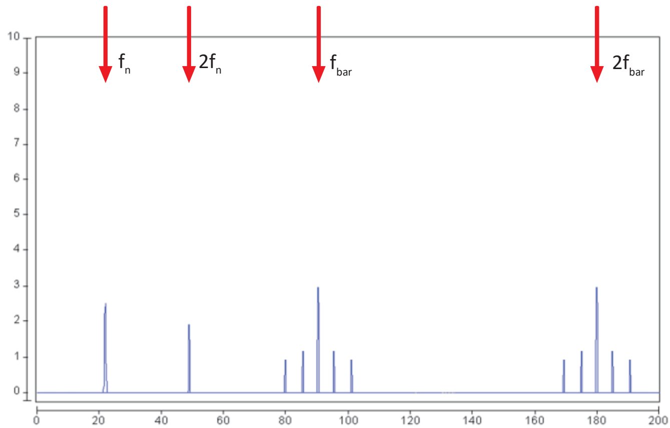
f_n: rotational frequency
n_{bar} : number of rotor bars
p : number of poles

Other electromechanical faults

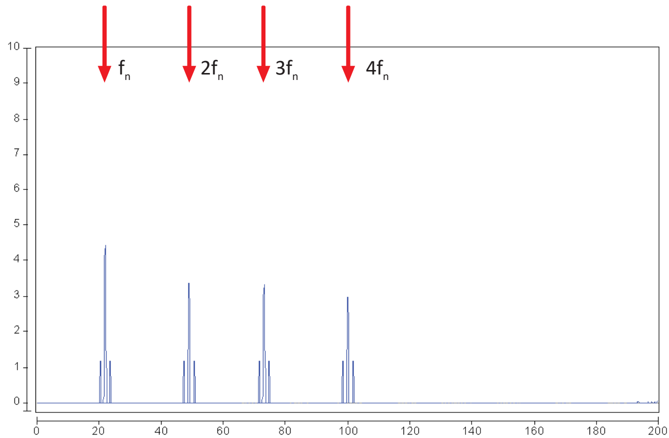
Uneven heating of the rotor due to unbalanced bar currents will cause the rotor to warp (bow). This causes unbalance with the characteristic FFT signature. It can be identified because the symptoms disappear when the motor is cold.



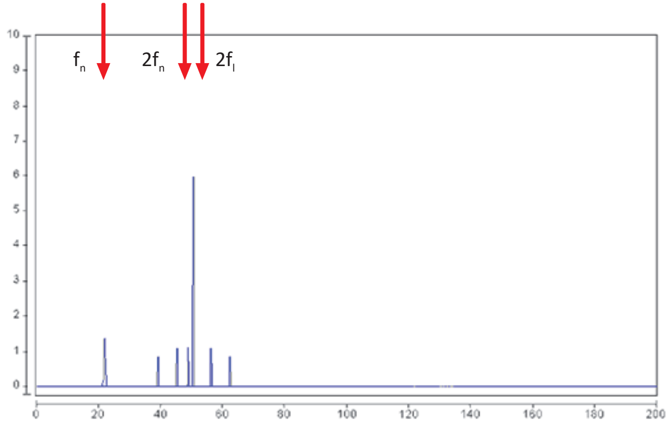
Rotor bow



Loose rotor bars: f_{bar} and $2f_{bar}$ with $2f_l$ sidebands; $2f_{bar}$ can be higher; 1x and 2x can be present



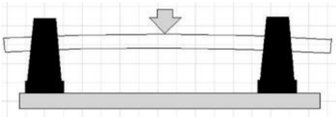
Cracked rotor bars: 1x and harmonics with sidebands at f_p; high resolution spectrum required; possible beating signal

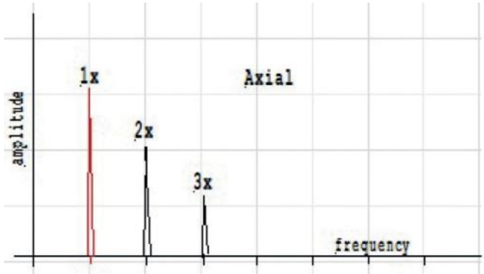


Loose connections: $2f_l$ excessive signal; electrical phase problems; correction must be immediate

Bent rotor or shaft

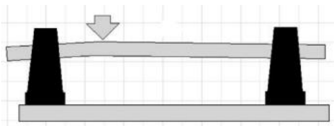
Bent shaft at center

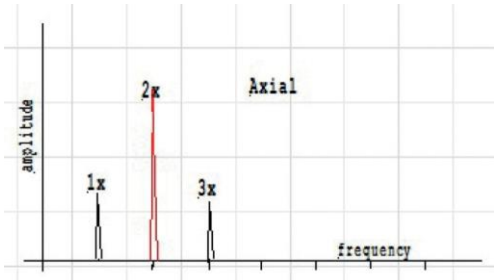




Bent shafts generates high radial and axial loads. Axial vibration shows up at 1x, 2x and 3x components. 1x is dominant if the bend is near the shaft center; and a 180 degree phase shift in the axial direction. Phase measure- ments are essential in this diagnosis.

Bent shaft at one end





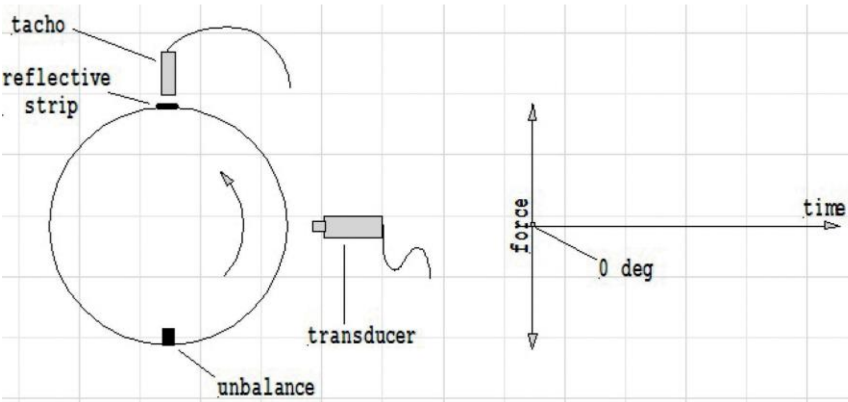
Bent shafts generates high radial and axial loads. Axial vibration shows up at 1x, 2x and 3x components. 2x is dominant if the bend is near the shaft center; and 180 degree phase shift in the axial direction. Phase measure- ments are essential in this diagnosis.

Phase

Phase measurements are a very useful tool for diagnosis of a number of common rotating machine conditions, such as misalignment, unbalance or bent shafts. It is therefore necessary to have an understanding of the measurement technique and what in fact you are measuring. Phase is a measure of the time difference between two sine waves.

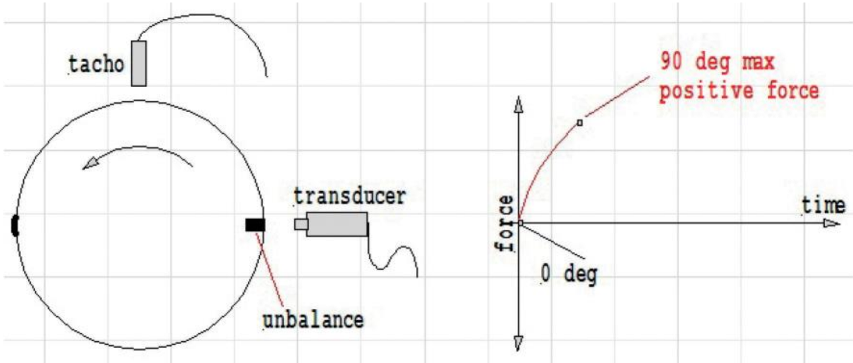
For clarity of explanation, we use in the following examples a vibration transducer to sense the imbalance force and a light sensitive tachometer (“tacho”) and a reflective strip attached to the shaft to sense shaft position.

The phase angle is the angle in degrees that the shaft travels from the start of data collection to the position when the vibration transducer measures the maximum positive force of imbalance.



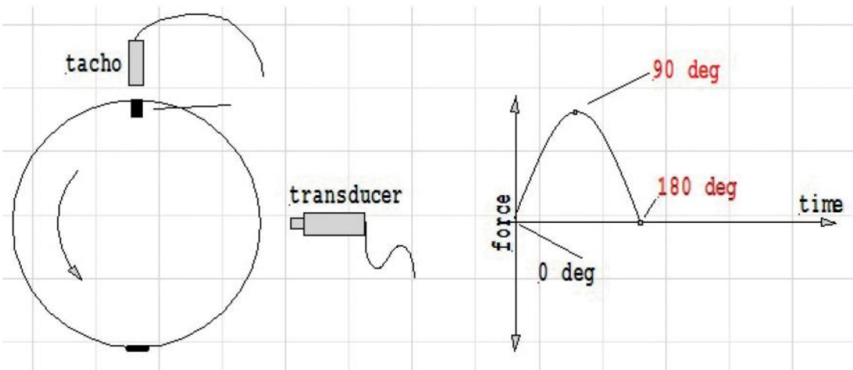
Shaft at start

The tacho senses the reflective strip and starts the data collection. At this point phase = 0.



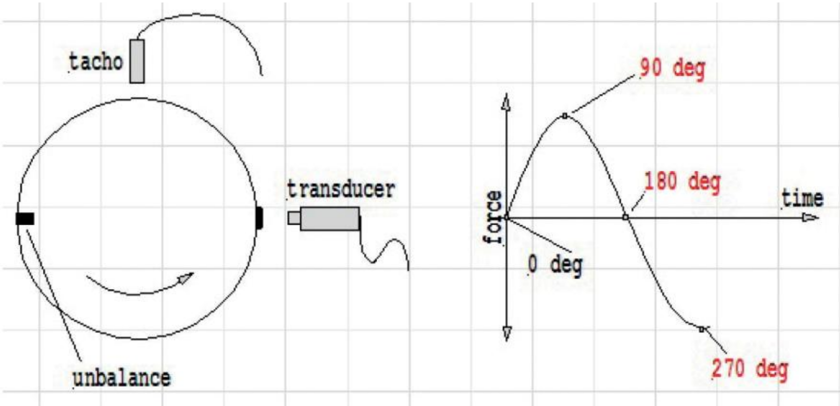
Shat rotated through 90 degrees

The imbalance force has rotated through 90 degrees. At this point the imbalance force produces the highest positive reading at the transducer. As the imbalance is traveling towards the transducer, its force is considered to be in a positive direction.



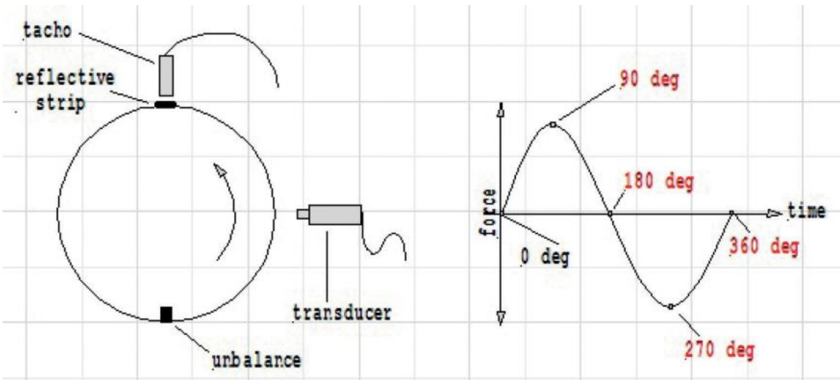
Imbalance travels a further 90 degrees

The force experienced by the transducer is zero.



Imbalance travels a further 90 degrees

The imbalance is now opposite the transducer. At this point the force produced is at its highest negative reading from the transducer. Force is considered to be in the negative direction.



Imbalance travels a further 90 degrees

The imbalance force has completed its 360 rotation and the force experienced by the transducer is again zero.

In the preceding examples the mounting angle between the transducer and tacho for simplicity is shown as 90 degrees. This is not an absolute neces- sity. They can be mounted in the same plane or indeed 180 degrees apart. The key is the use of the tacho and transducer together to initialize and measure the phase shift of the machine. The examples also use a simple static unbalance to explain the principle of phase measurement.

Phase is a key component of FFT diagnosis. Without carrying out phase measurements, it is often impossible to accurately distinguish between faults such as imbalance, misalignment, bent shafts or other low frequency problems, which manifest themselves in 1x, 2x or 3x of the machine fun- damental frequency.

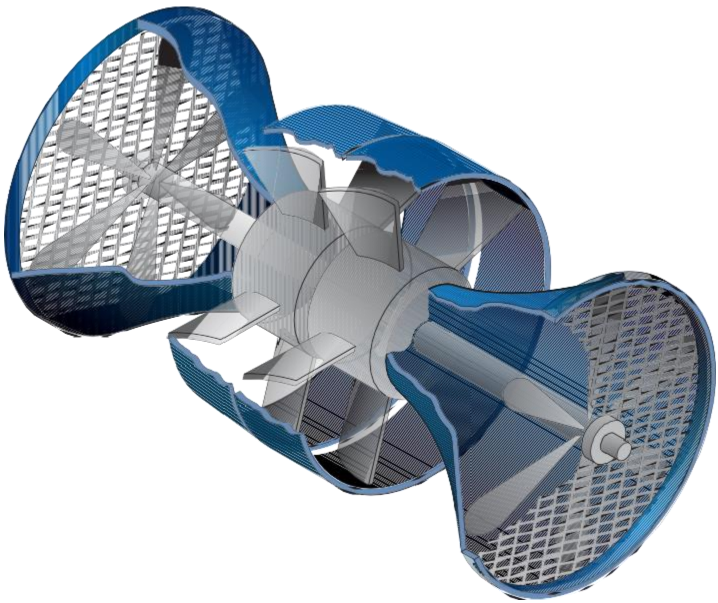
PRUFTECHNIK data collectors and vibration analyzers

The vibration analysis functions and methods described in the previous pag- es can all be performed by the PRUFTECHNIK **VIBXPERT® II** system. Essen- tially VIBXPERT II is a modular system which can be configured to perform a number of vibration related tasks including the following:

- Route-based data collection
- Dual channel FFT analysis
- Time waveform analysis
- Dynamic Balancing

Section 3

Dynamic balancing



Balancing standards

In accordance with ISO standard 1940 – 1973 (E)

"Balancing is the process of attempting to improve the mass distribution of a body so that it rotates in its bearings without unbalanced centrifuge forces".

Our focus in this handbook is to look at balancing standards for simple rotating machines with rigid rotors. And how, by using calculation and portable balancing systems, such as the PRUFTECHNIK VIBXPERT II Balancer, these balancing standards can be achieved.

As with shaft alignment, balancing of rotating machines will contribute significantly to improving machine reliability and hence improve production profitability.

Unlike with shaft alignment, where if necessary, it is possible to align a machine set to a zero alignment condition, it is not possible to balance machines to a zero unbalance state. Even after balancing, machines will continue to have some residual imbalance. Using modern systems it is possible to reduce machine unbalance to very low levels. It is however not economic to pursue very low unbalance levels on most standard machines. To what extent unbalance should be reduced, and where the economic compromise between pursuing lower unbalance levels and accepting what has been achieved, is a subjective issue. ISO standards can however be used as a good guide to acceptable unbalance conditions.

Standards use machine operating speed and rotor mass to establish guide levels which can be applied. Balance quality is classified in a range from G 0.4 to G 4000. For most standard rotating machines, such as fans, pumps and motors, the range we should be concerned with is from G 6.3 to G1. Depending on individual plant operation and commissioning standards, some plants will specify a balancing standard within this range. The ISO standard does not intend that balance standards are used to determine acceptance test for specific rotor types. They are more designed as in indi- cation of unbalance acceptability based on experience and historical evalu- ation of machine operation.

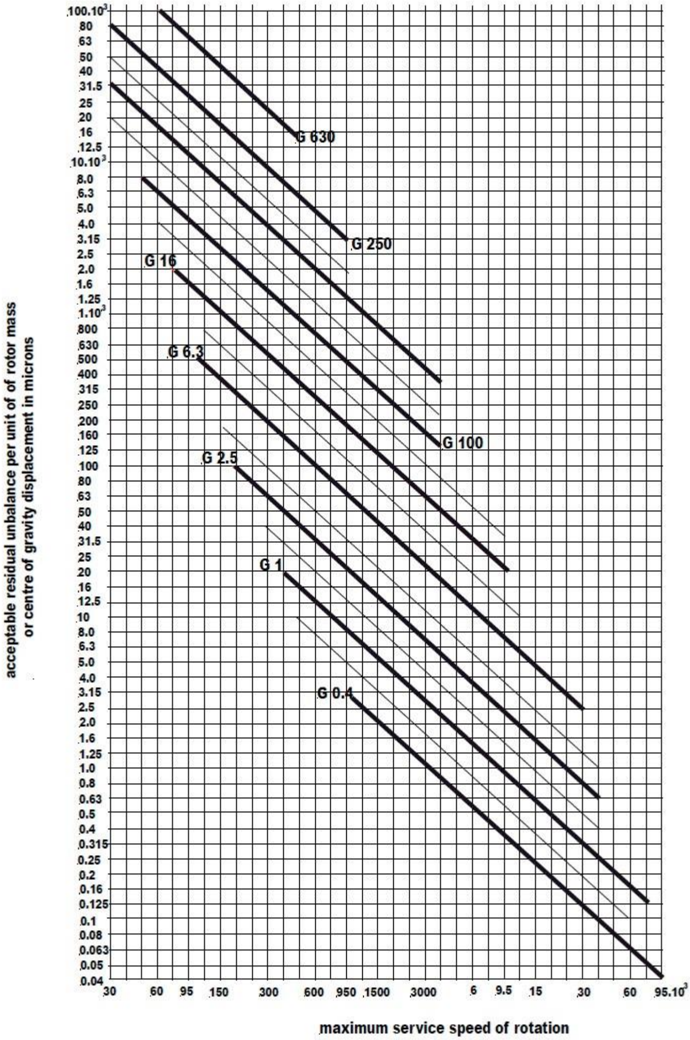
In general, the larger the rotor mass the greater the permissible unbalance. Permissible unbalance **U** is related to rotor mass **m** to give the specific permissible unbalance of the rotor as per the ´given formula.

$$e = U/m$$

General classification for standard machines according to ISO 1940

Balance quality grade G	Examples of balancing bodies or machines
630	Crank gear rigidly assembled four-stroke engines and elastically mounted ships diesel engines
250	Crank gear rigidly mounted, high-speed 4-cylinder diesel engines
100	Crank gear rigidly mounted, high-speed diesel motors with six and more cylinders
40	Automobile wheels, rims, wheel sets, drive shaft crank gear elastically mounted, high-speed four-stroke engines with six and more cylinders
16	Crank gear components of automobile, truck and locomotive engines, crank gear of six and more cylinder engines with special requirements´´
6.3	Fans, flywheels, centrifugal pumps, machine construction and machine tool construction parts
2.5	Impellers of jet power plants, gas and steam turbines, turbo blowers and generators
1	Tape recorder and phono drives, grinding machine drives
0.4	Precision grinding machine anchorages, shafts and disks, gyroscopes

Practical experience shows that for rotors of the same type the permissible specific unbalance varies as the speed of the rotor changes. This produces a chart of permissible unbalance against rotor speed:



Source: ISO 1940-1973 (E)

Using the balance grades as shown in the guide table an estimate of the acceptability of the unbalanced condition of a rotor can be established.

ISO 1940-1973 (E) suggests a variation as follows can be used to when measuring and assessing the balance quality of a rotor compared to the standard table:

Balance quality grades	Permissible deviation
G 2.5 - G16	± 15%
G 1	± 30%
G 0.4	± 50%

The graph on the previous page is a simplified version of the ISO standard graph. In bold are the most commonly used balance grades for standard machine types. For standard motors, pumps and fans, the most commonly used balance grades are in the range of 2.5 and 6.3. In exceptional cases grade 1.0 can be used. This however is an exacting standard for such applications.

Before embarking on corrective action for unbalance, it is essential to eliminate any extraneous causes of vibration such as looseness, rubs or misalignment. Analysis as described in the previous section will help establish the primary source of vibration. This should be attended to first.

Correction methods

By far the major method of correcting rotor unbalance is by use of a specifically designed balancing machine or portable balancing system, such as the PRUFTECHNIK VIBXPERT II Balancer.

The use of static balancing machines and the types available is not a subject covered in this handbook. Here we are focused on in-situ balancing as addressed by most portable systems. It is useful however to look at a method of determining unbalance in-situ without the aid of a dedicated balancing system. This will help to establish basic balancing requirements and will most likely underline the usefulness of investing in a balancing system that eliminates the rather complex calculations that are necessary.

In-situ balancing should be performed under normal operating conditions, and at normal speed. Should this be precluded by unbalance having be- come excessive already, so that normal operation would be too dangerous, pre-balancing must first be performed at reduced speed to lower vibration severity, followed by final balancing at normal speed.

The locations on the rotor at which trial weights and balance weights are fixed in the balancing planes must be freely accessible. Dismantling of major components in between trial runs may alter external influences in such a manner that the measurements of the individual runs are no longer comparable.

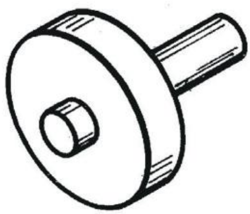
An appropriate measuring point, identical with all runs, should be identified at the location where the vibration measurement instrument indicates the highest vibration value.

With all procedures, vibration severity can be measured in the form of vibration severity $\mathbf{v}_{eff}$, displacement amplitude **s**, or acceleration amplitude, **â**. For the purpose of simplicity, the following discussion is based on v_{eff} only, further abbreviated to v.

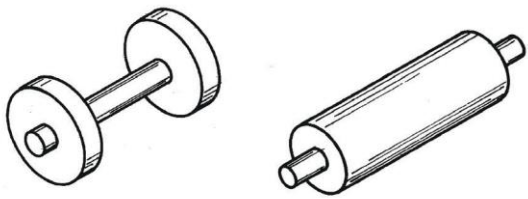
Just as with car wheels, balancing is performed in the main by adding balance weights, less frequently by removing weight. Hence it is a matter of determining the location where the balance weight must be added, and how heavy it must be.

There are several procedures for determining location and amount of the balance weight. But depending on the geometry of the rotor to be balanced, there are basically two methods for correcting rotor unbalance which we shall consider here.

Single-plane balancing for mechanical assemblies having, essentially, one rotor whose thickness is smaller than its radius, e.g. the majority of fans, belt pulleys or half-couplings. With this method, the mass center plane is called **balancing plane**.



Two-plane balancing is designed accordingly for use with machines hav- ing two or more rotors, or having one rotor whose width exceeds its radius. With this method, the two **balancing planes** are the left and the right side, or the front and the rear side, respectively.



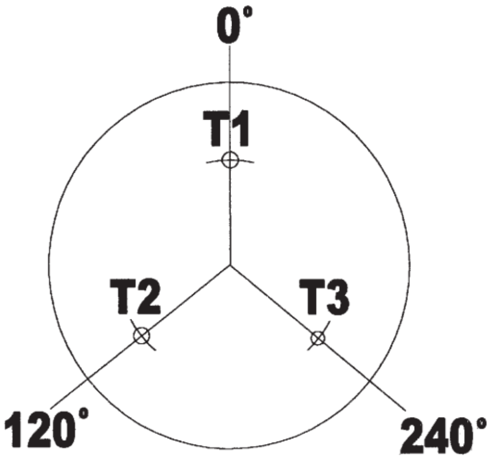
Single-plane balancing

To be able to define the position of the balance weight, the rotor is graduated anticlockwise in degrees, from 0º to 360º. The graduation can be applied to the front face or to the side of the rotor, depending on where the balance weight is to be added later on. In the event of a cover permitting access to the rotor in one particular place only, a 0º mark is applied first, and from this mark the other angles can be measured by conversion of radius and circumference.

There are several procedures again for both single-plane and two-plane balancing. Set out below are two procedures for single-plane balancing. For ease of understanding each one is explained on the basis of an example.

Single-plane three-point balancing
The first procedure is the **single-plane three-point**. The only accessories required are a trial weight and a pair of compasses.

As can be implied from the name, three test points, T1, T2, T3, for attachment of a trial weight are marked out on the rotor, at 0º, 120º and 240º.



A convenient size of the trial weight can be determined by means of the following formula:

$$MT = G * s / r$$

MT: Trial weight
G: Weight of the vibrating parts
s: Displacement amplitude of vibration
r: Distance between the trial weight and the rotational axis

It is of course sufficient for the trial weight to be only approximately this size. 1.5 times or half the calculated weight also is permissible.

As a first step, a trial run is made without a trial weight, to record the vibra- tion severity VO. Assume the result is

$$VO = 8.0 \text{ mm/s}.$$

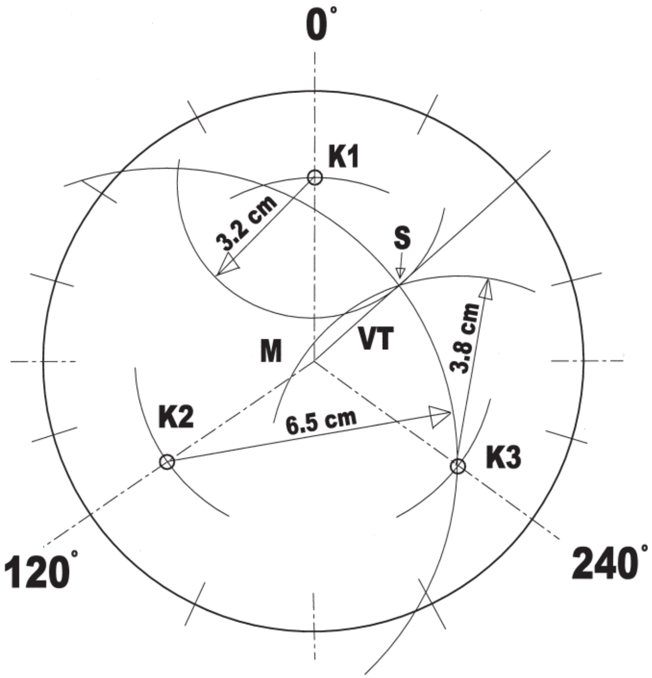
Care must be taken to ensure that VO is measured at the rotational fre- quency.

Then the trial weight MT is fixed to the points T1, T2, T3 in turn, and each time a trial run of the machine and a measurement of vibration severity, V, is performed. Assume the measuring results are

$$V1 = 6.4 \text{ mm/s}$$
$$V2 = 13.0 \text{ mm/s}$$
$$V3 = 7.6 \text{ mm/s}$$

The trial weight in this example is MT = 22.5 g.

The measurement can best be evaluated on an prepared sheet showing a basic outline graduated in degrees as shown on the following page.



Values		Results	
without trial weight VO	= 8.0	VT = 5.0	WA = 307 deg
Trial weight at 0 deg V1	= 6.4		MA = 36 g
Trial weight at 120 deg V1	= 6.4		
Trial weight at 240 deg V1	= 6.4		
Trial weight MT	= 22.5		

A suitable scale is selected for the purpose: 1 cm corresponds to 2 mm/s. A circle is drawn with M as its center and VO as its radius. In our example, this would mean: r = 4.0 cm. The points of intersection of this circle with the three legs 1, 2 and 3 are the points K1, K2 and K3.

The next step is to draw circles around these points: with a radius of V1 (i.e. 3.2cm) around K1; with a radius of V2 (i.e. 6.5 cm) around K2; and with a radius of V3 (i.e. 3.8 cm) around K3. It does not matter if these circles do not fit on the paper in their entirety. The crucial point is the common intersection of all three circles, the point S.

The angle WA represented by the leg M to S gives the angle at which the balance weight MA must be fixed to the rotor; in this case,

WA = 307º.

The length of the segment M to S is measured and denoted by VT. In the example,

VT = 5.0 mm/s.

As a general rule, the balance weight is fitted at the same distance r from the center as the trial weight. The balance weight MA is therefore calculat- ed according to the formula

MA = V0·MT/VT = 8 x 22.5 / 5

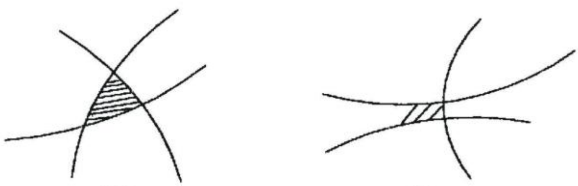
The result obtained in our example is:

Correction weight MA = 36 g, located at 307 degrees from 0

If the balance weight is to be fixed at a shorter or greater distance R than from the axis, the balance weight must be calculated as follows:

MA(1) = r / R x MA

In most cases the measurements will not intersect at one point as shown in our example; intersection S has been somewhat idealized. Most instances will resolve a small triangle (detail a) or in exceptional circumstances there may not even result a complete triangle (detail b). In these cases a mean value should be selected from the hatched portion of the intersect area.



a b

Should the triangle or the hatched area for once be very large, vibration is not attributable to unbalance but to other defects. In that case, another vibration analysis must be carried out, preferably backed by shock pulse measurement.

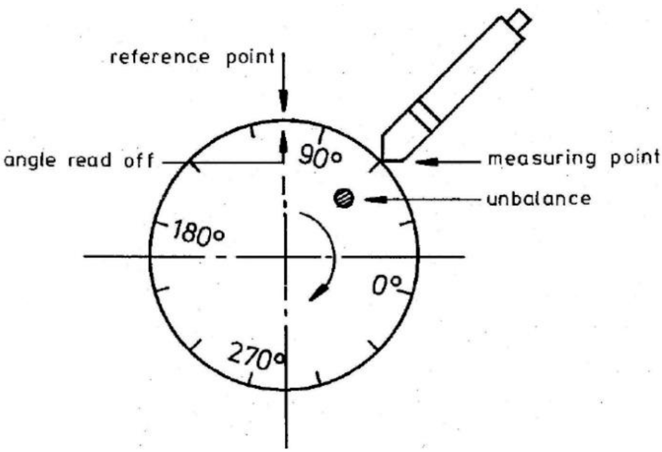
Single-plane stroboscope method

The second procedure considered here for single-plane balancing is the **single-plane stroboscope method**. The additional instrument required is a stroboscope which can be synchronized with the machines rotational frequency via a vibration analyzer. This method is more expensive than the previous method. But it has the advantage that just two trial runs are required instead of four, and that evaluation can be made both graphically and by calculation.

For balancing, the rotor is again graduated anticlockwise in degrees, from 0º to 360º. During measuring, this graduation is viewed by the light of the synchronized stroboscope so that the machine appears to be stationary.

The first run is made without a trial weight. The parameters to be recorded are the vibration severity VO and – with the aid of the stroboscope – the angle WO of the occurring vibration SO. Care must be taken to ensure that VO is measured selectively for the rotational frequency.

For measurement of angles, a reference mark is fixed, e.g. at the top, which is then used for both trial runs and where the angle is read off.



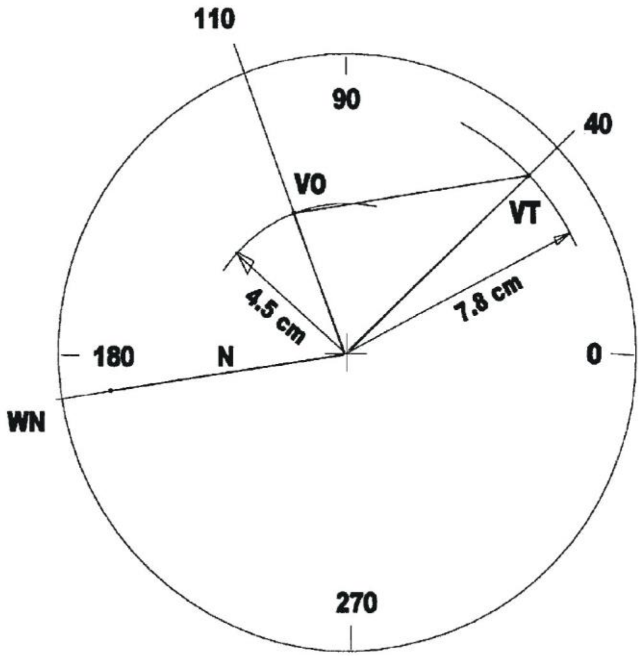
In the second trial run, a trial weight MT is attached at 0º, at a distance r from the rotational axis. Then the vibration ST is recorded, with its vibration severity VT and its phase angle WT, just as in the first trial run. Both, the measuring point for the vibration pick-up and the reference mark for measurement of angles, must remain unchanged in the process.

The following recorded values are assumed as an example:

SO: VO = 9 mm/s; WO = 110º ST:
VT = 15.6 mm/s; WT = 40º
Trial weight MT = 20 g

Correction by graphical method

The first step is to plot SO and ST to a suitable scale in a circle graduated in degrees, the final points being O and T. The arrow from T to O is then shifted parallel to the center. Its length is measured and is denoted by N, its angle is WN.



The amount of balance weight MA is calculated as follows:

$$MA = SO \cdot MT/N$$

Its angular location is

$$WA = WO – WN\ WA$$

The result is

$$= 280º, MA = 12\ g.$$

Correction by calculation

For an evaluation by calculation, complex numbers must be used. Unbalance and vibration may be considered complex numbers in polar co-ordinates, and equally the balance weight.

Conversion of these complex numbers into Cartesian co-ordinates will yield the expressions below.

The given quantities are: VO, WO, VT, WT, MT.

VO = 9 mm/s; WO = 110º; VT
= 15.6 mm/s; WT = 40º; MT
= 20 g

Stage 1: conversion of SO and ST into Cartesian co-ordinates.

SOX = VO·cos (WO)	SOX = 9 mm/s·cos (110º) = -3.08 mm/s SOY
= VO·sin (WO)	SOY = 9 mm/s·sin (110º) = 8.46 mm/s STX =
VT·cos (WT)	STX = 15.6 mm/s·cos (40º) = 11.95 mm/s
STY = VT·sin (WT)	STY = 15.6 mm/s·sin (40º) = 10.03 mm/s

Stage 2: calculation of SO – ST

NX = SOX – STX	NX = -15.03 mm/s
NY = SOY – STY	NY = -1.57 mm/s
$N^2 = (NX)^2 + (NY)^2$	$N^2 = 228.4$ mm/s

Stage 3: calculation of the balance weight

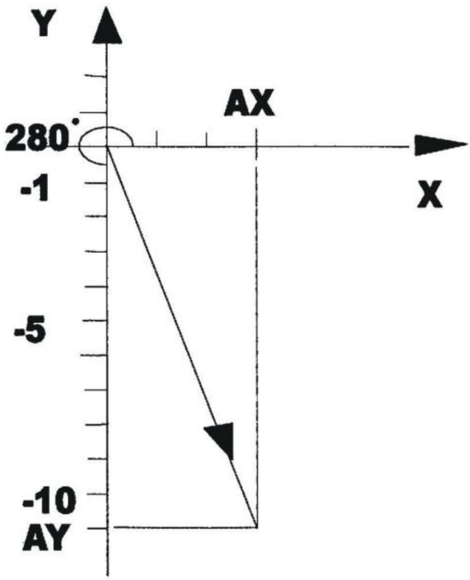
AX = (SOX·NX + SOY·NY)·(MT/N²)	AX = 2.89 g AY
= (SOY·NX - SOX·NY)·(MT/N²)	AY = -11.56 g

Stage 4: conversion of the balance weight into polar co-ordinates MA

= $(AX)^2$+ $(AY)^2$ MA = 11.9 g
WA = √[$\tan^{-1}$(AY/AX)] WA = -76º

With respect to WA, allowance must be made for the fact that a calculator will indicate the principle value of $\tan^{-1}$(AY/AX), and also that when AX = 0, WA is not defined at all. It is therefore advisable either to perform an extra graphical evaluation (doubling as a further check) to definitely identify the angular location, or to plot the balance weight in Cartesian co-ordinates. The exact value of WA is then obtained by adding or subtracting appropri- ate multiples of 180º.

This is the case in our example where the result of both the graphical eval- uation and the plotting of the Cartesian co-ordinates is about 285º. 360º must thus be added to the previously calculated value to obtain the exact value of WA = 284º.



Just as with the single-plane three-point method, the calculated balance weight **MA** must be attached at the same distance **r** from the center as the trial weight. In case another spacing is preferred, the balance weight must again be converted according to the principal of the lever.

With the single-plane stroboscope method, calculation is as follows:

If U is the unbalance, the first measurement yields

$$\text{(i) } SO = a \cdot U$$

with a complex parameter a.

In the second measurement, the trial weight T is added to the unbalance U so that the result of the measurement is

$$\text{(ii) } ST = a \cdot (U+T)$$

From (i), a = SO/U, and inserting this into (ii) yields

$$ST = SO \cdot (U+T)/U$$

Resolving to U yields

$$U = (SO \cdot T)/(ST-SO)$$

The balance weight A must be opposite to U:

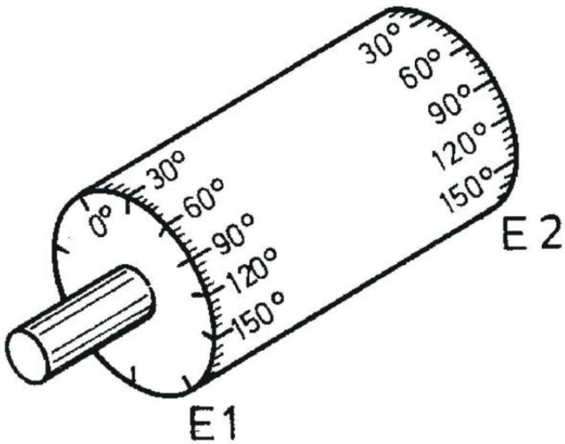
$$A = (SO \cdot T)/(SO-ST)$$

Two-plane stroboscope method (devised by A Wahrheit)

If the rotor assembly concerned has either two rotors or one rotor whose width exceeds its diameter, two-plane balancing methods must be used.

We will review one two-plane balancing method and will show calculations to obtain balance correction weights and location.

The two ends of the rotor represent the two balancing planes E1 and E2.



This method is similar to the single-plane stroboscope method, especially in that identical equipment is required, i.e. a frequency-selective vibration measuring instrument and a stroboscope. A graduation in degrees is again applied to both balancing planes, the positions being identical on both planes.

A total of three trial runs must be made; the first one without a trial weight and the other two with a trial weight added. The trial weight is always at- tached at 0º, first in the balancing plane E1 and then in the balancing plane E2. With all three trial runs, either balancing plane is measured in respect of vibration severity and, with the aid of the stroboscope, phase angle.

Twelve recorded values are thus obtained, and the 13th value is the trial weight.

Without trial weight:

E1: $V10 = 14.7$ mm/s	$W10 = 56º$
E2: $V20 = 12.0$ mm/s	$W20 = 75º$

With trial weight at E1:

E1: $V11 = 27.3$ mm/s	$W11 = 33º$
E2: $V21 = 14.1$ mm/s	$W21 = 55º$

With trial weight at E2:

E1: $V12 = 18.5$ mm/s	$W12 = 45º$
E2: $V22 = 23.9$ mm/s	$W22 = 44º$

Trial weight: MT = 60 g

On the basis of the 13 values recorded, a calculation is then made to determine the angles in both balancing planes at which the balance weights are to be fixed, and also the value of those weights. Four values are thus obtained: MA1; WA1 and MA2; WA2. The calculating procedure is as follows (refer to the chart on the next page):

Step I

The measuring results are entered in the top portion of the evaluation sheet.

Step II

Columns I and II are completed as specified. The highest result of operation I is underlined, the other results of 1 are then divided by this value, multiplied by 10 and entered in column VX on the right. The angles resulting from operation II are moved directly to the column WX. Should an angle exceed 360º, however, 360º must be subtracted until the result is in the range from 0º to 360º.

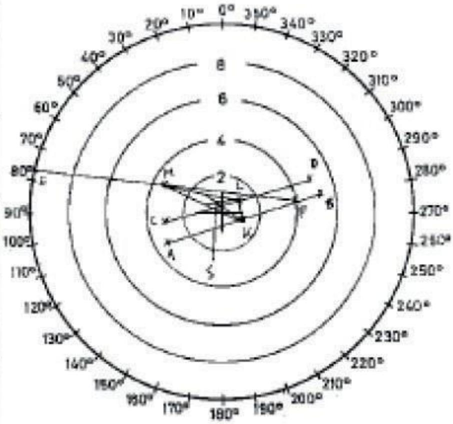
The points A to F are plotted with amount and angle into the circle. The radius of the circle corresponds to 10. The amounts and angles of K (center of AB), L (center of CD) and S (center of gravity of the triangle KLM, M center of E F) are read off and entered in the respective spaces.

Balancing — Two-plane stroboscope method

Machinery: ⁄ Date: ⁄

Values:	Plane E1 amount angle	Plane E2 amount angle	Trial weight:
1. without trial weight:	V10 = 14 W10 = 56	V20 = 12.0 W20 = 75	MT = 60 g
2. trial weight at E1:	V11 = 22.3 W11 = 33	V21 = 14.4 W21 = 55	WT = 0°
3. trial weight at E2:	V12 = 18.5 W12 = 45	V22 = 23.5 W22 = 44	

I	II		VX	WX
V12 · V20 221	W12 45 +W20 75 120	A	3.4	120
V10 · V22 351	180° +W10 56 +W22 44 280	B	5.38	280
V10 · V21 201	W10 56 +W21 55 111	C	3.18	111
V11 · V20 328	180° +W11 33 +W20 75 288	D	5.42	288
V11 · V22 654	W11 33 +W22 44 7	E	10	77
V12 · V21 261	180° +W12 45 +W21 55 280	F	4.00	28b



VK = 1.33 WK = 250	VL = 1.11 WL = 290	VS = 0.67 WS = 20

III	$\frac{1}{3} \cdot \frac{MT}{VS}$ = 29,85 (Z1)	360° − WS = 340° (Z2)

Balance at E1		Balance at E2	
MA1 = Z1 · VK	WA1 = Z2 + WK	MA2 = Z1 · VL	WA2 = Z2 + WL
MA1 = 39.7	WA1 = 230°	MA2 = 33.13	WA2 = 270°

Prüftechnik

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Step III
Determines two auxiliary quantities, Z1 and Z2, which are eventually used to calculate the sizes and angular locations of the balance weights. The principals of the two-plane stroboscope method are as follows:

As with the single-plane stroboscope method, the unbalances in the balancing planes E1 (U1) and E2 (U2) as well as the trial unbalance T are defined as complex numbers.

Vibration severity levels without trial weight are as follows:
S10 = a·U1 + b·U2 S10 = (V10; W10)
S20 = c·U1 + d·U2 S20 = (V20; W20)

Where a, b, c and d are complex parameters, b and c represents the effects of the unbalances on the other balancing plane.

Since the equations contain six unknown quantities (a, b, c, d, U1, U2), four more equations are needed. They are obtained from the two trial runs with trial weight.

S11 = a·(U1 + T) + b·U2 S11 = (V11; W11)
S21 = c·(U1 + T) + d·U2 S21 = (V21; W21)

S12 = a·U1 + b·(U2 + T) S12 = VG12; W12)
S22 = c·U1 + d·(U2 + T) S22 = (V22; W22)

These give the following solutions:

U1 = {[(S10 x S22) - (S20 x S12)] / [S20x(S12 - S11) + S21x(S10 - S12) + S22x(S11 - S10)]} x T U2 =

{[(S11 x S22) - (S10 x S21)] / [S20x(S12 - S11) + S21x(S10 - S12) + S22x(S11 - S10)]} x T

With regard to the balance weights M1 = -U1 and M2 = -U2.

It is obvious from the preceding pages, that calculating unbalance corrections even on simple rotors in both one and two planes involves a series of complex graphical and/or mathematical calculations. The balancing meth- ods reviewed are based on the assumption of ideal linear conditions, which is rarely encountered in reality. Corrections may therefore require a second and in extreme cases a third balancing procedure to achieve the balance quality required.

There is however a more straightforward method of in-situ dynamic balanc- ing which substantially reduces the time required to achieve an acceptable balance quality and all but eliminates the complex calculations of the pre- vious methods.

Dedicated balancing systems, such as **VIBXPERT® II Balancer,** performs balancing in one or two planes. It graphically describes balance weight corrections and locations or allows the operator to select suitable mount- ing locations, e.g. on the blades of a fan, and automatically compensates weight requirements to accommodate this.

The time saved and the improved accuracy from such systems makes the initial investment well worthwhile

Trial weight calculation

Most suitable trial weights are small screw cramps that are easy to install and to remove at any location desired. Screws or bolts can be used as well, provided the respective locations are fitted with the tapped holes required. Additionally, an exactly weighed quantity of a special putty can also serve as a trial weight.

It goes without saying that the trial weights must be weighed, and secured to the rotor carefully. If balancing work needs to be done fairly frequently, a set of trial weights could be prepared and kept in readiness.

Instruments, such as **VIBXPERT® II Balancer**, described in the next pages will calculate the trial weight to be used and will identify the location. If however the unbalance is to be calculated or resolved graphically it is necessary to accurately calculate the amount of trial weight to be used as follows.

A convenient size of the trial weight can be determined by means of the formula below, where MT is the trial weight, G the weight of the rotor, s the displacement amplitude of the vibration, and r the distance of the trial weight from the center.

$$MT = G \bullet s / r$$

If a bolt is to be fitted as a balance weight, the respective tapped hole must be deducted from the weight of the bolt.

Balancing safety

When performing in-situ balancing on rotating machines, it is essential that operators are aware of the in plant and local HSE safety requirements for working on the machines. And must be mindful of the implications and safety requirements when mounting balance weights to machines. Guide lines supplied below should be the minimum precautions taken.

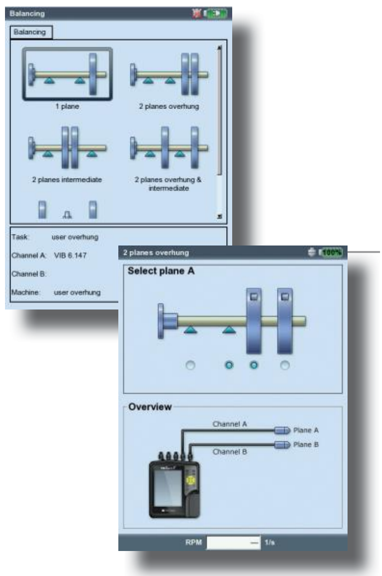
- Isolate machines to avoid accidental switching on when work is being carried out.
- When assembling measurement components ensure that no fixtures or cables or other objects project within the vicinity of rotating machine parts.
- Follow manufacturers directions when attaching balance weights.
- Always operate within the maximum permissible rpm of the rotor.
- Always pay particular attention to the calculation of the trial mass to be used for the initial balance run. Oversize trial masses can have grave consequences for machines and operator alike.
- During trial and balancing runs ensure no operators are within the radial vicinity of the machine.
- Close all access guards and doors to the rotor before switching on the machine.
- Do not exceed the permitted number of machine start ups for any given time.
- Before balancing begins determine the cause of unbalance and where possible remove any material which may be caked on to the rotor. Weld any cracks in impeller blades or replace the impeller if this is not possi- ble.

PRUFTECHNIK balancing instruments

The principle of operation of **VIBXPERT® II Balancer** is similar to that of resolving balance by graphical or calculation method. Each method requires the following:

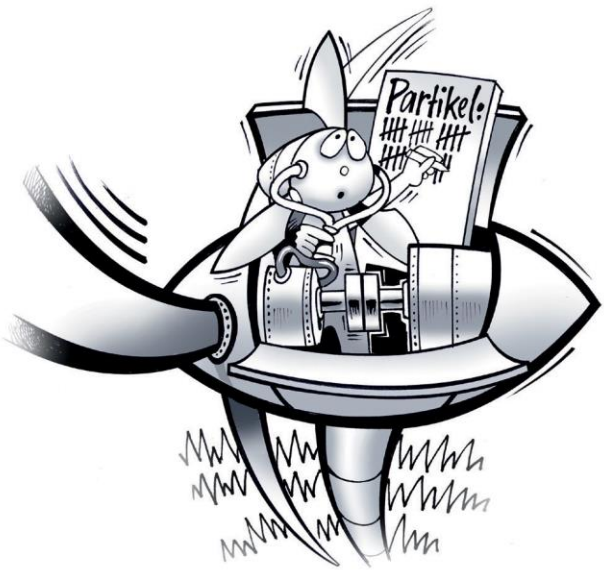
- A preliminary machine measurement to establish current unbalance condition
- Either one or two trial measurements with trial weights, depending on number of planes to balance; one plane requires one trial run.
- A trim run to correct rotor unbalance; it may in some cases be necessary to carry out two trim runs to achieve final balance condition.

The fundamental difference between **VIBXPERT® II Balancer** and manual balance methods is that the instrument carries out all necessary calculations and reports corrective action to be taken. Different machine type configurations can be initially selected to establish either single or two plane balancing requirements.



Section 4

Wear Debris Analysis



An overview

Wear debris analysis (WDA), like previously discussed methods of maintenance actions which are designed to improve machine operating life, is just one section of a comprehensive maintenance regime. On its own it will not prove to be an all encompassing solution for improving plant operating performance. WDA however does provide in some instances a very good early warning system for incipient machine failure.

As its name suggests the concept of WDA is that it is used to measure debris in machine lubrication systems. A lack of debris is usually a good indication that little or no component wear or breakdown is taking place. Conversely a significant amount of debris is a good indication that some- thing is taking place within the machine system that should not be happen- ing. The type and size of debris will further indicate the source and extent of the component failure.

Of course all components wear over time and there will hardly ever be a situation where no debris is found in a lubrication sample. What is import- ant is the size and increase of debris encountered. WDA data trending is therefore used extensively to measure over time the ongoing condition of a machine component, such as a bearing. It is of course perfectly pos- sible to obtain a reasonable indication of a component condition by one “snapshot” sample analysis. A significant amount of debris will indicate progressive and often terminal component wear requiring almost immedi- ate intervention before machine failure. Little or no debris will indicate the component is in OK condition.

Actually it’s not quite that simple. Debris found in lubrication samples needs to be analyzed, so that the particle size and content can be used as a guide to the source of the debris and component wear. WDA was initially heavily used in the mining industry, where airborne particles of coal dust were often ingested into machinery contaminating lubrication mediums for bearings, gears and other “sliding” components. WDA was a very good early indicator of the level of contamination and the extent to which these abrasive airborne particles were damaging the sliding components of the machines. In many cases WDA was the first line indicator of possible ma- chine component failure.

Care should however be taken, as what you see is not always what you get. The accuracy of readings and analysis depends entirely upon the qual- ity of the sample taken. Samples taken a long distance from the target component can be contaminated “down the line” as can samples taken in “unclean” containers or with already contaminated sampling systems. As a consequence, data can be corrupted and meaningless before the analysis takes place. Care has to be taken when considering sample location and sampling instruments and methods.

Review of WDA methods

Generally there are two types of WDA methods that are employed in in- dustrial applications: **off-line analysis** and **on-line (or in-line) analysis**. These methods represent distinctly different approaches to the application of WDA. On-line analysis, as its name suggests, is incorporated into the system process. It yields more immediate results than the more commonly adopted off-line method, but on-line analysis relies heavily upon particle count measured against particle size. Detailed analysis of the debris content is generally not possible in on-line analysis.

On-line analysis

Within On-line analysis methods there are a number of different system options designed to catch or measure specific wear debris particles.

- **Filter blockage** – where flow passes through a filter with a known pore size, changes in flow characteristics indicate an increase in particle size causing an in line blockage of the filter.
- **Magnetic attraction** – a magnet is used to attract debris particles into a trap, further on-line measurement is then performed to assess particle volume.
- **Optical** – detectors are positioned within a light source focused within the flow line. These detect particle size either by diffraction of the light source or by using a photo diode to detect a drop in the light intensity measured.
- **Film ware** – fluid is projected at a surface coated with a thin conductive layer, as debris wears the film surface away the resistance across the film increases. This indicates developing debris levels and consequent wear problems.

- **Ultrasonics** – an ultrasonic beam is focused into the fluid, an echo sig- nal is measured as it hits debris. The echo changes (increases) as more debris is encountered.

Within each sub group there are a number of different techniques and a differing number of advantages for each system used. Depending upon the specific requirements of the particle analysis required, the engineer will have to establish the type of on-line system required. If for example only ferrous debris is to be analyzed, then a magnetic on-line system should be used. If however none ferrous components such as white metal or phosphor bronze bearing debris is to be measured, a device to sense these materials is essential. Likewise, if all debris is to be measured, then a simple filter blockage system can be employed to deliver the required debris data. All on-line systems however suffer from the universal problem (with the exception of the filter block system) in that they are analysis specific. They focus on targeted types of debris analysis rather than the broad spectrum analysis possible with off-line WDA.

In one area however, on-line analysis does offer a distinct advantage over off-line analysis in that it makes possible WDA in machines and systems that would not normally be accessible to humans without considerable ef- fort or potential danger. Applications such as wind turbine gearboxes and marine thruster gearboxes are typical machine systems which cannot easily be measured for wear debris without on-line analysis.

Off-line analysis

This method is more universally used in general industrial applications and it does offer the analyzer much greater scope for evaluation of a wide variety of debris. As noted earlier however, considerable care has to be taken in obtaining the debris sample and storing it for later analysis.

Off-line analysis can more or less be sub-divided into three types of anal- ysis methods. **Patch analysis**, **ferrography** and **particle concentration.** Within each, there are a number of methods of sample analysis possible and scope for greater or lesser degrees of detailed analysis.

Patch analysis is probably the most common method of off-line analysis of debris. A sample of the extracted lubricant is passed through a filter mem- brane, debris is deposited on the filter for examination. The advantage of this method is that all debris types are deposited on the filter allowing for a comprehensive analysis of debris if required. This can be achieved in a number of ways ranging from the very rough to the highly detailed as follows:

- **Rough analysis:** Color analysis – A comparison is made between the filter with the debris deposit and a new clean filter. Generally this anal- ysis is used to establish grade of contamination ranging in three or four standard conditions of good, acceptable, and warning.
- **Rough / intermediate analysis:** Standard comparisons – This uses a standard set of filters made in accordance to ISO 4406 level of debris concentration or cleanliness. A comparison is made between the debris filter and the ISO filters again to establish good, acceptable or warning levels of debris concentration. This type of system is often found in the many portable WDA analysis kits available.
- **Intermediate analysis:** Particle counting – Usually carried out on a fil- ter membrane with a grid etched on to the surface. The total quantity of debris particles are counted in any one grid area, the total particle count then being multiplied by the number of grids. This system is classed here as intermediate simply because the size and distribution of debris can be further checked using a microscope.
- **Intermediate / advanced analysis:** Using visual inspection – A de- tailed observation of debris particles is carried out. This allows analysis of shape, size, type and concentration of debris contamination. Micro- scope examination, different types of lighting and operator experience will determine accurately the level and general composition of contami- nation of the off-line sample.
- **Advanced analysis:** Using a scanning electron microscope – This pro- vides access to highly detailed evaluation, analysis and reporting of de- bris sample concentrations including the specific elements of the debris in each analysis sample.

Particle concentration – This method can be carried out using either visual or by magnetic methods of particle analysis. Usually the sample is transferred into an optically clear glass vessel. The sample contents are then placed on a shaker to uniformly distribute the debris throughout the sam- ple. The debris is then evaluated either by an optical device or by magnetic induction to view particle size, type and distribution. Obviously when using magnetic induction only ferrous particles can be observed in the sample.

This analysis method is usually used in conjunction with more detailed laboratory analysis of particle content.

Ferrography – As its name suggests, this analysis method concentrates on ferrous and other magnetic particles. It is extremely useful in establishing the content of ferrous particles in a sample, but virtually useless in estab- lishing other non magnetic particles such as phosphor bronze or white metal. The resulting slide taken using this method will allow identification of ferrous wear particles. When the slide is heated, ferrous particles can be compared to a standard wear particle color chart to establish actual debris particle content of the sample.

Assessing the results from WDA analysis

As with all methods of condition based maintenance, a level of on plant experience is necessary to accurately establish whether intervention is necessary on a particular item of equipment. If for example using a rough or intermediate analysis of a WDA sample, the color of the slide is significantly different from the control sample or slide then its pretty well a “no brainer” that at the very minimum, further and more detailed analysis should take place. This then becomes a two stage WDA analysis strategy. A second WDA sample should be taken and sent to a laboratory for detailed analysis. Depending on the report content action can then be taken to prevent dam- age to machine components.

Rough or intermediate analysis then becomes an important link in the WDA maintenance strategy. Whereas the maintenance manager may have previously dismissed this technique as not good enough for the plant, it does provide an easily obtained and readily available source of plant operating condition evaluation. Its added advantage is that it requires very little training of the operative to compare samples with pre-established reference slides. It is in this area that the many portable oil analysis laboratories provide a good introduction product to WDA.

When the exercise of WDA is extended to a detailed analysis of a debris sample the usual route is to send the extracted sample to a specific laboratory. Analysis will be comprehensive and should be accompanied by a detailed report of the sample including an overall spectrograph of the sample together with a description and analysis of each element found in the sample. This should include a list of the proportions of each element in the sample.

Where laboratory analysis is a routine procedure carried out, there should also be a trend graph of samples over time available. Depending upon the laboratory, and the price you are prepared to pay for analysis, it is also possible to have an indication of the source of any significant increase of a particular debris element and thus any component that may be showing signs of advanced wear. Without a trend graph for example, the laboratory analysis of a sample is no more than a snap shot of the present lubrication condition of the machine. It gives no indication of any change in operating state and is therefore of only limited use or can identify machines only in an advanced state of component failure.

It is of course very important that each sample sent for analysis is accurately labeled with location, machine, date and time of the obtained sample. As noted previously it is also important that the sample is taken from the same location each time in order for the analysis to have any serious validity.

Sampling procedure and location

It stands, saying again, location and repeatability are all important with WDA sampling. The best possible arrangement is to have the sampling point installed as a integral part of the machine system, located in a pres- surized part of the system. This will provide a guaranteed repeatable sample process and hence reliable results from WDA.

Where this is not possible, the sample has to be obtained from a location that is repeatedly accessible such as an oil reservoir or sump. It has to be accepted that the results may be compromised by other contaminants but as with all condition monitoring something is better than nothing.

Where it is possible to retrofit a sampling point, is ideal, and there are many options available. Each manufacturers sample point will provide a secure drip free, clean and easily attachable probe location. All manufacturers will also supply sample probes to fit their systems.

If the operator has to rely on sampling from a non fixed point, then it is necessary to use an extraction syringe attached to a sample bottle. This will enable the operator to suck the sample into the bottle. Before taking a sample in this way, it is necessary to ensure that the sample tube is thoroughly flushed with clean (same) lubricating fluid as is to be extracted. Care must be taken to ensure that the sample tube is suspended in the fluid and not sitting in the lower settled debris in the bottom of the reservoir.

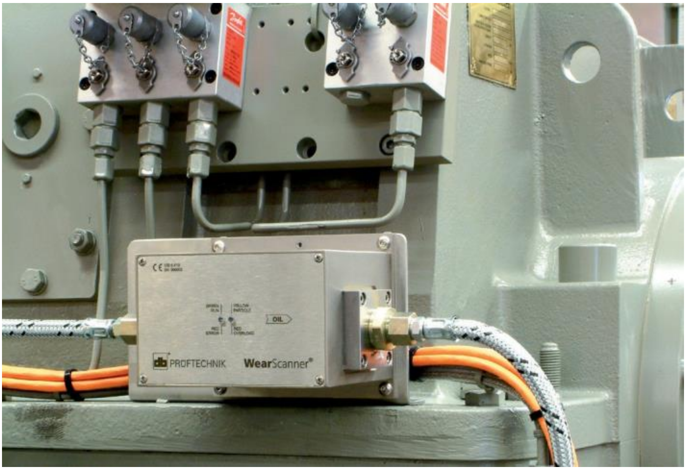
It is not good practice to simply dip the sample bottle into the reservoir to obtain a fluid sample. This potentially introduces a number of additional contaminants that may corrupt the fluid sample being taken.

The forgoing information should be read as a guide only to the possibilities of introducing a WDA system to your plant. The bibliography at the end of this book lists further essential reading that will assist the introduction of a successful WDA regime.

PRUFTECHNIK have introduced an on-line WDA system designed to give early warning of bearing and gearbox damage in critical equipment such as wind turbine and marine thruster gearboxes. The following notes describe the system and its operating principal.

PRUFTECHNIK WDA system

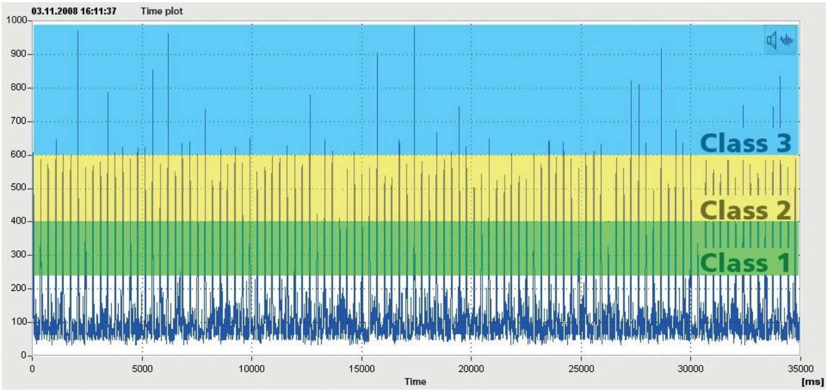
WEARSCANNER® monitors particle distribution in lubricating oil. The sys- tem detects metal particles in the circulating lubricating oil, records them in real-time and classifies them by size in accordance with ISO 16232.



When the particle count in a particular size class exceeds a defined thresh-old, the system indicates an alarm via the switching signal. The internal ring buffer can accommodate one year of data when values are stored at two minute intervals. They can be read out at any time for trend analysis or when thresholds are exceeded. If the system is integrated in a data network, it transfers measurement data to the system control via Modbus. When the WEARSCANNER® is used in combination with the VIBROWEB® XP Online Condition Monitoring System, the data is transferred to the Monitoring Center by e-mail.

The particle count is influenced by the oil temperature, flow rate, air and water content, and darkening (especially when using optical methods). The patented WEARSCANNER® method, based on the eddy current principle, is not influenced by these restricting parameters.

How WEARSCANNER® works

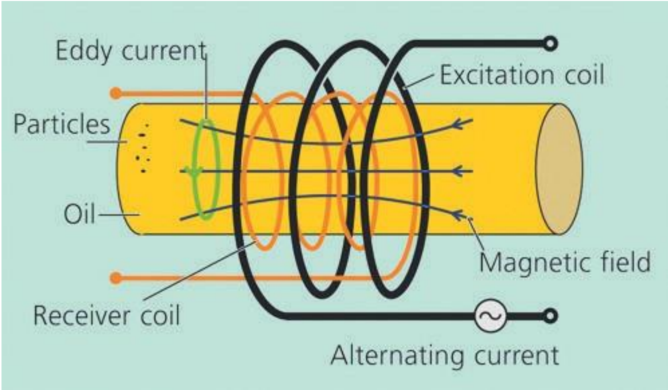


- Each peak represents a particle that flows through the sensor tube.
- The amplitude indicates the size of the particle.
- The peaks are counted in set time intervals – and the number of peaks per time interval is transmitted via ModBus TCP.
- The size categories – three classes in this case – are configured for each particular situation.
- The time resolution and scan rate can be selected.
- The sensitivity can be adapted by adjusting the gain, power and filter to the machine application.

The eddy current measuring principle

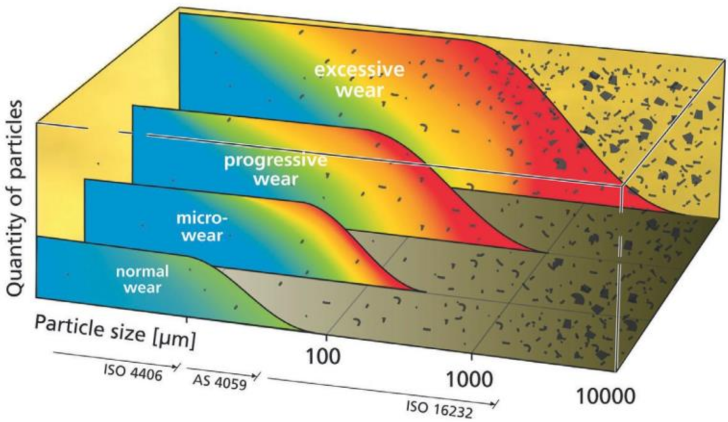
An **excitation coil** that generates an alternating magnetic field induces eddy currents in the oil. A **receiver coil** detects the resulting eddy current density.

Electrically conductive, i.e. ferritic and non-ferritic contaminants in the oil change the current flow between the excitation and the receiver coils. In or- der to detect even minuscule particles at high flow velocities WEARSCAN- NER® utilizes the fast reacting differential coils method with two receiver windings connected in opposition.



Particle size distribution in oil

As a general rule, the greater the particle size, the greater the damage.



Section 5

Resources

About PRUFTECHNIK

PRUFTECHNIK is an international company involved in the design, manufacture and sale of measurement and diagnostic systems used for laser alignment, condition monitoring, and nondestructive testing.

Originally founded in 1972 in Munich by Dieter Busch, the company first acted as a marketing company for a range of metal detector systems and bearing testing equipment. As the company became established, a new range of its own designed eddy current testing systems was developed, fol- lowed over the next few years by the world's first laser shaft alignment sys- tem **OPTALIGN**. This was later followed by vibration analysis and dynamic balancing systems; all designed with the unique PRUFTECHNIK approach to simplicity of operation and application.

PRUFTECHNIK head office is still located in Germany, but has established subsidiaries in the UK, Belgium, Netherlands, France, Singapore, Italy, Spain, Canada, U.S.A., Turkey, Japan, Russia, China, Poland, India, Brazil and Dubai. In addition to this, PRUFTECHNIK has a network of more than 70 sales partners throughout the world.

For PRUFTECHNIK being close to customers is the essential ingredient to its past and future success. Many of the current products are the result of long established working relationships with its customers. Being close to the market is more than a slogan.

In addition to the unrivaled range of products, PRUFTECHNIK has an international training, service and support organization that provides product training, consultation services, and specialized services in areas such as turbine alignment, installation, commissioning and planning of condition monitoring systems.

PRUFTECHNIK holds more than 200 world wide patents together with 100 registered trademarks for its innovative range of products.

Further Reading

Shaft Alignment

- Shaft Alignment Handbook, 2nd edition, John Piotrowski, ISBN 0-8247-9666-7
- Maintenance Fundamentals, Keith Mobley, ISBN 0-7506-7151-3
- Introduction to machinery analysis, John Mitchell 1993, ISBN 0-87814- 401-3
- Infrared Thermography and Laser alignment technologies, 1994, Infra-spection Institute, Frank Pray and Bruce Bortnem
- Drives and Seals, M J Neal, ISBN 0 7506 0981 8

Vibration Analysis

- The Vibration monitoring Handbook, Coxmoor publishing, ISBN 1 90189 200 X
- International conference on condition monitoring, Coxmoor publishing, ISBN 1 90189 214 X
- Management guide to condition monitoring in manufacture, Institute of Production engineers, ISBN 0 85510 037 0
- Vibration Spectrum analysis, Steve Goodman, 2nd Edition, ISBN 08311 30881

Balancing

- Dynamic balancing of rotating machinery, JB Wilcox, ISBN 0273429590
- Machinery Vibration Balancing, Victor Wowk, ISBN 00707 1938 1
- Balancing accuracy of rigid rotors No 8, Dott.Ing Luigi Buzzi, booklet, III edition 1989, translated by P F Kercher

Wear Debris Analysis

- Wear Debris Analysis, Brian Roylance, Trevor Hunt, ISBN 1 90189 202 6
- A Handbook of Oil Analysis, Augustus H Gill

ISO standards

- Mechanical vibration – Evaluation of machine vibration by measure- ments on non-rotating parts: ISO 10816 - 3
- Field balancing equipment – Description and evaluation ISO 2371
- Balancing quality of rotating rigid bodies ISO 1940.-1973
- Cleanliness of components of fluid circuits ISO 16232-1:2007

including:

  - Method of extraction of contaminants by ultrasonic techniques
  - Particle mass determination by gravimetric analysis
  - Method of extraction of contaminants by ultrasonic techniques
  - Particle sizing and counting by microscopic analysis
  - Particle sizing and counting by automatic light extinction particle counter

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Productive maintenance technology